DOT/FRA ORD - 77/13X

1977 Technical Proceedings
14th Annual Railroad
Engineering Conference

"R&D and RAILROADING: 1977"

DEPARTMENT OF TRANSPORTATION
Federal Railroad Administration

March 1978

Technical Report Documentation Page

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

4. Title and Subtitle	5. Report Date
14th Annual Railroad Engineering Conference Proceedings, "R&D and Railroading: 1977"	March 1978
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No.
9. Performing Organization Name and Address Department of Transportation Federal Railroad Administration Office of Research and Development Washington, D. C. 20590	10. Work Unit No. (TRAIS) 11. Contract or Grant No.
12. Sponsoring Agency Name and Address Department of Transportation Federal Railroad Administration Office of Research and Development Washington, D. C. 20590	13. Type of Report and Period Covered Conference Proceedings October 18 - 20, 1977 14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

This report constitutes the proceedings of the three day railroad engineering conference held at the University of Southern Colorado on October 18 - 20, 1977. Conference papers were presented from the Federal Railroad Administration, Office of Research and Development, the railroad industry, and the Association of American Railroads. Generally, the papers covered a review of the R&D activities in the railroad industry during 1977. A tour of the Transportation Test Center Facilities was also included.

17. Key Words	18. Distribution Statement
Railroad Technology	Document may be purchased from the National Technical Information Service 5285 Port Royal Road Springfield, Virginia 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
none	none	418	

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

TABLE OF CONTENTS

FOURTEENTH ANNUAL RAILROAD ENGINEERING CONFERENCE

R&D AND RAILROADING: 1977

PAGE

SESSION III THE STATUS OF RAIL SAFETY RESEARCH

SESSION IV REPORT ON ASSOCIATION OF AMERICAN RAILROAD'S R&D ACTIVITIES

SESSION VI REPORT ON STATUS OF FAST & RAIL DYNAMICS LABORATORY

SESSION VII TRANSPORTATION TEST CENTER

EVENING SESSIONS

EXECUTIVE SUMMARIES

Executive Summaries have been included at the beginning of each session for the convenience of the reader.

PROCEEDINGS OF THE 14TH ANNUAL RAILROAD ENGINEERING CONFERENCE UNIVERSITY OF SOUTHERN COLORADO PUEBLO, COLORADO, OCTOBER 18–20, 1977

"R & D AND RAILROADING: 1977"

U.S. DEPARTMENT OF TRANSPORTATION FEDERAL RAILROAD ADMINISTRATION

The opening session of the 14th Annual Railroad Engineering Conference was called to order by Robert E. Parsons, Associate Administrator, Research and Development, Federal Railroad Administration. Mr. Parsons served in the capacity of conference chairman. Chairman Parsons introduced Dr. Richard Pesqueira, President of the University of Southern Colorado.

Dr. Pesqueira welcomed the delegation on behalf of the University.

Chairman Parsons then gave some introductory remarks and outlined the conference format, after which he turned the meeting over to Dr. Donald Spanton, Chairman of Session I, "The Status of Freight Systems, R&D.

Robert E. Parsons
Associate Administrator for Research and Development
Federal Railroad Administration

Robert E. Parsons was appointed to his present position in March, 1975. Previously he served as Director of the Secretary of Transportation's Research and Development Plans and Resources Program. He began his federal career in 1964 as a value analysis engineer for the Federal Aviation Administration. Two years later, he was appointed chief of the Analysis and Control Division, and from 1969 to 1971, he was Acting Deputy Director, Office of Supersonic Transport Development. Prior to his federal experience, Parsons was with the Martin Company and the Cincinnati Milling Machine Company.

Parsons received his B.S. degree in Mechanical Engineering from the University of Cincinnati and his M.S. from Drexel Institute of Technology. He is a Registered Professional Engineer and has been active in professional and civic organizations. Special career recognitions have included the Secretary of Transportation's Award and the Department of Transportation's Meritorious Achievement Award.

SESSION I THE STATUS OF FREIGHT SYSTEMS, R&D

OVERVIEW OF FRA'S FREIGHT SYSTEMS R&D

BY

A. J. Bang

P. Olekszyk

Executive Summary

This paper describes the mission, goals and objectives of the Office of Freight Systems and briefly summarizes the research, development, test and evaluation (RDT&E) activities of the past year. 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.

In performing its mission the Office of Freight Systems within the Office of Research and Development serves the rail community and the public through the conduct of RDT&E activities designed to stimulate the advancement of rail freight technologies. The Office is organized into two divisions. In the Analysis and Evaluation Division there are currently two subprogram areas of activity; specifically (1) Analysis and Evaluation and (2) Rail Dynamics Laboratory. In the Freight Service Division there are presently four subprogram areas of activity; namely: (1) Classification Yard Technology, (2) Equipment Performance Analysis, (3) Energy/Environment, and (4) Intermodal Systems Technology.

A brief description of each subprogram's goals and objectives, recent accomplishments and current project undertakings are presented. Performing organizations and industry participation are also cited in the paper.

Selected representative examples of research relating to each subprogram area are presented in the following conference papers:

a. Research in Freight Car Dynamics.
b. FAST Mechanical Equipment Test Results to Date--Future Plans.
c. Rail Dynamics Laboratory--Performance Requirements & Hardware Configuration.
d. Optical ACI--A New Look.
e. Truck Performance-Friction Snubber Force Measurement System.
f. Energy Conservation-Multiple Unit Locomotive Throttle Control.
g. Intermodal Cars--New Developments.

Research results expected to become available within the next year or so are also outlined for the reader.

ENERGY CONSERVATION--MUTIPLE UNIT LOCOMOTIVE THROTTLE CONTROL

BY

M. E. Jacobs

Executive Summary

During unit train tests performed on the Burlington Northern and Union Pacific railroads, significant fuel savings were realized by using a semi-automatic throttle control device or "fuel saver" system to take one or more units of the locomotive consist off line when the available power and tractive effort exceeded the demand. This procedure effectively lowered the horsepower per ton ratio of the train, improved power management in various terrains, and decreased the rate of fuel consumption. A prime ingredient for the effective use of such a device was the operating locomotive engineer and the manner in which he handled his train.

Within each of the two independent test series, test comparisons were made on the basis of fuel consumed with and without use of the fuel saver in trains of similar configurations, trailing gross tons, and operating speeds. For the particular set of operating conditions tested, the average fuel savings reached 12.4% for the unit TOFC train test and 9.8% for the unit coal train test at average speeds of 50 and 25 MPH respectively. The savings in the unit coal train test were an average of approximately zero fuel savings recorded for the loaded coal train and the striking 21.5% fuel savings recorded for the unloaded coal train.

In both test series there were significant pattern changes in the histograms of throttle position versus time between the data base and fuel saver tests. As might be expected, operating in fuel save dramatically increased the total time accumulated in the more efficient eighth throttle position with corresponding decreases in the lower throttle positions.

In addition to changes in train handling strategy, several other trends were observed:

* For any particular operating route segment, the fuel savings ranged from zero to much greater than 20%.

* With respect to the motive power, it was found that there is a threshold horsepower per ton assignment at which the fuel saver concept can be effectively employed.

* Long periods of dynamic braking tended to decrease the fuel saver effectiveness while undulating terrain and ascending grades tended to increase the fuel saver effectiveness.

* With respect to speed, a potential exists for greater fuel savings when using the fuel saver system at increased operating speeds.

Of major importance to those concerned with maintaining scheduled train service was the fact that operating in fuel save did not affect travel times or average operating speeds. Although there were no difficulties experienced in either the lead or trailing units of the locomotive consists, any maintenance or mechanical problems which might develop can only be evaluated after extensive usage of the system combined with continual monitoring of the results. In light of the potential savings to be accrued, the economic advantage of the fuel saver concept can hardly be ignored.

INTERMODAL CARS - NEW DEVELOPMENTS

BY

J. R. Blanchfield

M. A. Kenworthy

Executive Summary

Intermodal railcars used for piggyback trailer and container transportation are becoming the subject of increasing design interest in response to a recognized need to improve efficiency and to provide superior levels of service. Although the optimal piggyback railcar design requirements will not be defined until the FRA's Intermodal Systems Engineering Program has been completed, work in the private sector is underway to investigate and test a variety of new railcar ideas and concepts. From these designs it is apparent that two common goals are the reduction of weight and aerodynamic drag. The ride quality and dynamic stability characteristics of lighter weight cars are key issues which are being investigated. The importance of good ride quality is a result of the type of cargo to be carried in the trailers. High value manufactured goods are fragile by nature and cannot be subjected to harmfully high levels of vibration during transportation. Rail car stability, especially the avoidance of truck hunting at speeds above 50 miles per hour must be achieved; not only to protect the cargo from vibration, but also to minimize wheel and rail wear and the chances of derailment. This paper describes the features of several new intermodal railcar concepts. It also reports on a cooperative government industry test program designed to quantitatively define and measure the ride vibration characteristics of current and experimental intermodal flatcars. Under the program, the procedures, equipment and analytical techniques suitable for evaluating car performance under both controlled test and actual service conditions were developed and successfully employed. Preliminary results indicate that substantial weight savings can be achieved without adversely affecting ride quality or dynamic stability.

OPTICAL ACI -- A NEW LOOK

BY

R. L. Wiseman
H. C. Ingrao
W. F. Cracker

Executive Summary

When Automatic Car Identification (ACI) was introduced over a decade ago, it was considered to be a major breakthrough for improving railroad service, operating efficiency, and car utilization; the Nation's railroads, however, have recently been reconsidering its continued use. This paper describes a research program undertaken to provide the railroad industry with quantification of the potential for improvement of its Optical ACI (OACI) system. The program involves the studies of car presence detectors, performance and cost improvements to the OACI scanner system, an analysis of OACI label properties and label life, and a model to evaluate various car identification enhancement alternatives.

The research effort is intended to specify the means for obtaining increased OACI system accuracy at lower costs. The major part of the effort addressed the improvements possible through the use of advanced technology to design, build, and test pre-prototype hardware, and to develop a "firm" specification of the scanner system performance limit.

Recent results of this research clearly indicate that significant improvement in the OACI scanner system performance and costs are achievable. Laboratory tests indicate that the scanner system readability accuracy with the existing labels can be increased at least 6% from its nominal value of 88%-91%. This increase is obtainable through the use of charge-coupled devices and micro-processors which also enable a 40% reduction in the initial scanner system purchase costs and a 33% reduction in yearly maintenance costs. This fully modified scanner system will have a readability of 94-97% with a reduction in initial purchase costs from the range of $40,000-$54,000 down to $27,000. The scanner system yearly maintenance costs would also be reduced by $1700 per year, yeilding a $3400 yearly cost including wheel sensor maintenance. Alternatively, a scanner system performance of 92 to 95% is achievable. Through a $4500 field retrofit cost for a more modest modification; however, maintenance costs would not be improved.

The program also resulted in an assessment of the underlying causes for label deterioration and a label life estimate. A number of non-reversible cause of OACI module degradation over the years is the change in physical properties of the upper layer of Scotchlite due to solar radiation. Also, a reversible cause of OACI standard module degration is the loss of gloss of the module surface due to the natural environment or to abrasion. This loss of gloss can be corrected by maintenance or module redesign. Data from two sources (the label material manufacturer and the Canadian National Railways) leads to a 17 year estimate for the label's operational life. In the area of extending label life, TEFLON overlay on the modules is completely effective in protecting the Scotchlite base material. Also, it has been determined that labels of new design using materials not affected by solar radiation and optically matched to the scanner can be readily developed.

A number of new capabilities for the improved OACI scanner system are also described. A users' guide for an OACI evaluation model is also identified. In sum, the research results from this program will assist the industry in its determination on any future investment in ACI.

TRUCK PERFORMANCE -- FRICTION SNUBBER FORCE MEASUREMENT SYSTEM

BY
G. R. Fay
K. L. Cappel

Executive Summary

In seeking to improve railroad productivity through the improvement of equipment performance, a great deal of emphasis in the Freight Service Division of the Office of Freight Systems R&D has been placed on the characterization and improvement of the performance of the three-piece, friction snubbed, freightcar truck.

In 1974, the Federal Railroad Administration awarded a contract to the Southern Pacific Transportation Company for the performance of Phase I of the Truck Design Optimization Project (TDOP) which had the objective of quantitatively characterizing the performance of this general purpose freightcar truck. The data generated from this project can be used as a base for the evaluation of new or modified designs aimed at improving such performance parameters as ride quality, lateral stability, and curve negotiation.

TDOP Phase I instrumented the American Steel Foundries "Ride Control" truck, with constant snubbing friction, and the Barber S-2 truck, with load dependent snubbing friction, to measure accelerations, normal contact forces at the roller bearing adapters and relative linear and angular displacements between the side frames and bolster. Relative rotation between truck and carbody bolsters was also measured. It quickly became apparent during testing that instrumentation was not available to measure the forces transmitted through the spring loaded friction shoes or wedges between the side frames and bolster. While the forces were calculable, it was clear that this approach would be cumbersome and expensive in computer time. Therefore, in November 1974, a preliminary design concept of a transducer system to directly measure these forces was submitted by Wyle Laboratories.

The resulting assembly contains five force blocks capable of measuring two of the three moments applied by the friction shoes. Space limitations in the side frames precluded a transducer configuration capable of measuring a yaw movement, however, the increased column load due to yaw can be measured. Friction Snubber Force Measurement Systems have been installed in the side frames of both the Ride Control truck and the Barber S-2. To be capable of running across the country under cars in revenue service, the modified, instrumented trucks had to be able to withstand normal shock loads, resulting in stresses below the fatigue limit as specified in AAR M-203-65. Therefore, all side frames were modified, properly reinforced, stress relieved, and stress tested to assure compliance.

Calibration tests were conducted on the Friction Snubber Force Measurement System to demonstrate performance of the transducers in the truck under some simulated operating conditions without reproducing all aspects of the rail environment. Full scale tests are planned as part of TDOP Phase II recently awarded to Wyle Laboratories, Colorado Springs Division. Acquisition of these data will aid in the validation of mathematical simulation of truck performance, will complete the characterization of the general purpose freight car trucks, and will provide a technical baseline for the evaluation of special purpose trucks to be accomplished in Phase II TDOP.

RESEARCH IN FREIGHT CAR DYNAMICS

BY

N. T. Tsai
E. H. Law
N. K. Cooperrider

Executive Summary

The research work reported here is conducted by the Clemson and Arizona State Universities under sponsorship of the Federal Railroad Administration. The objective of the project is to develop analytical tools and techniques to analyze the dynamic behavior of railroad freight cars. The models and analytical techniques developed can be used to determine the causes of undesirable freight car dynamic behavior that contribute to vehicle derailment, damaged lading and the premature failure of track and vehicle components.

The approach used in this program is an integrated analytical experimental evaluation approach, involving theoretical modeling, correlation or validation of models and analysis of the model behavior. Numerical techniques were developed to deal with arbitrary wheel and rail head profiles and other non-linear characteristics of rail freight vehicles. Several computer models of different complexity have been developed for vehicle behavior on tangent track, during curve entry, and in curve negotiation. Together with the models, four different methods of solution, namely, linear method, quasi-linearization method, numerical integration method and hybrid simulation method were used. The wide range of theoretical models thus developed are intended to meet the requirements of given types or problems such as hunting, forced response, and curving behavior.

Validation techniques were developed to evaluate the numerical models by comparing the results with that of a full scale field test. Some of the innovations developed for the rail vehicle test are the application of hydraulic excitation system during the test and the random decrement technique in analyzing the test data. The test was planned and carried out by the Association of American Railroads and the Union Pacific Railroad. The validation process is in progress.

After validation, the models will be utilized to examine current vehicle and track maintenance procedures. The models will be supplied to the railroad industry for use in evaluating possible modifications of current freight car trucks, and in exploring new design concepts.

OVERVIEW OF FRA'S FREIGHT SYSTEMS R & D

BY

A. BANG

P. OLEKSZYK

This paper presents an overview of the freight systems research and development activities in the Office of Research and Development 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 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 reorganization enabled FRA to respond to the rail industries R&D needs. There was at that time an emerging awareness that the Nation's rail system required 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 system.

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.

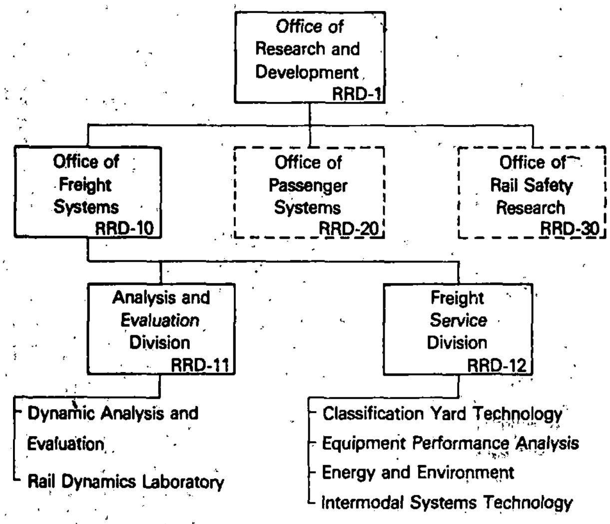


FIGURE 1. OFFICE OF FREIGHT SYSTEMS-ORGANIZATION

The Analysis and Evaluation Division mission reads, "to plan, implement and sponsor the development of specialized research facilities and conduct

---

Arne J. Bang is the Chief, Freight Service Division, Office of Freight Systems (R&D) for the FRA. He received his B.A. (1966) from San Jose State University; M.S. (1968) in Systems Management, University of Southern California.

Phil Olekszyk has served as the Research Engineer/Division Chief, Office of Freight Systems for the FRA since 1972. He received his BSME from Michigan State University (1959) and his MBA from Case Western Reserve University (1968).

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.

The Freight Systems 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, Colo. 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:

- o 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.
- o Manage an evaluation program on the Facility for Accelerated Service Testing (FAST) to produce results aimed at reduction of procurement, operating, and maintenance costs of rolling stock and track.
- o Evaluate available Maintenace-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.

- o 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 three ply and five 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.
- o 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 des-

cribing 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."

o 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 of over 60,000 miles. The recent status of this project will be presented in the accompanying paper entitled, "FAST Mechanical Equipment Test Results to Date Future Plans."

o At the time of this writing, a request for a MOW Equipment Evaluation Plan 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.

FIGURE 2. EXAMPLE OF LADING PROBLEM BEING INVESTIGATED

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

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

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

- o Suspension Characteristics
- o Rock and Roll
- o Component Stress
- o Component and Vehicle Natural Frequencies
- o Analytical Model Validation
- o Adhesion
- o Ride Comfort
- o Acceleration
- o Braking
- o Lading Response
- o 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:

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

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

- o 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 the label readability performance of the optical scanner system to 97% from the current level of 80% and indicates a potential label life of 17 years.
- o 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 264051.
- o A "Research Plan for EMC (Electromagnetic Compatibility) Study of the Communication and Control Systems in a Railroad Classification Yard" was provided to inter-

ested parties in the industry in June 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 and 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. 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.

- After concluding the laboratory verification (see figure 3) of optical ACI scanner system improvements, four final technical reports will be made available to the industry. This research was begun in cooperation with the Research and Test Department, AAR and the Rolling Stock Committee of the Railway Progress Institute.

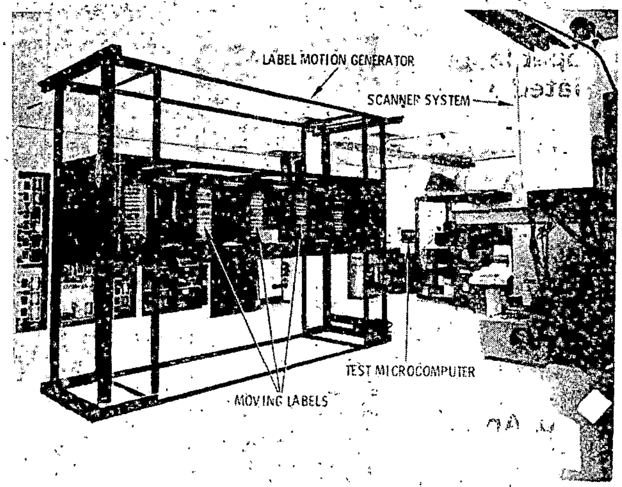


FIGURE 3. TSC LABORATORY SETUP FOR TESTING OACI SCANNER SYSTEM IMPROVEMENTS

- 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, which began 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.

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 <u>Equipment Performance Analysis</u> area the major program objectives are to:

- o Increase railroad profitability through the reduction of lading damage which can be attributed to dynamic phenomena associated with rolling stock suspension systems.

- o Develop those technologies that will help to reduce costs occurring from the wear and maintenance of rail vehicle components.

- o Quantify the information necessary to provide economic-based performance data and specifications upon which sound investment decisions may be reached.

- o 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:

- o 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.

- o 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 in Spring 1978. The coupling work was coordinated with the AAR Advanced Coupler Concepts Program while the braking effort was coordinated with the AAR Brake Equipment Committee.

- o 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.

- o 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 subcontractor 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.

- o In light of the recommendations stemming from the previously mentioned braking and coupling technology assessments, investigatory research will be initiated in the following areas; electropneumatic brakes, friction material versus wheel wear, automatic coupling concepts, load sensing devices, disc brakes, and wheel thermal capacity. These studies will examine both technical and economic aspects. Reports will be made to industry as they become available.

- o 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.

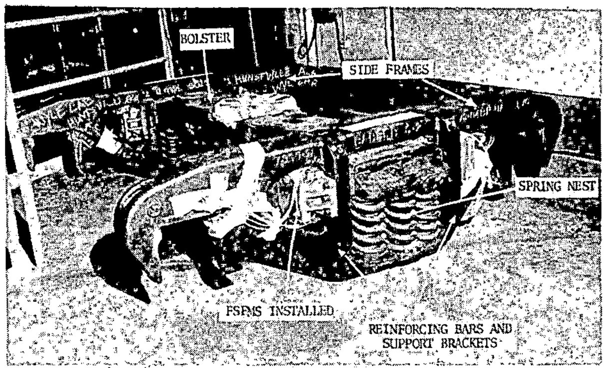


FIGURE 4. STANDARD BARBER S-2 TRUCK EQUIPPED WITH NEW FRICTION SNUBBER FORCE MEASUREMENT SYSTEM (FSFMS).

o 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. The report on the full-scale validation of TOFC/COFC wind tunnel data should become available in spring 1978.

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:

o Exploit and improve the inherent energy and environmental advantages of the rail mode.

o Assist the railroad industry with practical guidelines and test procedures contributing to overall noise abatement and the reduction of noxious emissions.

o Develop and demonstrate energy conservation techniques having economic as well as technical payoffs.

o Quantify areas where locomotive improvements will produce further performance efficiencies.

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

o 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 April 1978. Representative of work accomplished in this program area and related to this subject is the accompanying paper, "Energy Conservation--Multiple Unit Locomotive Throttle Control."

o In conjunction with the Intermodal Systems Technology subprogram, a wind tunnel aerodynamic drag study was completed on TOFC/COFC equipment. 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). The quantification of fuel consumption in various rail freight operations, including TOFC/COFC was also concluded. Another 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 (Accession PB 273 277).

FIGURE 5. FRA RESEARCH ENGINEER LOGGING OPERATIONAL DATA DURING REVENUE OPERATIONS IN THE EVALUATION OF A MULTIPLE UNIT LOCOMOTIVE THROTTLE CONTROL DEVICE.

- o 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 April 1978. Recommendations for further research and suggested alternatives will be incorporated in the report.

New and continuing research underway at this time includes:

- o 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.

- o Another contract, 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.

- o Establishment of a technical assistance agreement with Lawrence Berkley Laboratory, 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.

- o The research 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.

- o 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 <u>Intermodal Systems Technology</u> area the major program objectives are to:

o Support the rail industry's efforts to increase its market share in freight transportation, especially in high revenue traffic.

o Promote realization of superior levels of service to be achieved through an optimized truck/rail/ truck intermodal system.

o 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.

o 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:

o 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.

o With the cooperation of the Atchison, Topeka and Santa Fe Railway, Pullman-Standard, Trailer Train, American Steel Founderies, 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 intermodal 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 rerun 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 August 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.

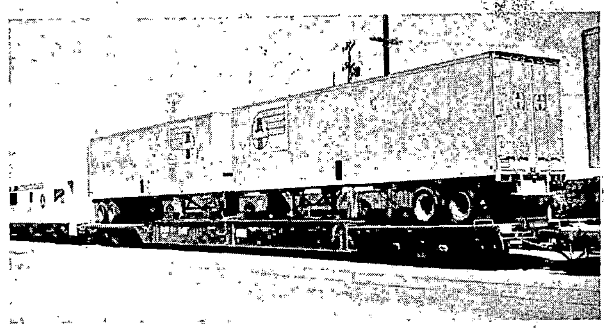


FIGURE 6. INSTRUMENTED TRAILERS AND "SKELTON CAR" ALONG WITH FRA DATA ACQUISITION CAR, T-5 USED IN THE LIGHTWEIGHT FLAT CAR EVALUATION PROJECT

New work underway includes:

o 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, Peat, Marwick, Mitchell & Company and A.T. Kearney Inc. are 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 in-depth 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 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 April 1978. The Phase II effort should be completed in June 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

The Office of Freight Systems manages a diverse RDT&E program to promote the continued viability of the Nation's railroads. Much of the program structure relates to recent publications that attempt to definitize the research needs of the industry. The availability of resources limits what 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. 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.

We would also like to acknowledge those people that have participated in the unselfish donation of time, material, and expense in support of various projects including the management and personnel at the Transportation Systems Center and the Transportation Test Center.

ENERGY CONSERVATION--MULTIPLE UNIT LOCOMOTIVE THROTTLE CONTROL

BY

M. E. JACOBS

During unit train tests performed on the Burlington Northern and Union Pacific railroads, significant fuel savings were realized by using a semi-automatic throttle control device or "fuel saver" system to take one or more units of the locomotive consist off line when the available power and tractive effort exceeded the demand. This procedure effectively lowered the horsepower per ton ratio of the train, improved power management in various terrains, and decreased the rate of fuel consumption. A prime ingredient for the effective use of such a device was the operating locomotive engineer and the manner in which he handled his train.

INTRODUCTION

Reducing fuel consumption in rail freight transportation has become increasingly cost effective. As the price of diesel fuel continues to spiral upward, substantial investments in improved locomotive maintenance practices, operating efficiencies, and control devices to decrease fuel consumption have become a necessity.

Recognizing this need for increased fuel conservation, the Federal Railroad Administration sponsored a research study by J. N. Cetinich entitled Fuel Efficiency Improvement in Rail Freight Transportation*. This report presented an excellent discussion of how to design train operating policies specifically to conserve fuel while continuing to provide desired schedule and service performance. In addition to the presentation of an overall operating policy for the rail industry, the author discussed nine items characterizing the ideal diesel road locomotive from the standpoint of fuel efficiency. Accordingly, the ideal diesel locomotive would:

1. Be easily maintained
2. Have 3000 horsepower
4. Be four axle
5. Be turbocharged without a parts catcher
6. Use low pressure drop engine air filters
7. Have controllable cooling fans and air compressor disengagable when not needed
8. Have clean cut-off fuel injectors
9. Have a built-in control logic to automatically take individual units in a locomotive consist on and off line.

With respect to the last item, the objective of such a control device would be to keep a working turbocharged consist at its most efficient seventh or eighth throttle position as much of the time as is operationally feasible by reducing to number one throttle those units in excess of the normal operational requirements. This procedure effectively lowers the horsepower per ton ratio of the train and decreases the rate of fuel consumption. In a practical field application, the number one throttle position is selected in preference to the idle position in order to maintain the dynamic brake capability of the units selected for throttle reduction. Because of the

*Available from the National Technical Information Service; Springfield, VA. 22161; NO. PB 250673.

Marilynne E. Jacobs has served as Research Engineer in Mechanical Systems, Office of Freight Systems (R&D) for the FRA since 1976. She received her BSME (1968), University of Delaware; and her MSME (1972) - Mechanical Engineering, Fairleigh Dickinson University.

principles involved in using the control device, a decrease in fuel consumption can be expected for those trains operating on level grades, on lesser uphill grades, and on lengthy downhill runs where the number of operational units in the locomotive consist is most likely to exceed the power requirements.

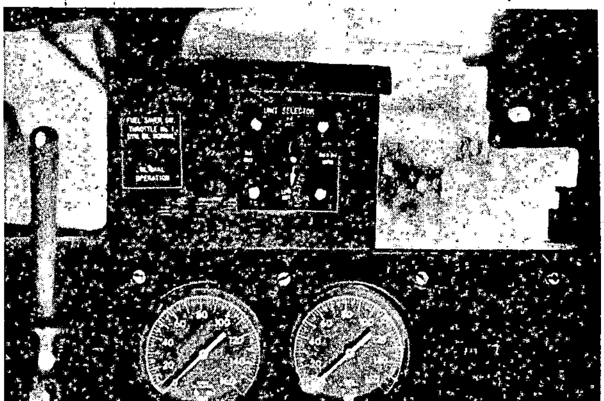


FIGURE 1. FUEL SAVER CONTROL BOX ON LOCOMOTIVE CONTROL STAND

The objective of this paper is to quantify the actual fuel savings resulting from the usage of one such device in an operating locomotive consist. Commonly referred to as the "fuel saver" system, the device itself is amazingly simple. It consists of a control box mounted on the control stand in the lead unit of the consist (figure 1) and a "fuel saver set up switch" located on the isolation panel of each unit in the consist (figure 2).

The electrical wiring is accomplished through two available pins in the jump cable between the individual locomotives. It should be emphasized that the locomotive remote control capability of the system functions through the interconnecting jump cables of the consist and is not radio frequency controlled.

TEST DESCRIPTION

The two test series presented in this paper, Table 1 and Table 2, involved two distinctly different train configurations operating in two distinctly different rail environments. In the

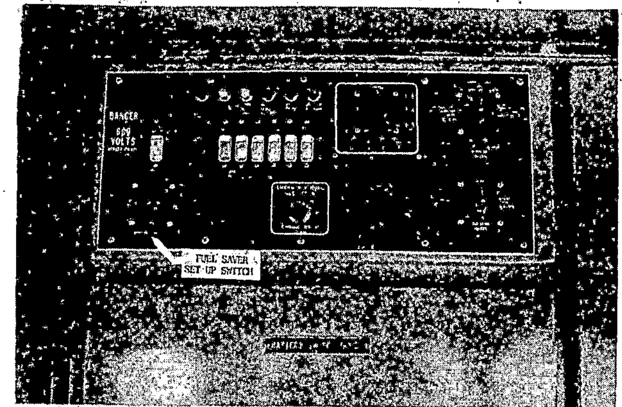


FIGURE 2. "FUEL SAVER SET UP SWITCH" ON LOCOMOTIVE ISOLATION PANEL

first test series, conducted on the Burlington Northern, four turbocharged SD 40-2 locomotives pulled a slow, heavy 14,000-ton 110 car unit coal train across the predominantly level 682 miles between Lincoln, Nebr., and Metropolis, Ill., at an average speed of 25 MPH. This was in marked contrast to the second test, a high priority "Super Van" TOFC train of the Union Pacific. Powered by two EMD DD 40's and one SD 40-2 for a total of 16,200 hp, the 2,500-ton 30 car Super Van reached an average speed of 50 MPH in spite of the extremely variable and somewhat mountainous 1,519 mile terrain between North Platte, Nebr., and Los Angeles, Calif. The advantages of testing dedicated unit trains operating between points A and B were the predictable operating speeds and the relatively constant trailing gross tons and number of cars per train. This type of test effectively eliminated the three most predominant variables encountered in testing manifest freight locomotive consists: speed, trailing gross tons, and number of cars.

For both the BN and UP test series, one round trip was conducted with the fuel saver system "off" as a control or data base (D.B.) test and one round trip was conducted with the fuel saver system "on" designated in the data analysis as the fuel saver (F.S.) test. The accumulated mileage per round trip totaled an impressive 1,364 miles for

OPERATING CONDITIONS*	LOADED COAL TRAIN TEST ZONES 1 TO 6			UNLOADED COAL TRAIN TEST ZONES 6 TO 1		
	D.B. TEST	F.S. TEST	% DIFF.	D.B. TEST	F.S. TEST	% DIFF.
1. TOTAL HP (SD 40-2, FOUR)**	12000	12000		12000	12000	
2. NUMBER OF COAL CARS	110	110		110	110	
3. SCALE WEIGHT OF CARS, TONS	14,368	13,958	- 2.8	3370	3375	+ 0.1
4. AVER. CABOOSE WT., TONS	27	27		27	27	
5. TRAILING GROSS TONS, TGT	14,395	13,985	- 2.8	3397	3402	+ 0.1
6. HP/TGT	.83	.86	+ 3.6	3.53	3.53	
7. MILES TRAVELED	682.1	682.1		682.1	682.1	
8. % TOT. MILES/GRADE RANGE						
a. LEVEL 0 ± 0.49%	79.4			79.5		
b. ASCENT 0.50 - 0.70%	9.5			8.2		
0.71 - 1.50%	0.9			1.7		
1.51 - 2.50%	0.0			0.0		
c. DESCENT 0.50 - 0.70%	8.3			9.2		
0.71 1.50%	1.9			1.4		
1.51 2.50%	0.0			0.0		
9. TOTAL TIME IN MOTION, HR.	27.90	29.15	+ 4.5	27.54	23.90	- 13.2
10. AVER. SPEED, MPH	24.4	23.4	- 4.1	24.8	28.5	+ 14.9
11. NUMBER OF CREWS	6	6		6	6	0
12. NUMBER OF STOPS	31	27	-12.9	32	31	- 3.1
13. AVER. % TIME IN FUEL SAVE		***			***	
14. AVER. % TIME/THROTTLE NO.						
a. IDLE (IN MOTION)	8.0	9.0	+12.5	6.1	7.2	+ 18.0
b. THROTTLE #1	7.7	10.7	+39.0	12.5	8.1	- 35.2
c. THROTTLE #2	13.9	7.0	-49.6	20.0	6.2	- 69.0
d. THROTTLE #3	12.4	6.3	-49.2	11.7	5.2	- 55.5
e. THROTTLE #4	10.7	4.6	-57.0	13.1	7.4	- 43.5
f. THROTTLE #5	10.1	6.9	-31.7	9.5	5.8	- 38.9
g. THROTTLE #6	7.5	5.7	-24.0	5.5	5.4	- 1.8
h. THROTTLE #7	4.3	4.7	+ 9.3	5.0	7.7	+ 54.0
i. THROTTLE #8	25.4	45.1	+77.6	16.6	47.0	+183.0
15. FUEL CONSUMPTION, GALS.	7566	7700	+ 1.8	6278	4927	- 21.5
16. TGTM/GAL.	1297.8	1238.8	- 4.5	369.1	471.0	+ 27.6
17. GAL./1000 TGTM	0.77	0.81	+ 5.2	2.71	2.12	- 21.8

* TEST DATES: FEB. 14-22, 1977
** SAME LOCOMOTIVE CONSIST FOR ALL TESTS
*** INCOMPLETE DATA

TABLE 1. BN UNIT COAL TRAIN SUMMARY OF FUEL SAVER TEST RESULTS

the BN unit coal train and slightly double that or 3,038 miles for the UP unit TOFC train. The BN unit coal train test series actually comprised two tests. For the outbound leg, the coal train was loaded. After dumping the coal at the end point of Metropolis, the train returned on the inbound leg empty. All testing proceeded within the normal operational framework of each railroad.

Scale weighing both the coal and the cars insured less than a 3% variation in trailing loads per test for the loaded unit coal train. However, such information was not readily available for the unit TOFC trains. Instead, the gross tons per car were determined by adding the tare weight to the estimated trailer plus lading weight supplied by the shipper. As noted in Table 2, the UP west or outbound TOFC fuel saver test for zones #1 through #8 included an average value for both the number

OPERATING CONDITIONS	WESTBOUND TEST ZONES 1 TO 8			WESTBOUND TEST ZONES 1,2,&4			EASTBOUND TEST ZONES 1,2,&4		
	D.B. TEST	F.S. TEST	% DIFF.	D.B. TEST	F.S. TEST	% DIFF.	D.B. TEST	F.S. TEST	% DIFF.
1. TOTAL HP	16,200	16,200		16,200	16,200		16,200	16,200	
2. TRAIN MAKE UP									
a. LOADED CARS	34	31*		34	32		46	31(21BOX)	
b. EMPTY CARS	1	2		1	2		1	2	
c. TOFC CARS	ALL BUT 3	ALL		ALL BUT 3	ALL		ALL BUT 6	10	
3. AVER. CABOOSE WT., TONS	27	27		27	27		27	27	
4. TRAILING GROSS TONS, TGT	2501	2500*	0.0	2501	2627	+ 5.0	3233	3155	- 2.4
5. HP/TGT	6.48	6.48	0.0	6.48	6.17	- 4.8	5.01	5.13	+ 2.4
6. MILES TRAVELED	1519	1519		605	605		605	605	
7. TOTAL TIME IN MOTION, HR.	31.38	29.70	- 5.4	11.70	11.21	- 4.2	11.03	11.36	+ 3.0
8. AVER. SPEED, MPH	48.8	51.5	+ 5.6	51.9	54.5	+ 4.9	55.3	53.4	- 3.5
9. NUMBER OF CREWS	8	8		3	3		3	3	
10. NUMBER OF STOPS	8	9		3	4		3	4	
11. AVER. % TIME IN FUEL SAVE									
a. LEAD POWER UNIT	0	**		0	**		0	0.0	
b. 3RD POWER UNIT	0	**		0	**		0	55.6	
c. 4th,5th POWER UNITS	0	47.4		0	59.7		0	60.7	
12 AVER. % TIME/THROTTLE NO.									
a. C-DYN. BRAKE, IDLE	30.3	42.9	+41.6	23.5	23.0	- 2.1	25.6	26.4	+ 3.1
b. THROTTLE # 2	4.1	3.5	-14.6	5.0	3.3	-34.0	3.3	3.7	+12.1
c. THROTTLE # 3	6.4	3.4	-46.9	5.2	6.5	+25.0	6.3	3.0	-52.4
d. THROTTLE # 4	8.5	3.5	-58.8	10.6	6.3	-40.6	9.2	3.0	-67.4
e. THROTTLE # 5	8.6	2.8	-67.4	5.8	3.4	-41.4	7.3	7.7	+ 5.5
f. THROTTLE # 6	9.6	4.1	-57.3	7.1	5.9	-16.9	13.5	6.1	-54.8
g. THROTTLE # 7	9.6	7.5	-21.9	11.2	9.2	-17.8	9.7	5.5	-43.3
h. THROTTLE # 8	22.9	32.3	+41.0	31.6	42.4	+34.2	25.1	44.6	+77.7
13 FUEL CONSUMPTION, GAL.***	12,145	10,641	-12.4	5274	4491	-14.9	4617	4240	- 8.2
14 TGTM/GAL	312.8	356.9	+14.1	286.9	353.9	+23.3	423.6	450.2	+ 6.3
15 GAL/1000 TGTM	3.20	2.80	-12.5	3.49	2.83	-18.9	2.36	2.22	- 5.9

* AVERAGE: CAR EXCHANGE AT MID TRIP POINT.
** USED BUT NOT RECORDED
*** WESTBOUND INCLUDES SOME DERIVED DATA

NOTES

1. TEST DATES: MARCH 29, 1977 TO APRIL 5, 1977
2. SAME LOCOMOTIVE CONSIST AND ORDER FOR ALL TESTS

LEAD, #2	#3	#4, #5
DD40	SD 40-2	DD 40
6600hp	3000hp	6600hp

TABLE 2. UP UNIT TOFC TRAIN SUMMARY OF FUEL SAVER TEST RESULTS

of cars and the total trailing gross tons due to a bad order car exchange at the mid trip point. The number of cars varied by one and the tons varied from 2,372 to 2,627 for an average of 2,500 tons.

The test parameters recorded via trip logs and analog chart recorders included the following:

* Times
* Milesposts
* Fuel consumption per locomotive
* Refueling readings at trackside fueling racks
* Throttle position vs. time
* Speed vs. time
* Average % time in fuel save per locomotive
* Number of crew changes and stops
* Fuel oil temperature (pump-up and return), °F
* Lube oil temperature at the oil pump, °F
* Traction motor exhaust air temperature, °F (BN only)
* Alternator current vs. time (BN only)

* Spectographic lube oil analysis (BN only)
* Lube oil additions (BN only).

To aid in the subsequent analysis, the above data was supplemented with track profiles, track diagrams, and mileage tables. All speeds, temperatures, throttle positions, and alternator currents were recorded for the lead locomotive only. In addition, a set numeric order for manually recording all other pertinent locomotive data was established and adhered to throughout the tests.

The locomotives assigned to both test consists had all been screened for potential problems in regularly scheduled 15 or 30 day inspections just prior to testing. Hence their performance characteristics and fuel efficiencies were considered to be typical of the average locomotive operating under similar conditions. The designated lead and trailing power units never varied from one test to the next and were set up to operate in the fuel save mode either individually or in a preset combination. Although all SD 40-2 locomotives could be operated independently in fuel save, both power units of the 6,600 hp DD 40 were wired to simultaneously reduce power when in fuel save. In this case, the DD 40 represents a special class of locomotive. The decision to monitor the two power plants as a single unit was based on the prevailing route profile grades and the relatively high track speeds.

To record diesel fuel consumption to the nearest gallon, two calibrated volumetric flow meters were installed in each of the four locomotives of the BN unit coal train and in each of the five power plants of the three locomotive UP unit TOFC train. The difference in meter readings between the supply line and the return line to the fuel tank indicated the fuel consumed per locomotive. The meter readings were recorded manually at the end of each test zone as well as for any delay encountered. Because of the number of crew changes per test and the importance of the locomotive engineer in evaluating the performance of the fuel saver system, a test zone was defined as that distance traveled before a crew change occurred.

In addition to the on-board meters, trackside tank refueling readings were also recorded, where possible, to determine a comparability factor between the on-board meter readings and the quantity of fuel supplies to each fuel tank. For the purpose of these tests, the BN installed in the pump line of their refueling racks an accurate high volume flow meter calibrated by the Nebraska Bureau of Weights and Measures. The most recent calibration dates of the UP trackside refueling meters were not known, but the meters themselves were less than a year and a half old.

As an indicator of the variation in train handling techniques with and without use of the fuel saver system, throttle positions versus time were recorded continuously on the BN tests and at discrete time intervals on the UP tests using a millivolt versus time recorder wired to the various solenoid valve combinations.

In the course of each test it was found that the on-board locomotive speed recorders were considerably inaccurate for speeds less than 20 MPH and greater than 35 MPH. Standard procedure per crew change involved calibrating the speed recorder with wristwatch and milepost to correlate indicated recorder speed with the actual track speed. Therefore, it was not possible to continuously monitor speed versus distance as a means of comparability between any data base and fuel saver test series. Instead, average trip times in motion per crew change were calculated by matching the start and end times per crew change with the analog brush charts recording locomotive throttle positions versus time. The average operating speeds per test zone were then calculated by dividing the known distance traveled by the total test time the train was moving.

To accurately record the desired temperatures, all of the iron-constantan thermocouple leads were checked for breaks and precalibrated prior to testing. The sensing elements inserted into the various fluids through drain plugs or special fixtures varied from a multi-twisted wire to a dip stick configuration to a completely compensated insulator sleeve emersion thermocouple. Temperature recording methods included direct readings from a pyrometer at the turn of a switch (BN) and continuous readings at discrete time intervals using a millivolt versus time chart recorder with temperature versus time paper (UP).

Located on the back of each fuel saver system control box were hour counters to accumulate actual time in use to the nearest tenth of an hour. The data was available but unfortunately was not recorded in all of the fuel saver tests on a per test zone per locomotive basis. On the West or outbound leg of the UP fuel saver test, the third fuel saver system was inoperative. While repairs were being made, both the first power unit of the lead DD 40 locomotive and the third SD 40 unit were manually isolated to simulate fuel saver test conditions. Therefore, time in fuel save per locomotive was not available for this type of situation.

Effective use and operation of the throttle control device was highly dependent on the skill of the locomotive engineer. Skill in this instance was indicated by the engineer's *ability to* match the use of the fuel saver to the track profile and the power requirements. For each fuel saver test on the BN and UP, the locomotive operating engineer was instructed by on-board test personnel to keep the locomotive consist at the seventh and eighth throttle positions as much of the time as possible. The fuel saver switches were employed to reduce power where necessary without sacrificing track speeds or operating schedule times. Because of the numerous crew changes on both the BN unit coal train and the UP unit TOFC train, the time and number of locomotives in fuel save varied considerably.

Looking at the East or inbound UP fuel saver test summary of results presented in Table 2, note that only three of eight test zones of data have been presented. Test zones #5 to #8 were eliminated because the assigned fuel saver test train was mixed freight and did not match the data base TOFC train in configuration or number of cars. Midway through the test route, the train was changed but again it was mixed freight with only one third TOFC.

DATA REVIEW AND ANALYSIS

Within each test series conducted on the BN and UP, test comparisons were made on the basis of the fuel consumed with and without use of the fuel saver in trains of similar configurations, trailing gross tons, and operating speeds. The two methods employed to compare fuel efficiencies included the evaluation of the percent decrease in fuel consumed and the calculation of the ratio of one thousand trailing gross ton miles per gallon of fuel (1,000 TGTM/GAL). An increase in the ratio of 1,000 TGTM/GAL denoted an increase in the fuel efficiency.

BURLINGTON NORTHERN UNIT COAL TRAIN

With 79% of the route miles at less than 0.5% grade, the overall average percent decrease in fuel consumed round-trip was 9.8%. This figure represented an average of approximately zero fuel savings recorded for the loaded coal train and the striking 21.5% fuel savings recorded for the unloaded coal train (Table 3). Though the average fuel saver test speed for the unloaded case was 15% greater than the data base run, it must be remembered that the percent difference technique is deceiving for low numbers and that the actual difference was only 3.7 MPH from one test to the next.

Significant pattern changes were exhibited in the average percent time spent per throttle position between the data base and fuel saver tests. As shown in figure 3, operating in fuel save dramatically reduced the accumulated hours in throttle positions #2 and #5 by 30-50% and 40-70% respectively for the loaded and unloaded coal trains. However, the time spent in throttle position #8 almost doubled and tripled with increases of 78% and 183%

TEST ZONE	TYPE OF TEST	MILES	SPEED, MPH ZONE AVER	SPEED, MPH % DIFF	% TIME IN FUEL SAVE PER LOCOMOTIVE 2ND	3RD	4TH	FUEL CONSUMED, GALLONS PER LOCOMOTIVE LEAD	2ND	3RD	4TH	AVER.	CONSIST TOTAL	PERCENT DIFF.
(LOADED) 1	DB	141.6	27.8	-11.9				234	383	542	532	422.8	1691	+10.9
	FS		24.5					558	327	370	620	468.8	1875	
2	DB	113.1	32.1	-10.0				219	195	281	287	245.5	982	+ 5.0
	FS		28.9					388	176	188	279	257.8	1031	
3	DB	117.4	26.2	- 3.4				357	295	423	410	371.2	1485	-11.4
	FS		25.3					405	254	330	326	328.7	1315	
4	DB	69.4	23.2	- 0.9				117	113	147	158	133.7	535	+24.1
	FS		23.0		TEST ZONES 1 to 5:			228	137	137	162	166.0	664	
5	DB	134.4	19.4	+ 5.7	24.7	39.0	28.5	336	330	463	482	402.7	1611	- 2.6
	FS		20.5					531	386	314	338	392.2	1569	
6	DB	106.2	21.7	- 6.9				292	250	363	357	315.5	1262	- 1.3
	FS		20.2		TEST ZONES 6 to 6:			418	269	269	290	311.5	1246	
(UNLOADED) 6	DB	106.2	21.8	-12.8	39.6	56.1	56.1	174	164	239	249	206.5	826	- 8.8
	FS		19.0					381	171	80	121	188.2	753	
5	DB	134.4	19.3	+ 8.3	*	*	*	145	162	250	271	207.0	828	- 9.3
	FS		20.9					465	132	55	99	187.7	751	
4	DB	69.4	18.5	+54.6	*	*	*	156	134	198	141	157.2	629	-46.1
	FS		28.6					109	75	72	83	84.7	339	
3	DB	117.4	43.5**	-11.0	*	*	*	305	252	362	424	335.7	1343	-15.6
	FS		38.7					482	175	231	246	283.5	1134	
2	DB	113.1	33.4	+26.9	*	*	*	314	253	367	367	325.2	1301	-15.4
	FS		42.4					372	210	258	261	275.2	1101	
1	DB	141.6	33.2**	+13.6	*	*	*	317	263	384	387	337.7	1351	-37.1
	FS		37.7					386	98	173	192	212.2	849	
AVERAGE VALUES														
LOADED 1 to 6	DB	682.1	24.5	- 4.5	*	*	*	1555	1566	2219	2226	1891.5	7566	+ 1.8
	FS		23.4					2528	1549	1608	2015	1925.0	7700	
UNLOADED 6 to 1	DB	682.1	24.8	+14.9	*	*	*	1411	1228	1800	1839	1569.5	6278	-21.5
	FS		28.5					2195	861	869	1002	1231.7	4927	

* USED BUT NOT RECORDED **BASED ON AVAILABLE DATA

TABLE 3. BN UNIT COAL TRAIN: FUEL CONSUMPTION PER TEST ZONE

respectively for the two test cases. Due to time gaps in the paper tape recordings, only three of six test zones of representative throttle data have been presented in figure 3.

Looking at the loaded coal train test results (Table 3), the differences in fuel consumption ranged from an increase of 24.1% or 129 gallons for test zone #4 to a decrease of 11.4% or 170 gallons for test zone #3. The average speeds of 25 and 23 MPH for these two test zones were similar and the number of stops were identical. However, the number of route miles per grade range were distinctly different.

In the loaded coal train test for test zone #4, 90% of the route miles were essentially level at $\pm$ 0.5% grade as opposed to 73% for test zone #3. From figure 4, the relative time spent in the lower throttle positions for test zone #4

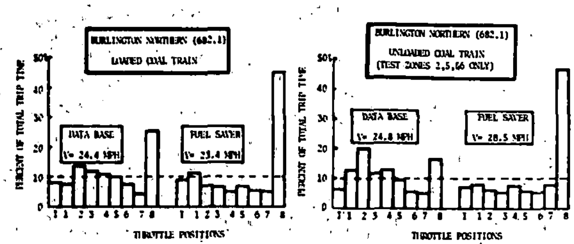


FIGURE 3. HISTOGRAM OF TIME VERSUS THROTTLE POSITION

indicated that most of the 90% "level" route miles were actually descending with increased periods in dynamic brake. Fuel saver usage in this situation was not as effective as for test zone #3, where the terrain was characteristically more undulating. Examining the histogram for test zone #3 more closely (figure 4), it was found that the recorded fuel savings for this zone were obtained by reducing the time spent in throttle notch #5 followed by smaller reductions for positions #1 and #2. The net result was more efficient power usage in the eighth throttle position and 11.9% fuel savings in spite of the heavy 14,000-ton trailing load.

For the unloaded coal train, there were fuel savings on every test zone. As shown in Table 3, the decreases in consist fuel consumption ranged from 8% to 15% on four out of six test zones. However, for test zones #4 and #1, the fuel savings exceeded 30%. For these two test zones as well as for test zone #2, the fuel saver test operating speeds were significantly higher than the data base tests, yet the fuel consumed was definitely less for the fuel saver tests. Combining this fact with higher ratios of ton miles per gallon and extensive shifts in throttle usage patterns, figure 5, indicated a possible trend toward greater fuel savings when using the fuel saver system at increased operating speeds for the 3.5 horsepower per ton power assignment.

Installing calibrated meters in the pump lines of the trackside fueling racks enabled a direct comparison between the quantity of fuel added to the locomotive fuel tanks and the actual fuel used as recorded by the on-

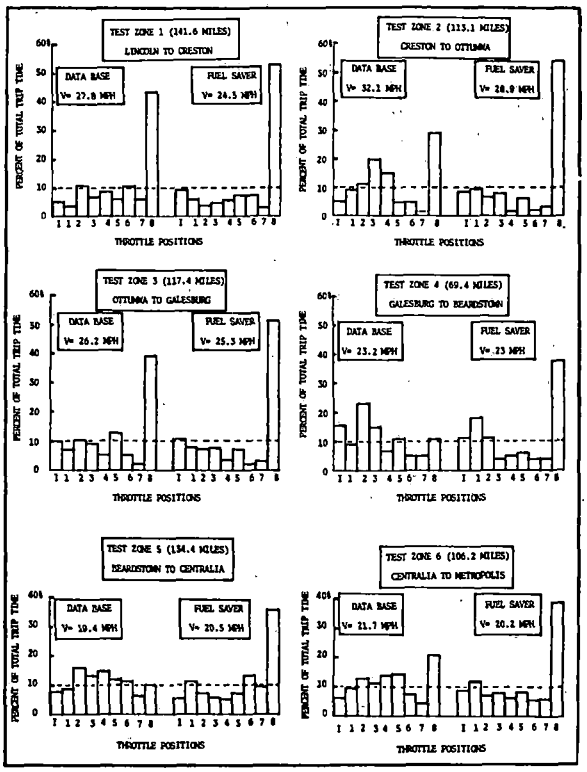


FIGURE 4. BN LOADED COAL TRAIN HISTOGRAM OF TIME VERSUS THROTTLE POSITION PER TEST ZONE

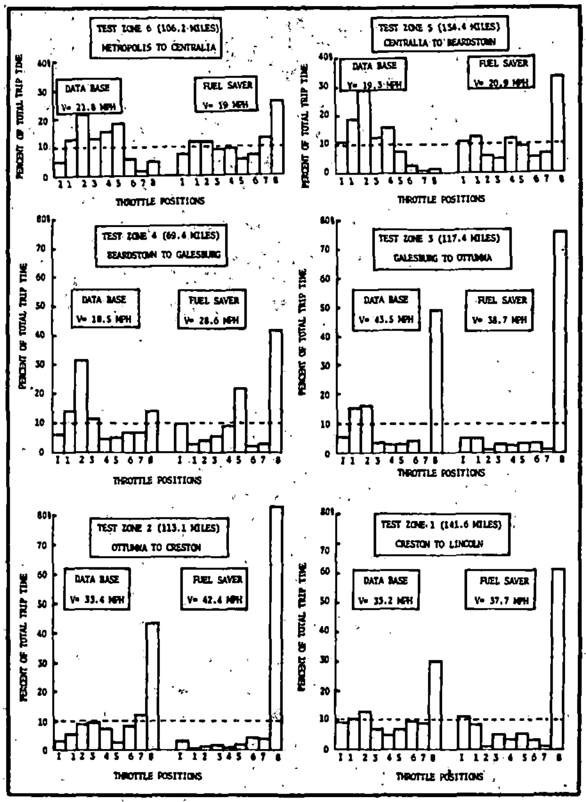


FIGURE 5. BN UNLOADED COAL TRAIN HISTOGRAM OF TIME VERSUS THROTTLE POSITION PER TEST ZONE

board fuel flow meters. The percent variation between the two fuel recording methods was extremely small; less than one percent. The advantage of the on-board meters was that fuel consumed in-motion could be differentiated from fuel consumed when the train was stopped. Therefore, a variation in the number and length of stops between tests could be effectively eliminated as a test variable by considering only the fuel consumed when the train was in motion. The fact that the stops occurred would of course be recorded and evaluated in the overall locomotive operational performance.

Using a sampling technique for recording fuel and lube oil temperatures (Tables 4 and 5), rather than continuous monitoring, indicated two trends. First, for the ambient conditions tested, the average temperatures changed very little per round trip after 55 and 53 hours in motion with the same number of crew changes and a similar number of stops. Second, the average temperatures were slightly elevated for the fuel saver tests as could be expected with increased time in the higher throttle positions.

More specifically, for the data base round trip test (loaded plus unloaded train route), the average temperatures ranged from 90°F to 100°F respectively for the pump-up and return fuel oil lines with the lube oil at 162°F. For the fuel saver round trip test, the average pump-up and return fuel oil temperatures ranged from 95°F to 108°F respectively, while the lube oil temperature increased to the 172-178°F range.

UNION PACIFIC UNIT TOFC TRAIN

Examining the aggregate test results in Table 6 for the eight test zones in the Westbound direction indicated an

TEST* SAMPLE NUMBER	DATA BASE TEST (TRIP TIME=27.90 HR.)				FUEL SAVER TEST (TRIP TIME = 29.15 HR.)			
	FUEL OIL TEMPERATURE, °F			LUBE OIL	FUEL OIL TEMPERATURE, °F			LUBE OIL
	PUMP UP	RETURN	DIFF. °F	TEMP., °F	PUMP UP	RETURN	DIFF. °F	TEMP., °F
1	85	90	5	140	90	105	15	180
2	85	98	13	140	90	110	20	180
3	90	110	20	157	95	105	10	180
4	95	120	25	175	95	105	10	180
5	85	100	15	170	95	115	20	180
6	85	95	10	165	100	105	5	180
7	90	100	10	175	100	120	20	180
8	80	90	10	165	95	105	10	180
9	95	105	10	170	100	105	5	180
10					90	105	15	170
11					95	110	15	170
12					100	115	15	180
AVER.	88	101	13	162	95	109	14	178

TABLE 4. BN LOADED COAL TRAIN: FUEL AND LUBE OIL TEMPERATURES

TEST* SAMPLE NUMBER	DATA BASE TEST (TRIP TIME=27.54 HR.)				FUEL SAVER TEST (TRIP TIME=23.90 HR.)			
	FUEL OIL TEMPERATURE, °F			LUBE OIL	FUEL OIL, TEMPERATURE, °F			LUBE OIL
	PUMP UP	RETURN	DIFF., °F	TEMP., °F	PUMP UP	RETURN	DIFF., °F	TEMP., °F
1	90	95	5	150	100	105	5	150
2	85	90	5	150	95	110	15	175
3	90	95	5	155	90	100	10	150
4	90	100	10	160	95	105	10	185
5	100	110	10	175	100	115	15	185
6	100	110	10	170	95	105	10	175
7	95	100	5	175	100	115	15	185
AVER.	93	100	7	162	96	108	12	172

TABLE 5. BN UNLOADED COAL TRAIN: FUEL AND LUBE OIL TEMPERATURES

TEST ZONE	TYPE OF TEST	MILES	SPEED, MPH		% TIME IN FUEL SAVE PER LOCOMOTIVE			FUEL CONSUMED, GALLONS							
								PER LOCOMOTIVE						CONSIST TOTAL	FUEL SAVED %
			ZONE AVER	% DIFF	LEAD	3RD	4TH,5TH	LEAD	2ND	3RD	4TH	5TH	AVER.		
(WEST) 1	DB	221	49.6	+ 7.0				552	497	515*	498	534*	519.2	2596	-23.5
	FS		53.0		0.0	**	45.6	523	460	556	200	248*	397.4	1987	
2	DB	173	53.6	+10.2				335	290	305*	292	315*	307.4	1537	- 9.6
	FS		59.0		0.0	**	58.0	358	311	266	203	252*	278.0	1390	
3	DB	135	40.5	+20.7				225	179	193*	178	199*	194.8	974	-20.3
	FS		48.9		**	**	54.4	131	256	228	72	89*	155.2	776	
4	DB	211	52.6	- 2.4				267	208	226*	207	233*	228.2	1141	- 2.4
	FS		51.3		**	**	75.4	316	286	348	73	91*	222.8	1114	
5	DB	207	60.5	- 3.4				331	283	290*	261	300*	293.0	1465	+ .6
	FS		58.5		**	**	45.2	299	428	290*	196	243*	291.2	1456	
6	DB	243	45.8	+ 3.1				368	287	321*	312	332*	324.0	1620	-12.3
	FS		47.2		**	**	19.4	279	333	320*	218	271*	284.2	1421	
7	DB	170	49.3	+ 4.6				357	294	313*	291	323*	315.6	1578	-14.2
	FS		51.5		**	15.1	30.3	306	331	281	193	243	270.8	1354	
8	DB	159	38.1	+11.5				282	227	244*	228	253*	246.8	1234	- 7.4
	FS		42.5		**	48.1	50.8	303	266	224	140	210	228.6	1143	
(EAST) 1	DB	221	51.9	+ 2.9				277	213	235	209	260	238.8	1194	-14.0
	FS		53.4		0.0	58.0	67.6	327	279	209	65	147	205.4	1027	
2	DB	173	59.0	- 7.0				299	266	270	256	301	278.4	1392	-13.8
	FS		54.9		0.0	57.1	60.3	315	230	234	184	237	240.0	1200	
4	DB	211	55.0	- 5.7				504	367	395	350	415	406.2	2031	.9
	FS		51.8		0.0	51.6	54.0	565	503	381	231	333	402.6	2013	
AVERAGE VALUES															
(WEST) 1 TO 8	DB	1519	48.8	+ 5.6				2717	2265	2407	2267	2489	2429.0	12,145	-12.4
	FS		51.5		**	**	47.4	2515	2671	2513	1295	1647	2128.2	10,641	
(WEST) 1,2,4	DB	605	51.9	+ 4.9				1154	995	1046	997	1082	1054.8	5274	-14.9
	FS		54.5		**	**	59.7	1197	1057	1170	476	591	898.2	4491	
(EAST) 1,2,4	DB	605	55.3	- 3.5				1080	846	900	815	976	923.4	4617	- 8.2
	FS		53.4		0.0	55.6	60.7	1207	1012	824	480	717	848.0	4240	

*DERIVED DATA **USED BUT NOT RECORDED

TABLE 6. UP UNIT TOFC TRAIN: FUEL CONSUMPTION PER LOCOMOTIVE PER TEST ZONE

overall average decrease in fuel consumption on the order of 12% at an average speed of 50 mph. Individual test zone savings for this direction ranged from zero to a high of 23%. Due to a problem in matching the Eastbound fuel saver test train with the TOFC data base train, only selected test zones in this direction have been presented for analysis. Though the Eastbound comparison fuel saver test train was only one third TOFC, the fuel savings still averaged 8%. The trailing gross tons and speeds were similar for all test zones with the only marked variation in number of cars occurring in the Eastbound direction.

Test zone #3 was eliminated from the Eastbound data comparisons because the average test zone speed for the fuel saver test exceeded the data base test by 30%. In this test, there was a reluctance on the part of the locomotive engineer to use the fuel saver system under the misapprehension that track speeds were going to be sacrificed as part of the test criteria. As might be expected, operating at eighth throttle more of the time without reducing power in the trailing units increased the fuel consumption and the average operating speed for that test zone.

As on the Burlington Northern, fuel usage on-board the locomotives was compared to the actual fuel added to the fuel tanks from the trackside refueling racks. Unfortunately, the method of refueling and the refueling personnel were not controllable elements of this test series. Consequently, the comparability between the two fuel recording methods varied from 2% to greater than 15%.

In the Eastbound direction, all on-board fuel meters were operational for both the data base and fuel saver tests. Such was not the case in the Westbound direction. Due to clogging in the "pump-up" fuel meters of two of the five power plants, it was necessary to derive some of the fuel data presented in Table 6. By determining the percent deviation from the average locomotive fuel consumption in the Eastbound direction, it was possible to develop coefficients to predict individual locomotive fuel consumptions for the Westbound data base test. This, of course, assumed that the individual locomotive performance characteristics within the consist were essentially constant throughout the 1,519 route miles. Evaluating these fuel consumption coefficients on a per test zone basis (Eastbound) indicated that this was indeed the situation with only a few exceptions in test zones #4, #5, and #8. Note that not all of the Eastbound data was presented in this paper for reasons previously discussed.

Only a slightly different technique was used to derive fuel data for the fifth power plant operating Westbound in fuel save. Due to continual variation in the number of locomotives in fuel save at any given time, it was not possible to predict individual locomotive fuel consumptions from the consist average. However, the fourth and fifth power plants were wired to be simultaneously operated in and out of the fuel save mode. With similar duty cycles, and again assuming consistent performance characteristics throughout testing, the fuel consumed by the fifth power plant in the Westbound direction was derived from the parallel performance of the fourth and fifth power units. Both the data base and fuel saver test results (Eastbound) were included in the derivation.

As shown in figure 6, the total time accumulated per throttle position varied considerably between the data base and fuel saver tests. The letter "C" denoted an idle-dynamic braking sequence frequently encountered in the somewhat mountainous terrain. For test zones #1 through #8 Westbound from North Platte to Los Angeles, major decreases of 50% to 70% were recorded for throttle positions #3 to #5 accompanied by 40% increases in the idle-dynamic brake and throttle #8 positions. The pattern shift of time versus throttle position for the three Eastbound test zones was slightly different. Though the time in throttle #5 varied little, significant decreases of 40-70% occurred in positions #3 and #4 as well as in #6 and #7, with a corresponding increase of 78% in throttle #8. Individual throttle histograms per test zone have been presented in figures 7 and 8.

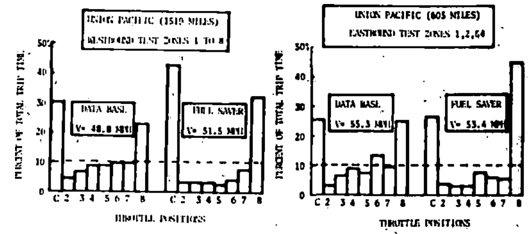


FIGURE 6. HISTOGRAM OF TIME VERSUS THROTTLE POSITION

In the Westbound direction, test zones #2, #3 and #8 exhibited higher operating speeds for the fuel saver tests, but lower fuel consumption per test zone when compared to the data base tests. This same trend was observed on three of six test zones evaluated on the BN unloaded coal train tests.

To determine if there were any significant changes in the fuel and lube oil temperatures while testing, these temperatures were continuously monitored at the rate of one set of readings a minute. Breaks in the thermocouple leads were common and were difficult to avoid. All available temperature data was condensed to illustrate the distribution at the completion of regular time intervals per test zone (Tables 7 and 8). Data for the pump-up fuel oil temperature was not available. After 30 hours in motion Westbound, the maximum return fuel oil temperatures averaged 128°F both with and without the fuel saver, ranging from a low of 117°F to a high of 138°F. For all eight test zones, the fuel saver return fuel oil temperatures were consistently similar to the data base comparison levels. The same trend was observed for the limited lube oil temperature data where the maximums ranged from 173°F to 184°F for the data base and fuel saver comparisons. Therefore, for these test conditions, operating at higher track speeds in fuel save did not affect the fuel and lube oil temperature levels.

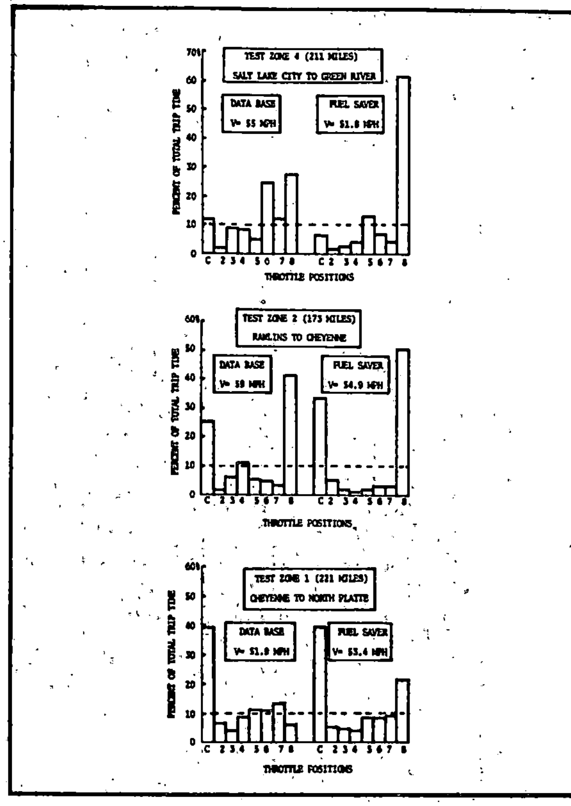


FIGURE 7. UP UNIT TOFC TRAIN (EASTBOUND) HISTOGRAM OF TIME VERSUS THROTTLE POSITION PER TEST ZONE

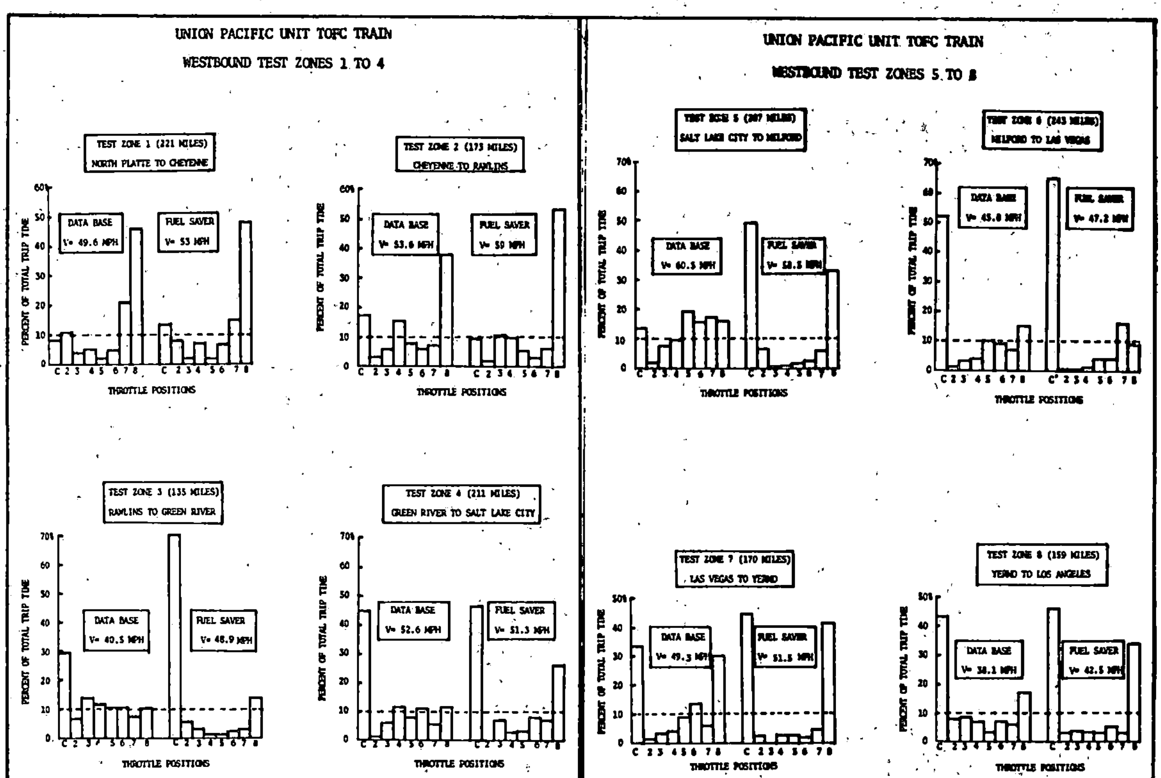


FIGURE 8. WESTBOUND: HISTOGRAM OF TIME VERSUS THROTTLE POSITION PER TEST ZONE

TEST ZONE	TYPE OF TEST	RETURN FUEL TEMPERATURE, °F: PERCENT OF TRIP TIME COMPLETED 0	10	20	30	40	50	60	70	80	90	100	MIN. TEMP. °F	MAX. TEMP. °F	DIFF. °F
(WEST) 1	DB	110	129	130	132	125	127	131	130	114	104	115	104	132	28
	FS	112	132	130	132	130	129	132	115 STOP	127	129	119	112	132	20
2	DB	111	129	129	122	114	121	121	122	115	116	108	108	129	21
	FS	108	128	129	118	112	123	118	125	123	115	115	108	129	21
3	DB	99	118	114	114	102	100	111	99	109	106	103	99	116	19
	FS	97	117	109	114	85	96	91	88	86	83	88	83	117	34
4	DB	NONE	114	115	118	107	106	108	106	105	105	109	105	118	13
	FS	88	119	121	121	101	90	95	98	101	105	100	88	121	33
5	DB	102	115	123	120	122	113	115	117	123	125	114	102	125	23
	FS	101	105	123	111	127	107	109	115	119	106	110	101	127	26
6	DB	111	123	125	119	120	117	124	119	133	124	119	111	133	22
	FS	110	125	127	114	111	112	113	114	116	115	123	110	127	17
7	DB	114	132	128	136	135	122	126	128	128	137	128	114	137	23
	FS	114	132	122	122	136	121	115	115	133	137	128	114	137	23
8	DB	124	123	134	127	128	122	122	123	122	123	122	122	134	12
	FS	117	138	137	137	120	116	119	134	121	121	121	116	138	22
(EAST) 1	DB	105	110	116	117	108	119	123	123	126	129	113	105	129	24
	FS	109	111	112	120	109	118	129	131	129	127	115	109	131	22
2	DB	103	109	122	111	118	122	124	116	109	109	109	103	124	21
	FS	96	113	108	103	123	123	117	132	116	111	110 STOP	96	132	36
4	DB	110	125	127	127	132	131	131	117	120	122	115	110	132	22
	FS	116	134	121	138	137	138	137	135	128	132	121	116	138	22

TABLE 7. UP UNIT TOFC TRAIN: RETURN FUEL OIL TEMPERATURES

TEST ZONE	TYPE OF TEST	LUBE OIL TEMPERATURE, °F: PERCENT OF TRIP TIME COMPLETED 0	10	20	30	40	50	60	70	80	90	100	MIN. TEMP. °F	MAX. TEMP °F	DIFF °F
(WEST) 1	DB	158	184	180	184	180	180			-	-	-	158	184	26
	FS	157	180	180	178	173	173	174	155 STOP	172	179	-	155	180	25
2	DB														
	FS	-	-	-	-	154	173	165	161	172	-	143	143	173	30
3	DB														
	FS	133	177	163	172	155	150	142	137	135	134	139	133	177	44
4	DB														
	FS	140	176	175	175	156	154	148	156	161	166	155	140	176	36
5	DB														
	FS	162	163	182	169	177	160	164	164	173	156	160	156	182	26
6	DB														
	FS	160	176	-	-	-	-	-	-	-	-	-			
7	DB														
	FS														
8	DB														
	FS														
(EAST) 1	DB	-	156	167	171	158	168	169	174	173	176	157	156	176	20
	FS														
2	DB														
	FS														
4	DB														
	FS														

TABLE 8. UP UNIT TOFC TRAIN: LUBE OIL TEMPERATURES

SUMMARY

During unit train tests performed on the Burlington Northern and Union Pacific railroads, significant fuel savings were realized by using a semi-automatic throttle control device or "fuel saver" system to take one or more units of the locomotive consist off-line when the available power and tractive effort exceeded the demand. This procedure effectively lowered the horsepower per ton ratio of the train and decreased the rate of fuel consumption. For the particular set of operating conditions tested, the average fuel savings in percent reached 9.8% for the unit coal train tests, and 12.4% for the unit TOFC train tests. A prime ingredient for the effective use of such a device was the operating locomotive engineer.

On a per test zone basis within each of the two test series, the fuel savings ranged from zero to considerably more than 15%. However, for three out of six test zones on the BN loaded coal train tests, the fuel consumed actually increased. In this particular situation, the 0.8 horsepower per ton power assignment was below the threshold at which the fuel saver concept could be effectively employed.

In both test series there were significant pattern changes in the average percent time spent per throttle position between the data base and fuel saver tests. As might be expected, operating in fuel save dramatically increased the total time accumulated in the eighth throttle position. As a result, there were significant reductions recorded for the average percent time spent in throttle positions #2 through #5.

In the TOFC tests and in the unloaded coal train tests, several of the individual test zones exhibited higher operating speeds for the fuel saver tests but lower fuel consumptions when compared to the data base tests. Combining this fact with higher ratios of ton miles per gallon and extensive shifts in throttle usage patterns indicated a possible trend toward greater fuel savings when using the fuel saver system at increased operating speeds for the 3.5 and 6.5 horsepower per ton power assignments.

For the ambient conditions tested, the fuel and lube oil temperatures changed very little per round trip after 30 and 50 hours in motion with the same number of crew changes and a similar number of stops. At the lower 25 mph operating speed of the unit coal train, the average temperatures were slightly elevated for the fuel saver tests. Such was not the case for the 50 mph unit TOFC train where the fuel and lube oil temperatures were consistently similar to the data base comparison levels.

For the conditions encountered, testing of the fuel saver system did not affect the total test times or the average operating speeds. Although there were no difficulties experienced in either the lead or trailing units of the locomotive consists tested, any maintenance or mechanical problems which might develop can only be evaluated after extensive usage of the system, combined with continual monitoring of the results.

ACKNOWLEDGMENTS

The cooperative efforts of the Burlington Northern and Union Pacific railroads were sincerely appreciated during the planning and execution phases of the fuel saver test series. Test coordination and equipment installation expertise was provided through the staff of the Transportation Systems Center at Cambridge, Mass. At the completion of the BN test series, the tedious reduction and evaluation of all test data was accomplished through the diligent efforts of the OAO Corporation.

INTERMODAL CARS--NEW DEVELOPMENTS

BY

J. R. BLANCHFIELD

M. A. KENWORTHY

Intermodal railcars used for "piggy-back" trailer and container transportation are becoming the subject of increasing design interest. In response to mounting fuel costs and competitive pressures, a number of new car designs have been developed, some to the hardware stage. From the designs it is apparent that two common goals are the reduction of tare weight and aerodynamic drag. The ride quality and dynamic stability characteristics of lighter weight cars are key issues which are being investigated. This paper describes the features of several new intermodal railcar concepts. It also reports on a cooperative government industry test program designed to quantitatively define and measure the ride vibration characteristics of current and experimental intermodal flatcars. Under the program, the procedures, equipment and analytical techniques suitable for evaluating car performance under both controlled test and actual service conditions were developed and successfully employed. Preliminary results indicate that substantial weight reductions can be achieved without adversely affecting ride quality or dynamic stability.

INTRODUCTION

The objectives of this paper are to review developments in the area of intermodal railcar design which represent the beginning of a new cycle in the evolutionary process and to describe a method of testing and evaluating the dynamic performance of new car prototypes under actual service conditions.

CAR DESIGN OBJECTIVES

As the backbone of the intermodal hardware system the flatcar has become the subject of increasing design interest due to a number of trends. In combination, these trends have produced several distinct objectives for all new flatcars which are:

- o Reduced train resistance
- o Ability to carry 45-foot trailers
- o Improved service capabilities
- o Reduced acquisition and maintenance costs
- o Reduced man-machine interaction
- o Reduced clearance requirements

The driving forces behind these objectives can be readily traced to:

- o Increased fuel costs
- o Shortages of motive power
- o Increased cost of investment capital
- o Tightened competition within and between modes
- o Need to extend service into northeast metropolitan area

Consideration of design changes is appropriate at this time because there is a need to produce new intermodal cars to replace those that have reached the end of their useful life and to pro-

James R. Blanchfield is the Research Manager for Intermodal Systems, Freight Service Division, Office of Freight Systems (R&D) for the FRA. He received his B.S. (1961) Mechanical Engineering, from Carnegie Mellon University.

Michael A. Kenworthy has served as Engineer, Engineering, Testing, and Analysis Division, ENSCO, Inc., since 1976. He received his B.S. (1970) Virginia Polytechnic Institute; M.S. (1972) Virginia Polytechnic Institute & State University.

vide the fleet expansion needed to handle an annual 10 to 12% growth rate in intermodal carloadings. A production rate of 6,000 new cars a year may be necessary.

INTERMODAL IMPORTANCE

The importance of intermodal traffic to the railroads is based on its revenue potential. Shippers of high value manufactured goods pay the highest revenue rates, but they demand prompt, reliable, damage free service in return. Dedicated, run-through intermodal trains appear to be the only way the railroads can meet the service requirements.

Such service is proving to be well worth the effort, for its revenues can assure the long term economic viability of the railroads. It is likely that within a few years many railroads will derive 20% or more of their total revenues from intermodal traffic. Recently it was reported that one major railroad has already attained the 25% mark. Noting that in terms of traffic volume, intermodal loads account for only 15% of the total, shows that its revenue potential is remarkably high. As a result of its earning power the investment in new intermodal equipment appears attractive. The opportunity will soon exist to introduce designs that are better suited to the needs of the times. Although the design requirements have yet to be defined, and probably will not be until the Federal Railroad Administration (FRA) Intermodal Systems Engineering Program has been completed, a number of concepts have been formulated within the industry toward achieving the basic objectives cited above.

DESIGN CONSIDERATIONS

The design of an intermodal car is a challenging task. The current car presents a deceptively simple image. It has been doing its job well for the past 20 years and when it is superseded the new equipment must be superior in meeting the needs of the industry.

Will the new design be evolutionary or revolutionary? For good reason the industry has been cautious and slow to accept revolutionary hardware. At the present time the new concepts being proposed range from alterations of the current design to radical new approaches to the idea of moving trailers and containers by rail.

In connection with FRA's Intermodal Systems Engineering Program , new ideas are being solicited for study and evaluation. Each will be subjected to a comprehensive assessment of its merits based on the interrelationships between the car and the other components in the intermodal system.

The systems aspect must be emphasized. In order for the railroads to benefit from the intermodal potential, the cost of providing the service must be minimized. A well-matched, efficient, and cost effective hardware system will be essential. Each element's characteristics must be established in concert with those of the other equipment items toward maximizing the system's overall performance in terms of service and return on investment. This means that all the equipment, facilities and sub-systems used between the shipper's and consignee's loading docks must be considered. The intermodal rail car is certainly one of the major elements of the system. Typical considerations for a new intermodal car include:

Size(s) of trailers/containers to be carried

- o Number of loads per car
- o Method of loading
- o Tare weight per load
- o Aerodynamics (resistance and stability)
- o Number of axles
- o Ride quality and stability
- o Method of connection between cars
- o Vertical and lateral clearances
- o Automation of load securement devices

- o Cost of acquisition and maintenance
- o Durability
- o Compatibility
- o Interchangeability

NEW CONCEPTS

A review of the proposed concepts has disclosed a variety of ideas concerning these design considerations. Variations in the ranking of priorities is apparent reflecting tradeoffs made by the respective designers. However, without commenting on each concept's merits with respect to the design considerations, it is interesting to observe in what areas changes have been proposed in some of the better known concepts.

The Santa Fe "Six Pack" concept utilizes articulated joints with 2-axle trucks at the five intermediate points of the six-unit car sets. This arrangement reduces by two-thirds the number of couplers and brake reservoirs in a typical 60 trailer train. A box type structure serves as the longitudinal member and supports cantilevered aprons for the trailer wheels. The slenderness of the center structure allows it to be straddled by the landing legs of the trailers eliminating the need for their adjustment. Car section length will correspond to the length of trailer to be carried resulting in a minimal spacing between successive trailers and lower aerodynamic drag. Twenty-eight inch diameter wheels contribute to a 16-inch reduction in vertical loaded height. Compared to conventional equipment, the Santa Fe design has achieved a weight reduction of 34%. The cars are intended only for captured service and require lift type loading capability at the terminals.

The Trailer Train Company is considering modifications to their conventional car which will permit the carriage of two 45-foot trailers. They have also reportedly been looking at a new configuration that would carry single 45-foot trailers. In both cases, lift type loading operations appear to be required.

Pullman-Standard utilized a skeleton type longitudinal frame in their design for a lightweight intermodal container car which resulted in the construction of two experimental lightweight cars in 1969. Both cars carried two 40-foot or four 20-foot containers. They incorporated several innovations in load securement devices, but their primary achievement was a 17% to 31% reduction in weight compared to the conventional cars.

The Southern Pacific has received for testing, a unique design wherein two 40-foot containers are carried in a stacked configuraton such that their overall height is 18'-6" above the rails. A car length of 63 feet saves 26 feet of train length for each pair of containers. Using articulated joints between cars a weight savings of 40% may be achieved over conventional cars.

The Bimodal Corporation's concept is to eliminate the need for a rail car completely by constructing trailers with a rail wheel system included in addition to the highway suspension components. Trains would be made by coupling numbers of such trailers directly together using devices included in the trailers. With elimination of the flatcar, the net to tare ratio for the rail mode would improve dramatically and there would be no problem with tunnel or bridge clearances.

The Paton Corporation has a concept which reduces the rail-based equipment to a minimum configuration. Low tare weight, high utilization rate, flexibility and the ability to operate through the restricted clearance areas of the Northeast region have been cited as its predominant features.

LIGHTNESS vs. STABILITY

A common attribute of all the new concepts is an improvement in the net to tare weight ratio toward the goals of improving fuel efficiency, increasing locomotive productivity and increasing component service life.

Of particular concern to the industry are the ride quality and dynamic stability characteristics of lighter weight rolling stock. The reduced mass of the carbody, combined with the possibility of a higher loaded center of gravity, could render such a car unacceptable.

However, because of the major payoffs in intermodal efficiency that could result from lighter weight rolling stock, the effort to acquire and provide reliable, quantitative information on the relationship between car weight and dynamic performance to those who will be working on new car concepts and performing system trade-off studies is very important. The capability to measure and characterize the ride vibration environment of intermodal cars will also aid in the evaluation of specific design alternatives once a basic concept or configuration has been established.

Under a cooperative FRA/industry research program, the procedures, equipment, and processing techniques have been developed and tested toward providing this capability. Intended to yield information about car behavior in the operational environment, the techniques can be applied to the evaluation of all of the new car designs.

FRA-INDUSTRY TEST PROGRAM

The program was originally planned to evaluate the performance of two experimental lightweight intermodal cars produced by Pullman-Standard in 1969. For comparison, a conventional Trailer Train TTAX car was included in the evaluation. Both lightweight cars were originally configured to carry containers only with landbridge operations in mind. At a later date, one of them was modified to carry trailers by adding fixed stanchions and support plates for the trailer wheels. Both cars carry two loads, as does the conventional flatcar. However, one of the lightweight cars weighs 47,800 pounds compared to 69,300 pounds for the conventional car resulting in a savings of 31%. The lightweight trailer car weighs 57,100 pounds for a savings of 17%.

DESCRIPTION OF TESTS

The Lightweight Flatcar Evaluation program consisted primarily of two types of tests whose geographic locations are shown in figure 1. The first test, referred to as the Vehicle Dynamic Characterization (VDC) test, was conducted in a controlled environment to provide information on the dynamic or elastic nature of flatcars. The second test was conducted in a revenue service environment to quantify the acceleration environment actually experienced by the flatcars and loads. This test is referred to as an Over-the-Road (OTR) test.

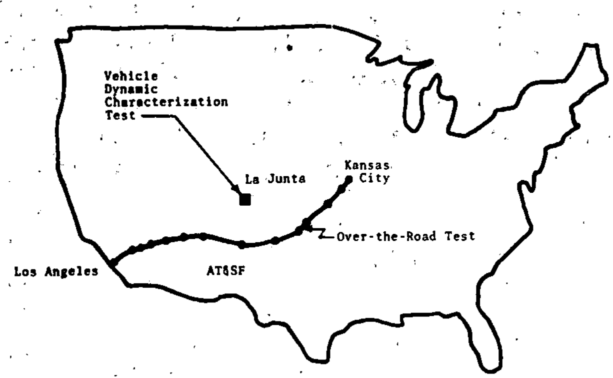


FIGURE 1. TEST ZONE LOCATIONS

The test consist was comprised of the three instrumented test flatcars and the FRA/DOT Data Acquisition Vehicle T-5. The test vehicles included a conventional TTAX (973799) and two lightweight flatcars designed by Pullman-Standard. One lightweight flatcar (TLDX 62) was designed for container service only and the other (TLDX 61) was modified for trailer service only. The test matrix included empty, half loaded, and fully loaded configurations for both the VDC and OTR test series.

The VDC test was conducted on two test zones near La Junta, Colo. The test zones were established on a one mile section of class 3 tangent track and a three mile section of class 5 tangent track. Accelerations were measured on the vehicle and recorded at consist speeds of 10, 15, 20, 30, and 40 mph over the first test zone and at 40, 50, 60, 70, and 79 mph over the second zone.

The OTR test series was conducted on main line track between Argentine Yard, Kansas City, Mo., and Hobart

Yard, Los Angeles, Calif. Data were recorded in twelve 10-mile test zones representing a cross section of track classes and structures. During the OTR test, there was no control over consist speed, and as a result, speed varied from 20 to 79 mph during measurements. This method of testing includes the effects of train handling which were not included in the VDC test.

Signals recorded during the VDC and OTR tests consisted of speed, automatic location detection (ALD), and up to 120 accelerations. The latter of these required a number of ancillary components in addition to the accelerometer transducer as shown in figure 2.

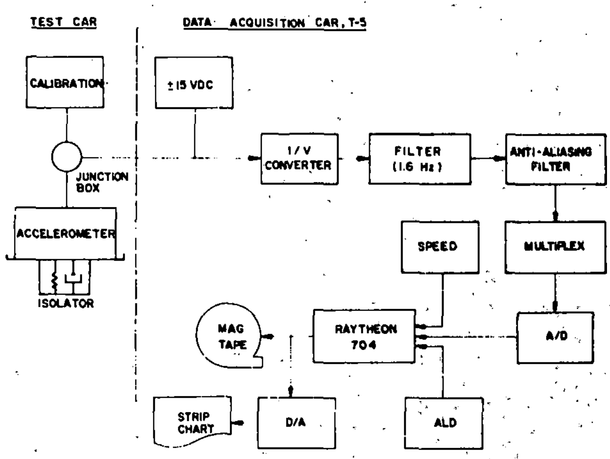


FIGURE 2. SCHEMATIC OF INSTRUMENTATION AND RECORDING SYSTEM

For the purpose of this evaluation program, 120 precision servo-accelerometers were mounted on the carbody, loads and axle journal bearings of the three test flatcars. 5g accelerometers were used on the car and loads while 30g accelerometers were used on the axles. An excitation voltage was supplied from T-5 and calibration signals were input at junction boxes located on each test vehicle.

Each accelerometer was mounted in a mechanical isolator for protection from high frequency accelerations of large amplitude present in the rail environment. Mismatched rail joints, for example, can produce impulses as large as a hundred g's. The isolators with a natural frequency of 150 Hz were designed to low pass accelerations. The signal from the accelerometer transducer was transmitted along the consist through a maximum of 250 feet of shielded cable in a current mode to avoid voltage drop due to line resistance.

In the Data Acquisition Car the signal was converted to a voltage. This signal was then filtered using a low pass single pole filter (-6dB/octave) with a corner frequency of 1.6 Hz. This filter is used to offset the effect of acceleration amplitude increasing with frequency and thereby produced an increase in the system resolution. Next the signal was anti-aliased using a 4-pole (-24dB/octave) Bessel filter with a corner frequency of 30 Hz. This type of filter provides a linear phase shift which is essential to data processing requiring phase synchronization.

The fully conditioned signal was multiplexed and converted to a 12 bit digital word at a rate of 128 samples per second. The digitized signal was stored in the on-board mini-computer (Raytheon 704) and buffered onto a magnetic tape. Selected channels were passed through the D/A converter and displayed on a strip chart recorder for real time examination of data. This system was also used to verify the data tapes after tests.

Data Reduction and Analysis

In order to analyze the acceleration environment of the flatcar/load system use is made of the technique of superposition. This method presupposes that the acceleration, a, at any point on the flatcar may be thought of as being the sum of contributions of rigid, a_r, and elastic body, a_e, accelerations.

$$a = a_r + a_e \qquad (1)$$

The components of a_r and a_e are referred to as the modes or modal coordinates. The use of modal coordinates offers a number of distinct advantages in the analysis of rail vehicles. First, the modal coordinates are conceptually easy to visualize and as a result are a great aid to the design engineer. Second, since these coordinates by definition are orthogonal or independent, phenomena such as cancellation and reinforcement do not obscure details of analysis. Third, modal coordinates may be used to obtain the actual acceleration level at any point on the vehicle.

The vehicle subsystems treated in this evaluation program are the load, carbody, and axle. The acceleration response of each subsystem is modeled as a separate freebody. A Cartesian coordinate system is established at the geometric centroid of each subsystem with positive x in the direction of travel, positive y to the left when viewed in the direction of travel, and z positive upwards. The rotational coordinates are θ, ϕ, and ψ about the x, y, and z axes respectively. Figure 3 illustrates the sign conventions for the carbody coordinate system. The load and axle coordinate systems are identical with the origin at their respective centroids.

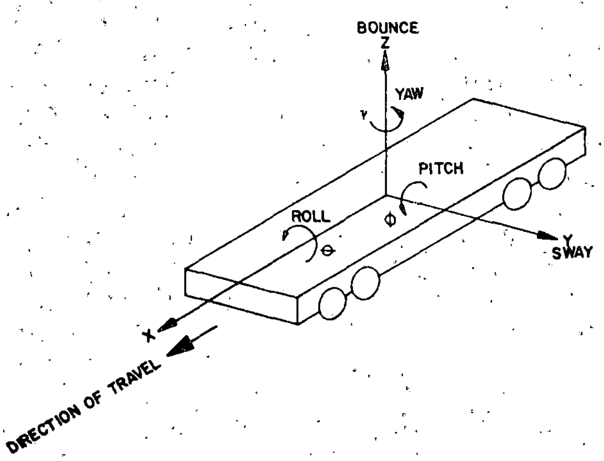


FIGURE 3. RIGID BODY MODAL COORDINATES

The rigid body modal coordinates are composed of three linear and three angular acceleration elements. Linear accelerations are parallel to the axes defined above and angular acceleration are about these same axes. For the purposes of identification, the acceleration along or about a given axis is indicated by a double dot over that coordinate, used to show double differentiation with respect to time. Furthermore, modal coordinates are subscripted with ^{0}. Thus the longitudinal modal coordinate is $\ddot{x}_0$; the lateral modal coordinate, referred to as sway, is $\ddot{y}_0$; and the vertical modal coordinate, referred to as bounce, is $\ddot{z}_0$. The angular modal coordinates are roll pitch and yaw denoted $\ddot{\theta}_0$, $\ddot{\phi}_0$, and $\ddot{\psi}_0$ respectively.

The contribution of rigid body modes to linear accelerations may be written as the modal coordinate in the given direction plus the contribution due to angular modal coordinates about the two remaining axes. Thus, denoting the rigid body acceleration components with the subscript r:

$$\ddot{x}_r(y,z,t) = \ddot{x}_0(t) + \ddot{\phi}_0(t)z - \ddot{\psi}_0(t)y \quad (2)$$

$$\ddot{y}_r(x,z,t) = \ddot{y}_0(t) + \ddot{\theta}_0(t)z + \ddot{\psi}_0(t)x \quad (3)$$

$$\ddot{z}_r(z,y,t) = \ddot{z}_0(t) + \ddot{\theta}_0(t)y - \ddot{\phi}_0(t)x \quad (4)$$

Note that the small angle approximation, cosine of the angle of deflection is approximately unity, has been made.

Next certain assumptions concerning the elastic body modal coordinates must be made. In the most general case an elastic body may bend and twist about any axis. Associated with each elastic deformation is an infinite set of modes or harmonics. Experience has shown that the more important contributions to the rail vehicle vibration environment occur below 30 Hz. Thus based on structural considerations and experience, it was determined for the carbody it was necessary to include only the first and second bending modes about the y-axis and first and second twist modes, more commonly referred to as torsion, about the x-axis. There are denoted α_1, α_2, β_1 and β_2 respectively and are illustrated in figure 4. The loads were found to require only the first bending mode about the z-axis. This mode is called lateral bending to distinguish it from the previous bending modes. The axle is treated as a purely rigid body so that the set of modal coordinates associated with the axle contain neither bending or torsion modes.

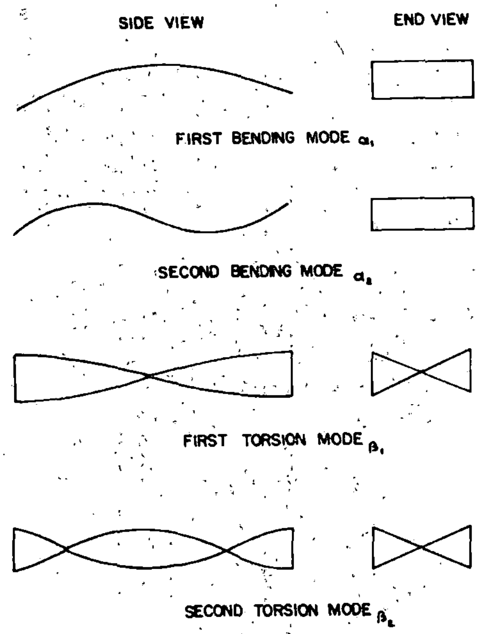


FIGURE 4. ELASTIC BODY MODAL COORDINATES

The shapes of the elastic modes are described by power law expressions. This is considered reasonable since the elastic deformations of rail vehicles are small in amplitude and result in relatively smooth shapes. Three terms are required to model the carbody first bending mode while two terms were required for all other modes. Based on considerations of symmetry these are written as:

$$f_1(x) = 1 + a_1x^2 + a_2x^4 \text{ for } \alpha_1; \quad (5)$$

$$f_2(a) = X + b_1x^3 \text{ for } \alpha_2; \quad (6)$$

$$g_1(x) = X + c_1x^3 \text{ for } \beta_1; \quad (7)$$

$$g_2(x) = 1 + d_1x^2 \text{ for } \beta_2 \quad (8)$$

where the coefficient a_1, a_2, b_1, c_1, and d_1 are referred to as the mode shape coefficients and are to be determined. The mode shape of the load lateral bending mode is similar to equation 5 but with a_2 set to zero. Elastic body contributions to carbody linear acceleration subscribed e are:

$$\ddot{y}_e(x,z,t) = [\beta_1(t)g_1(x) + \beta_2(t)g_2(x)]z \quad (9)$$

$$\ddot{z}_e(z,y,t) = [\alpha_1(t)f_1(x) + \alpha_2(t)f_2(x)] + [\beta_1(t)g_1(x) + \beta_2(t)g_2(x)]y, \quad (10)$$

Because of the assumption of small amplitude deflection, cross sections taken normal to the x-axis are not deformed. As a result, elastic body deformations make no contribution to longitudinal accelerations.

Equations 2 through 4 are combined with equations 8 and 9 to obtain expressions for the linear accelerations along the principal axes in terms of the modal coordinates. An example of this is illustrated by the vertical carbody acceleration

$$\ddot{z}_m(x,y,t) = A_0(t) \pm A_1(t)x^2 + A_2(t)x^4 + B_0(t)x + B_1(t)x^3 + C_0(t)xy + C_1(t)x^3y + D_0(t)y + D_1(t)x^2y \quad (11)$$

where

$$A_0(t) = \ddot{z}_0(t) + \alpha_1(t)$$
$$A_1(t) = \alpha_1(t)a_1$$
$$A_2(t) = \alpha_1(t)a_2$$
$$B_0(t) = \alpha_2(t) - \ddot{\phi}_0(t)$$
$$B_1(t) = \alpha_2(t)b_1$$
$$C_0(t) = \beta_1(t)$$
$$C_1(t) = \beta_1(t)c_1$$
$$D_0(t) = \ddot{\theta}_0(t) + \beta_2(t)$$
$$D_1(t) = \beta_2(t)d_1$$

and the subscript m denotes a modeled acceleration as opposed to a measured or observed acceleration. Expressions for the remaining components and masses can be written similarly. The time dependent coefficients defined in equation 11 are determined from measured accelerations.

Each mass was individually instrumented with a specified number of servoaccelerometers: 17 on each carbody, 8 on each load, and 5 on each axle. The transducer location and orientation are shown in figure 5, 6, and 7. A summary of the modes and measurements of each subsystem is given in Table 1. Each measurement of acceleration represents a single equation in terms of modal coordinates. Thus the set of 10 modal coordinates for the carbody is found using 17 equations. Similarly, the 7 load modal coordinates are found using 8 equations.

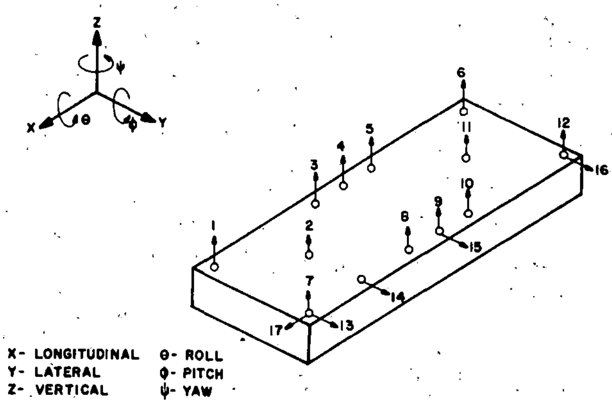


FIGURE 5. CARBODY ACCELEROMETER LOCATIONS

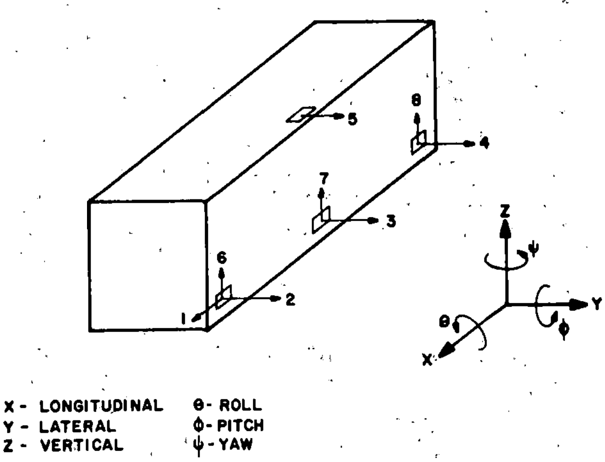


FIGURE 6. LOAD ACCELEROMETER LOCATIONS

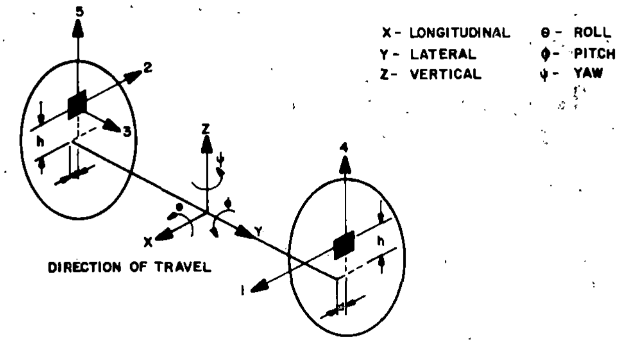


FIGURE 7. AXLE TRANSDUCER LOCATIONS

SUBSYSTEM	RIGID BODY MODES	ELASTIC BODY MODES	TOTAL MODES	MEASURED ACCELERATIONS
TRAILER	6	1	7	8
CONTAINER	6	1	7	8
CARBODY	6	4	10	17
AXLE	5 *	0	5	5

TABLE 1. SUBSYSTEM MODES AND MEASUREMENTS

These systems of equations are redundant, and the solution set of modal coordinates will not, except in the ideal case, satisfy the set of equations. For this reason the method of least squares is used to find that solution set which satisfies most nearly all the equations. The criterion of this method is that the sum of the squared errors be a minimum. This is accomplished by differentiating the expression for the sum of squared errors with respect to each modal coordinate individually and setting the result equal to zero. The result is a set of N equations in N unknowns which is solvable and yields the desired result. The set of equations corresponding to the number of measurements made is written as:

$$[X] \quad A = Z_m \qquad (12)$$

*Note that the axle pitch mode is not included since the axle itself coincides with the y-axis. Thus, pitch is simply wheel rotations.

DEPARTMENT OF TRANSPORTATION
FEDERAL RAILROAD ADMINISTRATION
WASHINGTON, D.C. 20590

ASSOCIATE ADMINISTRATOR

Dear Conference Attendee:

Over the last fourteen years, the Railroad Engineering Conference has earned a reputation as a highly effective symposium for exchanging technical information pertinent to the railroad community. We, in the Federal Raiload Administration have been proud to assist in sustaining this tradition during the past four years. In 1978, however, we will not sponsor our conference because the Sixth International Wheelset Congress will be hosted by the United States for the first time in its more than fifteen years of existence. The International Congress is composed of distinguished international technical experts and several hundred of these delegates will meet in Colorado Springs, Colorado, from October 23 through October 26, 1978. Although the Federal Railroad Administration's annual engineering conference will not be held this year, we intend to reconvene the series with the 15th Annual Engineering Conference in 1979.

If you were looking forward to participating in this year's Annual Railroad Engineering Conference as I was, I would like to take this opportunity to observe that the plans for the Sixth International Wheelset Congress give every indication that it will be a worthy substitute. The Congress is dedicated to the improvement of railway technology and safety; it serves as an international forum aimed at improving the contribution of wheels, axles and bearings in the rail system. To date, representatives from over twenty nations as well as from the United States plan to attend. Besides the formal program which promises insights into recent technological developments, the Congress offers a unique opportunity to informally meet with delegates possessing a broad range of application experiences and knowledge.

The first two technical sessions, on October 23, will be devoted to "The Behavior of Wheelset Materials". The third and fourth sessions, on October 24, will explore the "Behavior of Wheels in Service and Simulation Testing".

"Experience with Wheels in the FAST Loop" is the topic of session five, on October 25, which will conclude with a tour of the Transportation Test Center, including the Facility for Accelerated Service Testing (FAST) and the formal dedication of the new Rail Dynamics Laboratory (RDL). The final day of the Congress, October 26, will begin with presentations covering the "Theoretical Stress Analysis of Wheels" and finish with a session on "Roller Bearing Performance and Wheel Machining and Mounting".

I encourage you to attend and if you elect to do so, registration should be completed prior to July 15, 1978. Details, further information, and necessary forms can be obtained by contacting Mr. William H. Chidley, 230 North Michigan Avenue, Chicago, Illinois 60601, telephone number (312) 236-0294.

Sincerely,

Robert E Parsons

Robert E. Parsons
Associate Administrator
for Research and Development

cc: William H. Chidley

where X is the coordinate matrix of the polynomial, A is the vector of the coefficients defined in equation 11 and Z_m is the modeled acceleration vector. The condition of least squares is fulfilled by the expression

$$[X]^T \{Z_m\} = [X]^T \{Z\} \qquad (13)$$

where Z is the measured acceleration vector.

Substitution of equation 12 into equation 13 yields

$$[X]^T[X] \{A\} = [X]^T \{Z\} . \qquad (14)$$

Introducing the following definition

$$Q = [X]^T [X]$$

Equation 14 becomes

$$[Q] \{A\} = [X]^T \{Z\} , \qquad (15)$$

or finally

$$\{A\} = [Q]^{-1} [X]^T \{Z\} . \qquad (16)$$

Equation 16 thus defines the matrix operation required to convert measured accelerations on the carbody and load to the corresponding modal coordinate. The 5 axle modal coordinates are solved for using 5 equations in closed form requiring no fitting technique.

The A-vector for the carbody contains elements which are the sum of two modal coordinates (see equation 11), one rigid and one elastic. In order to uncouple these the condition of dynamic equilibrium is used. This condition states that the net force or moment due to an elastic deformation is zero. The result is an equation for each coupled element of the A-vector of the form

$$R_o(t) = \sum_{n=0}^{2} \frac{Mn}{Mo} A_o(t) \qquad (17)$$

where R_o is the coupled rigid body modal coordinate and A_n is the corresponding subset of the A-vector and M_n is defined by

$$M_n = \int_{-L/2}^{L/2} x^{2n} (x)dx$$

Here (x) is used in a more general sense to denote either a mass or polar mass moment of distribution. Detailed knowledge of the vehicle structure permits the uncoupling of rigid and elastic modal coordinates.

The time histories of accelerations measured on the carbody are thus reduced to modal coordinate time histories. As mentioned at the outset of this section, modal coordinates are useful in the analysis of rail vehicle performance, and the following data processing schemes are employed.

1. Goodness of Fit - A time series of the difference between each measured and modeled acceleration, z and z_m, is created. This series, as well as the z series, is Fourier transformed and the power spectral density (PSD) calculated. The power in the two PSD's is calculated from 0 to 30 Hz, and the ratio of residual to measured power obtained. This parameter indicates the percent error incurred in the model.

2. Power Spectral Density (PSD) of Modal Coordinates - The entire time series of modal coordinates are also Fourier transformed and PSD's formed. The root mean square (rms) is calculated from 0 to 30 Hz.

3. Positive Zero Crossing (PZX) Histograms - The time series of each modal coordinate is analyzed to determine the number of zero crossings with positive slope which lie in ranges or bins of given amplitudes. These results are tabulated in the form of numerical histogram on a mile by mile basis, i.e., one modal coordinate PZX histogram per mile.

4. Probability Density Function - The data for each modal coordinate are divided into 200 equal amplitude increments to cover the range from minimum to maximum amplitudes observed. The percentage of occur-

rences within each amplitude increment is then calculated and plotted to form a probability density function for that coordinate over the test run. In addition, the standard deviation, 95% level, 99% level and rms of the coordinate are calculated and printed out.

5. RMS Time History - The time history of each modal coordinate is Fourier transformed and a PSD calculated for each 4 seconds of time. The PSD is divided into octaves with center frequencies at 2, 4, 8, and 16 Hz, and the rms value calculated for each octave as well as the band from 0 to 30 Hz. These five values are then plotted as a function of time for the test run. The averaging time can be varied in increments of 4 seconds up to 16 seconds. In addition, speed is plotted on an adjacent graph for ease of analysis.

RESULTS AND CONCLUSIONS

The instrumentation and data analysis techniques developed for this program have proven highly successful in the evaluation of the dynamic performance of lightweight and conventional flatcars. In particular the use of modal coordinates provided clear, concise engineering results which correlate well with physical phenomena.

Shown in figure 8 is the effect of speed on the bounce of fully loaded flatcars. This result is representative of the acceleration environment in general; however, it should be kept in mind that bounce is only one of ten modes needed to completely describe the vibrational response of the carbody. This and other plots like it show conclusively that the lightweight and the conventional flatcars are quite comparable in performance. Futhermore, figure 9 shows the load bounce versus speed, and serves to illustrate that this conclusion can be extended to the loads. This is of primary importance in considering the economic performance of rail vehicles. Finally, the magnitudes of elastic deformation for both types of cars were generally equal which indicates that the lightweight flatcars are as structurally sound as their conventional counterparts.

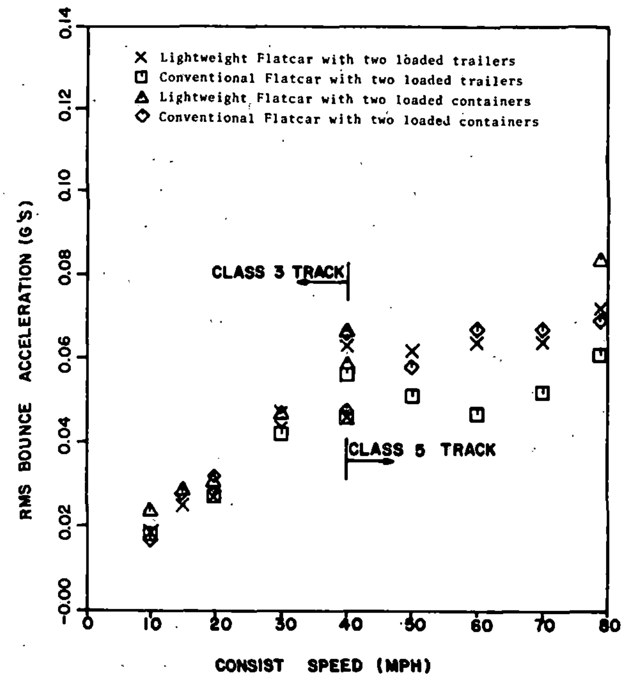


FIGURE 8. FULLY LOADED CARBODY ACCELERATION

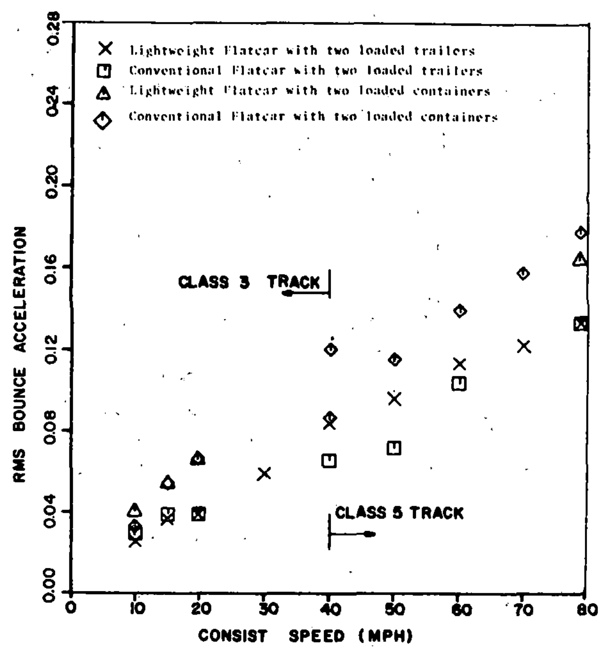


FIGURE 9. A-END LOAD ACCELERATION

Figure 10 shows the rms time history of the conventional carbody bounce mode. There are basically two points to note here. First, the first octave* with its center at 2 Hz is the largest contributor to the overall acceleration. This is anticipated based on considerations of carbody/load mass and the spring stiffness. Second, it is apparent that the acceleration level is speed dependent. In fact, figures 8 and 9 show speed to be the most important single parameter influencing the acceleration level. This serves to emphasize the comparable performance of the lightweight and conventional flatcars.

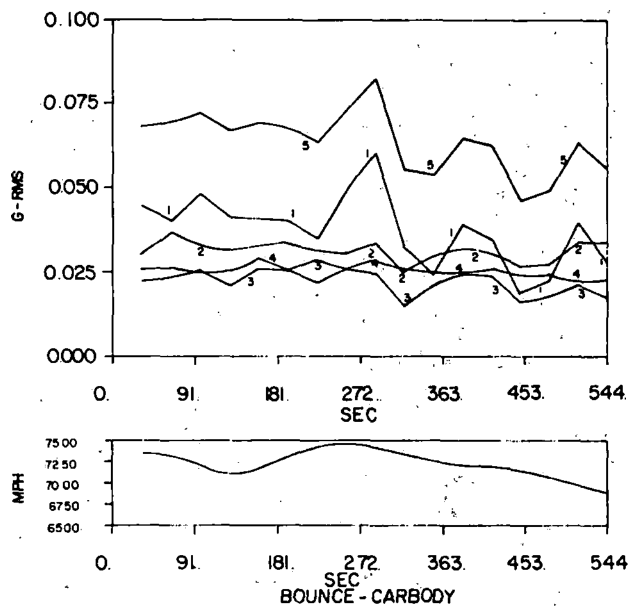


FIGURE 10. MODE TIME HISTORY

Results have also shown that fully loaded flatcars provide a better ride performance than other load configurations. In particular, it was found that the trucks of the conventional flatcar carrying a single load had a tendency to go into a lateral oscillation or hunting mode. This in turn had an extremely adverse effect on the acceleration levels in the loads.

Finally, data has been obtained which characterize the vibrational environment of containers and trailers during revenue operation. This data should prove beneficial to both the shippers and the railroads in evaluating the use of this mode of freight transportation.

Testing will continue during the upcoming year to provide similar data at specified points in the mileage accumulation of the test cars. This will allow a determination of the effects of component wear on the ride performance of these cars. This information is vital to engineers who will be designing and evaluating the future generation of intermodal rail cars.

ACKNOWLEDGEMENTS

The authors wish to thank Mr. L. P. Greenfield and R. Brodeur of Trailer Train Co., Mr. J. A. Angold and Mr. S. Green of the Santa Fe Railway Co., Dr. W. P. Manos of Pullman-Standard, Mr. Robert A. Love and Mr. Garth Tennikait of American Steel Foundries, Mr. N. A. Morella of National Castings Division of Midland Ross and Mr. Lorin Staten of the Transportation Test Center for their generous assistance in the conduct of this program. Dr. H. Weinstock of the Transportation Systems Center was instrumental in the formulation, planning and conduct of the program and his efforts deserve special recognition. Mr. Barney Ross of ENSCO, Inc. was responsible for coordinating and directing the data acquisition operations. He and the T-5 test personnel are to be thanked for their long hours and many miles of travel.

*In the rms time history plots the curves of the octaves are numbered consecutively corresponding to center frequencies of 2, 4, 6 and 8 Hz. The fifth curve is the 0 to 30 Hz band rms value.

OPTICAL ACI--A NEW LOOK

BY

R. L. WISEMAN

H. C. INGRAO

W. F. CRACKER

This paper describes a program which has been underway to provide the railroad industry with methods for the improvement of its Optical Automatic Car Identification (OACI) system. The program involves studies of car presence detectors, performance and cost improvements to the OACI scanner system, an analysis of OACI label properties and label life, and a model to evaluate car identification enhancement from the railroad's advanced consist information. The major part of the effort addressed improvements to the scanner system. These improvements involved the use of advanced technology to design, build, and test pre-prototype hardware to develop a "firm" specification of the scanner system performance limit. The results to date from laboratory tests indicate that the scanner system readability accuracy with the existing labels can be increased at least 6% from its nominal value of 88% to 91%. This increase is obtainable through the use of charge-coupled devices and microprocessors which will also enable a 40% reduction in initial scanner system purchase costs and a 33% reduction in yearly maintenance costs. The program also resulted in an assessment of the underlying causes for label deterioration, a label life estimate of 17 years, and a user's guide for each railroad's determination of the effectiveness of its own OACI data enhancement policy.

INTRODUCTION

This paper presents the recent results of efforts on the part of the Federal Government to explore the upgrading of the Optical Automatic Car Identification (OACI) system. The efforts have been under the technical support of DOT's Federal Railroad Administration (FRA) and are intended to specify the means for obtaining increased OACI system accuracy and wider application at lower costs. Although ACI is considered to be a major breakthrough for improving railroad service, operating efficiency, and car utilization, the nation's railroads have recently been reconsidering its continued use. Since its adoption on a national scale in 1967, the railroads have been faced with the problem of sustaining an effective maintenance program for the OACI scanner system and the labels mounted on 1.7 million freight cars. Depending on the operating life of the label and the levels of maintenance over the past five years, the OACI readability accuracy has varied from 78% to above 97%. While one railroad will contend that the lower accuracy can be greatly enhanced through correlations with their separately derived manual car identification records (advanced consist), another will argue that this degraded performance is unsatisfactory. The main reason for this difference of opinion

---

Robert L. Wiseman, OACI Task Force Manager, Department of Transportation, Transportation Systems Center. He received his BSE (1960) from City University of New York; MSAE (1968) from MIT.

Hector C. Ingrao, General Engineer, Equipment and Controls Branch, Transportation Systems Center. Ing. (1953) Facultad Ciencias Fisicomatematicas, La Plata University, Argentina. He is also an IEEE Senior Member.

William F. Cracker, Research Manager for Electrical Systems, Office of Freight Systems (R&D), since 1975. He received his BSE (1964) from Villanova University, MBA (1970), George Washington University; and his MSE-Electrical Engineering (1975), Arizona State University.

lies in the way each railroad utilized OACI in their Management Information System (MIS). Some railroads with their own maintenance program and fleets have derived significant benefits from greater efficiencies in their way-bill preparation, classification yard operation, and cargo identification yard operation, and cargo identification. However, since the nationwide benefits of ACI requires the cooperation of all the railroads, its fullest potential cannot be realized until a convincing case is made for improved performance at lower operating and maintenance costs and an attractive return on investment is demonstrated.

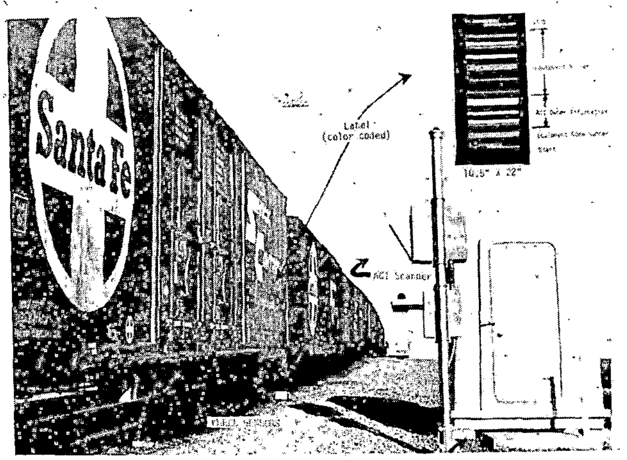


FIGURE 1. TYPICAL OACI INSTALLATION

BACKGROUND

Since 1890, when a patent was issued for a mechanical technique, the railroads have recognized a need for the automatic identification of the ownership and serial number of freight cars passing critical rail junction points. In 1967, the optical ACI was adopted as a viable technology after the Association of American Railroads (AAR) had developed specifications and tested the system in the field. Referring to figure 1, the OACI system was composed of three distinct elements: (1) A color-coded label; (2) An optical trackside scanner system; and (3) Wheel sensors to determine train presence and direction. When the train first approaches, a high intensity light source is turned on and begins a rapid vertical scan of the trackside with a set of rotating mirrors. The wheel sensors then identify the passing of each rail car which has labels mounted on both of its sides. The labels consist of 13 modules of retroreflective material like that used for markers illuminated by car headlights along the highways. Each of the thirteen modules has two stripes, which are colored white, red, blue, or black. The reflected light from these stripes is sensed slightly off the incident light axis and converted into electrical signals by photomultipliers. These signals are then decoded into three digits of car owner information and seven digits of the freight car/type number. If all of these digits and the fixed code of the label's "start" and "stop" modules are detected, this information is passed on to a label data processor for subsequent transmission to the railroad's local management information system computer. Validity for the 10 identification digits is checked by the return from a modulo-eleven binary coded parity module.

In the past, use of the OACI system's information has varied considerably, depending on its size and the special needs of the railroad. One very large network, managed by the Chicago Railroad Terminal Information System, Inc. (CRTIS), was developed as a joint railroad effort to serve 28 users over 7,689 miles of track connecting 100 freight car classification and support yards. or the last five years, the FRA and 12 railroads have been involved in a cost-shared CRTIS demonstration of the benefits of OACI in reducing clerical costs, car detention times, misroutings, and classification errors.

The results to date have had limited success due to: (1) a degradation in the quality of the labels; (2) a less than optimum scanner system performance; and (3) limited data enhancement. In 1975, concern over the slowly deteriorating OACI readabilities led to an industry request for an FRA sponsored CRTIS field test program which was conducted by the Department of Transportation's Transportation Systems Center (DOT/TSC).[1] The tests showed that the readabilities could be increased from the national average of 80% to 91.3% through recent improvements to the OACI scanner system by the equipment manufacturers. Although the field test sample of over 6,000 cars was sufficiently large, this result was subject to considerable controversy. Some railroads believed that the test site was not representative of their own OACI experience. Others reported that readabilities of higher than 95% were obtainable through a careful label washing program on their own captive fleets.[2] The problem was further complicated when each railroad tried to assess the readability effect in terms of the costs and benefits of their own operations. Very little technical information was available on the effective life of the labels and the underlying causes for their deterioration. The extent of readability improvements through advanced consist and multi-scanner correlations from both sides of a car had not been systematically defined in a form which could be interpreted by each individual railroad. These and other problems established a "wait-and-see" environment in which the static market for OACI systems precluded any major technological upgrading by the equipment manufacturers.

For the past year, the FRA has been working on four areas of the Optical ACI System to resolve its major issues and to provide a firm basis for the railroad industry's decision regarding its future use and deployment. These areas and their status are described as follows:

1. Car Presence Detector Studies have just recently been initiated to identify methods for improving the performance and reliability of the current wheel sensors and to investigate alternative techniques.

2. An analysis of the optical properties of the existing labels has been underway for the past year in order to estimate the label life and suggest methods for label improvement.[3] Preliminary results of this work were presented to the industry in June 1977 and are discussed in a later section of this paper.

3. An extensive effort to improve the performance and cost of the scanner system has been conducted over the past year by DOT/TSC. The results of a first stage of fabrication and testing of newly designed pre-prototype hardware in this effort were also presented to the industry in June 1977.[4] The major portion of this paper describes the advanced designs which will lead to a July 1978 specification of the scanner system performance limit. The description contains an analysis of the current system, an identification of the levels of improvements, the results of laboratory testing, and cost and sizing considerations.

4. An OACI System Alternatives Evaluation Model is being developed to provide each railroad with a tool for assessing their own approach to OACI. The model deals with the effectiveness of OACI enhancements from advanced consist information and multiplexed data from pairs of scanners. A user's guide for this model is being prepared and can be combined with an extensive Classification Yard Simulation Model developed by ARINC Research Corp. for the AAR.[5] The combination should provide a thorough assessment of ACI costs and benefits in the context of system enhancements, individual yard operations, yard configurations, and clerical requirements.

PART I: SCANNER SYSTEM IMPROVEMENTS

CURRENT SYSTEM DESCRIPTION

The existing OACI Scanner System is composed of optics of good commercial quality and 1969-vintage electronics components. The system was originally designed to identify 99.5% of new or properly maintained labels with an unusually low false alarm rate of less than 1 part in 250,000. However, label degradation in the form of dirt, damage, and other causes has reduced the light returns from some of the labels to the point where they are obscured by the system noise. For the purpose of identifying readability improvements with the existing labels, the scanner system (figure 2) may be divided into four parts: an optics subsystem; front-end amplification electronics; a detector (called a "standardizer"); and a label data processor.

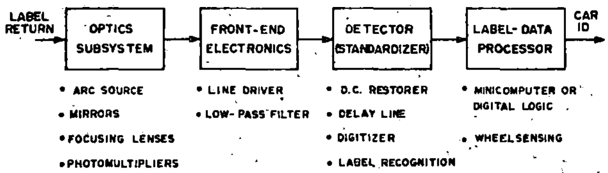


FIGURE 2. SCANNER SYSTEM COMPONENTS

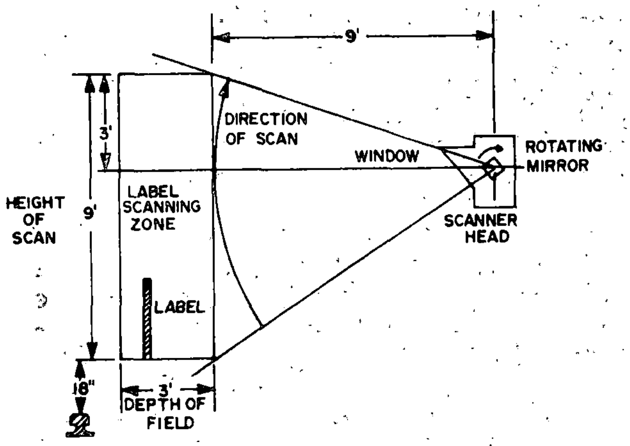


FIGURE 3. LOCATION ON SCANNER ZONE

The detection of a label starts when its vertical edge first appears in the plane of the label scanning zone shown in figure 3. The label is illuminated in this zone by a 7.5 inch circular beam of collimated light from a Xenon arc lamp within the scanner head. This incident light is swept upwards by mirror faces mounted on a spin cube which rotates at a rate high enough to ensure at least one scan of a label moving at 80 miles per hour. The rotating plane is tilted 7 degrees about the vertical axis to accent the labels retroreflective properties over non-label specular reflections which are dominant on the normal to the car side. The labels contain very small glass beads mounted on a silvered surface which reflect light back within a small 2 degree cone centered on the axis of the incident beam. This effect may be seen by sighting a label with a flashlight at angles which can be as much as 45 degrees off the normal to the label.

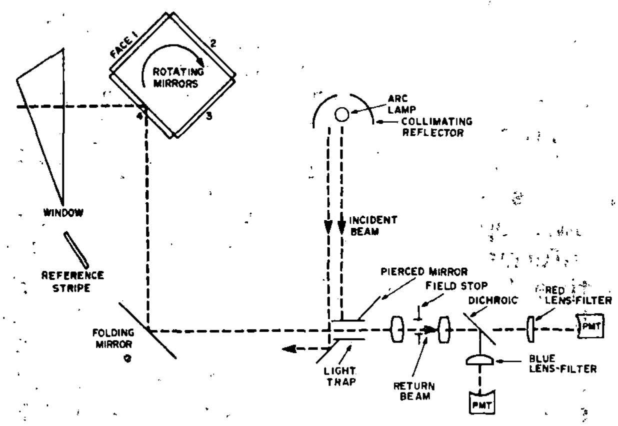


FIGURE 4. SCHEMATIC OF OPTICAL SUBSYSTEM

The optics subsystem is shown in figure 4 and contains a hole in a mirror which is also used to fold the incident light from the arc source on its way to the rotating mirrors. The hole admits the return light to a lens system which focuses it on the cathode of red and blue channel photomultipliers (PMT). Color separation is achieved by a dichroic mirror which passes the red band of light but reflects the blue band at 90 degrees. A white label module will therefore result in a PMT output on both channels. The red and blue filters are narrowband matched to the spectral characteristics of the label colors. The effective received field of view is established by a field stop which admits a horizontal slit equal to the entire length of a module but only one quarter of its height. This causes a triangular PMT output for each half-module color and a trapezoidal output for full module colors. An example of these waveforms and the color code interpretation is shown in figure 5. The color combinations will produce 16 possible logic states for each full module (four times four for each half module). Of these only 10 are used because of the restrictions that no bottom stripe will be black and that the red/blue and blue/red modules are respectively reserved for the "start" and "stop" modules. The label background between and around both edges of the modules is composed of low reflectance black anodized material. A background-to-module spacing ratio of 6 to 5 provides a near-zero return between the signal pulses of adjacent modules.

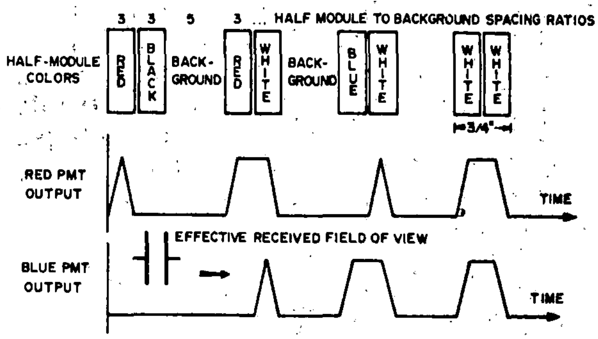


FIGURE 5. COMBINED RED AND BLUE CHANNEL COLOR CODE INTERPRETATION

Recalling figure 2, the red and blue photomultiplier outputs are each fed to a separate line driver/400-kHz low-pass filter combination with a dynamic range of 50 db. The line driver outputs present low impedance 30 mV to 10V label signals to the detector which is mounted in the air-conditioned Label Data Processor equipment hut. The hut also contains a power and signal interface box.

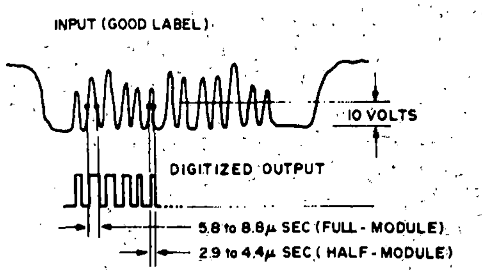


FIGURE 6. DETECTOR (STANDARDIZER) OPERATION

Since the purpose of the OACI system detector is to assure that only label analog signals result in an identification of a freight car, this device has been more appropriately called a standardizer. The standardizer eliminates false information from non-label reflections and assures the proper decoding of label signals through the use of DC Restorer circuitry, a delay line, a digitizer, and stripe/label recognition logic. As shown in figure 6, the triangular and trapezoidal photomultiplier signals arrive at the standardizer with rounded edges due to a non-ideal label reflectivity, and optical and electronic bandwidth limitations. DC Restorer circuitry first amplifies and level shifts the signal pulses so that they rise from a fixed DC reference level which is relatively independent of the slow variations in the outside ambient light. The ultimate objective is to convert these analog pulses into digital pulses with the same half or full module widths and relative spacing. This objective is realized through the use of a lumped constant transmission line which continuously tracks the instantaneous analog pulse amplitude over delayed time intervals. This delay line is shown in figure 7 and contains ten signal tap outputs with tap weight multipliers. The nine microsecond "times-one" tap and the two 0.5 taps assure that a digital pulse is formed from the half-amplitude points of the analog label pulse regardless of its peak amplitude (the reflectivity strength) or waveform width (the label distance from the scanner). Other taps further away from the X1 center tap provide amplitude guardbands which inhibit a digital pulse when adjacent module peak amplitudes or spurious noise spikes are more than ten times greater than the center peak. The center tap voltage is also inhibited when it goes below a 50 millivolt DC threshold or when it falls below 0.2 times a "crosstalk" signal from the center-tap of the other channel. The crosstalk signal prevents a white color decision when one color produces a signal less than 1/5th of the others.

The digitization of the red and blue label pulses is directly followed by two stages of Label Recognition Logic. The first stage examines the individual half-module pulse widths and determines the red and blue coincidence. These are assembled to arrive at a full-module numerical code which is checked for maximum pulse width and maximum distance from adjacent modules (which should always have at least one channel pulse in their first half module). If these conditions are not met, any preceding pulses are classified as noise and the circuitry is reset.

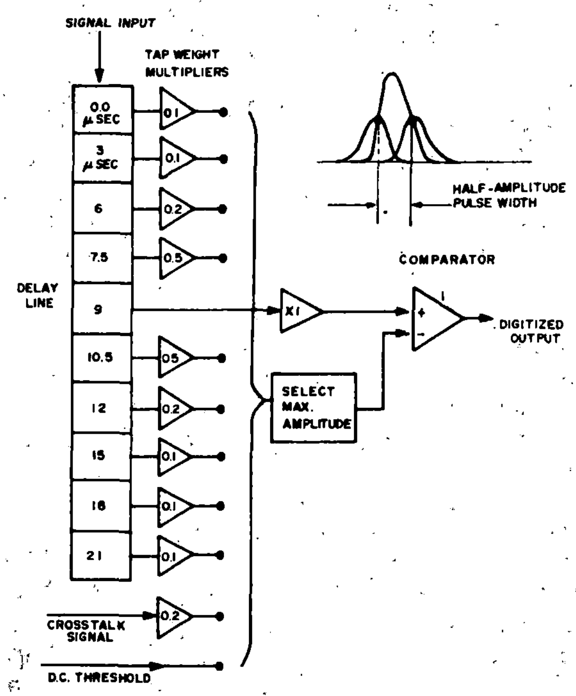


FIGURE 7. STANDARDIZER DELAY LINE OPERATION

The second stage of label recognition checks more global information about the label, verifying that the pulse train consists of a "start" module followed by 10 numerical modules, a "stop" module, and a "parity" module. If any module is missing, all preceding pulses are discarded as noise. If the pulse train satisfies all of the preceding conditions for a label, the label numerical codes are loaded into memory in the Label Data Processor.

The Label Data Processor verifies the label parity, checks for multiple scans of the same label, and uses the wheel sensor signals to identify each car and watch for cars with no labels. For each train, a list is then assembled and printed or transmitted to a remote computing site.

SCANNER SYSTEM IMPROVEMENTS

In its simplest form, the improvement of a scanner system to read more labels which have become degraded involves two tasks: (1) The dynamic range must be extended downward to read smaller optical returns; and (2) The effects of noise, either on the label or internally to the scanner must be reduced. That is, the system gains for new labels are maintained while improvements are made to identify very small signals from degraded labels in the presence of noise from three sources: (1) background noise from the label; (2) noise from the scattering of internal light; and (3) electronic noise. With this in mind, two stages of modifications to the scanner optics, electronics, and label detection subsystem were identified:

1. A <u>First Stage</u>, involved: (1) optics improvements; (2) wider dynamic range of the front end line driver; and (3) more stable thresholds for the existing standardizer. These modifications were intended to be an early package which could eventually be retrofitted in the field by the manufacturer at a cost of $4,500 (approximately 10% of the initial purchase price of the scanner system). The first stage hardware has been designed, fabricated, and tested in the DOT/TSC laboratory. The tests simulated the key aspects of the conditions in the field and were performed with a label population which was a representative selection of marginal and non-read labels provided by the railroads. These modifications were installed in a scanner system and directly compared to another scanner which had the manufacturer's latest improvements and a known readability of 91.3% established from field tests. The comparison revealed that the modifications produced a readability improvement[4] of over 4%.

2. Final Modifications are now being designed to replace the standardizer and the Label Data Processor minicomputer with a new detector and multi-scan correlation in a microprocessor. These modifications will build on the optics and line driver improvements of the First Stage and will take advantage of recent advances in signal detection techniques and integrated circuit technology. These modifications have been verified as feasible and, from the results of a detailed analysis, will produce an additional 2 to 2.5% increase in readability. They will also involve a major reconfiguration of the present four-box system with its air-conditioned hut into a two-box system mounted entirely on the scanner wayside pole. This and the use of integrated circuits will reduce the initial costs of the scanner system from the present $40,000 to $54,000 range down to an estimated price of $27,000. The system reliability and maintainability will also be improved, reducing the scanner maintenance costs from $5,100 to an estimated cost of $3,400 per year.

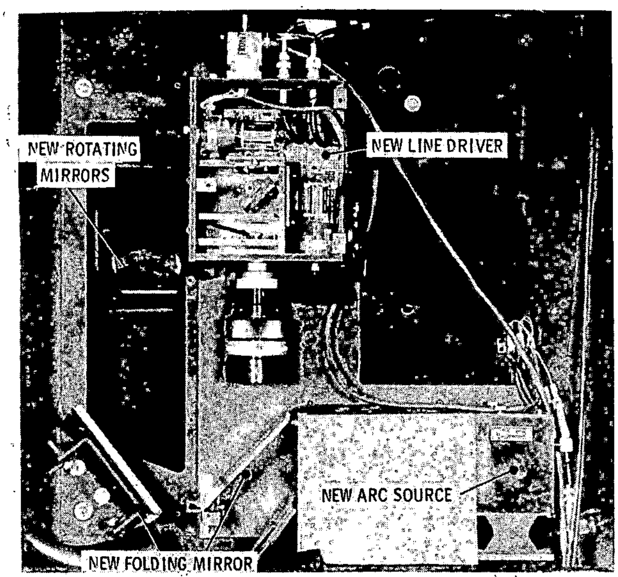


FIGURE 8. FIRST-STAGE OPTICS & LINE DRIVER MODS

The optics modifications and the line driver modifications are shown in figure 8 and are described as follows:

1. Optics Modifications

A. A new arc source manufactured by Varian, Inc., has been substituted to obtain a brighter rectangular beam of light on the label. The beam height has been reduced by a factor of two while maintaining an optical collecting and focusing system of the same F number as that currently used. The lamp assembly contains a secondary optical system with a well defined illumination beam which results in less internal light scattering. Although the new source is more expensive ($475 versus $375), it has a 60% longer lamp life and can be much more quickly replaced without any of the present requirements for special alignment time and skill.

B. A half-silvered mirror has been installed in place of the present pierced mirror. The new mirror and a larger, more expensive lens ($260 versus $60) increase the light returns on the photomultipliers.

C. Flatter folding and rotating mirrors have been installed (at negligible additional costs) to obtain better resolution and more repeatable scan-to-scan module pulse times from the label. These mirrors also operate in conjunction with a small lens to obtain a synchronization pulse with a .05 microsecond stability for the advanced detection processor applications. The sync pulse is obtained from a reference module placed inside the scanner and slightly below the bottom of its viewing window. This module can also be used for optical through-put checks and photomultiplier gain stabilization similar to that already provided in the present scanner.

2. Line Driver Modifications

The new line driver has an increased dynamic range of 80 db which should be sufficient for the weakest (1 millivolt) returns from very degraded labels. The driver has integrated circuit operational amplifiers in place of transistors to achieve a higher immunity to temperature and power supply variations. The optics modifications and the new line driver have resulted in a 3% readability increase and have reduced the internal light scattering to the point where the dominant noise (of approximately 3 millivolts) is from the background material and deterioration of the label itself. An additional 1% improvement was obtained through modifications to the existing standardizer at an incremental cost of $1,250.

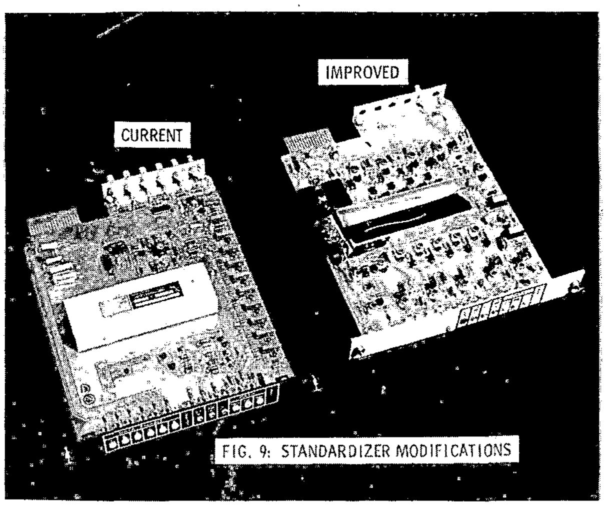


FIGURE 9. STANDARDIZER MODIFICATIONS

The new standardizer is shown along with the present one in figure 9 to indicate that the breadboard electronics were well-constructed and are direct plug-in replacements for the existing circuitry. The purpose of the modifications was to increase the dynamic range by a factor of 3 (from 46 db to 56 db) and to provide a stable threshold for degraded label signals in the region of 5 millivolts. The stability was obtained through the substitution of integrated circuit operational amplifiers for the transistor summers used in the tap weight multipliers (see figure 7). This substitution reduced the threshold temperature sensitivity by a factor of three (from 12 mv to 4 mv, 0° to 50° C) and resulted in a better immunity to power supply variations (from 15mV/V to l mV/V). Signal reflections in the delay line were also reduced through high impedance buffering at the inputs to the tap weight multipliers.

The final stage of modifications began with an assessment of the capabilities of the present standardizer and Label Recognition Logic. These subsystems were well designed to identify labels which had signal returns as low as 1% of those from a new label. However, a significant number of the degraded test labels received from the railroads had pulses more than 10 times lower (5 millivolts) than this threshold and, in some cases, were barely distinguishable from the background noise. In addition to the threshold limitation, the unusually strict requirement on false alarm rates had led to a design where partial label reads were discarded during each scan with no provision for scan-to-scan correlation. This situation, and recent advances in microcircuit technology, dictated a major modification of the system detector based on gated integrator or matched filter techniques. The new design started with the requirement for an accurate location of the label from which the signal energy could be accurately gated into an averaging circuit. The averaging circuit would be followed by a matched filter for module decoding which operated from an instantaneous estimate of the label's width and position in time. This design was complicated by three problems:

1. The pulse widths vary as the arc tangent of the ratio of the label height to its horizontal distance from the scanner.

2. The pulses have a wide variation in amplitude from module to module.

3. Vertical motion of the railroad car can cause a continuous shift in pulse location by as much as 1/6 of a pulse width during each successive scan.

These problems were solved with the following designs:

1. A Voltage-Controlled Oscillator (VCO) with a secant-squared function is used to vary the system clock rates. This results in the detection of pulse trains with an even spacing and width which is independent of vertical position.

2. A Label Locator is used to take advantage of the fact that the 13 regularly spaced label modules are a unique pattern on the railroad car side. The input to this locator is fixed-level digital pulse obtained from an adjustable threshold detection of the analog label signals. The output of the locator is the label location time, pulse widths, and an indication of the confidence in these two measurements.

3. A Signal Processor and Module Decoder uses the label locator output from the previous scan to perform a matched filter averaging of each module's pulse energy during the current scan. This information is then fed into a tapped delay line where the half-module and intermodule spacings are stored. The module is then decoded and finally assembled into a label identification and a confidence indication on each module's detected digit value.

4. A Microprocessor is used for multiscan correlation of the large number of label identifications and confidence indications obtained on each scan from the module decoder. For the train speeds usually encountered (speeds above 40 miles per hour are extremely rare) at least 10 identifications are available. The most likely identification and the confidence on each module are then stored as the car identification.

Figure 10 illustrates the method for locating a label with tapped analog delay lines made from charge-coupled devices (CCD's).[6] As the label's combined red and blue signals pass by in time, a set of windows matched to the label pattern are observed for the sum of colors in all windows. The sum has a triangular form which peaks when the label is exactly aligned with the windows. This analog label location technique requires an array of five such windows of successively smaller length and window size in order to match labels in the range of horizontal distances from the scanner. A breadboard version of a single array has been constructed and has located very degraded labels with pulses as low as 5 millivolts. The location accuracy was within 1/6th of a module width. As a backup measure, a second all-digital label locator utilizing the pulse symmetry was constructed and tested. This locator had the same accuracy as the analog version and could locate a significant number of the laboratory test labels which were not read by the scanner.

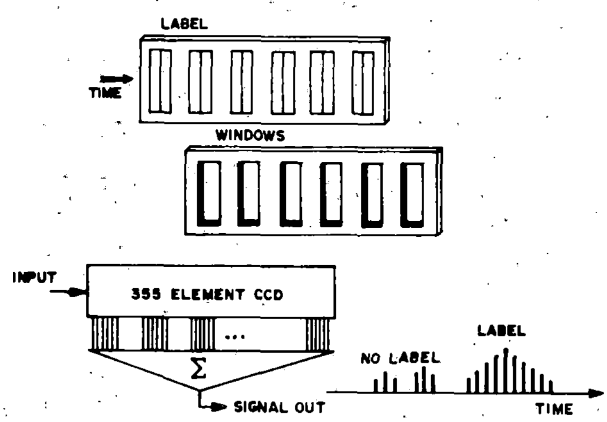


FIGURE 10. CHARGE-COUPLED DEVICE LABEL LOCATOR

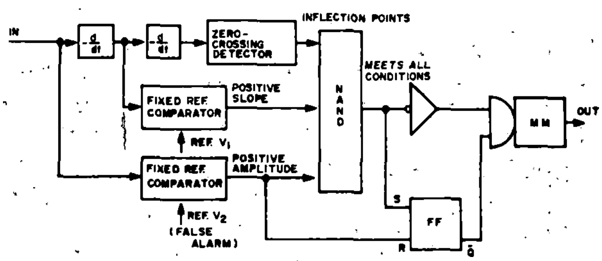


FIGURE 11. ADJUSTABLE THRESHOLD DETECTOR

The principle of adaptive threshold detection is illustrated in figure 11. Half-amplitude leading edge detection independent of the pulse amplitude is obtained through an "anding" operation of the signal amplitude, the sign of its slope, and its inflection points. All of these conditions will be met for band-limited signals at the half-amplitude point. The detector also has a threshold setting to reduce false alarms and contains a gated delay to suppress noise spikes.

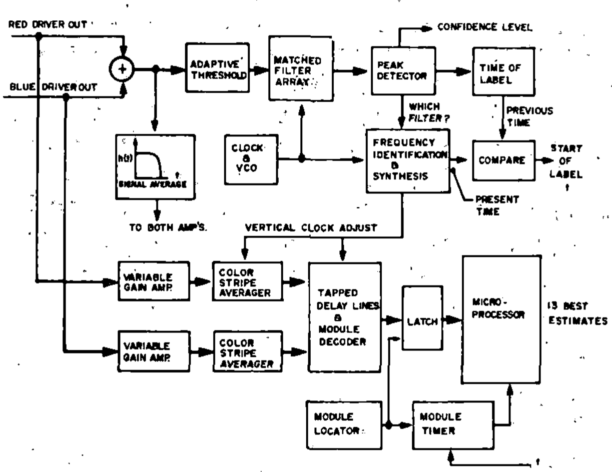


FIGURE 12. LABEL LOCATOR & POST DETECTION PROCESSOR

The full system block diagram for the label locator and the detection processing is shown in figure 12. The label locator is indicated on the top part of the figure where the combined red and blue channel digitized pulses are fed into the array of five matched filters. The highest amplitude of all of these filters is identified by a very fast (5 MHz) peak detector which has been designed and tested. The detection processor is shown in the bottom part of the figure and contains the signal averager with an h(t) impulse response which varies as the label length. The signal average dynamically varies the gains in the separate red and blue channels. The resulting signal is then passed on to the module detector which identifies each module and its associated detection confidence level.

The microprocessor is commercially available in a "Mil.Spec." version with a 0.5 microsecond clock cycle time. The input data is 16 eight-bit words per scan which includes 11 words for the car identification and the parity digit. Four thousand words of Random-Access-Memory (RAM) are required for the I/O buffers and 64 scans of data storage. The software program will occupy 4,000 words of Erasable Read-Only-Memory.

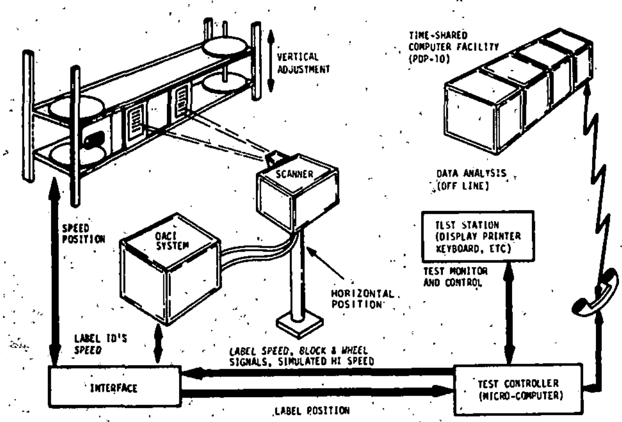


FIGURE 13. SYSTEM TEST FACILITY

SCANNER SYSTEM TESTING

Figure 13 illustrates the DOT/TSC laboratory test facility where the readability improvements were measured (a photograph of this facility is contained in an earlier paper in these proceedings entitled "Overview of Freight Systems R&D".) A label motion generator holding up to ten railroad labels was constructed to simulate horizontal label motion past the scanner system. Speeds up to five miles per hour were obtained from a chain-driver carrier. An extensive hardware interface and a microprocessor system were also developed for a test control and monitoring capability. These included an operator display and keyboard, control of label speed, and recording of the label read status on a printer and cassette tape. The freight car wheel sensor signals and speeds higher than five miles per hour were simulated. Off-line data analysis, comparing the scanner read values with the actual label identification, was performed on a large time-shared computer facility.

A test label population of 54 labels was carefully selected from 129 degraded[4] labels supplied by the railroads. The selection process involved the identification of the non-read and marginal read labels and a matching of the percent of non-read error causes (damage, dirt, bent backing plate, etc.) with the distribution of error causes in the field.[1] An analysis revealed that the error causes were independent of car type (box car, hopper, etc.). This enabled the tests to be performed at a fixed distance which was representative of the distances of car types in the national fleet. Representative speeds[4] were also obtained from field data.

The measurement of improvements in readability over the known 91.3% readability of the manufacturer's reference scanner system were based on a newly developed readability criteria called a Figure of Merit (FM). The FM was created from a close examination of the standardizer input signals and a knowledge of its central detection mechanisms. These examinations suggested a ratio of two terms: (1) The average value of the pulse heights in the red and blue channel; and (2) A denominator which was the sum of the ratio of the largest to smallest signal amplitudes in each channel. This Figure of Merit ratio represented the combined effects of thresholding and delay line reflectivity constraints in the standardizer. A functional relationship between values of the FM and known field readabilities was then established for the 54 test labels and was closely approximated by:

$$\text{Readability} = 1 - (\text{Figure of Merit}) \qquad (1)$$

A rank-ordered correlation of the read and non-read status of the 54 test labels in decreasing values of the FM showed significant results: A correlation of 0.97 was obtained in the determination of the readability of a single label from its reference scanner FM value. Moreover, the FM value for labels read by the reference scanner always established its readability at 91% regardless of their distance within the scanning region. This result enabled the increased readabilities of an improved scanner to be directly derived from the increase in the number of labels it read over the reference scanner. The increased readability established by equation 1 was easily obtained from a table look-up of the corresponding FM value for the improved number of reads.

PART II: OACI LABEL PROPERTIES

The Scotchlite engineering grade (commonly used on the highways) is used as the basic material in the manufacturing of OACI modules and consists of a superposition of eight layers with different physical properties. Figure 14 shows a schematic cross section of a standard module. The removable cover sheet (1), when peeled off, exposes a self-adhesive layer (2). The second layer (3) is a moldable cushioning coating. The reflector coating (4) in-

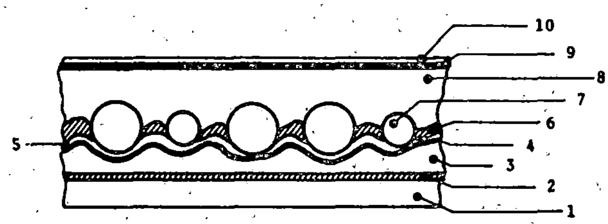


FIGURE 14. SCHEMATIC CROSS SECTION OF A STANDARD OACI MODULE

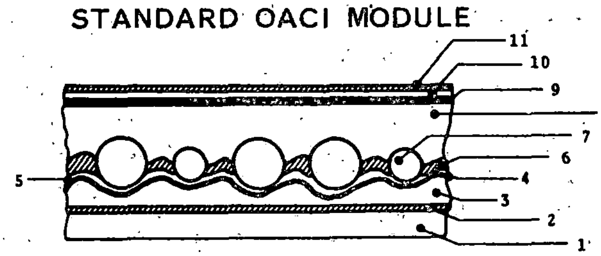


FIGURE 15. SCHEMATIC CROSS SECTION OF AN IST OACI MODULE

cludes metallic flake pigment particles; (5) contains a transparent color pigment; (6) contains a transparent color pigment in which is partially embedded a single surface layer of glass beads (7), which give retro-reflective properties when illuminated. There are upwards of 1550 beads per cm^2. The transparent coating (8) over the layer of glass beads bonds to and conforms to the exposed front surfaces of the beads and the binder coating (6). Finally, the silk screened (9) color-coded layer receives the protective coating (10).

Figure 15 shows a schematic cross section of an Improved Surface Treatment (IST) module which is similar to the standard label with the addition of a layer of fluorinated ethylene-propylene (FEP) Teflon 2 mils thick film (11) put on a layer (9) with a self-adhesive (10). The Teflon layer provides an inert surface which does not collect nearly as much dirt as the standard label, therefore considerably extending its life in the railroad environment.

To evaluate the optical properties of the OACI modules (blue, red and white), four parameters should be measured: (1) the wavelength λ max, for maximum retroflectance; (2) the bandwidth, between the 10% points; (3) the retro-reflected full beamwidth angle, A; and (4) the optical retroreflectance, G. The primary parameter of interest is the retroreflectance, G, which is proportional to the signal voltages at the detector input.

LABEL LIFE ESTIMATES

Previous estimates of label life have been made from visual judgments of label quality. The judgments have proven unreliable since the human eye does not see the labels in the same way the scanner system views them.[4] The estimates of the operational life expectancy in this paper have been based on field data from the Canadian National Railways and on weathering data supplied by the 3M Company at fixed test installations for Scotchlite modules which were exposed in a south-facing direction. The estimation of the OACI label operational life expectancy is complicated by the fact that a definition of operational life has not been formulated and that labels are: (1) installed with different degrees of quality control (especially substrate preparation); (2) exposed to different kinds of natural environment (i.e., solar radiation, rain, snow, etc.); (3) subjected to different railroad environments strongly dependent on type of car and cargo; and (4) subjected to different levels of maintenance. In order to properly assess 3, 8 the field Scotchlite weathering and OACI label data and translate it into operational life expectancy terms, an understanding is required of the causes and/or mechanisms which affect label operational life for different label structures (i.e., standard, IST, and standard overlayed).

For the OACI label operational life, the following definition applies: OACI label operational life is the time, T, required in a given environmental and maintenance condition to reduce the original retroreflectance, G, of any of the 13 modules to 5% of its original value. (This value is conservative and is consistent with the present and improved 1% minimum system voltage for the scanner system). Since the reduction in retroreflectance is affected by the environment, operational conditions, and/or level of label maintenance, it is important to qualify this definition with these factors.

The first source of data for the label life estimate was based on data from the 3M Company. 3M has conducted, over the years, weathering tests at their different test sites of the Scotchlite and OACI modules (standard and IST). These tests consisted of the measured G, along with spectral retroreflectances and chromaticity coordinates.

Figure 16 gives the results of tests at Texas (a) and Florida (b) for Scotchlite. The data consists of approximately 100 data points per year from 20 samples of each of the following colors: blue, green, red, silver, and yellow. An evaluation of the average solar insolation at ground level at both sites and, taking into account the different orientations (vertical and slanted at 45°), led to the conclusion that the ratio of total integrated solar radiation on the samples tested in Florida and Texas is approximately two. Field tests for standard OACI modules rendered similar results to the field tests of the Scotchlite material. By observation of the data given in figure 16, it is clear that to reach a given G in the Florida samples, it will take one-half of the time that it took in Texas. This suggests that the difference in solar radiation input is the cause. Further analysis of the Scotchlite also suggests that the reduction of G appears to be mainly due to changes in the bulk of layer #8 due to polymerization and on the surface of layer #10 (see figure 15) due to loss of gloss.

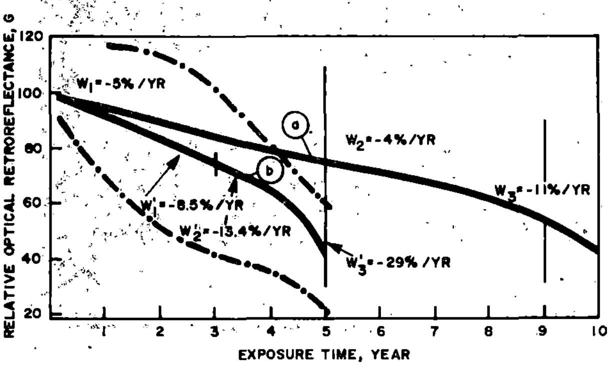


FIGURE 16. SCOTCHLITE RELATIVE OPTICAL RETROREFLECTANCE, G, VS. WEATHERING TIME AT: (A) TEXAS, SAMPLES ORIENTED VERTICALLY; (B) FLORIDA, SAMPLES ORIENTED 45° TO THE HORIZONTAL

Based on the limited data (9 samples over 20 months exposure) available, it appears that the IST modules do not weather any differently than the standard modules. That is, the Teflon material does not provide any protection against solar radiation.

The second source of data for life estimates was obtained from the CNR in-service evaluation of IST labels.[10] In February 1970, one IST and two standard labels were applied to each side of 20 "Dane Ore" captive fleet cars in service between Hamilton, Ontario, and the iron mines in Northern Ontario. These cars are exposed to a severe environment in passing through the mine and the steel plant to automatic ore loading and unloading operations. As a result of CNR laboratory measurements on some of these labels, and after five and seven years in-service, reductions in retroreflectance of 3.3% per year and 2.2% per year (a mean of -2.8% per year) were respectively obtained. The ratio of this value to the -4% per year observed by the 3M Company (figure 16a) for the Texas test site is 0.7, which is the estimated ratio of the yearly solar exposure on a vertical surface facing south in Texas to the region where CNR conducted the tests.

Since the tested CNR labels are seven years old, the 2.8% per year (4% per year x 0.7) adjusted rate from the 3M Company tests (figure 16) is applicable and is the same value obtained by the CNR. Based on the 3M data in figure 16 and yearly solar exposure adjustments, the IST CNR labels decay will increase up to 7.7% per year after a 10 to 11 year period. Allowing for an uncertainty on the order of 15%, <u>the estimated operational life of the IST CNR labels will be approximately 17 years</u>. This estimate is based only on module performance and not on other overriding mechanisms such as damage and deterioration of the background surface.

Simultaneously with the IST label evaluation, the CNR carried out the evaluation of overlays on in-service labels from the "Dane Ore" captive fleet. The experiment indicates that the self-adhesive on the Teflon has improved the standard label surface conditions and that Teflon improved the label life to the point where it had the IST performance.

LABEL ALTERNATIVES

Review of available OACI label operational data obtained by railroads and test data obtained by the 3M Company on the two different types of material (standard and IST) shows a wide range of railroad reports on label operational lifetimes. This apparent disparity can be explained on the basis of three factors:

1. Different definitions of OACI label operational lifetimes,

2. Non-uniform quality control on label assembly by different assemblers, and

3. Different characterizations of the label population and the railroad environment.

In cases where the IST label does not satisfy the operational needs, label alternatives can be suggested. These alternatives could be applicable for different car and service types as well as for given operational life expectancies. The main concepts developed in the FRA's OACI program and used in one of the new label designs are:

1. Use of materials practically not affected by solar radiation over a 15-year period;

2. Substituting automatic means of label construction for the present manual assembly methods;

3. Physically separating the addition of the label color from the retroreflective material (that is, the colors are added as a separate layer during construction);

4. Introduction of rugged modules which can be easily handled and inserted in the field; and

5. An outer layer of 5 mil thick abherent film (e.g., Teflon FEP to protect the module surface.

An alternative module design shown in figure 17 consists of a Plexiglass back plate with a cavity to receive the silver Scotchlite High Intensity Grade Module covered by a front plate which is made of the same material used for the blue and red filters in the scanners. The front and back plates are sealed to completely isolate the Scotchlite from the elements. A 5 mil thick Teflon FEP layer is applied for surface protection. The modules are dropped into cavities made in an aluminum back plate. Prototype modules of this design have been successfully made and photometrically tested. The retroreflectance is equal to the IST modules and the color match of the red and blue modules with the spectral transmittance of the respective scanner channels is perfect and does not change with time.

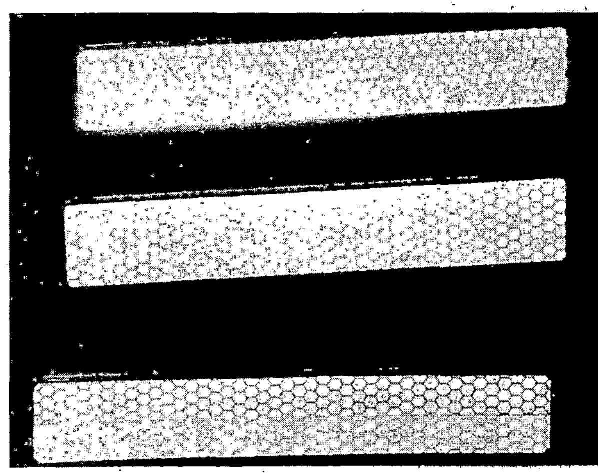


FIGURE 17. ALTERNATIVE MODULE DESIGN

PART III: <u>OACI SYSTEM ENHANCEMENTS</u>

The OACI scanner system improvements offer a number of new capabilities which can be used by the railroads in their OACI maintenance program and their management information system. These new capabilities can be summarized as follows:

1. Confidence levels on all digits of the car identification can be automatically correlated with "advanced consist" car identifications lists; even in cases where only a few digits are recognizable from a very degraded label.

2. The scanner system can provide train speed and direction data previously obtained from the wheel sensors. The speeds can be derived from the number of label locator outputs and the known synchronous speed of the scanner rotating mirror. Train direction can be obtained from the asymmetrical structure of the label "start" and "stop" modules.

3. A communications microprocessor can be installed in the scanner to permit multiple scanner polling by the management information system computer. The polling will enable a serial connection of large numbers of scanners systems at a reduced cost for communication lines.

4. Thorough system error and status checks of the scanner system and labels can be performed for maintenance purposes. This includes intercomparisons of scanner readabilities, the early identification of labels needing repair or washing, and more elaborate through-put checks on each scanner's operating condition.

The cost-effectiveness of these enhanced capabilities should be evaluated in the context of each individual railroad's operations. Quantitative information for this evaluation can be obtained by including the capabilities of the OACI Alternatives Evaluation Model in the Classification Yard Simulation model.[5]

SCANNER SYSTEM OPTIONS	READABILITY[1]	NEW SYSTEM[2] CAPITAL COSTS		FIELD[3] RETROFIT COSTS		YEARLY MAINTENANCE COSTS[4]	
		ONE UNIT	500 UNITS	ONE UNIT	500 UNITS	ONE UNIT	500 UNITS
1. FULLY MODIFIED SCANNER	94-97% [5]	$27K	$14M	$17K	$9M	$3.4K	$1.7M
2. PARTIALLY MODIFIED SCANNER	92-95%	$49K	$25M	$4.5K	$2.3M	$5.1K	$2.6M
3. MANUFACTURERS' LATEST MODIFICATIONS	88-91%	$47K	$25M	-	-	$5.1K	$2.6M
4. TYPICAL EXISTING SYSTEM	78-86%	$47K	$25M	-	-	$5.1K	$2.6M

NOTES

1. LOWER LIMITS INCLUDE WHEEL SENSOR AND MAINTENANCE PROBLEMS (ESTIMATED LOSS 3%); UPPER LIMIT IS FOR SCANNER SYSTEM ALONE
2. ONE-HUNDRED LOT BUYS
3. COSTS TO UPGRADE EXISTING SCANNERS IN THE FIELD
4. COSTS INCLUDE 40% MONTHLY WHEEL SENSOR MAINTENANCE AND 60% SCANNER MAINTENANCE
5. PERCENT READABILITIES EXCLUDE CARS WITHOUT LABELS AND MISAPPLIED LABELS

TABLE 1. OACI SCANNER READABILITY AND COSTS

CONCLUSIONS

This paper clearly indicates that significant improvements in OACI scanner system performance and costs are achievable. The extent of these improvements are summarized in Table 1 where the Final and First Stage of modifications are respectively listed as items 1 and 2 of the scanner system options. The table indicates the range of readability accuracies for each option and identifies one-unit and 500-unit initial purchase costs, field retrofit costs, and yearly maintenance costs. The final modifications will increase the scanner readability to 94%-97% enabling a reduction in initial purchase costs from the range of $40,000 to $54,000 down to $27,000. The scanner system yearly maintenance costs would also be reduced by $1,700 per year, yielding a $3,400 yearly cost after the wheel sensor maintenance is included. Alternatively, a scanner system performance of 92% to 95% is achievable through a $4,500 field retrofit cost for the first stage of modifications.

In the OACI label area, a number of significant conclusions are made. These are:

1. The main non-reversible cause of OACI module degradation over the years is the change in physical properties of the upper layer of Scotchlite due to solar radiation.

2. A reversible cause of OACI standard module degradation is loss of gloss of the module surface due to the natural environment or to abrasion. This loss of gloss can be corrected by maintenance or module redesign.

3. Data from two sources (the label material manufacturer and the Canadian National Railways) leads to a 17-year estimate for the label's operational life.

4. Teflon overlay on the modules has been completely effective in protecting the Scotchlite base material and extending its life.

5. Labels of new design using materials not affected by solar radiation and optically matched to the scanner can be developed.

In closing, the railroad industry has a large prior investment in the OACI system. It is believed that the improvement effort and the enhancements reported will assist the industry in making its decision on any future investment in automatic car identification.

REFERENCES

1. Ingrao, H. C., "Optical Automatic Car Identification (OACI), Field Test Program," Report No. FRA/ORD-76/249, U.S. Department of Transportation, Transportation Systems Center, May 1976.

2. "Optical Automatic Car Identification: An Evaluation of the Current System", Gellman Research Associates, Jenkintown, PA, June 1977.

3. Ingrao, H. C., "Optical Automatic Car Identification (OACI), Optical Properties of Labels," Report No. FRA/ORD-77/38.III, June 1977.

4. Wiseman, R., "Optical Automatic Car Identification (OACI), Scanner System Performance and Cost Improvements", Report No. FRA/ORD-77/38.I, June 1977.

5. Summary Report, "A Study to Analyze and Define Alternative Approaches to Automatic Car Identification", ARINC Research Corp., Annapolis, MD, June 1977.

6. Buss, D. D., Bailey, W. H., and Collins, D. R., "Analysis and Applications of Analog CCD Circuits", Proc. 1973 Int. Symp. Circuit Theory, Toronto, Ontario, Canada, April 1973, pp. 3-7.

7. Torrieri, D. "Adaptive Thresholding Systems", IEEE Transactions on Aerospace and Electronic Systems, Vol. AES-13, No. 3, pp. 273-279, May 1977.

8. Ingrao, H. C., "Optical Automatic Car Identification (OACI) Optical Properties of Labels", Final Report No. FRA/ORD-77/38.III, Cambridge Systems Corporation, Cambridge, MA. In preparation (1977).

9. Bennett, I., "Monthly Maps of Mean Daily Insolation for the United States", Solar Energy, Vol 9, No. 3 (1965).

10. Friesen, W., "Long Life OACI Label Tests", Canadian National Railways Report, Signals, Montreal, Canada, April 15, 1977.

ACKNOWLEDGEMENTS

The authors wish to express their sincere appreciation to the many individuals and their organizations that have contributed to the OACI Improvement Effort.

From the Transportation Systems Center, we would like to acknowledge the engineering contribution of L. Long, M. Yaffee, and A. Lavery.

Thanks are gratefully given to all of the members of the ACI Technical Committee especially the Chairman, D. Spanton, for their guidance and review throughout the improvement effort. Also, appreciation is expressed to the Research and Test Department of the Association of American Railroads, in particular C. Taylor, for their careful study of ACI alternatives and especially their economic methodology effort which led to a Classification Yard Simulation Model.

TRUCK PERFORMANCE - FRICTION SNUBBER FORCE MEASUREMENT SYSTEM

BY

G. FAY

K. CAPPEL

With all its truck and carbody instrumentation, Phase I of the Truck Design Optimization Project found that adequate transducer systems for the measurement of forces in the friction snubber were non-existent. This paper documents the design, testing and potential utilization of such a system, fabricated and tested by Wyle Laboratories.

INTRODUCTION

Within the last decade, an increasing amount of research has been directed toward improving the performance of the three-piece freight car truck. Truck manufacturers have designed and built several new truck configurations as well as added special purpose components aimed at improving such performance parameters as ride quality, lateral stability, and curve negotiation. There has been a great deal of testing by both industry and government for the evaluation of the comparative advantages afforded by modified or new trucks under a range of operating conditions. Of course, testing was conducted on the standard three-piece friction snubbed truck to establish quantitative performance characteristics as a base for the evaluation of new or modified designs.

While the conclusions drawn from many of the test results were very often contradictory, the methods of truck performance evaluation have steadily grown more rational. Much of the technological advance can be attributed to the parallel use of mathematical simulation and full-scale testing. The mathematical models used in simulations are necessarily idealized as they are linearized to reduce computing time, and because many of the nonlinear parameters in truck dynamics are not quantified.

In 1974, the Federal Railroad Administration awarded a contract to the Southern Pacific Transportation Company for the performance of Phase I of the Truck Design Optimization Project (TDOP) which had the objective of quantitatively characterizing the performance of the general purpose freightcar truck. The Contractor was primarily concerned with the evaluation of the two most commonly used three-piece trucks: the American Steel Foundries (ASF) "Ride Control" truck, and the Barber S-2 truck; the first incorporates constant snubbing friction, and the second, load-dependent snubbing friction. TDOP Phase I instrumented new 70-ton (63,502 kg) and 100-ton (90,718 kg) ASF and Barber trucks to measure accelerations, normal contact forces at the roller bearing adapters and relative linear and angular displacements between side frames and bolster. Relative rotation between truck and carbody bolsters was also measured.

It was soon recognized that instrumentation was not available to measure the forces transmitted through the spring loaded friction shoes or wedges between the side frames and bolster. In theory, it might have been possible

Grace R. Fay has been Research Manager, Equipment Performance Analysis, Office of Freight Systems (R&D), since 1975. She received her BSME, (1968) Howard University; Graduate Study, ME and Transportation Engineering, George Washington University.

Klaus L. Cappel is the Chief Design Engineer, Scientific Services and Systems Group for Wyle Laboratories. He received his A.B., (1940) University of California; Undergraduate Study at the University of London and Graduate Study at Stevens Institute of Technology and University of Delaware in Hydrodynamics.

to derive the friction forces by comparing measured accelerations and displacements with the calculated dynamics of a system without energy dissipation. It was also apparent that this approach would not only be cumbersome and expensive in computer time but would not lead to an accurate determination of the friction forces. In November 1974, a preliminary design concept of a transducer system to measure the forces between side frames and bolster was submitted by Wyle Laboratories for review.

ENGINEERING CONSIDERATIONS

Although the bolster-side frame connection is structurally and mechanically simple, it performs a multiplicity of functions:

- o Vertical support of the carbody weight through the spring nest.
- o Centering of the bolster between the side frames through lateral spring forces.
- o Partial isolation of the carbody from shock and vibration through the springs in both vertical and lateral directions.
- o Dissipation of energy in both vertical and lateral directions, through the friction wedges and wear plates.
- o Equalization of wheel loads on uneven track, by permitting relative pitch and roll displacement.
- o Transmission of longitudinal braking forces, also through the friction shoes
- o Limitation of excessive relative displacements through the bolster gibs.

Examining these functions it is important to note that the friction shoes are involved in four of the eight interactions between side frame and bolster. The non-rigid connection between side frame and bolster permits relative motion in six degrees of freedom and consequently transmits six generalized forces--three forces and three moments --between the friction shoes and wear plates.

The tapered surface of the friction shoes presses against the mating surface of the bolster, and the wedge action results in a normal force between the vertical shoe surface and the wear plate. This is generally referred to as the column pressure. The two shoes at each bolster end load each other. Relative vertical or lateral displacement of the bolster gives rise to corresponding friction forces. Braking forces are transmitted by increasing friction forces on the rear shoe unless the column load is exceeded which results in gib contact. Relative bolster roll, in which plan contact between friction shoe and wear plate is maintained, applies a roll friction moment on the side frame column.

The remaining two rotations, relative pitch and yaw, give rise to more complex interactions as both the slanting and vertical surfaces can no longer remain in plane contact with the bolster and wear plate, respectively. The resulting edge-to-surface contact is an important cause of wear, both in the bolster pocket and at the upper and lower edges of the vertical shoe surface. The high restoring moment in yaw, also in pitch, occurs when the side frame is yawed with respect to the bolster and plane contact at either the sloping or vertical surfaces of the friction shoes changes to contact at diametrically opposite wear plates or contact points in the bolster pockets. Rotation of this diagonal into the center plane of the side frame thus requires that the distance between opposing friction shoes be shortened. The friction shoes thus move closer together, and in so doing slide inward along the slanted mating surfaces with the bolster. This causes additional compression on the snubber spring, and since the vertical load has not changed, there is a slight rise in the bolster with respect to the side frame in the case of load-dependent snubbing. The potential energy of elastic deformation is merely redistributed between the suspension and snubber springs. Thus, yaw rotation increases the potential energy of the system by raising the weight carried by the bolster. For load independent snubbing, where the snubber spring is based on the bolster, the entire potential energy is stored in the snubber spring. In either case, the reactions due to skew are applied at diagonally opposite edges of the side frame column, thus providing a yaw restoring couple.

All of the load paths discussed above were considered in the design of the Friction Snubber Force Measurement System (FSFMS).

MANUFACTURING TECHNOLOGY AND CONCEPTUAL DESIGN

The technology applied to the FSFMS was available in the field of towing tank testing where "force blocks" are utilized to measure the forces and moments applied between a moving carriage and a towed ship model. A force block is a hollow, roughly cubical block of alloy steel mounted at opposite sides to the objects between which forces are to be measured. The other four sides are machined so as to leave short cantilever beams instrumented with strain gages to measure bending stresses resulting from shears applied at the mounting surfaces. Additional strain gages are provided for nulling stresses due to normal forces.

In measuring several degrees of freedom, the load path must pass through each transducer in turn. In other words, the transducers must be in series or cascaded. Cross coupling of signals is minimized by making the blocks very stiff in both shear and compression along all axes not used for measurement. In the case of the FSFMS, an additional requirement was symmetry of the load path in order to preclude unsymmetrical deflections that would alter the contact geometry between the friction shoe and the wear plate. In addition, all force blocks had to fit within the envelope of the side frame.

The original design concept of the FSFMS is shown in figure 1. The wear plate is welded to an adapter which in turn is bolted to a single vertical force block. The opposite face of the force block is bolted to an adapter to which a pair of lateral force blocks are mounted. Two normal force blocks are attached above and below, and are in turn bolted to an adapter rigidly mounted on the side frame. The adapter between the vertical and lateral force block divides the load path into two symmetrical sections which ensure that any tendency of the wear plate to tilt under unsymmetrical loading is minimized.

The five force blocks are capable of measuring two of the three moments applied by the friction shoes: The pitch moment is found from the differential loading of the upper and lower normal transducers, and the roll friction moment from the differential loading of the upper and lower lateral transducers. Space limitations precluded a transducer configuration capable of measuring a yaw moment; however, the increased column load due to yaw can be measured. An exploded view of the transducer as built is shown in figure 2. Space limitations in the side frame also required that the normal and lateral force transducers be combined into a single unit. Each individual transducer is compensated against cross-coupling between the vertical and normal transducers. This is due to the fact that the plane of the mounting adapter to which the lateral and normal force transducers are attached is offset from the plane of the wear plate adapter carried by the vertical transducer. A vertical friction force thus produces a moment which is resisted by equal and opposite normal forces. The forces making up this couple must be distinguished from unequal normal forces due to a vertical offset of the center of friction shoe pressure from the center of the wear plate which occurs with bolster displacement. The correction factors were established for each transducer assembly by calibration and must be used in the reduction of data collected in road tests.

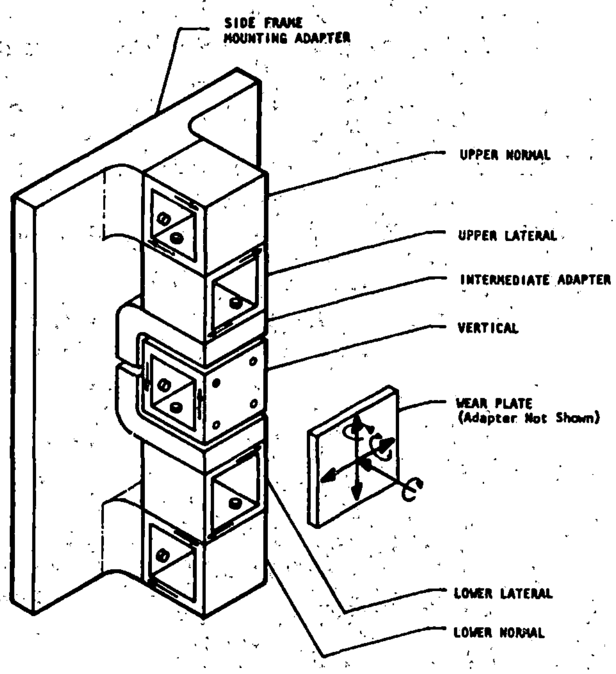


FIGURE 1. BASIC CONCEPT OF FORCE TRANSDUCER

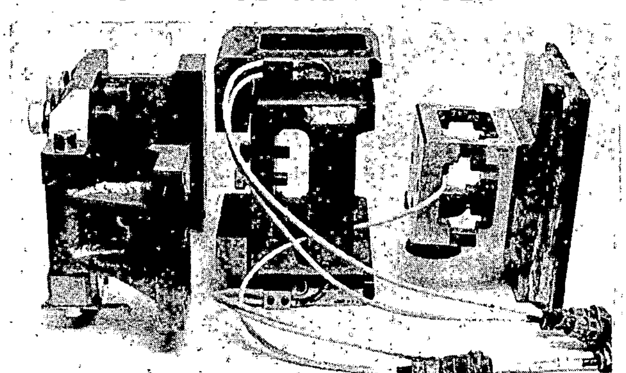


FIGURE 2. EXPLODED VIEW OF TRANSDUCER ASSEMBLY

MODIFICATION OF SIDE FRAMES

To be capable of running across the country under cars in revenue service, the modified, instrumented trucks had to be able to withstand normal shock loads, resulting in stresses below the fatigue limit as specified in AAR M-203-65. This required the side frame to be tested under a lateral load of 35,000 lb (15,876 kg) without exceeding the deflections listed in the standard. This led to a modification of the original design.

An opening in the center of each column of the side frames was required to accommodate the wear plate adapter which transmitted friction shoe forces to the transducer assembly mounted behind it. Originally, the entire column was to have been removed and replaced by two heavy welded steel bars. The bars would serve the double purpose of providing both reinforcement of the open center and a mounting surface. Removal of the entire column might have caused more deformation in the side frame than could have been corrected. Therefore, only the column web behind the location of the wear plate was removed after the reinforcing bars had been welded. The modified side frames were then stress relieved with the center opening of each side frame stabilized by diagonal braces to preclude distortion. A modified side frame of the Barber S-2 truck is shown in figure 3.

The critical dimension maintained in the modification of each side frame was the distance between wear plates: 17-3/4 in. (0.45m) in the ASF truck and 17 in. (0.43m) in the Barber S-2 truck. This spacing determines the column load with the given bolster and friction shoe geometry, and the spring characteristics. With the dimensions of the transducer stack between the wear plate and the mounting adapter flanges given, the offset between the face of the wear plate and the back surface of the reinforcing bars determined the spacing between the wear plates. Care was taken in the fabrication to prevent distortion.

A preliminary stress analysis indicated that the modified column when treated as a rigid frame with infinitely stiff girders had ample strength to resist a concentrated transverse force of 17,500 lb (7,938 kg) applied at the center of one of the reinforcing bars. This represented one-half of the specified 35,000 lb (15,876 kg). The ledges surrounding the wear plate where concentrated lateral force would be applied by one or the other bolster gibs were the weakest point in the modified column. Removal of the column web deprived the lip of a backup and caused the gib forces to be resisted by a portion of the lip in cantilever bending. A bar with tapered edges was welded to the inside of the cut to provide reinforcement, and the rear edges of the wear plate adapter were tapered to provide clearance in the reduced opening.

FIGURE 3. MODIFIED BARBER S-2 SIDE FRAME

TRANSDUCER CONSTRUCTION AND ASSEMBLY

The transducer components were machined from 17-4 pH precipitation hardening steel with a yield strength of 140,000 psi (965,266 x 10^3 N/m^2). Simultaneous application of a normal load of 6000 lb (2722 kg) and vertical and horizontal friction forces of 3000 lb (1361 kg) each would produce stresses of only about 20,000 psi (137,895 x 10^3 N/m^2). Therefore, an ample margin of safety is provided for unforeseen overloads.

The 35,000 lb (15,876 kg) lateral load does not pass through the force blocks and therefore posed no problem for the transducer design. A portion of this load, however, must be resisted by the U-shaped mounting adapter which is flange mounted on the column reinforcing bars and forms a structural tie across the column opening. The stiffness in the lateral load path through the mounting adapter was therefore lowered. As shown in figure 4, the thickness of one web between the transducer mounting plate and its flange was reduced so that it would act as a flexure. The opposite bracket is connected to the transducer mounting plate by a stainless steel pin assembled in self-lubricating bushings. This bracket transmits essentially all of the lateral friction forces from the wear plate to one column reinforcing bar. Under a lateral impact force high enough to decrease the distance between the reinforcing bars, the flexure will minimize the portion of the load transmitted through the mounting adapter.

The two-piece mounting adapter greatly eased the assembling of the transducers in the confined space of the side frames. Individual components were introduced one at a time and tightened with a torque wrench in a threadlocking compound. Some interferences between transducers and fillets in the side frame castings were found behind the column, above and below the cutout. Some of these interferences were due to variations between castings, and it was necessary to bevel the edges of the lateral and normal force transducers as well as the rear edges of the cutout. Modifications were also required in the same area of the lower two bolts of the mounting adapter in the case of the Barber truck to provide space for assembly. Figure 5 illustrates the completely assembled transducer in the ASF truck.

CALIBRATION TESTING

The tests conducted on the FSFMS were intended to demonstrate performance of the transducers in the truck under some simulated operating conditions without reproducing all aspects of the rail environment which would have required more complex and costly test equipment. Only vertical and lateral movements of the bolster were generated during testing, the latter displacement considered essential to prevent the formation of vertical grooves in the friction shoes. To minimize the hydraulic power required to move the bolster, only two springs were installed in each side frame. More springs should have been used to prevent rocking of the side frames about their roll axes; however, this motion demonstrated the capability of the transducer assembly to identify friction torques due to roll.

Figure 6 depicts the Barber S2 test setup which was, of course, identical for the ASF Ride Control Truck. An existing test frame was modified by adding four pedestals to support the pedestals of the side frames and to restrain them laterally. A beam simulating the carbody bolster was nested by a center plate in the truck centerbowl. The beam was raised and lowered by a pair of double-ended double-acting hydraulic actuators controlled by electrohydraulic servo valves. Linear Differential Voltage Transformers (LVDT's) mounted on the actuators provided position feedback. A third horizontal hydraulic actuator mounted on a bracket atop the test frame provided lateral motion of the simulated carbody bolster. A central frame guided the bolster beam in a vertical plane through grease lubricated rubbing plates. Vertical and lateral relative displacements between the bolster and each side frame were measured by LVDT's.

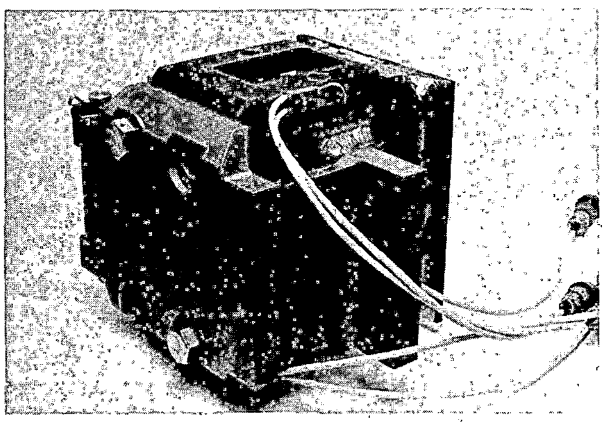


FIGURE 4. ASSEMBLY REAR VIEW

Outputs of all 20 force transducers, the four bolster-side frame LVDT's, and the three actuator LVDT's were recorded on four oscillographs. All calibration factors for the force transducers were established with the friction shoes out of contact with the wear plates. In the ASF truck, the pins that lock the shoes against the springs were left in place until after calibration was completed. In the case of the Barber truck, the bolster was lifted by crane to unload the friction shoe springs to a point where the shoes could be moved manually away from the wear plates.

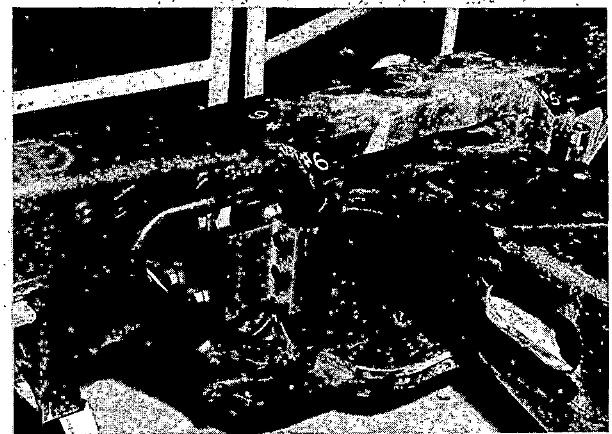


FIGURE 5. TRANSDUCERS IN ASF TRUCK

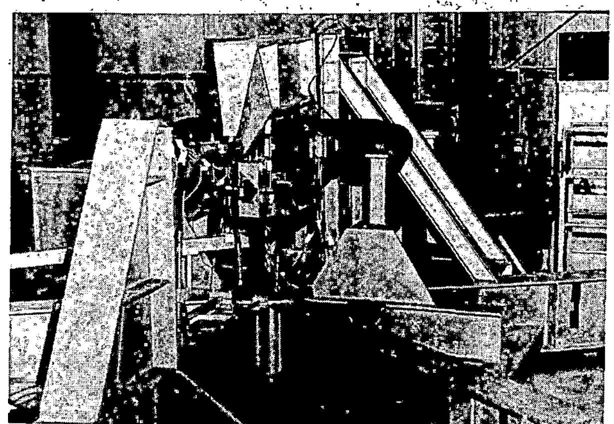


FIGURE 6. BARBER S2 TRUCK ON TEST STAND

For the first test series, the bolster was lowered until the springs were compressed to about half their travel. The bolster was then oscillated about this position through the amplitude of $\pm$ 3/4-inch ($\pm$ 0.19m) at a frequency of 0.1 Hz. Simultaneously, the bolster was displaced laterally through an amplitude of $\pm$1/4 inch ($\pm$ .006m) at a frequency of 1.0 Hz.

Next, a sine sweep was performed, with the frequency gradually increasing and the amplitude decreasing. During this test considerable wear was taking place at the friction shoe wear plate interface indicated by black powdery debris. The normal forces being measured were increasing beyond estimated levels, and gouging of the wear plates was noted. The sine sweep was terminated at 8.0 Hz as it was recognized that considerable time was required for the friction shoes to wear to service levels.

The trucks were then disassembled and the vertical surface of the friction shoes lightly ground to remove larger asperities so that the forces measured during the tests would be more representative of those occurring in service after wear-in. The first test described was repeated and the measured normal forces were in the expected range for both trucks. The sine sweep test was eliminated to prevent the localized wear at small amplitudes.

Static friction in the snubber system was checked by very slowly moving the bolster downward one inch from the centered position with the hydraulic actuators under manual control. Also, before disassembling the trucks for friction shoe grinding, the truck was forced out of tram and the increase in normal forces on the column measured.

TEST RESULTS

To reiterate, the main objective of these tests was to establish proper operation of the transducer system installed in the trucks. The objective was achieved. It was not intended to subject the truck to a full range of inputs such as would be observed on the track. The force distribution at the column is likely to change as the friction shoes wear and this phenomena should be monitored during future road tests. The following discussions are not intended to imply endorsement or critique of either truck design.

The highest recorded friction forces in the Barber S2 were approximately 1200 lb (544 kg) in both the vertical and lateral directions. The highest measured force at the lower normal transducer was about 7200 lb (3266 kg). Since the bolster was descending this must be corrected by subtracting the force due to the vertical friction moment, leaving a true lower normal force of about 6900 lb (3130 kg). The upper normal force is practically zero at this point or 275 lb (125 kg) with the correction factor. The total normal force is therefore about 7200 lb (3266 kg) and the friction coefficient is 0.167. The lateral coefficient appears to be about twice as high, but this may be due to a slight cocking of the friction shoes during the lateral bolster motion. The lateral friction forces were not as repeatable as the vertical and normal forces due more than likely to the rocking of the side frame which was supported only on two springs. Audible chatter was noticeable on the downstroke apparently due to vertical stickslip.

An attempt was made to measure the effect of forcing the truck out of tram by means of set screws at two adjacent pedestals. In this case, the normal column load is redistributed, increasing at the top and decreasing at the bottom, and the friction shoe moves downward, as expected, since the bolster is restrained by the actuators from moving upward. Because of the unsymmetrical distortion of the truck, there is some lateral sliding between bolster and side frames creating a friction force of about 200 lb (91 kg) each.

In testing the ASF Ride Control Truck, the highest vertical friction forces were 4500 lb (2041 kg) down, and 2500 lb (1134 kg) up. On the downstroke, the measured upper and lower normal forces were 7500 lb (3402 kg) and 5000 lb (2268 kg), respectively. Therefore,the corrected total normal force was 9910 lb (4495 kg) with a friction coefficient of 0.45. The lateral friction forces were about 4000 lb (1814 kg) and 2500 lb (1134 kg) on the downstroke, so the apparent lateral friction coefficient was 0.69. Vertical friction and normal forces were generally lower during the upstroke, but lateral friction forces were about the same in both directions. Therefore, the effective friction coefficient varied somewhat indicating some change in geometry which again may be due to the rocking of the side frames. There also appeared to be some rocking of the friction shoes indicated by a sharp rise in the lower normal and vertical friction forces as the direction of vertical motion reversed at the beginning of the downstroke. There was also heavy chatter implying additional energy dissipation and the distribution of the normal load was highly unsymmetrical with respect to the center of the wear plate.

It must be emphasized that the data discussed in the foregoing paragraphs are not necessarily typical of a friction snubber assembly worn in under actual operating conditions. These data are presented solely to illustrate the kind of information obtainable from the Friction Snubber Force Measurement System.

POTENTIAL UTILIZATION

Since the calibration testing of the FSFMS was not completed until March of 1977, utilization of the system in Phase I of the TDOP, for which it was designed, was not possible. To reiterate, the transducer system has been installed on two trucks commonly used in freight service in the United States. The most obvious difference between the two with respect to the snubbing force is the dependence on or independence of the load on the truck. A second difference relates to the change in snubbing friction as the truck parallelograms. The warp stiffness, and thus the friction force, is necessarily affected by the bearing width of the friction wedge which differs substantially in the two trucks. A third factor affecting snubbing friction is the frequency content and the vibrations applied to the side frame-bolster connection relating to the phenomenon of "breakout" friction.

All of the above suggest strong nonlinearities due to snubbing friction in the truck suspension system, the modeling of which is difficult and the effects of which on truck performance have not yet been quantified. Complete characterization of the general purpose freight car truck must involve the evaluation of these forces on both tangent and curved track, in both new and worn conditions. As part of Phase II, TDOP, recently awarded to Wyle Laboratories, both the Barber S2 and the ASF Ride Control trucks will be retested under various load conditions and on several track types with the transducer equipped side frames. In addition, quasi-static friction forces will be measured at frequencies for which inertial effects are negligible by using the calibration test setup and supporting the pedestals on load cells to measure the vertical and lateral reactions transmitted from the actuators, through the snubbing components, to the test frame. The actual forces on the columns can then be derived from the known applied vertical force and the wedge angle.

Acquisition of these data will allow more detailed specification of the test conditions to be met in testing for conformance to recommended performance guidelines (developed under TDOP); will aid in the validation of mathematical simulation of truck performance; will complete the characterization of the general purpose freight car trucks; and will provide a technical baseline for the evaluation of special purpose trucks to be accomplished in Phase II TDOP.

ACKNOWLEDGEMENTS

The authors wish to thank Mr. G. Tennikait, American Steel Foundries; Mr. R. Bullock, Standard Car Truck; Mr. B. Fowler, Project Engineer, Wyle Laboratories; and Modern Machine and Tool Company.

RESEARCH IN FREIGHT CAR DYNAMICS

BY

N. T. TSAI

E. H. LAW

N. K. COOPERRIDER

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 charac-

Dr. N. Thomas Tsai is Research Manager in the Analysis and Evaluation Division of the Office of Freight Systems (R&D) for the FRA. He received his BSME from Cheng Kung University, Taiwan, China (1960). Dr. Tsai received his MSME (1967) and Ph.D. in Mechanical Engineering (1969) from the University of Rochester.

Dr. E. Harry Law is Associate Professor of Mechanical Engineering at Clemson University. He received his Bachelor's degree in Aeronautical Engineering from Rensselaer Polytechnic Institute (1962), his Master's in Flight Mechanics from Princeton University (1965), and his Ph.D. in Applied Mechanics from the University Connecticut (1971).

Dr. N. K. Cooperrider is Professor of Mechanical Engineering at Arizona State University. He earned his B.S. degree (1963), his Master's (1964), and Ph.D. (1968) from Stanford University.

teristics, the use of a 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 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.

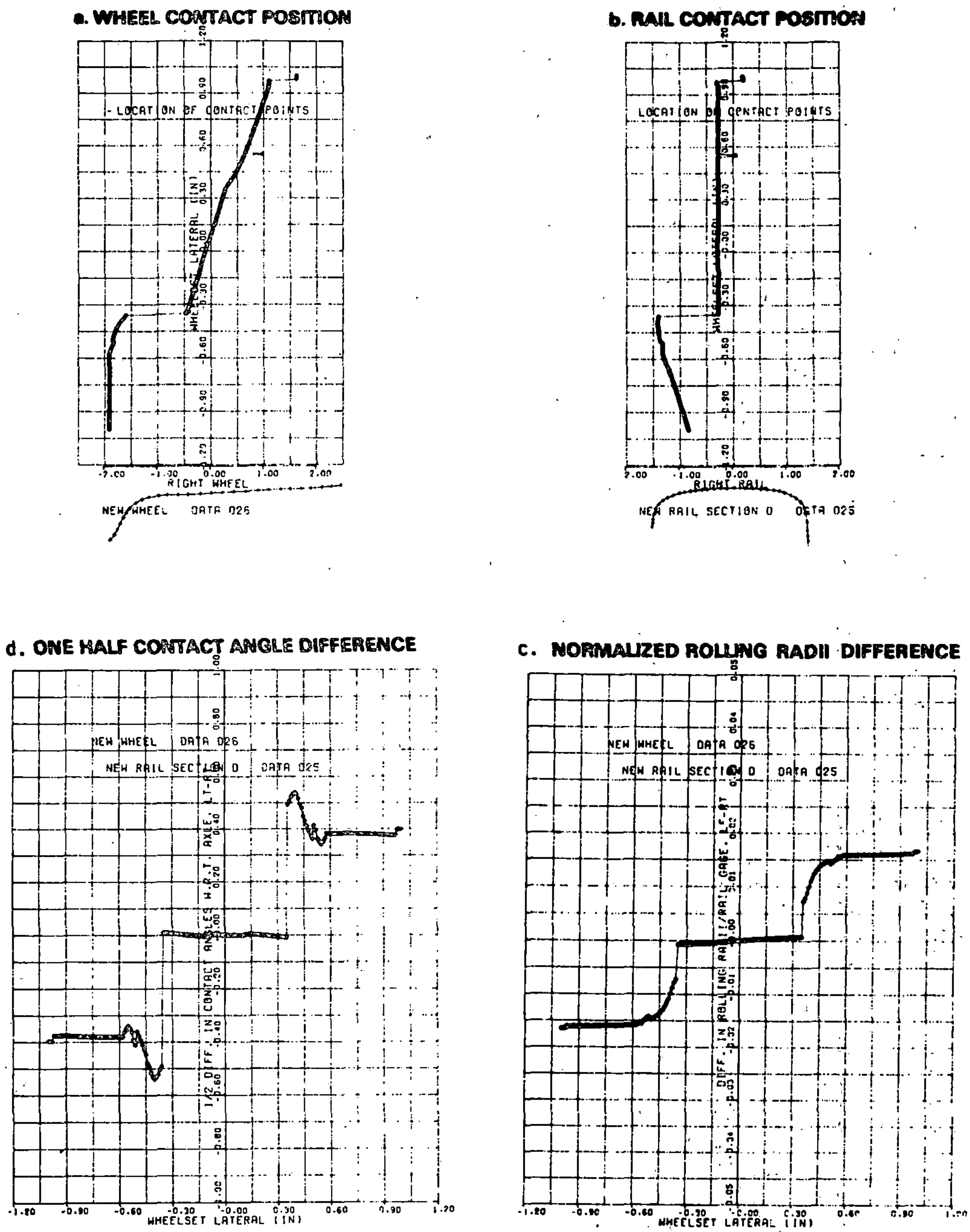
After analysis of the test results, we will compare the results from the various models and analyses with test results to evaluate the conditions and range of applicability of each model and analysis approach. For many purposes, analyses may be used that are inexpensive computationally as compared with 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



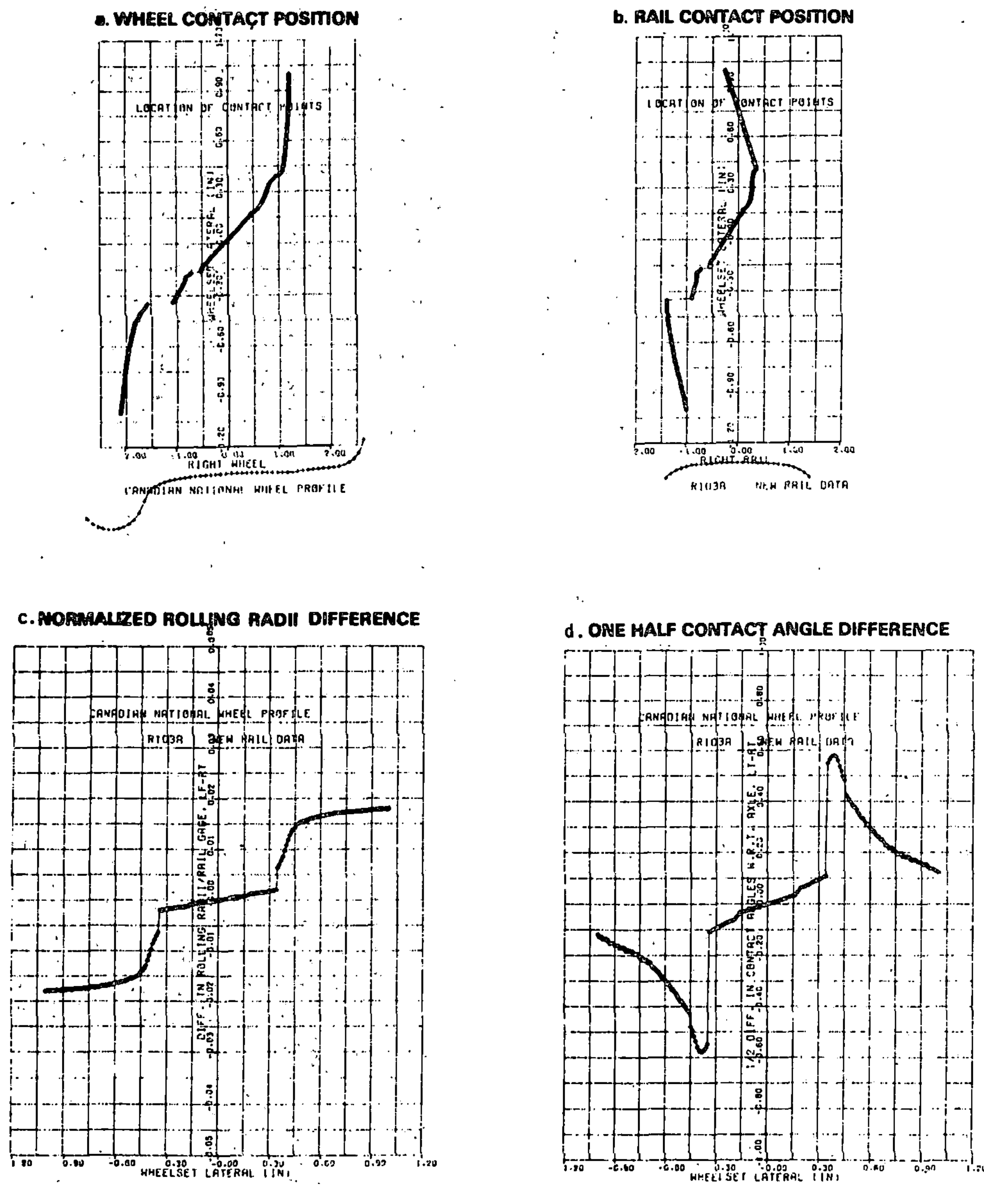
Wheel Gage 53 in

Rail Gage 56.5 in **Rail Cant** 0.025

FIGURE 1. WHEEL/RAIL CONTACT CHARACTERISTICS AND GEOMETRIC CONSTRAINT. FUNCTIONS FOR NEW WHEELS/NEW RAILS

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



Wheel Gage 53 in.
Rail Gage 56.5 in.

Rail Cant 0.025

FIGURE 2. WHEEL/RAIL CONTACT CHARACTERISTICS AND GEOMETRIC CONSTRAINT FUNCTIONS FOR CN "PROFILE A" WHEELS/NEW RAILS

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 movements. 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.[1]

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

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

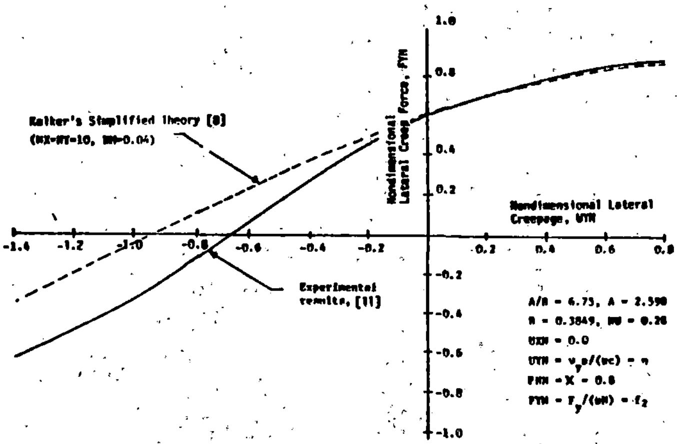


Figure 3. Comparison of Kalker's Simplified Theory with the Experimental Results of Figure 7, Reference [11].

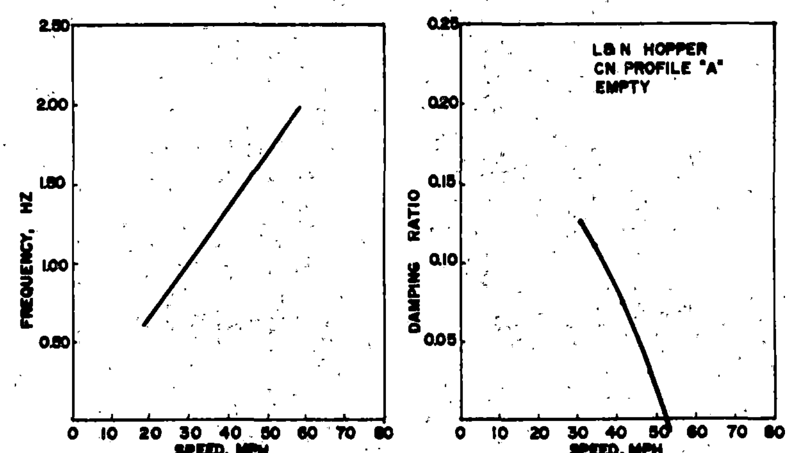


Figure 4. Theoretical System Damping Ratio and Frequency vs. Speed Relationships

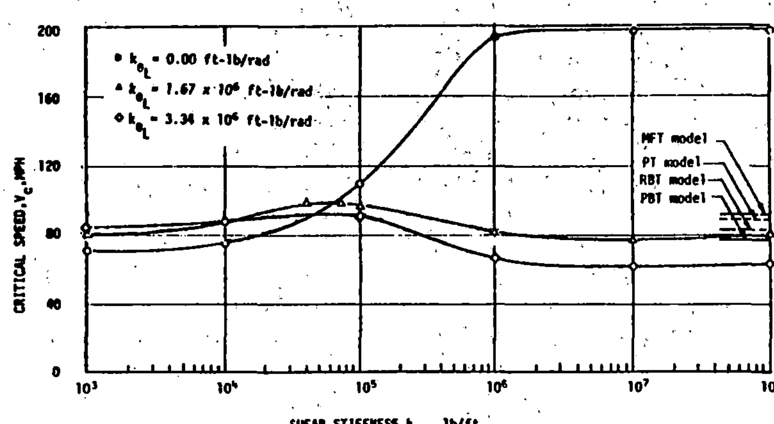


Figure 5. Effect of Shear Stiffness on Critical Speed of 11 DOF Model with Interconnected Wheelsets for Different Values of Bending Stiffness

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 of Hunting

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 over-damped 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. 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 judgment 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

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 wheel sets 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 wheel sets 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 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 five 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.

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.

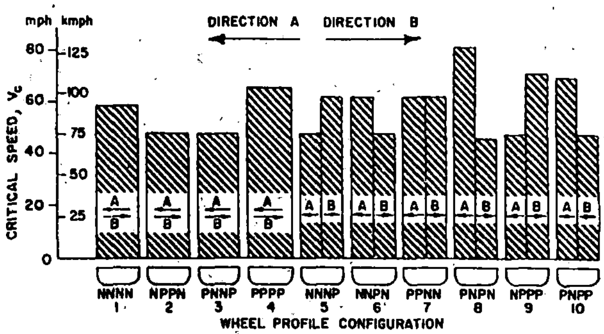


FIGURE 6. CRITICAL SPEEDS OF A NOMINAL EMPTY FREIGHT VEHICLE HAVING AXLES WITH DIFFERENT WHEEL PROFILES

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 judgment 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 [2] in (1, 2, 3)

[2]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.

represents the first efforts in this area and results obtained for a 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.

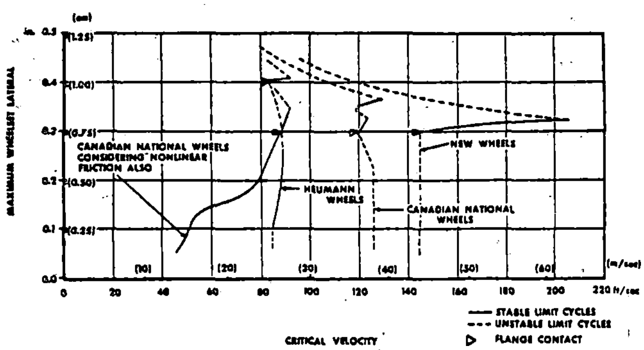


FIGURE 7. LIMIT CYCLE AMPLITUDE (WHEELSET LATERAL DISPLACEMENT) VS. SPEED FOR DOF EMPTY FREIGHT CAR

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 [3]. 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[4]. This is being performed by numerical integration on digital computers[5]. 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.

---

[3] 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.

[4] 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.

[5] We are using two models of digital computers: the UNIVAC 1110 at Arizona State University, and the IBM 370/3165-II at Clemson University.

This implies neglecting degrees of freedom such as bolster pitch and side-frame 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[6]. 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[7]. 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.

[6] 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.

[7] 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).

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.

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 nonlinear 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 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 one 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, quasilinear, 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 ground work 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 versus 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 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 and 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 mode 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.

The 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 (9, 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 the speed, and the stability margin available at lower speeds.

The rail freight car behavior at any speed can be loosely[8] 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

ζ_j - damping ratio for mode j

ω_j - frequency of mode j

ω_{nj} - undamped frequency of mode j

ϕ_j - phase angle for mode j

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

[8] This description is mathematically correct for linear systems, and can be used with caution for nonlinear systems such as the rail freight car.

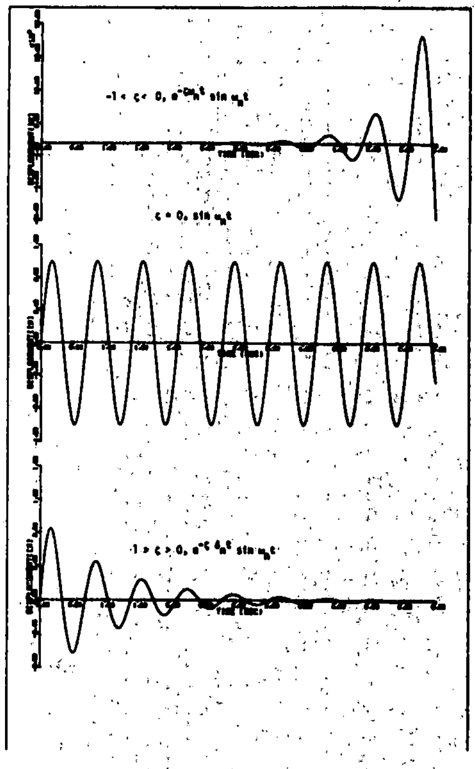


FIGURE 8. TYPICAL SYSTEM DYNAMIC RESPONSE

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 their difficulties to three factors: 1) the use of profiled wheels whose "effective"

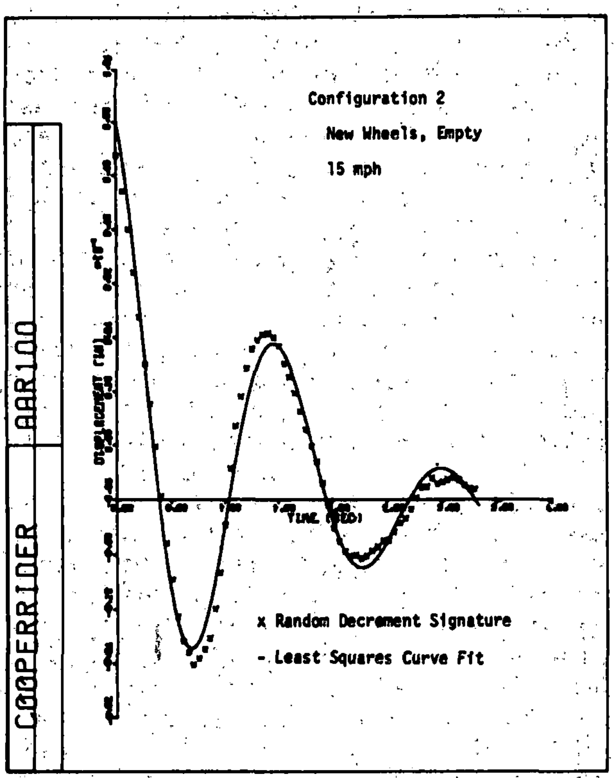


FIGURE 9. RANDOM DECREMENT SIGNATURE

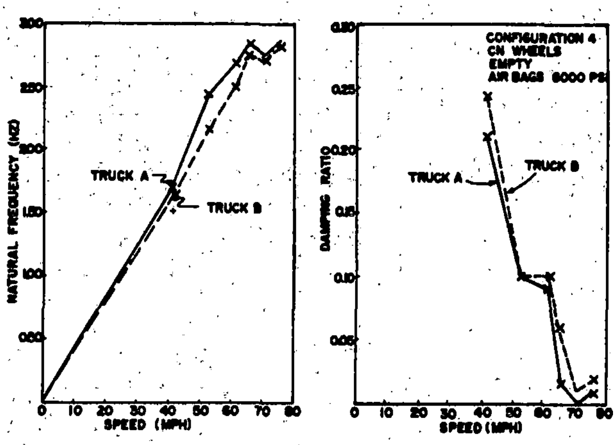


FIGURE 10. EXPERIMENTAL DAMPING AND FREQUENCY VS. SPEED RELATIONSHIPS

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

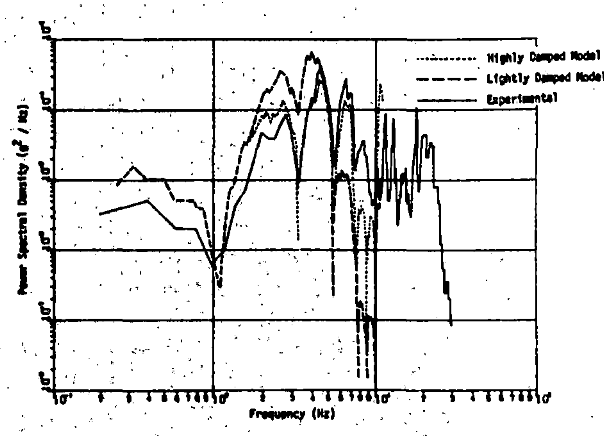


FIGURE 11. FREIGHT CAR BODY VERTICAL ACCELERATION SPECTRA FOR EMPTY VEHICLE AT 100 FT/SEC ON CWR [20]

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.

This 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 wheelsets later 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 vehicle. In addition, we were able only to measure rail head profiles at a limited number of stations and hence cannot 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 ensuring 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 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).

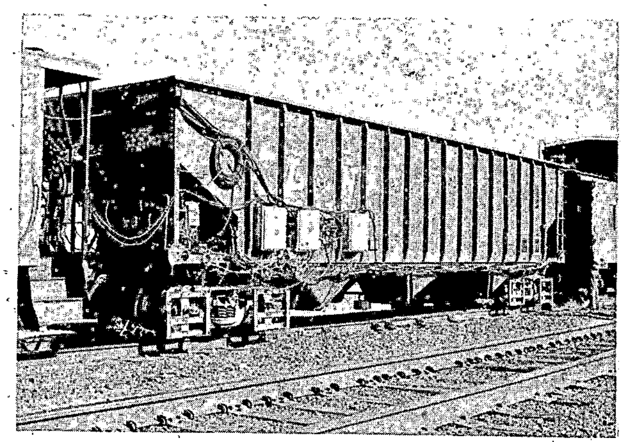


FIGURE 12. L&N HOPPER TEST CAR

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 (6x11 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"[9] 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 instrumentation 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 information.

The tests were conducted on both tangent and curved track sections on the Union Pacific Railroad. The AAR provided the test car, instrumentation

[9] 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.

Configuration	Wheels	Load	Side-Bearings	Truck Stiffner	Centerplate
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	6000 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 2. TEST VEHICLE CONFIGURATIONS

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, Calif., and Las Vegas, Nev. 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, Nev. 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 disasterously. 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 FRA Track Geometry Car 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.

Data Processing and Reduction

The test data collected during the field tests nearly filled 11 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.

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.

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

TABLE 3.

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.

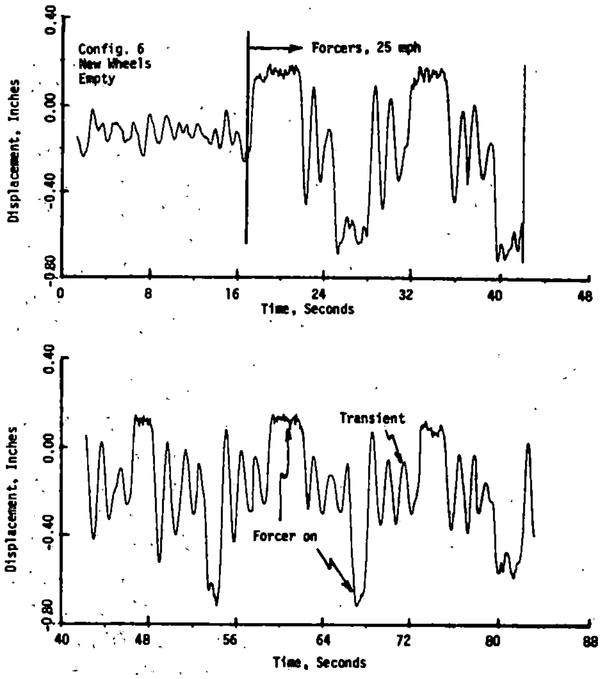


FIGURE 13. TRUCK LATERAL DISPLACEMENT USING HYDRAULIC FORCERS

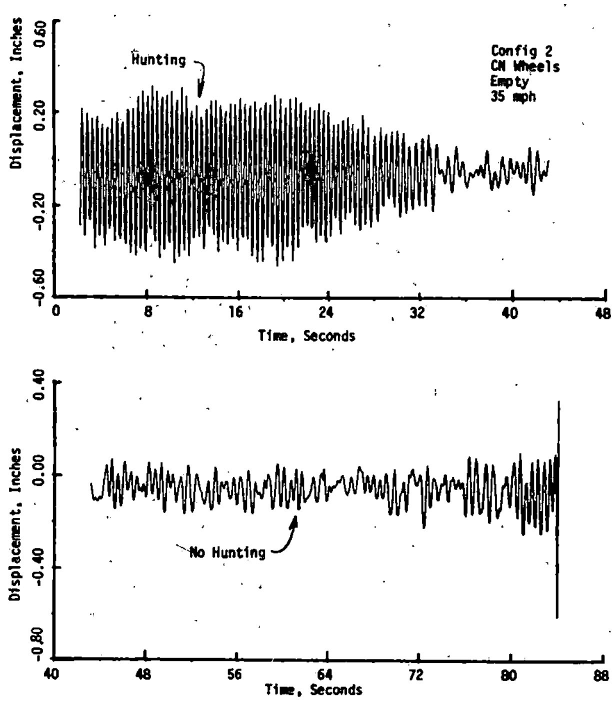


FIGURE 14. TRUCK LATERAL RESPONSE TO RANDOM RAIL IRREGULARITIES

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 tests 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.

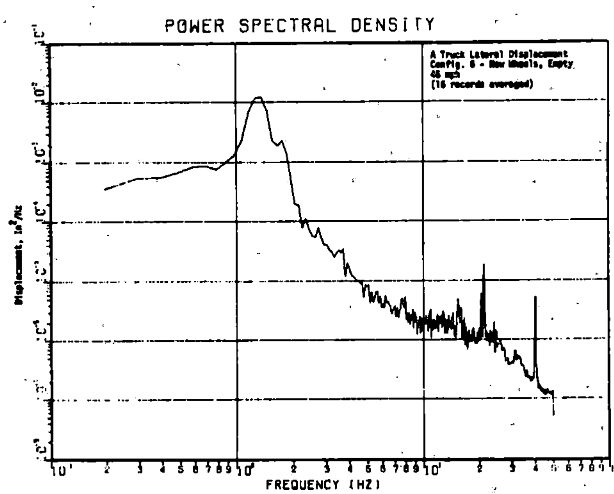


FIGURE 15. TYPICAL TRUCK LATERAL DISPLACEMENT PSD

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 I.

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 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.

DATAPLØT - prepares CALCØM 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.

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 designed 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 the behavior strongly. (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, particularly dry friction, will probably necessitate use of quasilinearization 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.

REFERENCES

1. Hedrick, J.K., Cooperrider, N.K. and E.H. Law, "The Application of Quasi-Linearization Techniques to Rail Vehicle Dynamic Analyses", Final Report for DOT-TSC-902, January, 1977.

2. Cooperrider, N., et. al., "The Application of Quasi-Linearization to the Prediction of Nonlinear Railway Vehicle Response", Proceedings of IUTAM Symposium on the Dynamics of Vehicles on Roads and on Railway Tracks, Delft, The Netherlands, August 1975, pp. 314-325.

3. Hull, R. and Cooperrider, N.K., "Influence on Nonlinear Wheel/Rail Contact Geometry on Stability of Rail Vehicles", ASME Journal of Engineering for Industry, Vol. 99, No. 1, February 1977, pp. 172-185.

4. Cooperrider, N.K., et. al., "Analytical and Experimental Determination of Nonlinear Wheel/Rail Geometric Constraints", Report No. FRA-OR & D-76-244 (PB-25290), December 1975.

5. Heller, R. and Cooperrider, N.K., "Users' Manual for Asymmetric Wheel/Rail Contact Characterization Program", Interim Report for DOT-OS-40018, to be published 1978.

6. Tuten, J.M., "Interactive Computation Procedure for Digitizing Wheel and Rail Profile Data", Informal Report for DOT-OS-40018, Department of Mechanical Engineering, Clemson University, June 4, 1976.

7. Hobbs, A.E.W., "A Survey of Creep", DYN/52, April 1967, British Railways Research Dept., Derby, England.

8. Kalker, J.J., "Simplified Theory of Rolling Contact", Deft Progress Report, Series C: Mechanical and Aeronautical Engineering and Shipbuilding, 1 (1973), pp. 1-10.

9. Kalker, J.J., "On the Rolling Contact Between Two Elastic Bodies in the Presence of Dry Friction", Ph.D. Thesis, Delft University of Technology, 1967.

10. Goree, J.G. and Law, E.H., "Users' Manual for Kalker's Simplified Nonlinear Creep Theory", Interim Report for DOT-OS-40018, to be published 1978.

11. Gilchrist, A.O. and Brickle, B.V., "A Re-Examination of the Proneness to Derailment of a Railway Wheelset", J. Mech. Engr. Sci., Vol. 18, No. 3, 1976, pp. 131-141.

12. Hadden, J.A. and Law, E.H., "Effects of Truck Design on Hunting Stability of Railway Vehicles", ASME Journal of Engineering for Industry, Vol. 99, No. 1, February 1977, pp. 162-171.

13. Hadden, J.A., "The Effects of Truck Design and Component Flexibility on the Lateral Stability of Railway Freight Vehicles", M.S. Thesis, Dept of Mechanical Engineering, Clemson University, 1976.

14. Tuten, J.M., Law, E.H. and Cooperrider, N.K., "Lateral Stability of Freight Cars with Axles Having Different Wheel Profiles and Asymmetric Loading", to be published, 1978.

15. Heller, R., Malstrom, C.W. and Law, E.H., "Hybrid Simulation of Rail Vehicle Lateral Dynamics", 1976, (accepted for publication in Simulation).

16. Heller, R. et. al., "Analog and Digital Computer Simulation of Coulomb Friction", Interim Report for DOT-OS-40018, to be Published 1978.

17. Newland, D.E., "Steering a Flexible Railway Truck on Curved Track," ASME Journal of Engineering for Industry, Vol. 91, Series B, No. 3, August 1969, pp. 908-918.

18. Boocock, D. "A Simplified Linear Analysis of the Steady-State Motion of Conventional and Cross-Braced Bogie Vehicles and Four-Wheelers on Curved Track", DYN 89, April 1968, British Railways Research Dept., Derby, England.

19. Cooperider, N.K. and Law, E.H., "Validation Plan for Freight Car Dynamic Analysis", Arizona State and Clemson Universities, June 15, 1976.

20. Fallon, W.J. Jr. "An Investigation of Railcar Model Validation," M.S. Thesis, Arizona State University, May 1977.

21. Fallon, W.J. Jr., Cooperrider, N.K. and Law, E.H., "An Investigation of Techniques for Validation of Railcar Dynamic Analyses," FRA Report, to be published, 1978.

22. Cole, Henry, A., Jr., "On Line Failure Measurement and Detection of Aerospace Structures by Random Decrement Signatures," NASA-TND-4503, 1968.

23. Weibe, D., "The Effects of Lateral Instability of High Center of Gravity Cars", ASME Trans., Vol. 90-B, No. 4, Nov. 1968, p. 462.

24. Anon., "A Comparison of Theoretical and Experimental Vehicle Behavior Using a 2 Axled Special Vehicle," Report No. 2, Question C116, Interaction Between Vehicle and Track, Utrecht, October 1972.

25. Fallon, W.J. Jr., "User's Manual, Random Data Analysis Programs," Arizona State Univ., Engineering Center Report No. ERC-R-76010, April 1, 1976.

26. Abbott, P., "Comparison of the Nonlinear Dynamic Characteristics of Barber S-2 and ASF Ride Control Freight Trucks," Martin-Marietta Corp., Report No. MCR-76-475, September 1976.

27. Orr, D.G., ed., "70 Ton Truck Component Data, Physical Restraints, Mechanical Properties, Damping Characteristics," Track-Train Dynamics Harmonic Roll Series, Vol. 2, 1974.

28. Cooperrider, N.K. and Law, E.H. "Program Plan for Field Test Validation of Lateral Freight Car Dynamic Analysis", Arizona State and Clemson Universities, July 5, 1974.

QUESTIONS SESSION I

Session Chairman -- Dr. Donald L. Spanton

Attendee: H. A. List, Railway Engineering Associates, Inc.
Attendee's Question: Is anything less than 100 percent identification really useful? Is car number, owner, type sufficient information? What about such things as lading and destinations?

Bob Wiseman: Less than 100 percent is useful if the OACI data is correlated with manual advanced consist data. One hundred percent readability is obtainable under the following conditions: (1) The label modules are originally properly applied. (2) The scanners are in good operating condition, and (3) The labels themselves are not degraded due to dirt or other causes. It is also possible to obtain nearly 100 percent through the use of multiplex scanners which independently read labels on both sides of the car. It has been my experience that except for certain very rare weather and solar radiation conditions good labels are consistently readable.

Attendee: H. A. List, Railway Engineering Associates, Inc.
Attendee's Question: What other kinds of things could you pack into a label?

Bob Wiseman: Currently the labels specify the car number, type, and owner and are permanently placed on the car side. The cargo in the rail car can be identified through the use of a separate label which is either mounted on the cargo container or placed temporarily on the car side. The scanner system presently has the capability to read more than one label per car; however, destination information changes too frequently to consider the use of additional labels placed on the locomotive. It would seem to me that destination information might best be obtained from a radio type data acquisition system such as a digital data link.
Attendee: E. J. Sierleja, Transportation and Distribution Associates.
Attendee's Question: Is there a criterion for determining a readability level at which the industry will implement MIS?

D. Spanton: The research engineers weren't really asked to address questions of that nature. This is an industry decision. The OT committee had specific recommendations on retaining or abandoning the present system. The Board of Directors of the AAR has a certain involvement and it is my understanding that recently a poll was initiated of all of the member railroads of the AAR; that's being handled, I believe, through Frank Danahy as Chairman of the Mechanical Committee. The poll has not yet been completed, and I presume it will be at least several months before all of the responses are in and appropriately analyzed. I don't know whether that answers your question adequately, but that's a summary of what we've seen in the press and what we understand.

Attendee: D. R. Sutliff, Association of American Railroads. Attendee's Question: Were the wear surfaces in a new or worn conditon for the coefficients of friction which were reported in your remarks?

G. Fay: They were ground so they were not in a fully worn condition, but in a simulated worn condition. Klaus, do you want to add something to that?

K. Cappel: I'll have to add that when we tried the runs with brand new, unworn shoes, we have some rather alarmingly high numbers. We were concerned about the integrity of the equipment so we stopped and ground the shoes off.

Attendee: R. W. Radford, Canadian National Railways:
Attendee's Question: Have any mea-

Donald L. Spanton has served as Director, Office of Freight Systems (R&D), for the FRA since 1976. He received his BS in MGT. Engr., (1952) from Rensselaer Polytechnic Institute; MS in Ind. Engr., (1958) Georgia Institute of Technology; Ph.D in Bus. Adm., (1972) American University; Registered Professional Engineer, State of Georgia.

surements been made to record incremental fuel consumption and incremental work done in the form of alternators, kilowatts per hour, or draw bar horsepower per hour to show if any thermo-dynamic advantage exists for the use of the fuel saver?

M. Jacobs: The initial test objective of the fuel saver concept was just to install the equipment, get it on the track, and see if there were indeed any fuel savings. In answer to your question, no, there were not any timed incremental measurements taken on board the locomotive. Measurements were made on the average of every half-hour or hour if possible. In high speed operation, measurements were not made until the end of a particular operating route segment when we had a crew change. In this situation, about 3½ to 4 hours elapsed between measurements. Currently, there are plans in FRA's Office of Freight Systems to develop a more accurate locomotive data acquisition package. This system probably will take about two years to develop and will definitely include the kilowatt-hour power measurements you mentioned as well as anything else that we can devise to define the fuel efficiency of the locomotive.

Attendee: H. A. List, Railway Engineering Associates, Inc.
Attendee's Question: What are the five modes in that matrix that you had up there that were ascribed to the "axles"?

M. Kenworthy: Thank you for the question; I had meant to cover that. The five modes were simply the rigid body modes because the axles are sufficiently rigid that we didn't need to consider elastic modes. The five modes were simply the three linear accelerations and two angular accelerations, pitch being deleted because the wheel rotates about the Y axis.

Attendee: Dr. L. Levine, Colorado State University.
Attendee's Question: Could you briefly itemize the four analytical techniques for investigating hunting motion.

N. Tsai: The four techniques we are using for the hunting motion are the linear technique or Eigen Value, the quasi-linearization, the non-linear integration, and the hybrid technique.

Attendee: B. Beetle, Abex Corporation.
Attendee's Question: The type of composition or metal brake shoe and its wear on the wheel tread had a marked influence on the hunting that was mentioned in Paul Garrin and Klaus Cappel's ASME paper in April of 1976. Has any of the testing that we've done confirmed the position on composition or metal brake shoes?

N. Tsai: As far as this program is concerned, we did not consider brake shoes in our modeling and application; we did not apply brakes during the test.

Attendee: E. Schwarm, Arthur D. Little, Inc.
Attendee's Question: Did you have any problem in dealing with the effect of head or cross winds in obtaining comparative data between two runs?

M. Jacobs: When we started the Burlington Northern unit coal train tests, we did start on a very windy day. There were gusts up to 30 mph. But, that was just in the area of Lincoln, Nebraska. When we got 1,000 miles down the road, we were testing in comparatively cold, still winter air. Because the tests were conducted within consecutive days of each other, we did have very good comparability.

D. Spanton: You might note that we also have a considerable amount of aerodynamic research going on here at the Test Center. We have some full scale tests going on to validate earlier wind tunnel model scale tests. The effect of aerodynamic quartering winds, head winds and so forth has been reported in preliminary fashion in some documents and will be further reported on in the future.

Attendee: E. Daily, Koppers Company.
Attendee's Question: What are you

doing to articulate, reduce, and get rid of truck hunting?

G. Fay: I can talk to you for about a half-hour on that. We just awarded the contract September 30 to Wyle Laboratories for 2.6 million dollars, to wrap-up Phase I first, that is, to finish the work that was done on the S-2 and the ride control trucks, and then to look at the Type II trucks which we hope will eliminate the hunting problem. We hope we will be able to show you, the industry, that it is really more economical to put a little bit more money into that truck in the initial phase of procurement, because it's going to save you in the long run in operating costs. We have a 32-page Statement of Work for Phase II which I would be glad to discuss with you.

Attendee: E. Dailey, Koppers Company.
Attendee's Question: Have the dimensions of containers been standardized?

D. Spanton: The ISO standards do exist, but I don't think it's quite fair to say that all containers are standardized at this point.

J. Blanchfield: No, there's quite a bit of activity in looking at different size containers for different purposes. A domestic container idea is being considered. It would be a container structurally lighter than the international containers, which have to be stacked as many as 7 deep. A domestic container would be lighter, perhaps have a better cube, and once the dimensions are standardized, then I think we may see some growth in the container field.

Attendee: E. Dailey, Koppers Company.
Attendee's Question: Is there an intent to enlarge the FAST loop beyond 4.8 miles?

D. Spanton: We didn't talk much about FAST this morning. You'll hear more about the details of what we have ongoing on Thursday. There is a wish to expand the FAST loop beyond 4.8 miles because it is very obviously limited in speed (namely 42 to 45 mph) with its current configuration; this does not permit us to investigate adequately speed phenomena, such as hunting, or conduct adequate braking tests. These are two very important things and I'm sure there are many more which do need to be investigated and which require a larger run of tangent track. The present configuration of four locomotives and 70 to 80 cars means that some portion of the train is in a curve all the time. For those of you who have consulted the budget documents, you will observe that in the Fiscal Year 1978 budget, we do have some funds for the design, in the sense of an A&E study, of a larger track. We do not presently have any authority to proceed with construction. We're working on it.

Attendee: M. Ephraim, ElectroMotive Division, General Motors Corporation.
Attendee's Comment: I just wanted to mention a few things in regard to Marilynne's presentation. Bob Radford has asked whether there was any attempt to try to measure horsepower hours, tractive effort, or some other parameters. There has been a test just recently completed on the Southern Pacific and the report will be out soon, which will cover some of those parameters. I might just say that we have made a number of studies which indicate that there is no thermal efficiency improvement used in the fuel saver. I do think, however, that there might be in the way of handling trains, some improvement. And, time does not allow, but it is possible by using the fuel saver under some conditions to improve fuel economy. It's not due to thermal efficiency improvements by using the fuel saver. Thank you.

Attendee: R.L. Bullock, Standard Car Truck Company.
Attendee's Comment: While the purpose of the paper is to introduce the development of a transducer system, the authors deviate from this purpose in presenting the test results, where they attempt to characterize the two

suspension systems just by one parameter, the friction coefficient. The two trucks are very different with respect to their damping characteristics and, therefore, test results may not relate to the difference in the conceptual designs. Just one data point, randomly picked, is used in calculating the friction coefficient and thus has little meaning unless the measuring system is proven to generate consistent and reproducible data which is not the case here. Was there not a wide scatter in the data?

Suspension systems are better characterized by an effective friction coefficient which would include normal changes in geometry, material damping and Coulomb damping. The energy loss method would be more reliable. Better mathematical simulations of suspension systems use either a describing function or can use actual experimental data. In either case for the variable damped truck, friction forces versus absolute spring position must be known. The transducer system introduced through this paper has not demonstrated the ability to furnish this valuable information.

In the section on design concept, the authors have very clearly stressed the importance of symmetric load distribution and the resulting geometry of the wear plate and friction shoe. However, in the results section it is stated that such a symmetric load distribution was not achieved. The authors have presented data from only one friction casting and wear plate assembly. This obviously makes the values of the friction coefficient very questionable. This problem could have been eliminated to an extent if the authors had presented the average of test results over all the friction shoe and wear plate assemblies present in the system. Also, further details, with regard to the calibrations involved to eliminate the effects of cross coupling between the vertical and normal force blocks, is desirable.

Another important aspect in the design of suspension systems is the material damping capacity or internal friction. The Barber S-2 trucks use cast iron friction shoes which have an excellent damping capacity compared to the steel shoes used in the ASF Ride Control Trucks. The difference in the chatter observed may be related to the difference in the damping capacities. The energy dissipated by internal damping will not be indicated by just considering the friction coefficient obtained using the FSFMS described in this paper.

K. Cappel and G. Fay: In response to your discussion, the main point to be made is that the intent was not to characterize a suspension system but rather to demonstrate the types of measurements afforded by the Friction Snubber Force Measurement System under conditions which cannot be equated to the full spectrum of the rail environment. We disagree that the selection of one data point was meaningless. The error due to cross coupling was on the order of 1% and when averaged over five cycles there was only a 4½% variation in the total work performed which includes the cross coupling effects. We feel the reproducibility is excellent.

Yes, there was scatter in the data. We feel this is due to low frequency of the input governed by equipment limitations. This should smooth out when we field test. For a half cycle (downstroke), the friction coefficient varied between .04 and .39 yielding an average of .25. The value of .167 cited earlier was simply an example of the type of measurements which can be made and was calculated at the point of the highest measured force at the lower normal transducer.

We must agree that the use of a total energy loss method is desirable. However, it was never the intent of this program to quantify this loss owing to the expense involved in the necessary laboratory equipment or the conduct of field tests. The shop tests were conducted at an arbitrary but known spring position. It should be pointed out that the Coulomb friction value would remain the same regardless of the load. It was never the intent, again, of the program to quantify the total energy dissipation of the suspension system which would include the internal damping capacity. We are not sure, however, that we agree with your statement regarding the contribution of this mechanism to the total energy dissipation.

You have misunderstood our reference to the importance of symmetric load distribution. The emphasis in the paper was placed on maintaining wear plate symmetry or in other words the FSFMS must have been rigid enough to preclude any tilt of the wear plate. Cocking of the friction shoe was not restrained and occurred just as it would under a similar service load.

We appreciate your comments and only wish more coordination could have been possible.

SESSION II THE STATUS OF PASSENGER SYSTEMS, R&D

OVERVIEW OF FEDERAL RAILROAD ADMINISTRATION
PASSENGER SYSTEMS RESEARCH & DEVELOPMENT

BY

M. B. Mitchell

Executive Summary

There is no question that, on a national basis, rail passenger service is here to stay. It may be that no single railroad can justify continuing passenger service at the expense of having its more profitable freight operations sit on a siding while a passenger train occupies the main lines; however, the government feels that a balanced transportation network must include some level of rail passenger service, especially in light of today's energy situation.

The Department of Transportation is deeply involved in rail passenger business, both through Amtrak and the Northeast Corridor Project. The Department was instrumental in preparing the legislative package establishing the National Railroad Passenger Corporation (Amtrak), the organization through which our national rail passenger goal is implemented. The Rail Passenger Service Act of 1970, as amended (84 Stat. 1327; 45 U.S.C. 541), created Amtrak to provide a balanced transportation system by improving and developing intercity passenger trail service. The Amtrak Corporation is built on a "for profit" basis, with investment capital and operating losses currently supported by Federal financing. Amtrak is governed by a 13-man board of directors, with the Secretary of Transportation being an ex-officio member.

The Secretary of Transportation is also charged with the responsibility for implementing the Northeast Corridor Improvement Project, as delineated in Section 703 of the Railroad Revitalization and Regulatory Reform Act of 1976 (4-R Act), which reads (paraphrased):

> The Northeast Corridor Improvement Project shall be implemented by the Secretary in order to achieve the following goals:
>
> (1) INTERCITY RAIL PASSENGER SERVICES.-
> (A) (i) Within five years after enactment of this Act, regularly scheduled and dependable intercity rail passenger service shall be established between Boston and New York, operating on a 3-hour and 40-minute schedule (including appropriate stops)........
>
> (E) Within two years after enactment of this Act, the Secretary shall submit to the Congress a report on (1) the financial and operating results of the intercity rail passenger service established under this section, (2) the rail freight service improved and maintained pursuant to this section, and (3) the practicability (considering engineering and financial feasibility and market demand) of establishing regularly scheduled and dependable intercity rail passenger service between Boston, Massachusetts and New York operating on a 3-hour schedule (including appropriate intermediate stops), and between New York and Washington operating on a 2½ hour schedule (including appropriate intermediate stops).

This report shall include a full and complete accounting of the need for improvements in intercity passenger transportation within the Northeast Cor-

ridor, and a full accounting as to the public costs and benefits from improving various modes of transportation to meet those needs. If this report shows (i) that further improvements are needed in intercity passenger transportation in the Northeast Corridor, and (ii) that improvements (in addition to those required by subparagraph (A) (i) of this paragraph) in the rail system in this area would return the most public benefits for the public costs involved, the Secretary shall make appropriate recommendations to the Congress.

Within six years after enactment, the Secretary shall submit an updated comprehensive report on the matters referred to in this subparagraph. Thereafter, if it is practicable, the Secretary shall facilitate the establishment of intercity rail passenger service in the Corridor which achieves the service goals specified in this subparagraph.

AN OVERVIEW OF PASSENGER TRAIN SYSTEM ACTIVITIES, INCLUDING RAILROAD ELECTRICFICATION

BY

R. A. Novotny

Executive Summary

The current status and future plans for three areas of railroad research and development are discussed.

Railroad Electrification. In the Railroad Revitalization and Regulatory Reform Act of 1976 (4-R Act) Congress requires a study of the potential benefits and costs of railroad electrification. The study is nearing completion and indicates that financially healthy railroads stand to benefit from electrification over the long-term, whereas marginal railroads are in no position to consider electrification in their scheme of improvements. Even for healthy railroads, the investment decision is difficult given the large amount of dollars required and the uncertainties associated with the investment. The technology is well developed in Europe, Russia and Japan and could be implemented in the United States with minimum technical risk. Operationally, electrified rail has the potential for offering improvements as a result of improved locomotive acceleration characteristics and reduced turnaround times although the practical realization of these benefits is very railroad specific and requires a detailed study on a case-by-case basis. Economically, the lower maintenance cost of the electric locomotive vs. the diesel-electric provides an improved return-on-investment for the railroad; if petroleum costs increase with respect to electrification cost over the years this would further enchance the electrification decision. However, the long-range nature of the investment (30 years for the rolling stock and fixed plant) coupled with the uncertainties of price of energy and traffic growth make an electrification decision for a railroad somewhat speculative. From a national benefit standpoint, the flexibility of the fuel source offers a strategic benefit in that it would render electrified railroads immune to the vagaries of imported fuel oil prices and supply. The Government is already involved in railroad electrification to some extent via the Northeast Corridor program, electrification of a test track at the Department of Transportation Test Center, and our financial interest (Sec. 606 of the 4-R Act) in the outcome of Conrail's study on electrification. The results of the FRA study will provide current information in railroad electrification for the consideration of Government decision-makers.

Advanced Technology. The High Speed Ground Transportation Act of 1975 set the stage for studies to determine the applicability of advanced technology for intercity ground transportation. A considerable amount of technical study and systems analysis was undertaken on a variety of technical features and system concepts. The initial work indicated a potential market for non-contact propelled and levitated vehicles operating at speeds between 150 mph and 300 mph. This led to the manufacture and test at the Transportation Test Center of three full-scale prototype systems exploring the practical aspects of air levitated and linear motor propelled vehicles. The testing proved the technical feasibility of air levitation and linear motor propulsion but indicated that the fixed plant investment for such systems was high. The lack of interest on the part of transportation planners with regard to implementing this advanced technology led to drastic cuts in the program plans and funds.

The current DOT program is limited to tracking the progress of foreign work, conducting a small amount of technical R&D, and continual assessment of the market place to determine where and when an advanced ground transportation application might be feasible.

Improved Passenger Service. The Rail Passenger Service Act of 1970 made the National Railroad Passenger Corporation (Amtrak) responsible for operating the major portion of rail passenger service within the United States. One goal of the FRA is to assist Amtrak in providing cost-effective and improved passenger service in those markets where demand warrants. In support of this goal, a train evaluation program has been structured to analyze advanced passenger equipment, both foreign and domestic, against the Northeast Corridor and against medium density corridors in the Amtrak route structure. Ten trains of interest (mostly foreign) have been identified and are being analyzed from a cost and performance standpoint to determine their usefulness to Amtrak in select corridor applications. The program includes a preliminary study assessment of the performance and the cost effectiveness of each train to be followed by an on-track evaluation of selected parameters (e.g., noise, fuel consumption, acceleration, track forces) if further corroboration of the train performance is deemed necessary. The program also identifies component and system areas that are candidates for R&D activity within FRA. Government personnel in eight foreign countries have been visited and agreed to cooperate by providing the information necessary to do the preliminary assessment of the trains. Reports on individual trains will be prepared with the first report available the latter part of 1977.

IMPROVED PASSENGER SERVICE COMPONENT RESEARCH AND DEVELOPMENT

BY

M. Clifford Gannett

Executive Summary

The Federal Railroad Administration Office of Passenger Systems (FRA/OPS) is responsible for the improvement of passenger equipment. Two major areas of interest are improved passenger car suspensions and electrical propulsion systems.

One of the major missions of the Federal Railroad Administration, Office of Passenger Systems FRA (OPS) is to promote and/or conduct research and development programs leading to continuous improvement of rail passenger service. These programs include development of better designs for car bodies and their associated electrical and mechanical systems. For the past five years, however, a major emphasis has been on development of better truck subsystems, i.e., trucks that will improve the ride quality of existing passenger cars. Reduction of truck maintenance has also been a prime consideration.

To illustrate the role of FRA (OPS) in truck development, this paper describes briefly two of our truck development programs. The intent is to describe the overall program and not to repeat detailed technical material presented in other papers.

OVERVIEW OF FEDERAL RAILROAD ADMINISTRATION PASSENGER SYSTEMS RESEARCH & DEVELOPMENT

BY

M. B. MITCHELL

INTRODUCTION

There is no question that, on a national basis, rail passenger service is here to stay. It may be that no single railroad can justify continuing passenger service at the expense of having its more profitable freight operations sit on a siding while a passenger train occupies the main lines; however, the government feels that a balanced transportation network must include some level of rail passenger service, especially in light of today's energy situation.

The Department of Transportation is deeply involved in rail passenger business, both through Amtrak and the Northeast Corridor Project. The Department was instrumental in preparing the legislative package establishing the National Railroad Passenger Corporation (Amtrak), the organization through which national rail passenger service is provided. The Rail Passenger Service Act of 1970, as amended (84 Stat. 1327; 45 U.S.C. 541), created Amtrak to provide a balanced transportation system by improving and developing intercity passenger train service. The Amtrak Corporation was established on a "for-profit" basis, with investment capital and operating losses currently supported by federal financing. Amtrak is governed by a 13-man board of directors, with the Secretary of Transportation being an ex-officio member.

The Secretary of Transportation is also charged with the responsibility for implementing the Northeast Corridor Improvement Project, as delineated in Section 703 of the Railroad Revitalization and Regulatory Reform Act of 1976 (4-R Act), which reads (paraphased):

The Northeast Corridor Improvement Project shall be implemented by the Secretary in order to achieve the following goals:

(1) INTERCITY RAIL PASSENGER SERVICES.-

(A) (i) Within five years after enactment of this Act, regularly scheduled and dependable intercity rail passenger service shall be established between Boston and New York, operating on a 3-hour-and-40 minute schedule (including appropriate stops).....

(E) Within two years after enactment of this Act, the Secretary shall submit to the Congress a report on (1) the financial and operating results of the intercity rail passenger service established under this section, (2) the rail freight service improved and maintained pursuant to this section, and (3) the practicability (considering engineering and financial feasibility and market demand) of establishing regularly scheduled and dependable intercity rail passenger service between Boston, Massachusetts and New York operating on a 3-hour schedule (including appropriate intermediate stops), and between New York and Washington operating on a 2½ hour schedule (including appropriate intermediate stops).

This report shall include a full and complete accounting of the need for improvements in intercity passenger transportation within the Northeast Corridor, and a full accounting as to the public costs and benefits

---

M. B. Mitchell has served as Director of the Office of Passenger Systems (OR&D) since 1975. Mitchell received his M.S. in Engineering from Oklahoma State University (1951). Mitchell served as Chief of Test Center and Demonstrations Division prior to his current position.

from improving various modes of transportation to meet those needs.

If this report shows (i) that further improvements are needed in intercity passenger transportation in the Northeast Corridor, and (ii) that improvements (in addition to those required by subparagraph (A) (i) of this paragraph) in the rail system in this area would return the most public benefits for the public costs involved, the Secretary shall make appropriate recommendations to the Congress.

Within six years after enactment, the Secretary shall submit an updated comprehensive report on the matters referred to in this subparagraph. Thereafter, if it is practicable, the Secretary shall facilitate the establishment of intercity rail passenger service in the Corridor which achieves the service goals specified in this subparagraph.

GOALS

Before discussing the organizational structure of the Office of Passenger Systems, it is helpful to review its six goals, (figure 1). The first is to provide the technology, both near and long-term, that will achieve maximum effective use of rail passenger systems in meeting the nation's transportation needs. This goal is more-or-less an umbrella for the general rail passenger research and development activity of this office. The day-by-day analysis of potential candidate train systems and/or components, along with analysis of viable corridor applications, is conducted under this charter. Cost effective rail passenger operation is predicated on meaningful research and development that will assure the overall system is reliable, easy to maintain, and secured at the lowest possible capital and operating cost.

The second goal is to provide technological data and advice to the Secretary to use in meeting his responsibility in connection with Amtrak. Not only is the Secretary a voting member on the Amtrak Board of Directors, a good portion of the funds Amtrak receives annually from Congress flows through the DOT. The Secretary must be aware of developments in rail passenger vehicle technology.

PASSENGER SYSTEMS GOALS

- PROVIDE THE TECHNOLOGY, BOTH NEAR AND LONG-TERM, THAT WILL PERMIT MAXIMUM EFFECTIVE USE OF RAIL PASSENGER SYSTEMS IN MEETING THE NATION'S TRANSPORTATION NEEDS;
- PROVIDE TECHNOLOGICAL DATA AND ADVICE TO THE SECRETARY FOR USE IN MEETING HIS RESPONSIBILITY IN CONNECTION WITH AMTRAK;
- PROVIDE DIRECT SUPPORT TO AMTRAK IN DEVELOPING NEW RAIL PASSENGER EQUIPMENT AS MUTUALLY AGREED TO;
- PROVIDE DIRECT RESEARCH AND DEVELOPMENT SUPPORT TO THE NORTHEAST CORRIDOR PROJECT OFFICE AS MUTUALLY AGREED TO;
- UPDATE AND ADVANCE THE TECHNOLOGY FOR WAYSIDE ELECTRIFICATION AND VEHICLE TRACTION FOR U.S. RAILROADS; AND
- MAINTAIN A TECHNOLOGY BASE AND KEEP ABREAST OF OF WORLDWIDE TECHNOLOGY DEVELOPMENTS IN ADVANCED SYSTEMS.

FIGURE 1.

The goal of providing direct support to Amtrak in developing new rail passenger equipment is amplified by a Memorandum of Understanding (MOU) that defines the general scope of responsibility and working relationships between the two organizations. This MOU is backed by a detailed Master Plan that not only displays the individual projects and schedule, but also lists the office with primary responsibility. These two documents have been very helpful in establishing a compatible working relationship and negating the possibility of duplicative research.

As with the goal defining the support to Amtrak, a similar agreement has been reached with the Northeast Corridor Project Office for direct support on near-term vehicles for NEC application and for installation of the electrification system for the Railroad Test Track at the Transportation Test Center. The working arrangements and relationships between the two offices are also displayed on the Master Plan.

The fifth goal is to update

and advance the technology in the areas of wayside electrification and vehicle traction. The FRA has a growing interest in railroad electrification for reasons of fuel efficiency, operations economy and environmental protection. We are proceeding with electrification studies and electrification research at the Test Center. In a related technical area, significant technical achievements have been recorded recently in the field of traction motors, and this office is now conducting feasibility studies in this area.

Finally, in the area of advanced systems, our mission goal has been reoriented to a more modest level. In past years, research and development achievements on advanced transportation vehicles outpaced the public demand and interest. The reestablished goal is to maintain a technical awareness of worldwide status in this field, and to alert planners when it is time once again to start appropriate research projects.

ORGANIZATIONAL STRUCTURE

The Office of Passenger Systems, one of three operating offices within the Federal Railroad Administration's (FRA's) Office of Research and Development (as shown in figure 2), consists of two divisions: the Passenger Equipment Division and the Passenger Systems and Facilities Division. This organizational structure maintains a proper balance between looking at the passenger business as a total system and responding to Amtrak on a one-by-one component basis. The systems aspect of passenger rail R&D supports the Northeast Corridor Projects (NECP) which will incorporate a major new train system. Every facet of the total system is being looked at critically before a decision is made on any single element. Amtrak, on the other hand, which integrated the passenger rail portion of many railroads into a single network within a very short period of time and started operation with equipment having an average age of some 20 to 25 years, requires quick response on equipment and maintenance issues. Until outdated equipment is retired and new designs implemented, Amtrak will continue to be burdened with a high-cost operation.

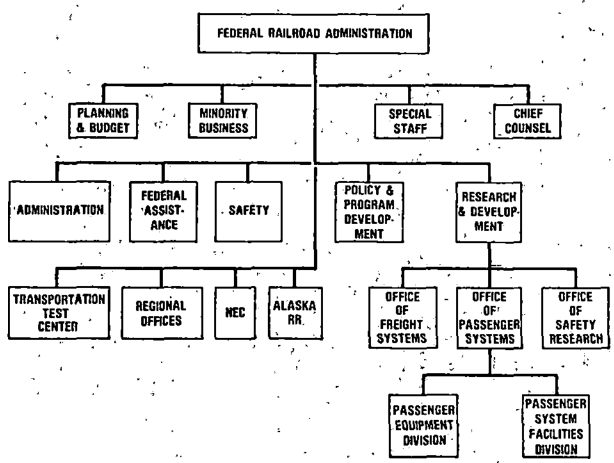


FIGURE 2.

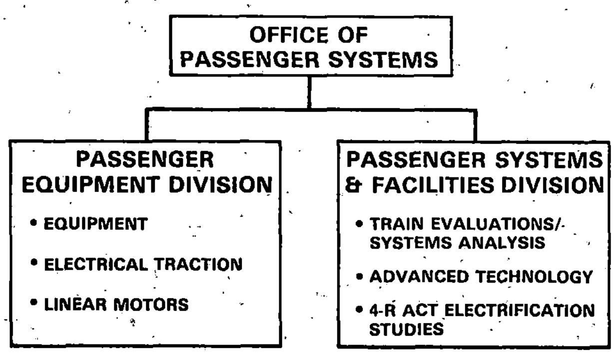


FIGURE 3.

The Passenger Systems and Facility Division is charged with planning, initiating, and conducting ground transportation research and development on a system level. The scope of activity includes both passenger train and advanced systems. The Passenger Equipment Division has a similar charter, except it works at the equipment level. As stated earlier, this matrix mix allows a complete system overview, while affording more detailed investigation into the technical aspects of equipment at the component level. It also allows quick response on near-term requirements without negating or disrupting long-range system planning and execution of major programs, such as Northeast Corridor Project and Improved Passenger Equipment Evaluation Program.

Major program and/or technology areas of the two divisions are shown in figure 3 and further amplified in the next section.

Improved Passenger Equipment Evaluation Program

The Improved Passenger Equipment Evaluation Program (IPEEP) is an outgrowth of the Improved Passenger Train (IPT) program initiated in 1973. The intended scope for the IPT was to develop a prototype train system for off-corridor application with provisions for converting the traction system to total electric (in place of the turbine or diesel-electric propulsion) for application in the NEC. During early phases of the program, it was determined that insufficient technical data was available for defining a rigorous performance criteria or design specification. One technical area with major shortcomings was that of ride quality. Others were concerned with trade-offs of propulsion systems, multi-unit versus locomotive-hauled trailer cars, articulation, and track forces. At the same time, it was recognized that available funding and personnel resources were inadequate for successfully conducting the program as originally planned. As a consequence, the program was redirected to solicit participation by the foreign railroad and supply industry in order to take advantage of technical achievements already accomplished by others, should they have potential application in the United States. Both the foreign and U.S. rail passenger train supply community has endorsed the concept and become active participants.

Electrification Report Submitted to Congress, I.E. 4-R Act

The FRA's Office of Research and Development has been assigned responsibility for railroad electrification R&D and preparation of the report on costs and benefits of railroad electrification to be submitted by the Secretary of Transportation to Congress next January. The Office of Passenger Systems is the lead organization for this effort.

The report is in response to the provisions of the *Railroad Revitalization and Regulatory Reform Act of 1976* (4-R Act).

In conducting the study on electrification, the capital and operating cost factors for converting a 26,000 mile high traffic density nationwide rail network to electrified operations has been reviewed. Potential diesel fuel savings, electricity consumption, and coal requirements were computed for the network on a nationwide and regional basis. The environmental effects (primarily emissions from fuels) were also computed nationwide, with and without the network in place. Potential benefits and impact were considered from both the national and railroad perspective, and as it would affect the electric utility system. Finally, because conversion to electrification requires a significant capital investment which must be recovered through savings in operating costs and service improvements, the study deals with R&D projects leading to low risk improvements.

Advanced Systems Program

Advanced System activity in past years included research and development that resulted in successful demonstrations of double-sided linear induction motors and a test-bed vehicle which ultimately set the steel wheelsteel rail land speed record of 255 miles per hours, a 2,500-horsepower tracked air cushion vehicle designed for speeds up to 300 miles per hour, a prototype 60-passenger version of a tracked air cushion vehicle tested and demonstrated at speeds up to 145 miles per hour, successful testing of a 8,250-volt power conditioning unit (PCU) for a 4,000-horsepower electric motor with application to conventional railroads, and a wayside power collection unit that was tested at speeds up to 300 miles per hour. The results of these programs significantly advanced the state-of-the-art of next-generation vehicle systems. However, recent ground transportation traffic demand analyses indicate that more emphasis must be placed on systems having more potential in the near-term. The advanced system activity, as a result, has been reoriented to provide access into developments being conducted by other countries, but stopping short of actual research and development projects.

Near-Term Vehicle Selection

The selection of rolling stock for application in the Northeast Corridor has been segregated from the IPEEP program owing to the 1981 time element imposed by the 4-R Act. Insufficient time exists before actual operation of the improved NEC system to allow development of a new generation of rolling stock. The "Near-Term Vehicle Selection" project entails review and evaluation of candidate locomotives for use in hauling Amfleet trailer cars and applicable multi-unit equipment. The evaluation includes actual running tests, cost estimation, equipment availability and reliability studies, and final vehicle selection and design modifications. The termination date of the project will depend on which vehicles are selected and which performance requirements are verified.

Electrification Program

The Electrification Program is structured to improve the efficiencies of electrical equipment reducing petroleum usage by our railroads. Near-term projects include system engineering studies directed toward adaptation of advanced foreign technology to U.S. application, and the development of models that will provide proper interfacing between railroads and electrical utilities. Substation and railroad/utiliy interface improvements are sought that will reduce peak power demands, improve phase balance, and reduce reactive power requirements. Improved catenary equipment is also being developed, as are improved AC traction equipment that will lead to improved motive power. Finally, a project is aimed at techniques for reducing EMI with wayside and public communication systems.

AN OVERVIEW OF PASSENGER TRAIN SYSTEM ACTIVITIES, INCLUDING RAILROAD ELECTRIFICATION

BY

R.A. NOVOTNY

Current status and future plans for three areas of railroad research and development are reported herein.

(1) Railroad Electrification The study to analyze railroad electrification, called for by the Railroad Revitalization and Regulatory Reform Act of 1976, is nearing completion. Conclusions and recommendations will be transmitted to Congress early in 1978. Preliminary findings and other government work related to railroad electrification are presented here.

(2) Advanced Technology. Results from the R&D program to develop an advanced ground transportation system for the United States are summarized. Future activity will; monitor the extensive work currently underway in foreign countries, undertake limited R&D on key aspects of advanced systems (particularly magnetic suspension and propulsion), and investigate opportunities for advanced ground transportation in the United States.

(3) Improved Passenger Train. Advanced passenger trains, both domestic and foreign, are being evaluated for the next generation U.S. passenger train. Reduced trip time, passenger comfort, and cost-competitiveness are the primary goals. Two train concepts are sought, an electrified train for Amtrak's high-density Northeast Corridor Program and a non-electrified train for other medium-density Amtrak routes.

RAILROAD ELECTRIFICATION

OVERVIEW

The Railroad Revitalization and Regulatory Reform Act of 1976 (4-R Act) directed the Secretary of Transportation to analyze and evaluate railroad electrification with emphasis on potential benefits, costs, and energy and environmental implications. The responsibility for conducting this study was assigned to the Federal Railroad Administration (FRA) which is now in the process of completing its work. The study covers four major efforts: (1) investigate the technology of railroad electrification; (2) estimate the capital and operating cost differential for electrified and diesel-electric railroads; (3) consider the energy and environmental issues; and (4) consider the options for government participation in rail electrification.

The final report will address the issues associated with railroad electrification to such an extent that the government will be able to determine accurately the national transportation and energy benefits that could accrue from widespread implementation of this technology. The report is scheduled for transmittal to Congress early in 1978.

Although the study report is preliminary in nature at this time, an overview of the main points can be presented:

0 Railroad electrification is a mature technology. It is in widespread use throughout Europe, and there have been recent applications in the United States using the most ad-

---

Richard Novotny graduated from Rutgers University (1957) with a B.S. Degree. Mr. Novotny joined the Department of Transportation in 1964. He has served as the program planning officer on the FRA's D.C. to Boston Northeast Corridor rail development program, and currently is Chief of the FRA Passenger Systems and Facilities.

vanced designs. There are no major technological barriers to electrification.

0 A primary benefit of electrification is financial. It offers an excellent opportunity for lower operating costs and higher return on investment (10 to 30% depending on the situation) for steady, high-volume traffic markets.

0 The primary risk of electrification is also financial. It requires an extensive, long-term investment (from $125,000 per mile for single track to $410,000 per mile for double track, depending on the installation) that will be a good investment decision only if the market for railroad service holds up for 20 to 30 years.

0 The energy implications of railroad electrification are strategic. Electrification permits the nation's rail system to rely on domestic energy sources, such as coal, rather than to be dependent on uncertain foreign supply. Electrification is not, however, justified solely by petroleum savings. Electrifying 26,000 route miles would save 1.7 billion gallons of oil per year, only about one and a half percent of the transportation industry's use.

BACKGROUND AND STATUS

Railroad electrification would complement existing diesel-electric locomotives (which generate their own onboard power) with a stationary power plant system as shown in figure 1. Electric locomotives can generate much higher horsepower than diesel-electric units, do not need fueling, require minimal servicing and maintenance, and can provide improved performance over diesels.

Prior to World War II, the United States led the world in electrified railroads, with its 2,500 electrified route miles constituting one-fifth of the world total. After World War II, the European nations, faced with rebuilding their fixed plant as well as replacing equipment, aided by the availability of hydro-electric power in mountainous regions in Italy, West Germany, Switzerland, Norway, and Sweden, and driven by the desire to use energy sources available within each country, undertook extensive electrification. North America, faced only with replacing worn motive equipment, adopted the diesel-electric locomotive units that now dominate their railroad motive power. Today, of all the major industrial nations in the world, only North American countries do not have sizable portions of track electrified (as shown by figure 2).

COMPONENTS OF AN ELECTRIFIED RAIL SYSTEM

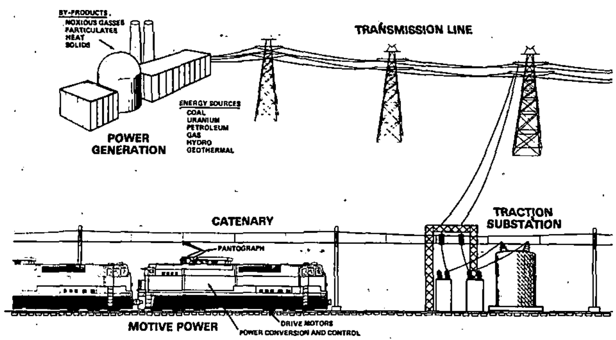


FIGURE 1: COMPONENTS OF AN ELECTRIFIED RAIL SYSTEM

COUNTRY	ROUTE MILES ELECTRIFIED	PERCENT OF TOTAL ROUTE MILES
Russia	22,780	27
France	5,520	24
West Germany	5,160	28
Italy	4,950	48
Sweden	4,350	61
Japan	3,860	29
Poland	2,180	15
England	2,070	17
Spain	1,970	23
Switzerland	1,790	99
Norway	1,420	54
Austria	1,320	39
Czechoslovakia	1,210	10
United States	1,162	0.5
Netherlands	1,010	52
Belgium	700	24
Portugal	470	27
Canada	-	nil
Mexico	-	nil

FIGURE 2. WORLD RAILROAD ELECTRIFICATION

Figure 3 shows the current extent of electrified U.S. railroad operations. The bottom three systems on this chart are operated by electric power utilities to transport coal from mines to power plants. These systems incorporate high-voltage, alternating current, and modern thyrister controls, and they demonstrate existing electrification hardware that is available in this country.

RAILROAD	LOCATION	ROUTE MILES	PROPULSION POWER
Illinois Central Gulf	Chicago, IL	37	1,500 Volts DC
Chicago South Shore & South Bend	Chicago, IL	76	1,500 Volts DC
Conrail/Amtrak			
Ex Erie Lackawanna	Hoboken, NJ	80	3,000 Volts DC
Ex Penn Central	New Haven, CT-Wash., DC & Phila., PA-Harrisburg	762	11 KV, 25 HZ AC
Ex Reading	Phila., PA	88	12 KV, 25 HZ AC
Muskingum Electric	Zanesville, OH	15	25 KV, 60 HZ AC
Black Mesa & Lake Powell	Page, AR	78	50 KV, 60 HZ AC
Texas Utilities	Monticello, TX	11	25 KV, 60 HZ AC
	Martin Lake, TX	15	25 KV, 60 HZ AC
Present total U.S. electrified miles		1,162	

FIGURE 3. CURRENT U.S. RAILROAD ELECTRIFICATION

CURRENT WORK

The government is vitally interested in railroad electrification because it offers the possibility of substantial long-term benefits and could have a significant impact on the strength of the nation's transportation system. This interest is reflected in the loan guarantee funds available under the 4-R Act; the Department of Transportation (DOT) plans to electrify the 14-mile, closed-loop track at the Transportation Test Center; the DOT's strong role in the Northeast Corridor electrification program; and this study on railroad electrification which is specifically called for by the 4-R Act.

Moreover, the railroad industry has a number of studies underway to clarify whether or not electrification is a fixed-plant improvement that railroads should undertake. For example, Conrail is considering a study to identify the costs and benefits of electrifying the Pittsburgh to Harrisburg, PA route. Amtrak and DOT, with funds made available under the 4-R Act, have plans to upgrade the 300 route miles of electrified line between Washington, D.C., and New Haven, Conn., and to electrify the 150 miles between New Haven and Boston. Detailed plans for electrification of the Northeast Corridor are being developed.

A cash-flow analysis model (see figure 4) has been constructed that can be used periodically to assess the investment rationale of railroad electrification as cost parameters (price of energy, catenary installation, etc.) and risk factors change. Finally, R&D opportunities are being identified that would reduce the investment risk to railroads that electrify, and would allow for an orderly transition should large-scale electrification be introduced into the United States over the next 30-40 years. Investigations to reduce investment risk would encompass areas such as power-factor improvements, catenary installation techniques, and low-cost catenary designs, while work to achieve an orderly transition to railroad electrification would cover such topics as safety, reliability, and electromagnetic interference standards.

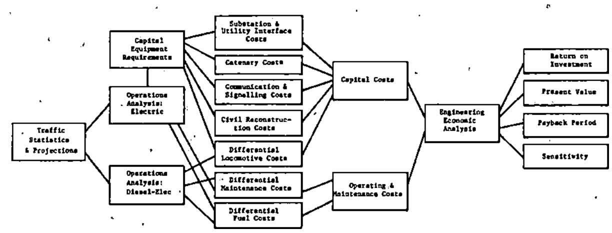


FIGURE 4. BLOCK-DIAGRAM OF THE DISCOUNTED CASH FLOW ANALYSIS MODEL FOR RAILROAD ELECTRIFICATION

ADVANCED TECHNOLOGY

HISTORICAL BACKGROUND

Following enactment of the High Speed Ground Transportation Act of 1965 (public law 89-220), the United States entered into a period of intensive research and development structured to ascertain the role of advanced technology for domestic intercity ground transportation. The program focused on high speed (150 to 300 mph) systems that not only could increase passenger comfort and safety, but would also improve efficiency and lower the cost of intercity ground transportation.

A considerable volume of technical study and system analysis was carried out for a range of theoretical concepts, including evacuated-tube vehicle systems, cable-suspended systems, and non-contact propulsion and suspension systems (air-cushion and magnetically levitated vehicles).

Linear motor propulsion and tracked air-cushion vehicles were selected to undergo full-scale development to assess actual cost and performance characteristics. Three major full-scale vehicles were developed and constructed as a part of this program. The first was the Linear Induction Motor Research Vehicle (LIMRV) depicted in figure 5. This vehicle embodied the features of a noncontact (linear electric motor) propulsion system with a conventional railroad truck design. During extensive testing over six miles of well-maintained railroad track, it yielded valuable technical information that led to development of a high-powered, low-weight, water-cooled linear motor propulsion system for a 300 mph Tracked Levitated Research Vehicle (TLRV). The LIMRV also proved the feasibility of operating conventional railroad trucks at very high speeds (255 mph achieved in 1975).

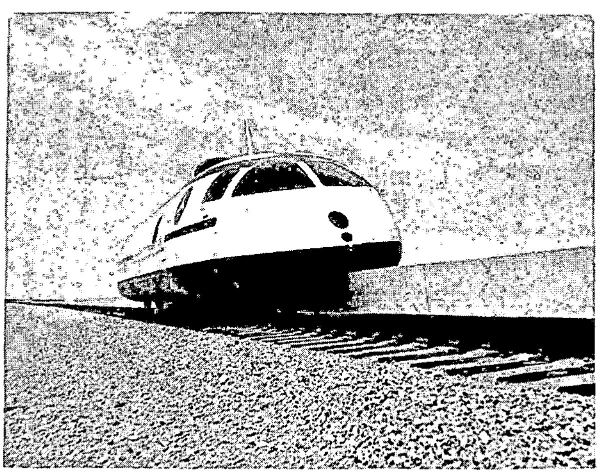


FIGURE 5. THE LINEAR INDUCTION MOTOR RESEARCH VEHICLE (LIMRV)

The next major advance was the construction and testing of a 300 mph, air-levitated, linear-motor-propelled Tracked Levitated Research Vehicle (TLRV) and associated guideway (figure 6). This TLRV work proved the feasibility of adapting advanced technology to ground transportation systems in such areas as noncontact air levitation and guidance, noncontact magnetic propulsion and braking, high-speed (300 mph) wayside power collection, and high performance solid-state electric power conditioning. It also demonstrated that vehicle and guideway technology could be put together into a workable system.

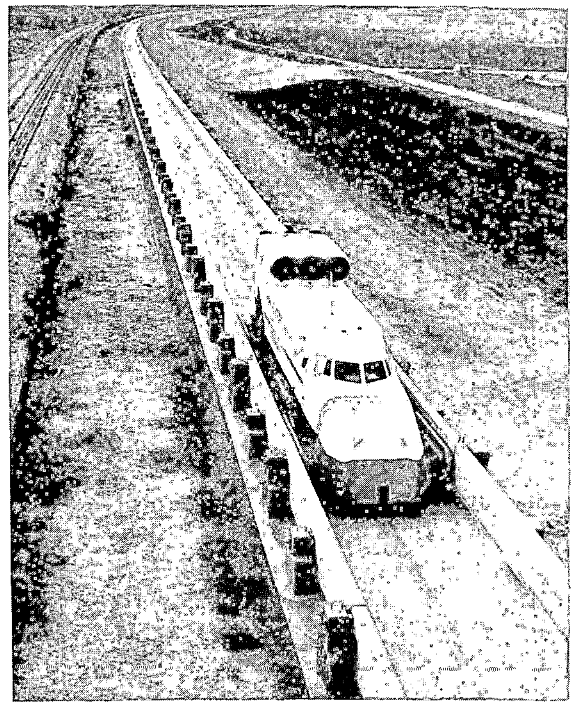


FIGURE 6. THE LINEAR-MOTOR-PROPELLED TRACKED LEVITATED RESEARCH VEHICLE (TLRV)

However, the TLRV program indicated that guideway costs were much higher than for existing conventional rail systems, and that switching vehicles into and out of guideways having a vertical reaction rail was more difficult than originally anticipated. Ultimately, the high cost (approximately $2,000,000 per mile of 1974 dollars) associated with closing the 22-mile TLRV guideway at the Transportation Test Center, along with an impending energy problem (propulsion requirements double when speed increases from 250 to 300 mph), led to demise of this program.

The third major element in the program was the Prototype Track Air Cushion Vehicle (PTACV) which was constructed to investigate advanced technology concepts for a passenger carrying vehicle having a more modest 150 mph design speed (figure 7). System technology was demonstrated at the Transportation Test Center in August, 1976. The program was subsequently shut down, suffering from renewed interest in conventional rail technology and an inability to compete economically at this time with existing auto, rail, or air intercity transportation systems.

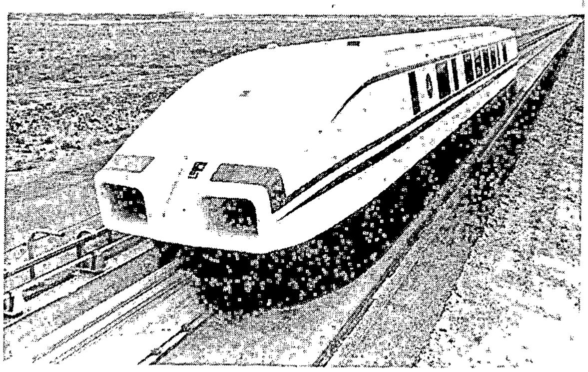


FIGURE 7. THE PROTOTYPE TRACK AIR CUSHION VEHICLE (PTACV)

(Note the power transmission railing to the side of the PTACV.)

STATUS

There has been significant progress in the technology associated with advanced ground transportation systems during the past 15 years, both domestically and abroad. Numerous alternative technological approaches have been carefully analyzed, and considerable experimental work has been performed. The following is a brief summary of the status of those systems that appear most promising for application to advanced ground transportation, based upon world-wide activity. Figure 8 shows some of the foreign-developed maglev research vehicles.

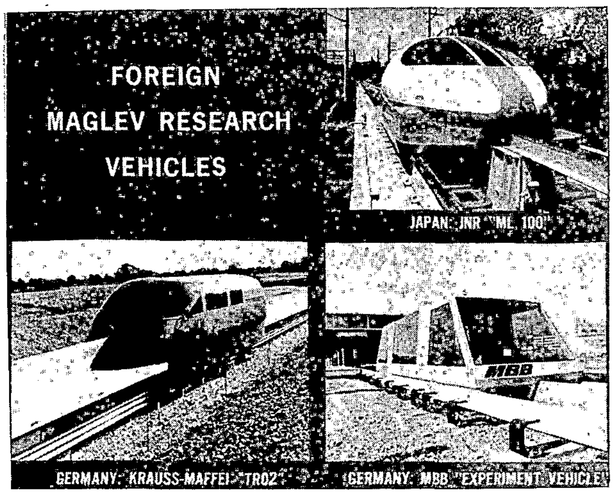


FIGURE 8. EXAMPLES OF MAGLEV RESEARCH VEHICLES DEVELOPED BY FOREIGN COUNTRIES

Air Cushion. Air cushion suspension technology is well-developed, and has operated at speeds to 150 mph in the United States using linear motor propulsion (the PTACV) and at 250 mph in France, using gas turbines for propulsion. Although technical feasibility is proven, the cost of the guideway and reaction rail structure must be reduced before such a system can compete economically with other modes of intercity ground transportation or very high patronage markets must be available to offset the high implementation cost.

Repulsion Maglev. The theory and analysis of repulsion maglev suspension, which relies on the repulsive force between like magnetic poles, is well advanced as a result of work in Japan and Germany. However, much work remains on the linear synchronous motor (the propulsion system usually associated with this concept), particularly in the area of sturdy, lightweight cryogenic hardware.

Attraction Maglev. Attraction maglev systems rely on the attractive force associated with opposite poles of a magnet to suspend a vehicle. The design and performance of attraction maglev suspension is well understood, and a system has been constructed and run at speeds up to 250 mph by the Germans. However, the integration of magnetic levitation and magnetic propulsion into a single unit requires additional investigation, as does the collection of power at high speeds. Work to resolve remaining technological problems is currently underway in Germany.

FUTURE DIRECTIONS

While the level of advanced technology R&D within the United States has been very low for the past several years, both Germany and Japan are actively studying and testing full-scale advanced transportation systems, with emphasis on magnetically levitated and propelled vehicles. Japan is particularly active and is expending tens of millions of dollars per year.

Currently, advanced technology R&D in the United States is limited to tracking the progress of foreign work, conducting a modest amount of technical R&D at the analytical/laboratory level, and continuing to assess the marketplace to determine where and when an advanced application might be economically feasib'.

With this approach in mind, a modally coordinated program is being developed that can evaluate new transportation concepts in much the same way a new train system would be judged. Both life-cycle costs and macro-economic benefits of new systems will be judged against an existing transportation system as a baseline. A block diagram of the approach is shown in figure 9.

IMPROVED PASSENGER SERVICE

OVERVIEW

The Rail Passenger Service Act of 1970 made the National Railroad Passenger Corporation (Amtrak) responsible for operating the major portion of rail passenger service within the United States. At present, Amtrak passenger service extends over 26,000 miles and links numerous urban localities (see figure 10).

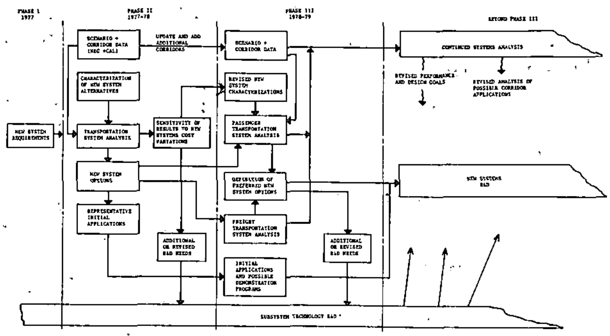


FIGURE 9. NEW SYSTEMS PROGRAM PLAN

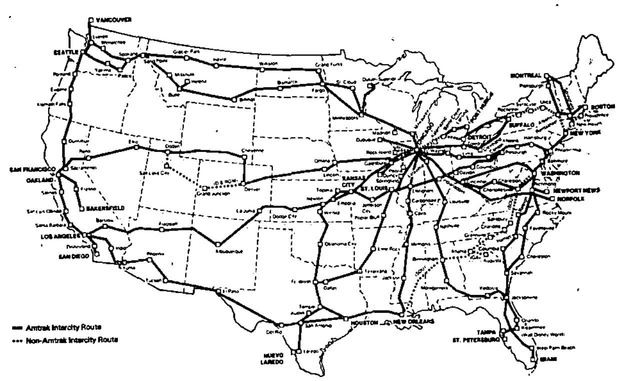


FIGURE 10. THE AMTRAK SYSTEM

The equipment fleet inherited by Amtrak was extremely varied and outdated, calling for an aggressive improvement program. Amtrak has initiated such a program which continues today. One goal of the FRA is to assist Amtrak in providing cost-effective and improved intercity passenger service in those markets where demand warrants.

In support of this goal, the FRA has structured a Train Evaluation Program to analyze advanced passenger equipment developments, both foreign and domestic, against medium and high-density corridors in the Amtrak route structure. The program is being coordinated with Amtrak and the FRA Northeast Corridor Project, and will provide an input for their equipment selection process.

Foreign technology will be stressed owing to the active passenger train R&D which has taken place in Europe and Japan over the past 15 years. Figure 11 lists the trains that have been selected for evaluation, and figure 12 describes their developmental status.

Analytical modeling techniques will be employed to ascertain the suitability of such train characteristics as speed on curves, top speed, acceleration, and braking for each corridor of interest. Other design features not dependent upon the corridor, such as passenger appeal and comfort, operational and passenger safety, maintainability, and developmental status, will also be assessed in detail. For each new train, estimates will be made as to patronage that would be generated, life-cycle costs, energy and environmental considerations, and operational flexibility as compared to existing trains that run in the corridors.

Country	Train Designation	Description
Canada	LRC	Diesel electric powered tilt train; 120 mph
England	HST	125 mph diesel electric locomotive powered train
England	APT	155 mph electric multiple unit powered tilt train
France	TGV-PSE	185 mph electric multiple unit powered train
Germany	ET-403	120 mph electric multiple unit tilt train
Italy	Y-0160	155 mph electric multiple unit tilt train
Japan	961	135 mph electric multiple unit
Sweden	X-15	110 mph electric multiple unit tilt train
Switzerland	Type III Coach	110 mph tilt car
USA	SPV-2000	100 mph diesel powered multiple unit train

FIGURE 11. TRAINS BEING EVALUATED FOR POSSIBLE EVALUATION

Advanced train technology will be evaluated for two distinct operational arenas within the United States. One is typified by the soon-to-be rehabilitated Northeast Corridor route that will be developed using the 1.75 billion dollar investment authorized in the Railroad Revitalization and Regulatory Reform Act of 1976. This operating arena will be characterized by high density traffic, excellent right-of-way (continuous welded rail, clean ballast, good ties, high-speed turnouts, few at-grade crossings, and fencing for safety and privacy), a modern signal system, the latest in electrification (25 kilovolt, 60 Hertz), a minimum of slow-orders caused by outmoded civil constraint such as bridges and tunnels, improved terminal facilities, and ownership by Amtrak. In effect, this operational arena offers a rail passenger transportation system similar to the best in England, not quite as advanced as the proposed Paris-Lyon line in France, but nonetheless a major step forward for intercity rail passenger service in the United States.

Country	Train	Status
Canada	LRC	Prototype in test. Production units being mfgd.
England	HST	In revenue service.
England	APT	Production units being mfgd. Start revenue service late 1978.
France	TGV-PSE	Being mfgd: Start running test mid 1978; revenue service late 1981.
Germany	ET403	Prototypes in test and demonstration.
Japan	961	Start running test in mid 1978.
Italy	Y-0160	In revenue demonstration service.
Sweden	X-15	Start running test in late 1977.
Switzerland	Sig III cars	Start public demonstration late 1977; revenue service in mid 1978.
USA	SPV-2000	Start running test in late 1977.

FIGURE 12. CANDIDATE TRAIN DEVELOPMENT STATUS

The second arena is characterized by medium-density passenger traffic, no electrification, right-of-way optimized for freight operation and leased by Amtrak for passenger service, and a mix of freight/passenger train traffic heavily dominated by freight.

FOREIGN RAIL TECHNOLOGY

Foreign passenger rail developments offer technology applicable to both arenas. Sizeable percentages of foreign rail systems are electrified, providing a strong technical base for high-speed (to 185 mph) pantograph/catenary dynamics, reliable and safe system design concepts, and solid-state electrical propulsion.

Certain countries are improving rail passenger service (trip time, ride comfort, amenities, and so on) while operating on the existing track structure. These efforts have resulted in such improvements as optimally designed truck suspension systems (for both ride quality and track maintenance reasons), car-body tilt systems that permit increased speed through curves, high acceleration and braking characteristics, light-weight construction techniques, and interior designs that offer quiet, well-ventilated, and aesthetically acceptable features.

Foreign developmental activities stress the systems approach to new technology development, and continually evaluate the dollar invested in rolling stock improvements against the dollar obtained in revenue as a result of more reliable, more comfortable, and faster service. Safety, maintainability, and reliability are also an implicit part of the engineering methodology.

An evaluation of foreign passenger train technology is an excellent base from which to launch our own R&D activity, and our program has been structured accordingly. However, in adapting foreign technology to American use, special care is being taken to avoid trying to mate the elephant with the hare. Differences between the foreign operational arena and our own are being carefully analyzed and placed in proper perspective. These include differences in design requirements (such as buff-load, car-body widths, allowable axle loads, and track conditions), the fact that rail as a transportation mode in foreign nations is as dominant as auto or air travel in this country, and the fact that foreign rail systems are nationalized.

TECHNICAL APPROACH

The Train Evaluation Program is divided into three phases. Phase I, now underway, is a paper evaluation of the performance and cost factors for each particular train when operating on a given domestic route. This evaluation will be based upon data supplied by the train manufacturer, and inspection of the corridors of interest, and information provided by Amtrak and the FRA Northeast Corridor Office.

For each corridor of interest, service and physical data, as well as plans for future improvements, are being gathered on such features as right-of-way, stations, and signal and communications systems. Likewise, technical data is being collected and evaluated for each train of interest. Those train characteristics that impact such major features as ride quality, reduced trip time, and cost are receiving special attention. For example, one means for reducing trip time is to increase the allowable speed through curves. This opens a host of technical and institutional areas to be addressed, including wheel-rail force interactions, car overturning forces, government regulations, ride quality, and railroad operating practices. Of particular importance is the three-inch unbalance rule now in effect in the United States. Such technical innovations as tilt systems and advanced truck design, coupled with worldwide variations in the criteria for allowable speed through curves, makes determination as to whether relief from the three-inch unbalance rule is warranted a high-priority objective of this program.

The Phase I analysis will provide a measure of the benefit that could accrue if a new train were introduced into an Amtrak corridor. Those trains showing promise will be carried into Phase II which will feature an on-site evaluation of parameters warranting verification via test and observation of actual hardware. Phase II activity will entail visits to foreign countries having trains of interest to witness and/or make measurements on such factors as wheel/rail force, ride comfort, interior/

exterior noise, energy consumption, and maintainability.

At the conclusion of this activity, a determination will be made on whether or not to bring a foreign train to the United States for testing and possible demonstration in the domestic rail environment. Such a demonstration would constitute Phase III of the program.

Figure 13 depicts the method that will be used to provide a measure of the cost required to introduce a new train system into a corridor. A key element in this analysis is the additional patronage that would be generated as a result of the new equipment reducing trip time. Trains that exhibit essentially the same schedule speed as today's Amtrak service are candidates for elimination in Phase I. A diesel-electric locomotive hauling Amfleet cars is the base-case train for off Northeast Corridor comparisons, whereas the Metroliner is the Northeast Corridor base case.

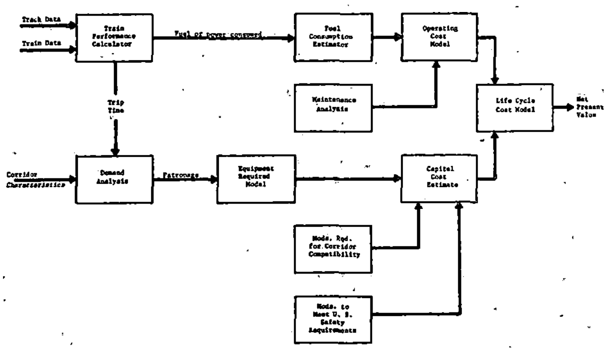


FIGURE 13. LIFE-CYCLE COST ANALYSIS FLOW DIAGRAM

Besides the cost-analysis for each train, other criteria (see figure 14) not amenable to costing, will be analyzed. In the area of passenger comfort, for example, ride quality and interior noise levels of new train equipments will be compared to the base-case trains.

STATUS

Work began in March 1977, with selection of the system contractor. The trains and corridors to be evaluated were selected after review of the literature and discussions with Amtrak. FRA, Amtrak, and contractor personnel have visited the appropriate foreign countries and obtained a commitment to participate on a no-cost basis for Phase I. The extent and circumstances of Phase II and III activity will be developed later. The data required for the Phase I evaluation have been received from a majority of the participating countries, and the first train evaluation is scheduled for completion in December, 1977. Phase I activity is scheduled for completion within the next four months, with the entire program to be finished approximately one year later.

- PASSENGER APPEAL
- PASSENGER COMFORT
- ENERGY CONSUMPTION
- OPERATIONAL FLEXIBILITY
- ON-TIME SERVICE CAPABILITY
- MAINTAINABILITY
- DEVELOPMENT STATUS
- MANUFACTURER'S CAPABILITY
- AVAILABILITY
- SPECIAL FEATURES

FIGURE 14. PASSENGER TRAIN EVALUATION CRITERIA

IMPROVED PASSENGER SERVICE COMPONENT RESEARCH AND DEVELOPMENT

BY

M. CLIFFORD GANNETT

The Federal Railroad Administration Office of Passenger Systems (FRA/OPS) is responsible for the improvement of passenger equipment. Two major areas of interest are improved passenger car suspensions and electrical propulsion systems. These topics are discussed in two separate sections of this paper.

TRUCK SUSPENSION PROGRAMS

One of the major missions of the Federal Railroad Administration, Office of Passenger Systems FRA (OPS) is to promote and/or conduct research and development of rail passenger service. These programs include development of better designs for car bodies and their associated electrical and mechanical systems. For the past five years, however, a major emphasis has been on development of better truck subsystems, i.e., trucks that will improve the ride quality of existing passenger cars. Reduction of truck maintenance has also been a prime consideration.

To illustrate the role of FRA (OPS) in truck development, this paper describes briefly two of our truck development programs. The intent is to describe the overall program and not to repeat detailed technical material presented in other papers.

The Budd Company Truck-Design Study was the first phase of the truck improvement program. The objective of the program was to develop methods for improving the ride quality of Metroliner railcars. The essential part of the design study was the development of a dynamic model which simulated the combined car body/truck system and displayed overall performance in response to realistic track inputs. The model was configured by carefully controlled, dynamic, laboratory experiments and used as a tool to evaluate recommended modifications to existing hardware and new truck designs.

The first phase produced good results and served as the basis for subsequent hardware development programs. Phase II called for development and testing of prototype trucks for existing Metroliner cars that would improve ride quality over the Washington, D.C., to New York, N.Y., route at speeds up to 160 mph. Vought Division of LTV Aerospace Corporation (LTV) in association with Swiss Industrial Company (SIG) won the contract. The second phase confirmed the results of the first phase studies and showed that hardware meeting first-phase design requirements produced predicted improvements in ride quality. The LTV/SIG truck program helped to evaluate several innovative concepts. The prototype trucks met the basic objectives, but proved to be difficult to maintain and too expensive.

Therefore, a third phase was initiated to modify existing Metroliner trucks by adding airspring secondary suspension and damping similar to the concept developed on the LTV/SIG phase while meeting the performance characteristics established during the first phase. General Steel Industries (GSI) was awarded a contract to develop this alternate approach by modifying the original GSI trucks installed on the Metroliner cars. The modified trucks were expected to achieve the same ride quality improvement as achieved by the LTV/SIG prototype trucks, except that the maximum speed requirement was reduced from 160 to 130 mph. A running test and an accelerated proof-of-principle test was completed during the period May-July of this year. Preliminary analysis shows that the modified trucks do, in

---

M. Clifford Gannett is Chief, Passenger Equipment Division Office of Passenger Systems. He is a Graduate BSME, from the University of Michigan.

fact, improve ride quality up to speeds of 130 mph. A preliminary evaluation of the changes made to the trucks indicates that the modified GSI trucks will be less expensive and easier to maintain.

A concept evaluation program is being sponsored by FRA to evaluate the performance of radial trucks for high-speed passenger service. A pair of radial trucks is to be designed by General Steel Industries under subcontract to Amtrak. The radial truck concept will be evaluated by an extensive FRA test and analysis program. The radial trucks are scheduled to be manufactured in the next 12 months and the tests are scheduled to be completed in the following 15 months.

PHASE I DESIGN STUDY

The truck-design study performed by the Budd Company with James Herring as principal investigator provided the basis for the truck improvement program and is the element which ties the various projects together.

The essential part of the study was the development of a dynamic model of the Metroliner car and its truck subsystems. Independent linear models of the lateral and vertical dynamics were developed; the linear models were coupled to represent the combined car/truck system driven by simulated track inputs. The model simulation was verified by comparison with actual shaker tests performed at the Budd Company Laboratories and Battelle Memorial Institute. The laboratory results were analyzed and compared to results obtained from the dynamic model to help refine and confirm the model. Over-the-road tests were performed by ENSCO, Inc. and the test results were used to characterize track inputs. After the model was designed, simulation was used to evaluate and develop possible methods for improving overall ride performance by modifications to existing trucks or design of new trucks.

Modifications to existing trucks, recommended as a result of the study, were limited to making changes to truck secondary suspension and adding damping to the anchor rods, the vertical primary and the secondary suspension. The modifications were restricted for the following reasons: (a) Controlled modification of the lateral primary suspension is difficult to accomplish because of the existing journal box construction; (b) Increasing vertical stiffness would increase wheel-to-rail dynamic forces; (c) Decreasing stiffness would increase car body roll. However, the study indicated that significant improvement could be achieved if the constraints described in (a), and (c) above were eliminated.

The studies of new truck designs indicated that vertical truck frequency should be made as low as practical provided stabilizing devices are added to prevent excessive car body roll. The simulation also indicated that lateral truck frequency should be as low as practical. However, lateral motion of the car body and dynamic stability restrict the reduction of lateral stiffness. Figures 1 and 2 illustrate the performance of a standard Metroliner car/truck system as compared to the predicted improvement in performance of a low frequency truck.

PHASE II - LTV/SIG TRUCKS

The second part of the Metroliner Truck Development Program called for fabrication and testing of experimental prototype trucks conforming to the specifications established by the Budd Company truck design study. LTV/SIG won the contract with proposed modifications to an experimental truck built by SIG for Swiss Federal Railway. The LTV/SIG team did an extensive design analysis which is well documented in a three-volume report (FRA 76250). The LTV/SIG truck is shown in figure 3.

The basic characteristics of the design are:

(a) Improved ride quality (comfort).

(b) Either General Electric or Westinghouse traction motors can be used.

(c) Maximum operating speed with

existing Metroliner car is 160 mph.

(d) Braking is adequate to stop a Metroliner car from 130 mph without dynamic braking.

The basic design features shown in figure 3 are:

(a) H-frame construction connecting wheelsets and support motors and gear boxes.

(b) Sumiride airspring for secondary suspension.

(c) Airspring bolster is supported by four rollers located in the H-frame and is connected to the H-frame by a center pin with yaw dampers.

(d) Four bellcranks are used to connect the wheelsets to the frame and to provide primary suspension.

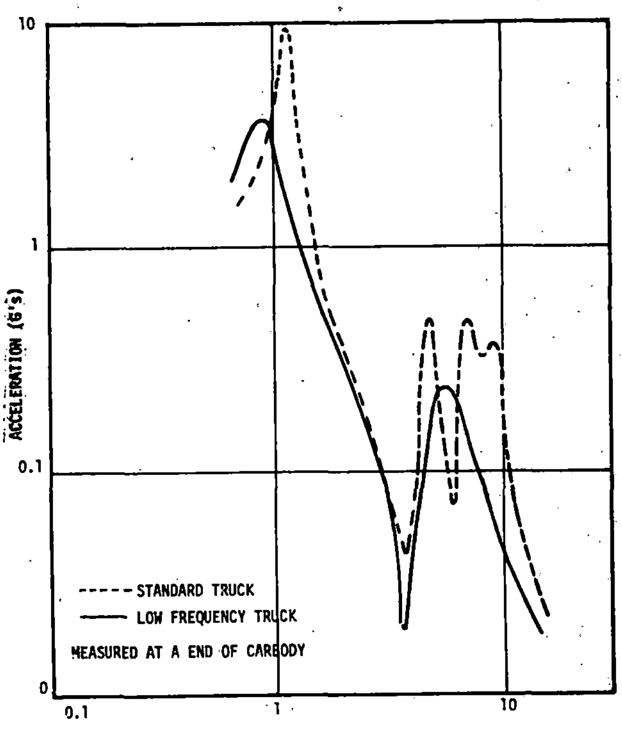


FIGURE 1. VERTICAL ACCELERATION DUE TO VERTICAL INPUT

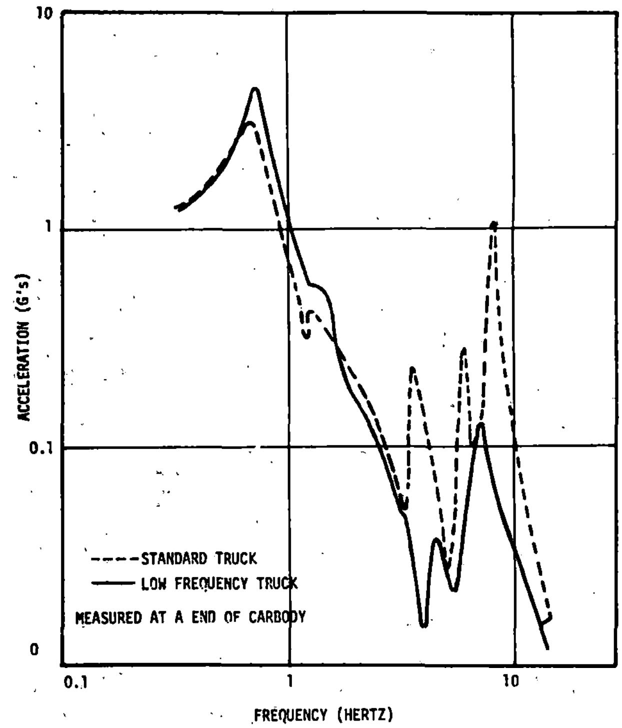


FIGURE 2. LATERAL ACCELERATION DUE TO LATERAL INPUT

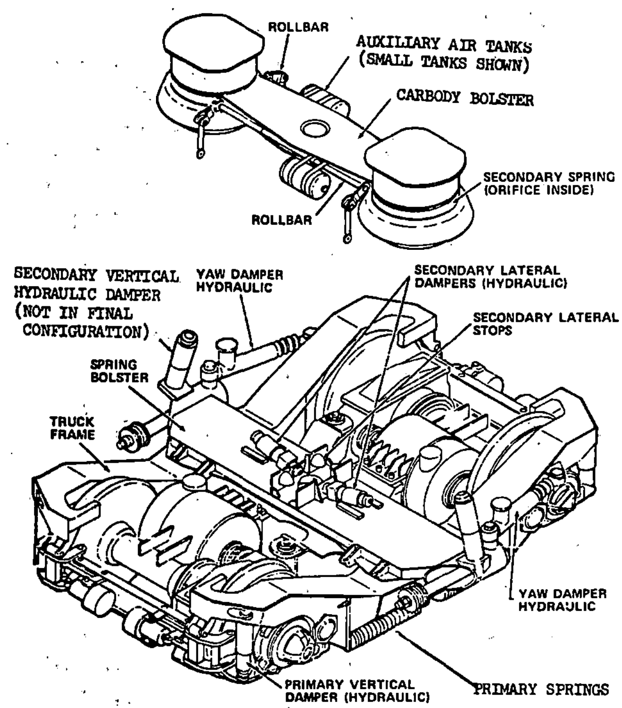


FIGURE 3. LTV/SIG TRUCK - PRIMARY AND SECONDARY SUSPENSION

Track suspension interacts with car body dynamics and is acted on by track inputs. Therefore, the selection of an optimum suspension configuration is a series of tradeoffs to select the best combinations of many conflicting factors. The two major criteria for the LTV/SIG design effort were safety and ride quality. The flow chart in figure 4 illustrates the LTV/SIG method for selecting the suspension configuration and analytically determining the optimum parameters.

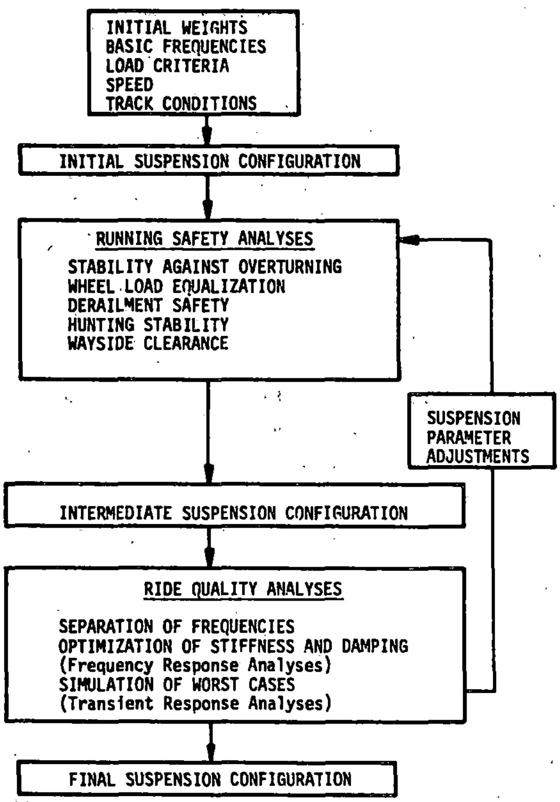


FIGURE 4. SUSPENSION ANALYSIS FLOW CHART

From the flow chart, it is evident that the analysis required extensive details of truck and car body characteristics, and track inputs. Analytical models were developed to determine car body overturning and wheel equalization forces, to describe lateral truck dynamics, and the interaction between wheel and rail. The analysis determined hunting stability and the optimum profile for Metroliner trucks.

Frequency response analysis, similar to that performed in the Budd Company design study was performed to evaluate variations in the suspension parameters and to evaluate the dynamic performance of the vehicle. Wayside clearance studies were performed to confirm adherence to clearance requirement.

After fabrication and fatigue testing, the prototype trucks were tested on a Metroliner snackbar coach (Car No. 850). Extensive shop, running and fatigue tests were performed. The trucks met all non-running test specifications. After they were tuned for optimum performance, the LTV/SIG trucks showed a modest improvement in ride quality as compared to standard Metroliner trucks.

PHASE III - GSI MODIFIED TRUCKS

The basic objective of this phase was to upgrade ride quality (comfort) of Metroliner service on existing Northeast Corridor track by improving existing trucks. This approach was evaluated because of the high cost of replacing existing trucks. The program was planned to use major elements of existing trucks and to improve the vertical and lateral ride of a Metroliner car as well as to make noticeable reductions in noise level. A modified truck is illustrated in figure 5.

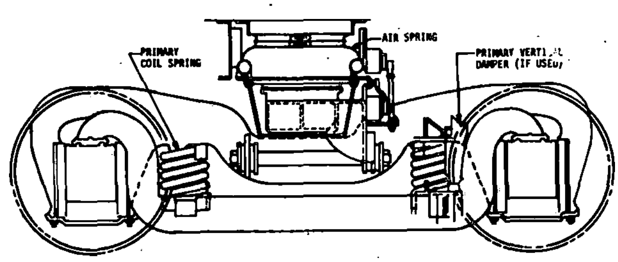


FIGURE 5. GSI MODIFIED METROLINER TRUCK

Major changes in truck suspension are:

(a) The elastomers with imbedded-steel-coil type primary springs are replaced with steel coil springs.

(b) Provisions are made for

adding friction snubbers or hydraulic dampers to truck primary suspension.

(c) Secondary springs are replaced by air springs with built-in orifices for damping.

(d) The standard bolster is replaced with a special bolster having an internal chamber to serve as an air reservoir for the secondary air suspension.

(e) The secondary vertical and lateral dampers are modified to make damping adjustable so as to achieve optimum ride comfort.

The GSI Metroliner phase was accelerated to demonstrate proof of principle by an early test. The full improvement phase, including additional fatigue and running tests, was scheduled to follow the accelerated test and to be subject to the results of the accelerated test phase. Shop and running tests similar to those run on the LTV/SIG trucks were run on the GSI prototype trucks. Tests were performed on two different airsprings. The primary design used the Sumiride airsprings similar to the LTV/SIG truck, while the other design used Firestone airsprings. The Sumiride air spring has a large diameter and requires that the car bolster be modified to accommodate it. The Firestone air springs are much smaller and can be fitted to the existing carbody bolster without modification, thereby reducing the cost of the overall modification. The preliminary test results indicated that the modified GSI trucks could produce modest improvement in ride quality when the Sumiride spring is used. However, ride quality using the smaller Firestone spring was virtually the same as the performance of a well-maintained existing truck. The Sumiride and Firestone modified trucks appear to promise less required maintenance than existing trucks. Therefore, the modified design appears to provide a net improvement in ride quality by maintaining the good ride for longer periods of operation.

PROGRAM STATUS

The Metroliner Truck Improvement Program is only one of the truck development programs sponsored and managed by FRA (OPS). Testing on the accelerated GSI Metroliner phase has just been completed and the results of same are not yet available. Both the LTV/SIG and GSI phases confirm the Budd Company study and indicate that ride quality can be improved. Quality improvement of vertical ride was greater than that obtained for lateral ride. The program indicates that the simpler GSI modified truck approach is capable of producing improved ride quality at speeds up to 130 mph and that the modified trucks will be easier and less costly to maintain than a more complex LTV/SIG design.

Both the LTV/SIG and GSI tests tend to indicate that track quality is the dominant factor in controlling ride quality. Equipment maintenance also plays an important role. When the existing Metroliner is in good condition, it rides reasonably well except for lurching produced by deviations in track alignment. While lurching irritates passengers and spills drinks, passenger fatigue is caused primarily by steady, low-frequency, dynamic exposure. None of the proposed changes eliminate lurching, but both the LTV/SIG and the GSI trucks with Sumiride air springs demonstrated improvement in the damping of the steady motion which contributes to passenger fatigue. A comparison of dynamic performance of the modified GSI truck versus the standard Metroliner truck is shown in figure 6.

The carbody requires extensive modification before it will accept the modified GSI trucks. Preliminary results are in the process of being analyzed. Analysis to date indicates that both the LTV/SIG trucks and the GSI modified trucks with Sumiride air springs produce equivalent improvement in ride quality (comfort). Therefore, overall cost must be evaluated before a final approach is definitized.

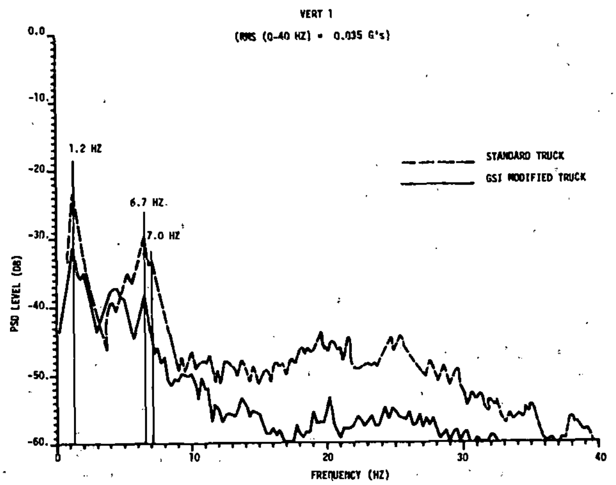


FIGURE 6. POWER SPECTRAL DENSITY VERSUS FREQUENCY -- STANDARD METROLINER TRUCK AND GSI MODIFIED TRUCK

RADIAL TRUCK PROGRAM

In addition to the truck improvement programs described in preceding sections, FRA has joined with Amtrak and private industry to pursue promising advanced systems. Present programs include evaluation of equipment designed and built outside the United States, developmental testing of radial truck concepts and development of new electrical propulsion systems. The propulsion system projects are also described in this paper.

The radial truck program is a concept evaluation program. These concepts are currently being used by Canadian National Railroad and South African Railroad in freight applications. Research has indicated that the good characteristics of the radial truck can be advantageously applied to high-speed passenger service. Under cooperative agreements with Amtrak, the radial truck concept is being evaluated; FRA (OPS) is responsible for planning and conducting a test program to evaluate the concept and its potential application to high-speed passenger service.

The radial truck concept has been around for a hundred years. A standard wheelset can negotiate a curved section of track without sliding only when the axle can assume a radial alignment with the truck as illustrated in figure 7. The distance traveled by each wheel must correspond to the length of the curved segment of rail; this requires that the wheels have a large conical taper (figure 7). To maintain high-speed stability with wheels having the large taper, the axle sets of the truck must be interconnected as illustrated in figure 7. Mr. Scheffel of the South African Railroad has demonstrated high-speed performance using a wheelset having greater than .5 to 1 effective taper. The claimed advantages of the radial truck are as follows:

(a) Less wear on wheel tread and flanges.
(b) Increased high-speed running stability.
(c) Better ride quality.
(d) Less rail wear.
(e) Less component wear.
(f) Lower noise level.
(g) Reduced traction power.
(h) Fewer derailments.

Established truck designers do not agree on the feasibility of the claimed advantages. The radial truck program will evaluate these claims and help to determine whether the concept currently being used in freight service can be applied to high-speed passenger service.

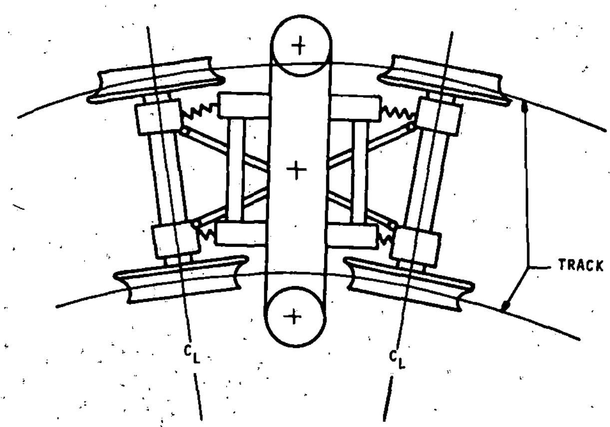


FIGURE 7. RADIAL TRUCK CONCEPT

ELECTRICAL PROPULSION R&D

BACKGROUND AND INFORMATION

Starting with the development of powerful, silent and pollution-free propulsion systems for levitated vehicles, the FRA has vigorously sponsored R&D related to electrical propulsion systems. The most successful demonstration of a high-power linear induction motor (LIM) was made at the Transportation Test Center using the Linear Induction Motor Research Vehicle (LIMRV) shown in figure 8. The LIMRV tests produced some notable results: a troublesome end effect in linear induction motors was evaluated and found tolerable; the use of a variable-voltage, variable-frequency power supply for an AC motor was successfully demonstrated; and a world speed record was set for steel wheel on rail vehicle (411Km/hr). Additional accomplishments related to linear induction motors included the advanced-design Tracked Levitated Research Vehicle (TLRV) LIM and its associated electronic power conditioning unit (inverter). This work resulted in advances in the state-of-the-art of high-power-density, liquid-cooled electrical machines and inverters.

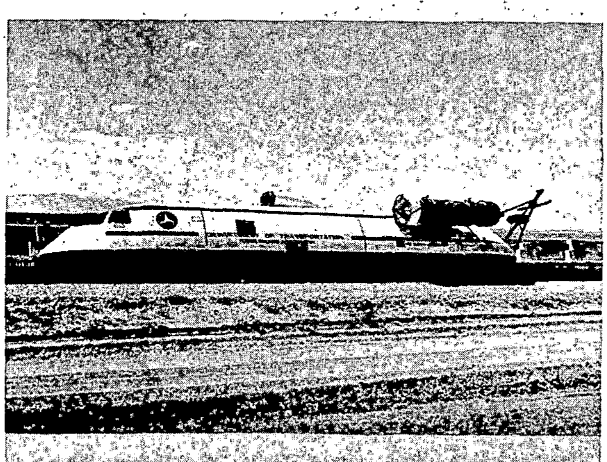


FIGURE 8. LINEAR INDUCTION MOTOR RESEARCH VEHICLE (LIMRV)

The initial government effort on linear motors goes back to 1965, the year in which The High Speed Ground Transportation Act was passed. In 1975, the emphasis on advanced systems changed. This resulted in a phasing out of the TLRV program and a redirection of the LIMRV program to 200 km/hr wheel on rail applications. The original doublesided LIM used on the LIMRV straddled a vertical reaction rail. This configuration is not suitable for rail applications, and thus work was shifted to single-sided LIMs (SLIM) utilizing a horizontal reaction rail having its top surface flush with the top of the running rails. Another new direction was the consideration of single-sided linear synchronous motors. Based on an initial suggestion made by Matthew Guarino in 1967, a feasibility study of the concept was carried out at Polytechnic Institute of New York (PINY). This study showed several types of synchronous motors to be major competitors to induction motors. Although they require a somewhat more complex reaction rail than induction motors, synchronous motors of equal output appear capable of operation with power converters having only half the volt-ampere rating.

The FRA linear motor work has thus evolved in orientation from application to advanced systems to railroading, and there has been an initiation of studies having immediate benefits to railroads. Some potentially important spinoffs of linear motor R&D have already been identified. These include high-performance, contactless, electromagnetic brakes and the use of linear motors to propel and brake cars in a classification yard.

Rotary motors rather than linear motors are likely to propel 200 km/hr trains of the future. From a theoretical point of view, rotary motors are a subclass of linear motors, and researchers steeped in knowledge of linear motor systems are in a good position to make major advances in rotary motor drive systems. Rotary drive systems of the future will use lightweight and compact induction or synchronous motors of rugged and simple construction. These must be powered by solid-state inverters, preferably of the variable voltage variable-frequency type.

The FRA Office of R&D deals not

only with advanced electrical propulsion systems, but also with problems related to railroad traction that are national in scope. This includes matters relating to public safety, health, and well being, such as the use of hazardous substances in electrical equipment, energy conservation, and electromagnetic interference resulting possibly from modern electric locomotives. Another aspect of the FRA's R&D efforts to increase railroad efficiency is to recommend equipment offering improved service with a minimization of first and maintenance costs. The FRA has an agreement of understanding with Amtrak to help them secure the best possible equipment by providing technical advice, product information, and help with specifications.

REVIEW OF PRESENT PROGRAMS

There are two major industry-based linear motor programs. The LIMRV work is being done by AiResearch Manufacturing Company, while exploratory development of linear synchronous and induction motors is being performed by the General Electric Company at their Corporate Research and Development facility in Schenectady, New York. Engineering support to FRA on both these programs is being provided by Transportation Systems Center (TSC) and MITRE/METREK. Selected support is being furnished by PINY and the Jet Propulsion Laboratory (JPL).

AiResearch recently completed tests with a modified version of the double-sided LIM using the vehicle and the vertical reaction rail shown in figure 8. Data was obtained for LIM operation with a varying number of poles (out of 10) excited. This was to gain a better understanding of end-effect phenomena in the motors. The vehicle and half the present motor are now being modified to permit testing a SLIM of the type depicted in figure 9. The reaction rail consists of .16" (4.1 mm) of 6061 aluminum laid on top of .88" (22 mm) of solid back iron. Thrust data similar to those obtained with the double-sided LIM are expected, except for effects produced by eddy currents and saturation in the back iron.

Another important consideration, and perhaps the most important, is that the SLIM structure produces a substantial normal force on the reaction rail. This force can be attractive or repulsive, depending on vehicle speed and the excitation frequency of the motor windings. Also, saturation effects in the back iron may have a more pronounced effect on normal force than on thrust. The data produced by the testing will be the first with such a large machine and will reveal information which is extremely difficult to obtain by analytical means. The performance of double-sided LIMs can be predicted reasonably well using a number of published theories, but the same cannot be said for prediction of normal and lateral forces in a SLIM. The normal force, especially, may significantly influence vehicle dynamics.

Experimental work on SLIMs is also being done at General Electric using the rotating wheel shown in figure 10. The wheel has a diameter of 54" (1.4 m) and is capable of tangential speeds to 400 km/hr. The wheel facility is well instrumented and a microprocessor is used to permit rapid and accurate acquisition of data. The 112 kW output SLIM being tested at GE has only 4 poles compared to the 10 in the megawatt-rated LIMRV/SLIM. The GE SLIM has a severe end effect, but nevertheless very useful for gaining an understanding of how SLIMs function. A solid iron wheel with an eighth inch (3.2 mm) band of aluminum forms the reaction rail. The tests with this configuration are almost complete and testing without the aluminum band will begin soon. These tests will yield information that bears directly upon the operation of a SLIM over conventional running rail. A possible outcome of these tests is the eventual realization of improved non-contacting linear brakes for use of passenger trains.

The SLIM testing at GE complements both the LIMRV/SLIM testing and the exploratory development of the linear homopolar inductor synchronous motor. The type of motor soon to be tested by GE is depicted in figure 11. The AC windings at both sides of the motor each resemble the windings in a SLIM.

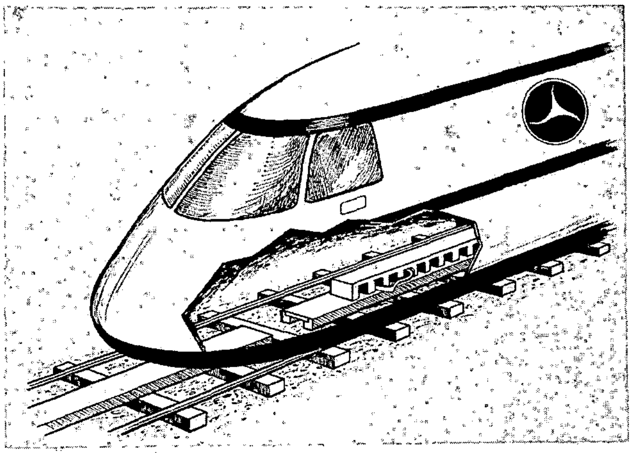


FIGURE 9. SINGLE-SIDED LINEAR INDUCTION MOTOR (SLIM) MOUNTED IN LIMRV

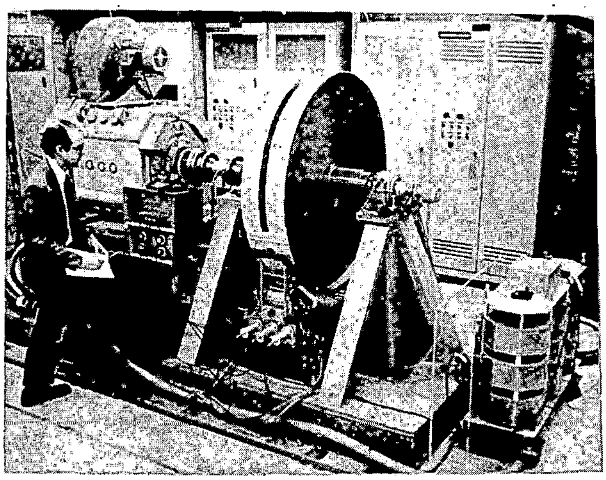


FIGURE 10. ROTATING TEST WHEEL AT GENERAL ELECTRIC COMPANY

Loosely speaking, the apparatus consists of two SLIM-like AC winding sets and a DC excited field winding. The field winding acting in unison with the iron track bars produces an undirectional (homopolar) undulating magnetic field. In terms of an observer fixed to the track, the DC field winding produces what appears to be a set of alternating north and south poles plus a bias field. The excitation of AC windings is such as to produce a traveling magnetic wave that is synchronized to the forward motion of the vehicle (stator). An observer on the track sees the magnetic field produced by the AC windings as stationary. This is the result of having a backward running (with respect to the stator) magnetic wave pulled forward mechanically such as to create a zero wave velocity with respect to the track. The net result of the AC and DC produced magnetic fields is that each produces a set of alternating north and south poles. The interaction of these sets of poles results in a thrust. Other results of this interaction are a strong attractive force and the possibility of obtaining a significant lateral guidance force.

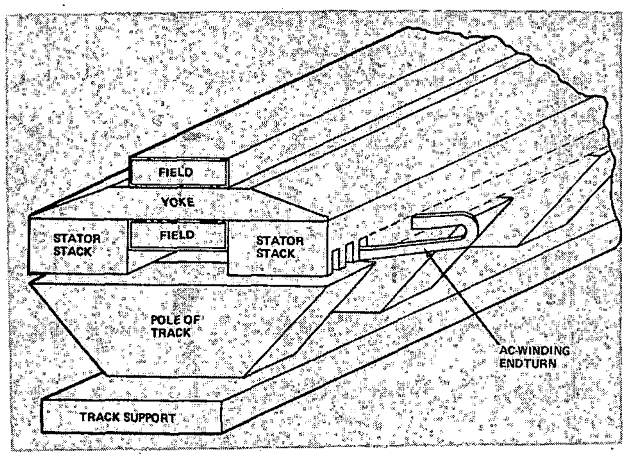


FIGURE 11. HOMOPOLAR INDUCTOR LINEAR SYNCHRONOUS MOTOR

The inductor motor is an important technology compared with SLIMs because it permits operation with larger air gaps and significantly better power factor. Factors going against the synchronous motor are that it is a bit more difficult to control than a SLIM and that the reaction rail is slightly more complex. Its advantages are primarily the result of its being able to produce leading volt-amperes (a SLIM cannot do this). A typical product of efficiency and power factor for a well designed SLIM is in the order of .4 - .5, whereas .8 - .9 is applicable to the inductor motor. This translates into the inductor motor requiring an inverter only half the size of that for a SLIM. Also, it is expected that control of normal force in an inductor motor is easier to accomplish than with a SLIM.

The advantages of inductor motors, and perhaps some unforeseen problems, will soon reveal themselves when a 112 kW power output model is tested on the rotating wheel. Test results will be compared both to predictions and to results obtained with a SLIM having an equal power rating. Also, such comparisons will be made on the basis of calculations for full-scale motors. Such an extrapolation from model tests to predictions of full-scale motor performance is not done routinely. General Electric will perform such an extrapolation for two systems to be specified by the FRA. Both PINY and MITRE/METREK will then assist the FRA in judging the GE assessment. After this, a decision will be reached by FRA as to the advanced developed potential of the motors.

Linear motor systems were initially considered exclusively because of a mandate to explore high-speed advanced systems. The emphasis has now changed to propulsion systems for trains traveling less than 250 km/hr. To make an orderly transition from advanced systems research, some programs were dropped and others redirected. Linear motor programs were redirected because of a conviction that this is an important new technology having significant benefits to railroading. Identifying and studying these benefits is now underway. Soon a report will be available that characterizes linear motors in such practical terms as typical force, weight, and power densities. These performance estimates will be based upon experimental results and refined calculations made and verified by numerous researchers. Potential applications of linear motors in railroading will then be studied in some detail. Finally, a site-specific study of one of the more promising applications will be made.

In addition to the linear motor work there is substantial FRA activity related to rotary motor drive systems, electrification equipment and problems, and the providing of test facilities for use by the railroads and industry at TTC. Substantial effort is being devoted at JPL to gathering on a worldwide basis, information related to modern electric traction equipment. This information will form a basis for judging the state-of-the-art and help FRA in deciding upon future courses of action related to new types of electrical traction equipment for use in the United States. Besides JPL, TSC is actively assisting FRA in such areas as electric drive systems, inverters, and linear motor brake systems.

PLANS FOR ELECTRIC PROPULSION R&D

The FRA involvement in electric propulsion R&D relates to those aspects of this subject where private initiative is either lacking, insufficient, or inappropriate. The linear motor programs are examples of risky technical ventures that could result in major advances in electric propulsion technology. Linear induction motors have already been identified as having potential for braking of high-speed passenger trains and as a booster-retarder in classification yards. As a result of FRA sponsorship of linear motor work, sufficient knowledge will exist to permit application of the technology without undue technical risk. Also, the linear motor programs have led to substantial technical innovations applicable to both linear and rotary motor propulsion systems. These include advanced methods for cooling motors and inverters, techniques for building high-power and high-voltage motors and inverters, and advancement of the state-of-the-art of inductor synchronous motors. The inductor motor in its rotary embodiment, for example, has been identified as a superior device for driving and being driven by the flywheel in an energy storage system.

The braking of high-speed trains using linear-motor type devices has been explored in some detail in France, Germany, and Japan. None of these systems are operational, and furthermore, powering of these systems on locomotive-propelled trains presents serious difficulties. As a result of FRA involvement, a new and low-cost technique has been found for powering the linear motor devices. Also, the wheel tests being conducted at General Electric are expected to help identify design problems related to the electromagnetic structures. Another aspect of FRA involvement in linear brakes has been to sponsor an applications-oriented study of linear brakes which was performed by A. T. Kearney, Inc.

The FRA has a strong interest in the operation of and improvement of thyristor-controlled all-electric locomotives using DC traction motors. Such matters as power factor and interference with signaling systems are of concern. Because of limitations on catenary design and operating voltage, getting sufficient power to a large train on the Northeast Corridor can present difficulties unless power conditioning equipment of advanced design is used. Another area of interest relating to electrification is the power distribution system, including the catenary-pantograph system.

The technology of the future for the propulsion of 250 km/hr passenger as well as freight trains is expected to be solid-state power converters coupled to AC motors of either the induction or synchronous type. Rotary motors for such applications are fairly well developed with the exception that the power density must be improved even further to reduce unsprung mass. The power converter is the really new ingredient in AC drive systems. These converters have become a realization because of major advances in high-power semiconductor device technology.

AC drive systems have been known for some time to be very desirable. However, no such systems are operational in the U.S. There are many reasons for this, such as cost, size, reliability, etc. The railroad environment and operational needs require that AC propulsion systems have affordable first costs and low operational costs because of superior performance, reliability, and serviceability. Development of these systems will be encouraged and facilities will be made available at TTC for testing the systems to verify that they can truly serve the railroads. One such facility will be the fully electrified 14-mile Railroad Test Track, which is scheduled for completion by 1979.

REFERENCES

1. Kalman, G. et al., "Linear Induction Motor Propulsion System", Ai-Research Manufacturing Company of California, report no. FRA/ORD-76/305, contract no. DOT-FR-40016; Dec. 1976.

2. Office of Research and Development, FRA, "Tenth and Final Report on the High Speed Ground Transportation Act of 1965", report no. FRA/ORD-77/27; May 1977.

3. Levi, E. "A Preliminary Evaluation of Electrical Propulsion by Means of Iron-Core Synchronously Operating Linear Motors", Polytechnic Institute of New York, report no. FRA/ORD-76/128, contract no. DOT-FR-30030; Jan. 1975.

4. Macie, T. W. "Selective Bibliography of World Literature on Electric Traction for Railroads (1970 - 1975 period)", Jet Propulsion Laboratory, report no. FRA/ORD 76-296, contract no. DOT-FR-30006 Amendment 4; Nov. 1976.

QUESTIONS SESSION II

Session Chairman -- M. B. Mitchell

Attendee: Frank Dean, Battelle Columbus Laboratories.
Attendee's Question: Cliff, you mentioned four bases for ride quality comparison that you used in the follow-on Metroliner evaluation program. I saw RMS acceleration, ISO, WZ factor, what was the fourth and of the four, which do you like best? Which do you think is the most meaningful?

C. Gannett: We measured exceedances to performance level. In other words, we were looking at the RMS and the exceedance level as peak acceleration performance and the ISO and the WZ to give us a more continuous averaging effect for ride quality. I really prefer the RMS which I think reflects what the passenger feels, but during his entire trip the averaging of the ISO and some of the others is worth doing, but I don't think it's as effective as RMS.

Attendee: R. W. Pinnes, Department of Transportation.
Attendee's Question: Based on the work you've done so far in electrification, what sort of a scenario would you project for the future of electrification in the United States?

R. Novotny: I have reacted like a pendulum over the past year and a half on how I feel about railroad electrification. I've swung from an extreme that says it's the best thing so far as a nation is concerned, to the other side that says it's really good for the railroad, and it's for their benefit. And, I'm straddling that fence right now, Bob. I think that it looks like the front-end money is so large and the risks are so involved that a railroad, even a railroad in good financial condition and the ones that have seriously studied it are in good financial condition, wouldn't undertake the investment. At the same time, the Government hasn't had its real crack at their position with regard to electrification. We're down at the level here in FRA where we're publishing findings and then it's going to work its way up through the Department into OMB and to Congress. And, we're working with the Energy Department too; there are energy implications, but we need a national viewpoint before you can get a balanced answer for that question.

M. Mitchell: If I may Mr. Pinnes, I would like to add my comment to those of Mr. Novotny's. As we conducted the study we found that 20 years represented a reasonable period of time for estimating the return on investment. Unfortunately, we have not found the crystal ball, ouija board or what have you, which enables us to look that far into the future. One cannot commit himself halfway on electrification. It is a major decision and we do not have sufficient answers at this time to make a final determination.

Attendee: B. Jeffcoat, Analytic Sciences Corp.
Attendee's Question: This is either for Mr. Novotny or Mr. Gannett. You mentioned the radial truck program. I wonder if you'd started looking at any of the other unconventional or radical truck designs?

C. Gannett: I'm not sure that I know of any other. We certainly are open to looking at all type trucks. Radials are the ones we know have been applied to freight cars but have never been tested in passenger car use.

B. Jeffcoat: Well, I know there's some odd designs abroad. There are scissors action trucks.

C. Gannett: Yes, there are trucks that are similar to SIG that are rigid design but most of them provide variations of the SIG features. I thought you were asking about something more advanced.

B. Jeffcoat: Well, I'll proceed on to the other question, which is how about active truck designs and some of the more unusual passive designs with the self-steering mechanisms and the like.

C. Gannett: We do have a university research contract looking at computer modeling to try to determine what is the ultimate capability of a passive truck, and how we might improve ride with an active truck. There has been very little work done with active trucks. LTV has a small unit that they've experimented with and as far as I know that's it, other than sort of semiactive things where you're regulating dampers and that sort of thing, there hasn't been a great deal of work done with active systems.

M. Mitchell: I think it will help for people to understand our interest in radial axle trucks. We are all aware of the four advantages of low wheel wear, low rail wear, less noise on curves and improved high speed stability. Since we are looking toward an increase in the speed of rail passenger equipment, we are mainly interested in the high speed stability on tangent track. We are presently in the conceptual phase of our program. If we are successful through this phase, we will broaden the scope of the program to include other applicable design concepts.

Attendee: N. Cooperrider, Arizona State University.
Attendee's Question: In your passenger equipment evaluation of European equipment, you mentioned you're including or taking a look at some of the economic factors concerned with the operation of this equipment. Do you, in that evaluation, include some of the costs of maintaining the roadbed which might differ with different kinds of equipment or are you considering the roadbed to be a fixed matter in that evaluation.

R. Novotny: The answer to that question is we do not include the maintenance cost of roadbed damage as a result of operating different equipment over it. I don't think we know enough about the wheel rail interaction to predict how it's going to effect damage to the roadbed and then the maintainability costs that would be associated with that. It's an area that we're operating in the fringes on. We know that the Europeans restrict their axle loads for the express purpose of reducing roadbed damage. But we're not accounting for that facet in the cost analysis that we're doing.

SESSION III THE STATUS OF RAIL SAFETY RESEARCH

HIGHLIGHTS RAIL SAFETY RESEARCH

By

Leavitt Peterson

EXECUTIVE SUMMARY

The FRA's Office of Rail Safety Research, together with similar industry organizations, exists because of perceived safety improvement needs. These needs, documented by safety statistics, as well as extensive interactions with railroad/union safety interest groups drive the research and development efforts. The office is organized to provide technical outputs in three major areas which encompasses the known R&D needs. The three divisions are Rail Vehicle Safety, Improved Track Structures and Inspection and Test Support. Rail Vehicle Safety is charged with conducting research for safety improvements associated with Rolling Stock, Human Factors and Accident Avoidance. The Track division has R&D responsibility for reducing the number of track caused accidents and improving track performance. The third division, Inspection, places research emphasis on track and vehicle inspection systems, safety life cycle methodology and the development of automated track inspection vehicles both as a research tool and for use by the Office of Safety. This report presents a panorama of research programs and projects as conducted by the Office Rail Safety Research within the framework of openness and cooperation with the railroad community.

ADVANCES IN RAIL FLAW DETECTION

BY

H. L. Ceccon

Executive Summary

In 1974 the Federal Railroad Administration (FRA) initiated a project to improve the rail flaw detection technology. The objective of the project is to improve the speed and sensitivity of current systems with long range goals of 25 mph operation for hy-rail systems and 40 to 50 mph for rail-bound sytems. The effort has concentrated on making improvements in three general areas; 1) ultrasonic sensor performance, 2) automatic controls and, 3) data processing. The sensor improvement centered around improving detectability primarily by adding transducers to conventional configurations. The development of automatic controls was directed toward automating functions which operators must perform on conventional systems. As a result, devices were developed to automatically maintain sensor position over the rail head and circuitry designed to maintain proper amplifier gain and signal gate positions.

Work to develop data processing equipment for specific modes of operation is underway. For hy-rail operation, a system is being developed which displays pertinent inspection data in a B-scan format on a T.V. monitor for operator interpretation. The development work on data processing systems for high speed rail-bound vehicles is directed toward a system which requires minimum operator interaction and provides real-time flaw classification.

Two systems are currently operational in the field which incorporate some of the devices described above. These systems are: 1) a hy-rail system with 15 mph inspection speed and 2) a railbound system with a maximum inspection speed of 25 mph.

SIGNIFICANT DEVELOPMENTS IN FRA TEST CARS, AS EXEMPLIFIED BY T-6

BY

Ta-Lun Yang

Executive Summary

The FRA has assembled seven railroad test cars since 1966; they are identified as T1 through T7. The T1/T3 consist served since 1973 as a track inspection car as well as providing a test bed for instrumentation development. Two more track inspection consists have been introduced during FY 1977 to augment the track measurement capability. The T2/T4 consist was equipped with track geometry measurement instrumentation in November of 1976; the system installed is similar to the current configuration of T1/T3. The latest track inspection car T6 was recently completed. The T6 instrumentation incorporates the latest developments in track geometry measurement as well as the latest technology in ultrasonic rail flaw detection. The on-board data processing capability was also expanded to allow the generation of exception reports for both geometry and rail flaw in real time.

The T6 track geometry measurements include gage, crosslevel, profile, alignment and curvature. Speed, distance and track location are also measured for correlation between data and track locations. The method of measurement for each of the parameters is summarized below.

Gage: Two truck-mounted, servo-controlled magnetic sensors are used.

Crosslevel: The gravity vertical reference is maintained by a continuously compensated accelerometer system.

Profile: An inertial profilometer system is used for high speeds and a multichord system measures at low speeds.

Alignment: The combination of an inertial reference system and the magnetic gage sensors is used.

Curvature: An inertial yaw rate gyro is mounted on the carbody.

Speed-and-distance: The output from an axle-driven tachometer is used to calculate speed and integrated for distance.

The above measurements provide the capability to inspect all of the parameters identified in the track geometry secton of the FRA Track Safety Standards. The track geometry measurement system is all-weather; it can perform track inspection at track speeds up to 110 mph under locomotive power.

Track geometry sensor signals are processed by an on-board computer to formats consistent with the FRA Track Safety Standards. These parameters are then compared against the thresholds given in the standards according to the specified track class. Exceptions are listed by track location in a report while the test is underway; the report is available at the conclusion of the inspection survey.

Rail flaw inspection equipment in T6 can be operated separately or simultaneously with the track geometry instrumentation at test speeds to 25 mph.

Multi-ultrasonic sensors are used on each rail. Ultrasonic pulses are directed into the rails. The occurrence of the loss of specific pulse echoes indicate structural anomalies. Sensor signals are first analyzed by analog time-and-gate logic circuits. This results in binary-type diagnostics as to whether or not a flaw might be present. Programs implemented in micro-computers then analyze these successive binary diagnostics from each rail to detect and classify flaws. Information on detected flaws is sent to the on-board computer which in turn records the information and lists the flaws by track location in the on-line exception report.

The track geometry and rail flaw instrumentation installed on T6 and the real-time analysis capability provide an excellent example of the significant developments in track inspection technology accomplished under FRA-sponsored research efforts during the last few years.

EXPERIENCE AND APPLICATION OF A HIGH-RAIL VEHICLE FOR TRACK MEASUREMENT BY A STATE DOT

BY

M. Sherfy

Executive Summary

The Iowa Track Inspection Program is based on the use of a high-rail track inspection vehicle. A description of the Iowa Track Geometry Car and a discussion of its capabilities is presented. Iowa is the only state operating this kind of equipment and has gained considerable experience regarding data formats, crew requirements, and scheduling requirements. This experience in the form of operational objectives and policies is contained within the paper.

Automated track geometry measurement vehicles represent a means for rapid data collection which is unsurpassed in its objectivity. This objectivity allows Iowa to make railroad planning and maintenance decisions based on actual data instead of subjective judgments or assumptions. Presented within this paper are two objective applications of track geometry data as developed by the Iowa Department of Transportation.

A Track Inspection Priority Rating System was developed to focus the inspection effort to those tracks most likely to incur track problems. The thrust behind this system is to guide the scheduling of visual inspections. A statewide Track Geometry Rating system which provides a numerical rating of track conditions has also been developed. These ratings assist in the programming of State and Federal rehabilitation funds so that the maximum benefit may be realized from project investments.

Under a research contract sponsored by the Federal Railroad Administration the Iowa Department of Transportation will be expanding the application and interpretation of track geometry data. Questions regarding vehicle weight, seasonal variations in roadbed conditions and optimum inspection speeds are currently being resolved. Research of this nature will assist in the development of appropriate techniques, models and simulations for the analysis of railroads.

FEASIBILITY OF ROLLING STOCK PERFORMANCE DETECTION
VIA
AN INTEGRATED MODULAR WAYSIDE APPROACH

BY

J. D. Ferguson

Executive Summary

The Federal Government has initiated R&D projects to help find solutions which will aid in reducing the occurrence of rail-vehicle caused accidents. FRA efforts to support research and development on automated trackside performance detection equipment and teachniques are underway to increase fault detection efficiency and reliability, and to utilize a larger data sample for decision making. The results of these efforts will be made available to industry to aid them in their performance detection activities.

One part of FRA's efforts is to sponsor research and development studies on rolling stock performance detection via an integrated modular wayside approach. This approach involves integrating various types of detection devices and equipments to sense, analyze, and display degraded performance of rail vehicle and rail vehicle components. Essential to this approach is the establishment of a research capability. Such capability will be established at TTC and will provide FRA with a tool for studying various approaches to integrating and evaluating new and existing types of detection devices. The sensors and equipment utilized in these studies will be those which will have the greatest positive impact on accident statistics and economic considerations.

EVOLUTION OF THE CONCEPT & POTENTIAL OF A RESEARCH LOCOMOTIVE AND TRAIN HANDLING EVALUATOR

BY

John T. Wilson
DSL Dynamic Sciences Limited

Executive Summary

The primary purpose of the research locomotive and train handling evaluator is to facilitate the collection of new objective information describing the interaction of an engineman with his train. Historically, field observation with supervisory personnel on-board the locomotive, has provided the primary source of engineman performance evaluation. Crew reports have served to indicate the acceptability or effectiveness of any design innovations affecting display or control functions. Since 1970 a series of five noteworthy locomotive crew training simulators have been introduced in North America. These have provided a simulated environment with medium to good realism, and consequently another situation in which engineman-train interaction could be observed. Performance under controlled conditions has been monitored on simulators, but the evaluation of performance has remained entirely subjective. Although the training instructor has to a great extent replaced the road operations supervisor, the influence of the human observer on operating procedures has remained.

The research evaluator concept defines for the first time, a system offering a high degree of environmental realism, while simultaneously providing for "on-line" objective evaluation of engineman performance. The evaluator is much more than a simulator in that the evaluation function is implemented by special programmed logic devices. The evaluation process is to take into account biotechnical data indicating the condition of the engineman. Also considered will be the current state of the dynamic forces in his train, data indicating the stability of any critical vehicles, and the state of subsystems affecting the safety or efficiency of operation.

Unlike the training simulator which presents many repetitious runs of relatively short duration, the evaluator concept provides for long duration "fatigue" runs. The facility to closely control all major aspects of the engineman environment will permit the acquisition of statistically significant data, describing the performance of a "control group" of enginemen with regular locomotive equipment. The effectiveness of other enginemen under a variety of circumstances and with access to diverse equipment innovations can be objectively assessed with respect to the control group. The total evaluator facility would include systems for the preparation of experiments, modular equipment to emulate design innovations, and systems support gear.

It is the author's conviction that much can be gained through the early development of the research evaluator. While a better understanding of the engineman's task and normal performance will result, this in turn will yield improved safety of operations, better use of human resources and improved economics of operation.

HIGHLIGHTS OF RAIL SAFETY RESEARCH

BY

L. A. PETERSON

INTRODUCTION

This paper will briefly present some R&D highlights of the office of Rail Safety Research. By doing so, I hope to introduce you to the research goals, programs and interactive and cooperative methods we, in R&D employ in conducting research efforts.

NEED FOR SAFETY RESEARCH

The Office of Rail Safety Research in FRA, together with other similarly oriented industry organizations exist because of perceived safety improvement needs--which are documented in historical records. Such "statistics"

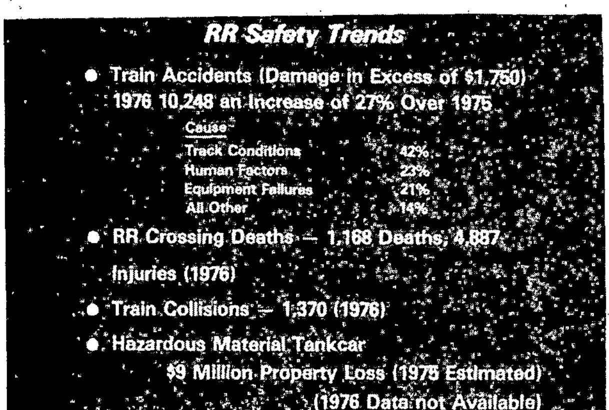


FIGURE 1. RAILROAD SAFETY TRENDS

as shown in figure 1, drive research and development efforts. Train Accidents, Railroad Crossing Deaths, Train Collisions and Tank Car Ruptures are highly visible evidence of serious problems. Fatalities, injuries, societal hazards and high costs are involved, and tend to compel some sort of remedial action--even though railroad safety as a whole may be comparatively better than other modes.

Further examination of typical records of fatalities and injuries, not necessarily restricted to railroad employees, confirm via figure 2,

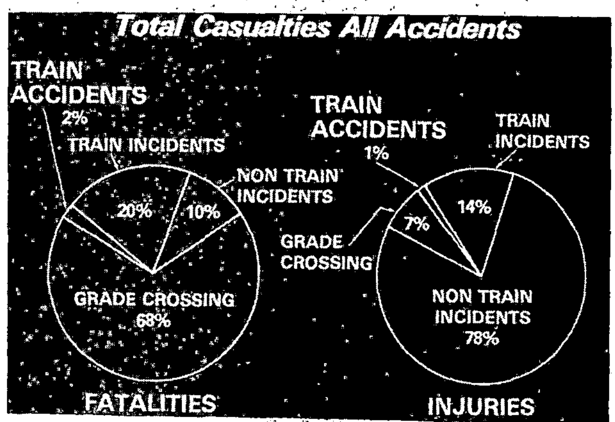


FIGURE 2. TOTAL TRAIN CASUALTIES

the relative importance of Grade Crossing accident reduction. Also the need to deal with train and non-train incidents (rather than the more spectacular accidents) is increasingly evident.

Similarly, as in figure 3, statistics limited to railroad employees confirm the importance of dealing with both train and non-train incidents in attempts to achieve fatality and injury reductions.

By presentation of statistics by another method of compilation (figure 4)--the major focus can be fixed on Transportation Department employees. At still a deeper level of examination, challenges for improving the safety of individual personnel work tasks are highlighted as in figure 5.

Leavitt A. Peterson is Director of the FRA's Office of Rail Safety Research. Leavitt holds B.S. and M.S. degrees in General and Industrial Engineering from the University of Illinois and Illinois Institute of Technology respectively. He is also a graduate of the Stanford University Transportation Management Program.

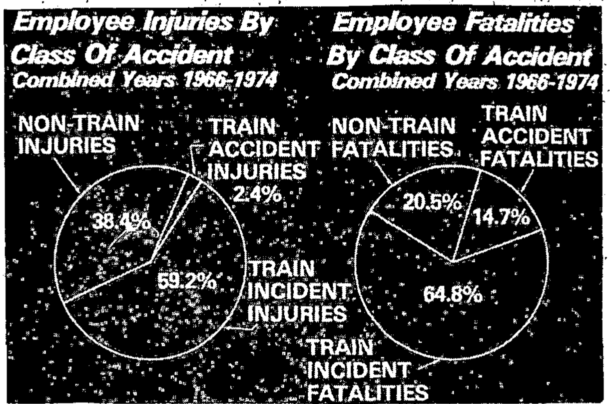


FIGURE 3. EMPLOYEE CASUALTIES

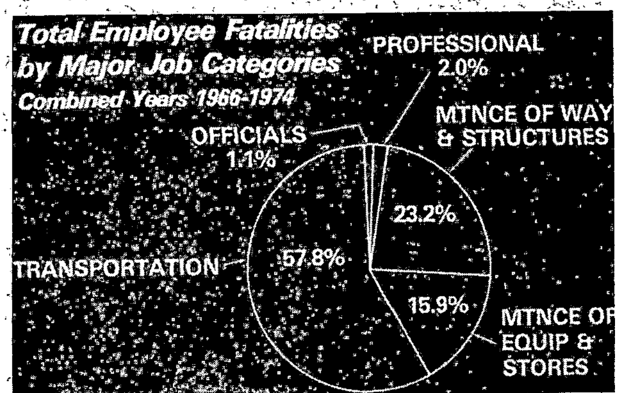


FIGURE 4
EMPLOYEE FATALITIES

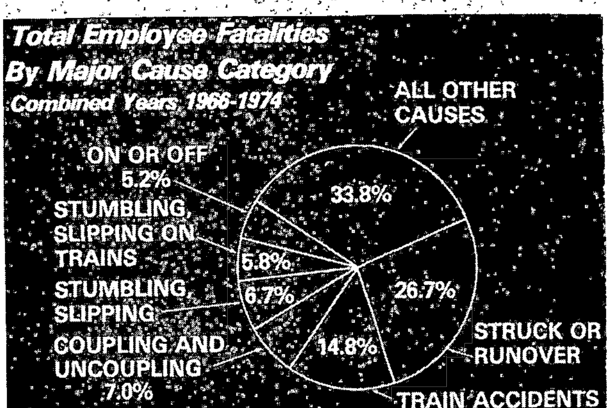


FIGURE 5
EMPLOYEE FATALITIES BY CAUSE

Being "struck or runover" is the most common cause category. (The "all other" listing is composed of many small items).

On the other hand, if direct costs (or damages) are considered, figure 6 illustrates that track caused accidents are foremost.

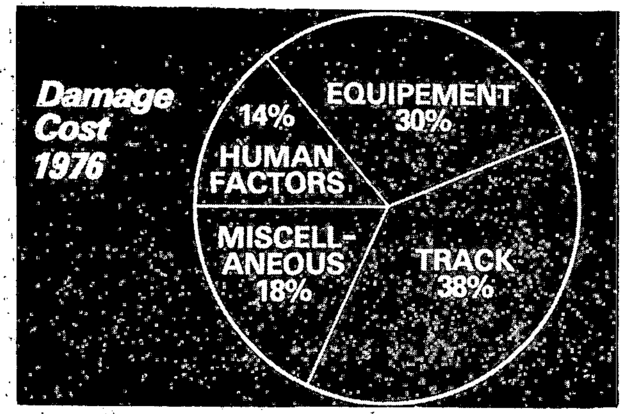


FIGURE 6.
DAMAGE COSTS

Incidentally, the results of figure 6 would be almost identical if the frequency of train accidents were shown.

A narrowed view of research needs can be obtained by examining the nature of accidents. For instance, in the Grade Crossing example of figure 7, it can be seen that over 15% of the time, the automobile hits the train rather than vice versa.

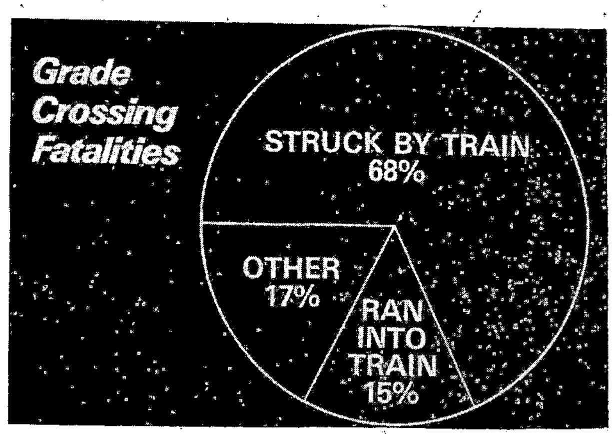


FIGURE 7.
CROSSING FATALITIES

One of the most useful inputs in guiding research priorities is an examination of the repeating and consistent leaders ranked by severity of consequences.

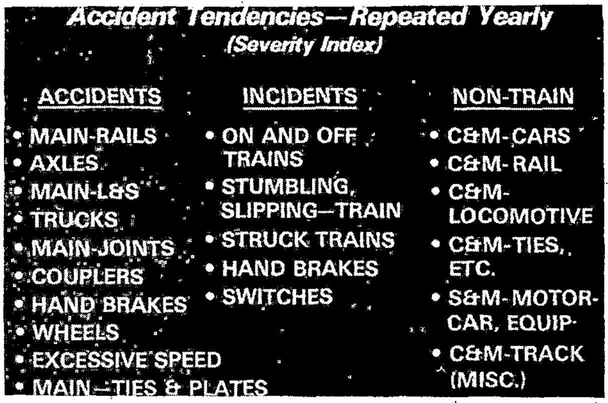


FIGURE 8. ACCIDENT TENDENCIES

Figure 8 shows the year-after-year "infection" of several "repeaters". "Main Line Rail Breakage", "Getting On and Off Trains" and "Construction and Maintenance Work on Cars" are the persistent casualty producers in the three categories shown. In summary (figure 9), a variety of measures may be employed in guiding research thrusts.

FIGURE 9. STATISTICAL MEASURES

While all deserve a measure of attention in addressing research needs, we are now becoming convinced that the fatality rate is a prime target for increasing amounts of Human Factors research concentration.

Routinely-collected "off the shelf" statistics now provide the broad coverage necessary to detect existing or emerging safety problem trends. However, collection costs and attendant processes to supply detailed data in all potential safety areas cannot be justified. As a result, these general indicators are not always sufficient to meet the more demanding requirements to guide the application of research resources. Accordingly, it appears that accumulation of supplemental safety data in identified major areas will always be a necessary and justifiable step in the conduct of effective research activities.

FRA RAIL SAFETY RESEARCH

The Office of Rail Safety Research is organized to provide technical outputs in three major concern areas --which encompass the identified R&D needs. In abbreviated form, the three divisions are:

. Rail Vehicle Safety
. Track
. Inspection

The office cooperates with a wide range of safety interest groups including the FRA Office of Safety, Industry, Unions, NTSB, the Rail Safety Research Board and the Locomotive Control Compartment Committee. The major programs under each of the three divisional responsibilities are as shown in figure 10.

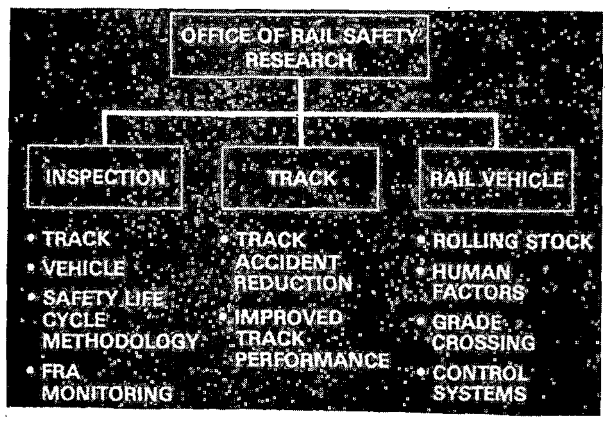


FIGURE 10. OFFICE OF RAIL SAFETY RESEARCH

RAIL VEHICLE SAFETY

The Rail Vehicle Safety Division is charged with research for greater safety associated with Rolling Stock, Human Factors and Accident Avoidance. Figure 11 shows sub-program emphasis.

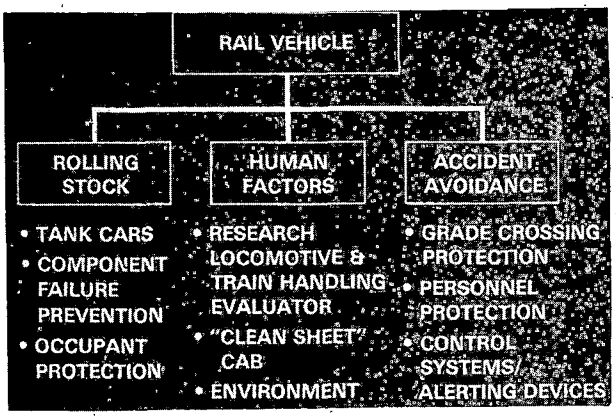


FIGURE 11. RAIL VEHICLE DIVISION

Note that Grade Crossing work is included under Accident Avoidance. Work in this division, and in the other two, has found the FRA as a participant in the total scope of research work performed--with AAR, RPI, U.T.U, B of L.E. among the specific interest groups taking prominent roles both in the conduct of work and the achievements realized.

Hazardous Material Tank Car research has been a mainstay productive effort. Progress has resulted through a continuing series of industry/ government cooperative interactions over the past several years. The highlights of this multi-year program are indicative of the general approach utilized in this and in other activities.

Early small scale tank car analysis and testing (figure 12) provided the basis for full scale tank car instrumentation (figure 13) and full scale pool fire tests (figure 14) taken to the ultimate failure destruction (figure 15). The results followed by comprehensive analysis (figure 16) provided information for generating improvement candidates which then were tested through use of developed/validated small scale plate testing facilities (figure 17) and full scale "torch" testing (figure 18).

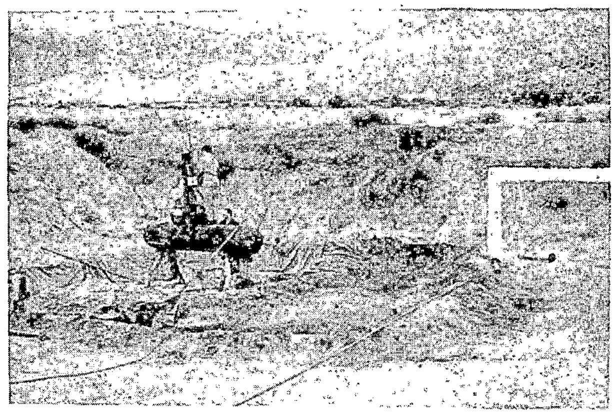


FIGURE 12. SMALL-SCALE TANK CAR TEST

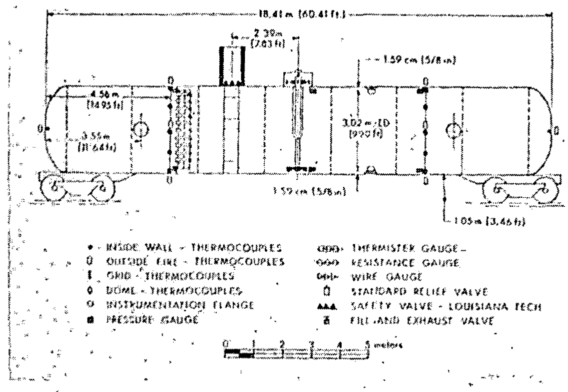


FIGURE 13. INSTRUMENTATION LAYOUT

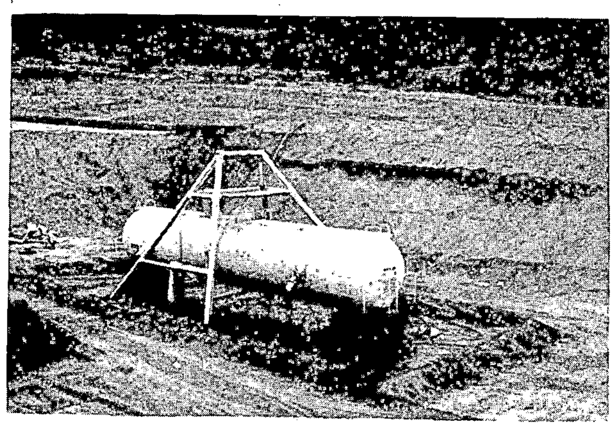


FIGURE 14. POOL FIRE TEST FACILITY

FIGURE 15.
POOL FIRE TEST

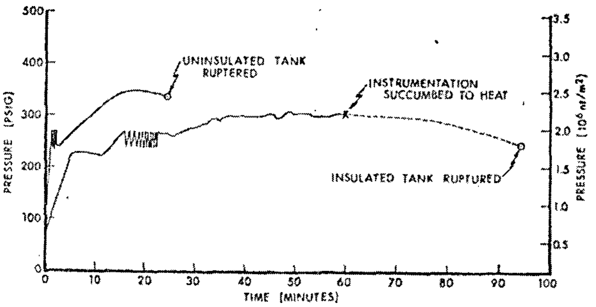


FIGURE 16. TEST ANALYSIS

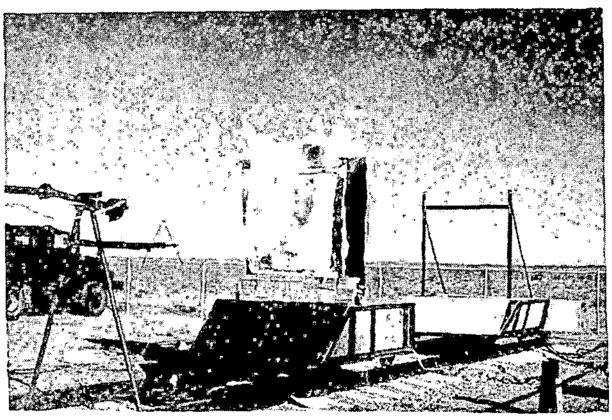


FIGURE 17. TORCH FIRE TEST

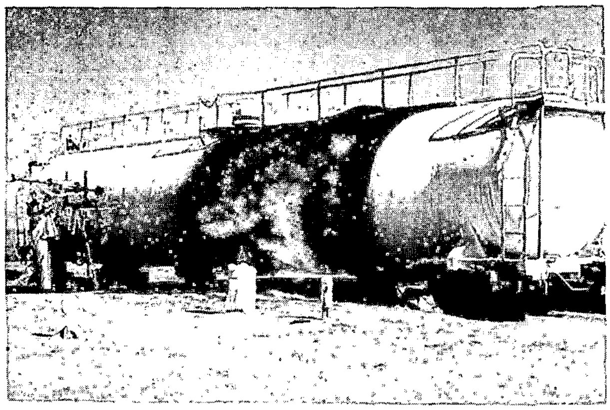


FIGURE 18. TORCH TEST

Such temperature tests were augmented by Impact Analysis and Testing (figure 19)

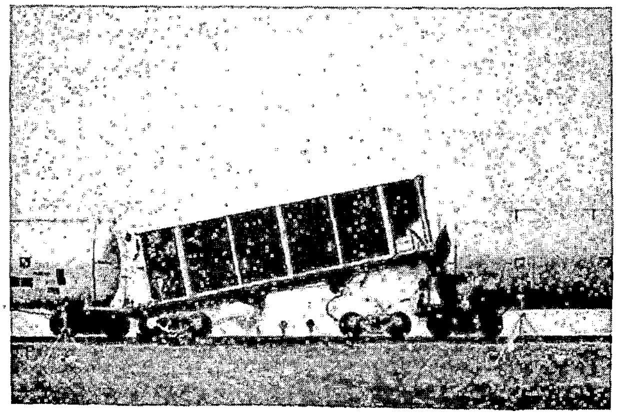


FIGURE 19. IMPACT TEST

to minimize puncture probabilities which eventually produce effective real life retrofit improvements (for incorporation on existing tank cars) such as the double shelf coupler of figure 20 and the thermally protected cars of figure 21.

Stipulation of protection needs in terms of performance rather than design specifications allows for varied methods of implementation, including the "coated" and "jacketed" versions of figure 21--and the less obvious variations of head shielding. Initial work has begun on other Hazardous Material

Rail Transport problems, including radioactive cargoes such as nuclear waste shipments.

In the Component Failure Prevention and Detection program, bearings and wheels (because of their historic safety impact) receive principal attention.

FIGURE 20. DOUBLE-SHELF COUPLER

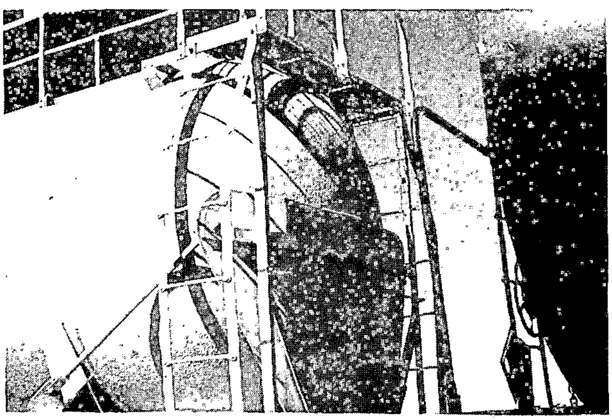


FIGURE 21. HEAD AND THERMAL PROTECTION

Figure 22 is a picture of a versatile dynamometer wheel testing facility located at one of our contractor's sites. It's designed to simulate a range of real-world conditions, including emergency stops from high speeds.

Figure 23 is a scene from a 120 MPH test using Cast Iron brake shoes.

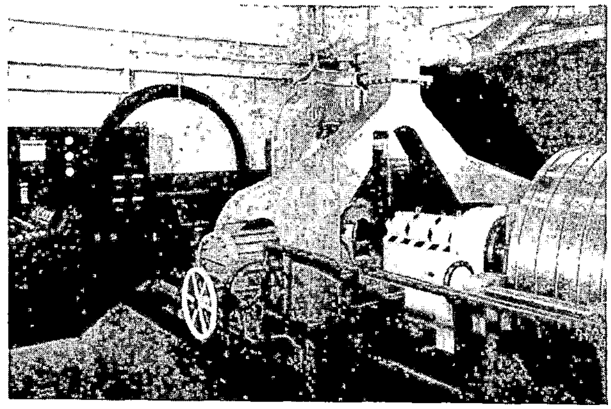


FIGURE 22. WHEEL TEST FACILITY

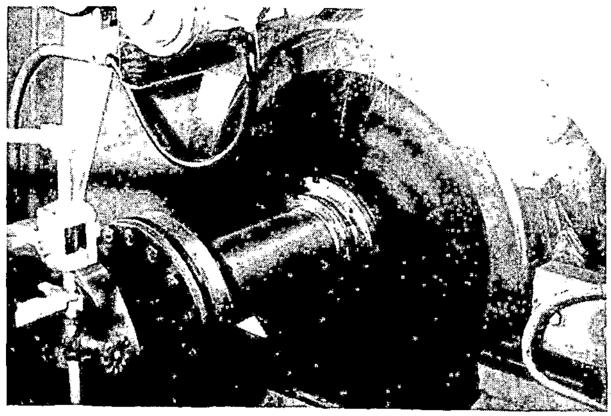


FIGURE 23. WHEEL UNDER TEST CONDITIONS

Such controlled experiments permit testing to destruction (figure 24) by drag braking and the subsequent possession of unique information made possible through wheel reassembly (figure 25) and precise reconstruction of the failure mechanisms.

Field testing validation of impact "models' here at the Pueblo Test Center have proven invaluable in both tank car and Personnel Protection studies. Figure 26 is an example of one of the results of a carefully designed series of such tests.

FIGURE 24. WHEEL TESTED TO DESTRUCTION

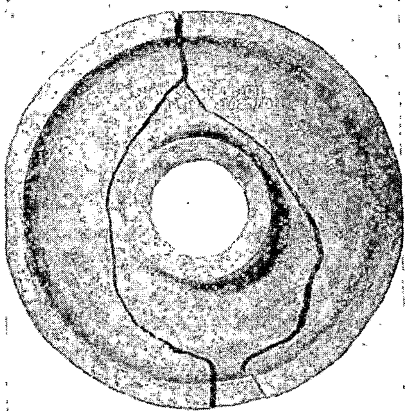


FIGURE 25. WHEEL REASSEMBLY

This 18 MPH rear-end collision test provided key insights in understanding collision forces and achieving the capability to predict impact dynamics and consequences.

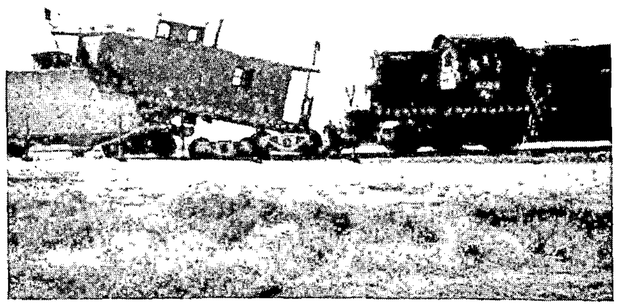


FIGURE 26. "REAR END" IMPACT TESTS

Other train "rear end" protection research activities included "conspicuity" testing--with significant contributions resulting from cooperation with the Illinois Department of Transportation and three Chicago areas railroads. Strobe lights and other marking devices were evaluated as to their utility, visibility and alerting properties.

Protection of occupants is accidents and in incidents of vandalism is another important endeavor. Representative of the "practical" nature of research conducted is a sequence of tests to evaluate the personnel protection characteristics needs and that afforded by available "glazing" materials (figure 27).

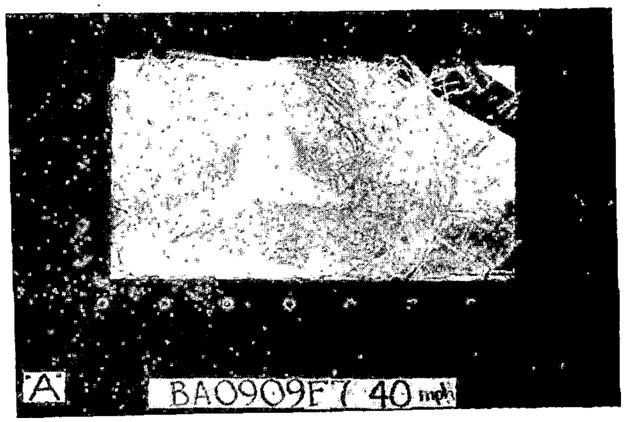


FIGURE 27. "GLAZING" MATERIAL TESTS

Both "ballistic" and "thrown or suspended object" tests were made. Results to date indicate that performance criteria can provide guidelines for use of certain sets of existing materials which will provide reasonable protection--even against the intrusion of cinder blocks.

Work to improve the safety and effectiveness of train handling controls and cab environment is continuing through another joint project with the industry, unions, and the Locomotive Control Compartment Committee.

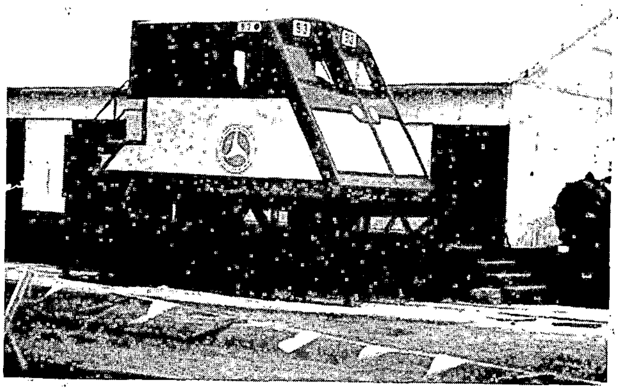


FIGURE 28. "CLEAN CAB" MOCK-UP

The "mock up" structure of figure 28 houses the improved interior cab concept of figure 29.

FIGURE 29. "CLEAN CAB"

This so-called "clean" cab incorporates several advanced cab alternatives--which were evaluated by experienced railroad locomotive engineers. Some of the more readily adaptable modifications to minimize personnel injuries are now being voluntarily implemented in new locomotives.

In addition, locomotive interior emissions and noise measurements have been made in the typical work conditions of several railroads.

FIGURE 30. EMISSION TESTS

The picture of figure 30 was taken at the site of one such environmental measurement testing on cabooses.

FIGURE 31. NOISE TEST

In figure 31, noise measurements are being recorded in a locomotive cab. Levels of noise and emissions under a variety of circumstances and most severe conditions (such as tunnels) have been recorded.

Several research activities are aimed at Grade Crossing protection extensions, including: alerting devices, activation means and constant warning time systems. Trade-offs in the cost effectiveness of the several alternatives found most attractive are still being assessed--including the efficacy of strobe lights as alerting devices at crossings. Locomotive Safety Performance evaluations are being made in both the laboratory and the field.

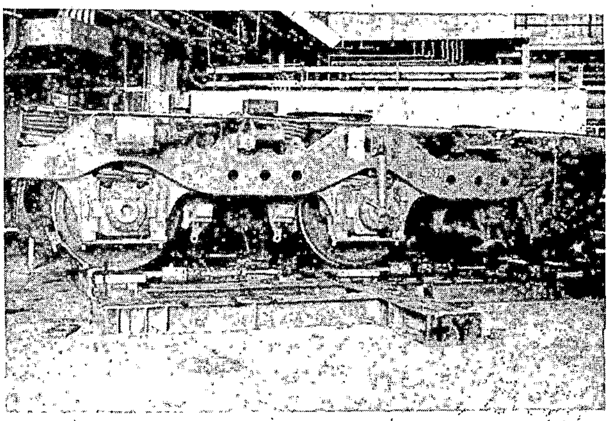


FIGURE 32. TEST FACILITY

Figure 32 shows a 3-axle locomotive truck being set up to be tested at Martin-Marietta. In this facility, the characteristics of a variety of locomotive trucks are determined in engineering terms so that comparison can be made and the potential effects of differences ascertained. Figure 33 shows a truck under test

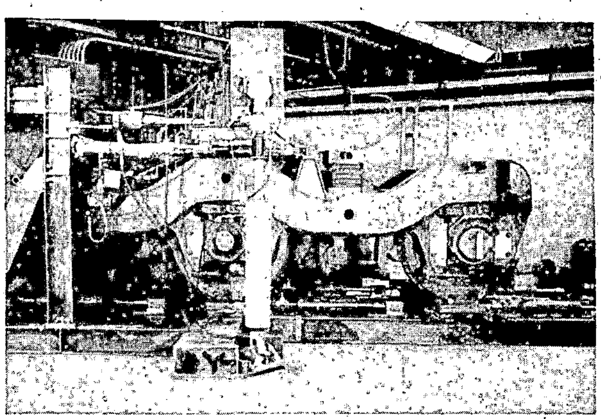


FIGURE 33. THREE-AXLE TRUCK

while figure 34 is a field example of dynamic train measurements being taken on a designated locomotive and consist.

FIGURE 34. FIELD TEST

Both "on-board" and "track side" measurements are necessary measurement requirements, as is track status/condition documentation.

Many locomotive cab improvement candidates will be subjected to a--" Research Locomotive and Train Handling Evaluator" installation if present research work is successful (figure 35).

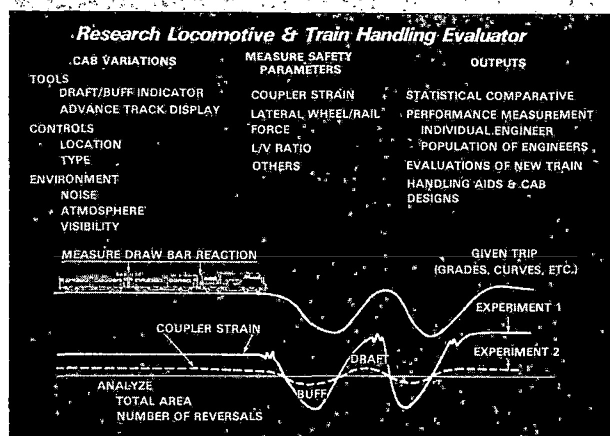


FIGURE 35.
FUNCTIONS OF THE RESEARCH LOCOMOTIVE AND TRAIN HANDLING EVALUATOR

The "evaluator" is an innovation which utilizes advanced "realism" generators of existing simulators but adds the dimension of <u>objective determination</u> of the merits of many types of improvement candidates.

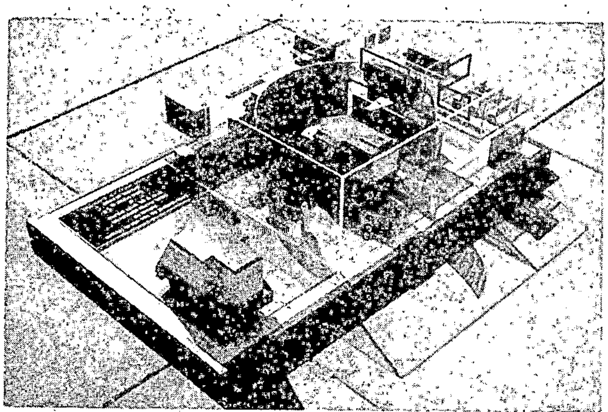


FIGURE 36. ARTIST CONCEPTION OF "EVALUATOR"

Figure 36 is an artist's concept of what the evaluator layout might be. Fabrication activities should begin within the next year. A subsequent presentation in this session will dwell more on the reasons for and potential benefits from this undertaking.

IMPROVED TRACK STRUCTURES

The next division, Track includes R&D responsibility for two major thrusts:

- Accident Reduction (due to track) and
- Performance Improvement

Figure 37 lists the major subprograms under the Track Research Division.

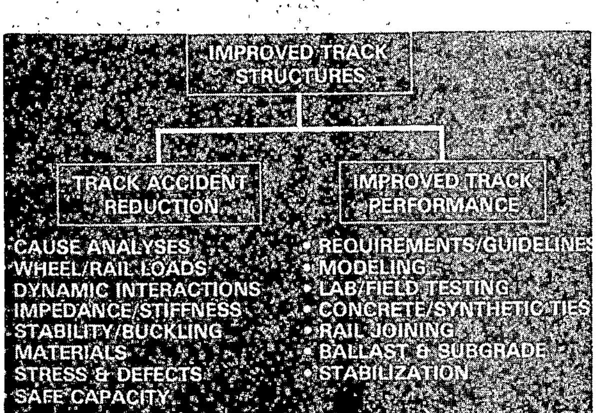


FIGURE 37. TRACK DIVISION

Track R&D resources are in a large part responsible for the research oriented trackage at the Transportation Test Center. A great portion of the division's activities are in support and utilization of FAST--to provide accelerated track answers which this division is relying on. A large number of track experiments are being conducted on this loop.

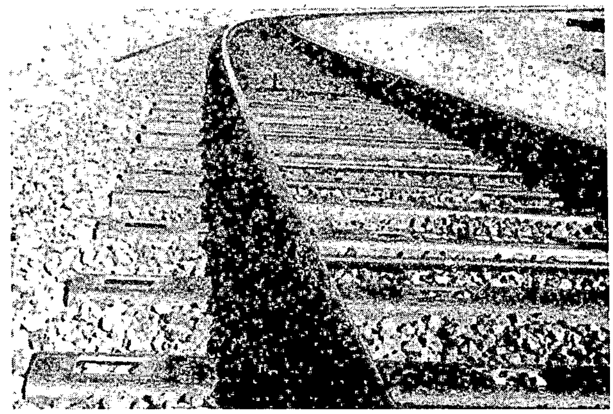


FIGURE 38. STEEL TIES

Typical of the components being tested are such things as steel ties (figure 38), --as well as (figure 39) wood and concrete ties--

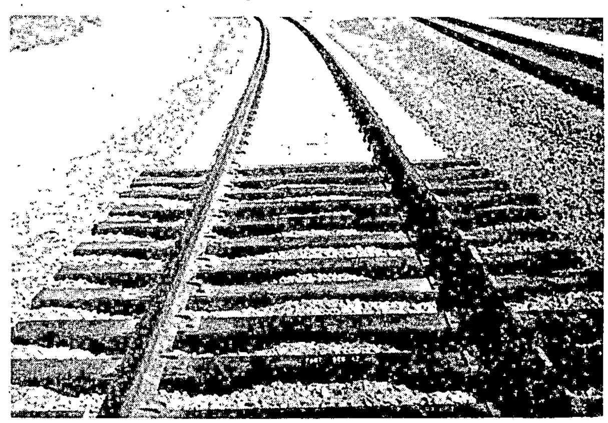


FIGURE 39. CONCRETE AND WOOD TIE EXPERIMENT ON FAST

and different fastener designs. Of course, rail wear rates and characteristics are not being neglected (figure 40).

FIGURE 40. RAIL WEAR

Results from the FAST experience are being integrated into findings from other areas by designated "Experiment Managers". Figure 41 represents one of the sources.

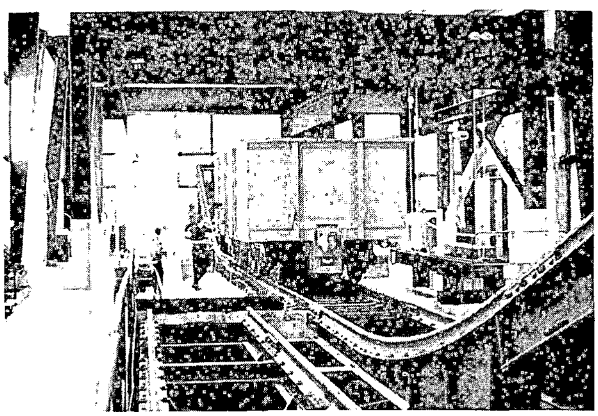


FIGURE 41. ROLLING LOAD LAB

It is the recently completed AAR track structure testing lab in Chicago. The potentials for railroad track stabilization have been investigated. Figure 42 is a Lime Slurry Pressure Injection Testing and Evaluation project in progress.

While a preliminary handbook was published in June 1977, additional validation is planned to more firmly substantiate the circumstances under which this process is most beneficial.

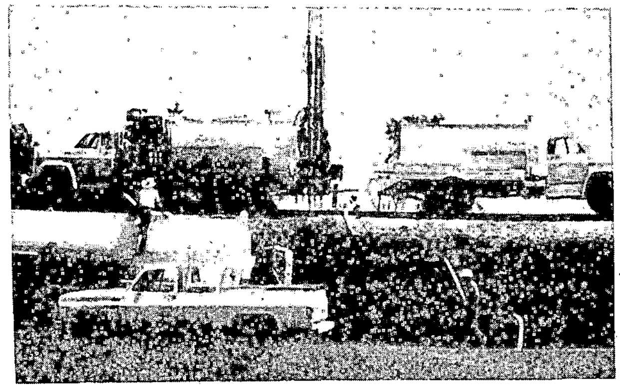


FIGURE 42. LIME SLURRY INJECTION PROCESS

Naturally, measurement of wheel/rail forces under varying vehicle, track and operating conditions is important to provide guidelines for track modification/maintenance practices which will result in <u>system</u> improvements.

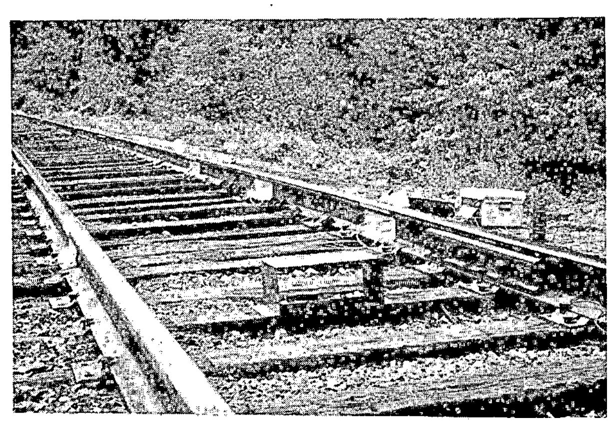


FIGURE 43. INSTRUMENTED TRACK

Extensive wheel/rail force measurements have been made at instrumented locations such as in figure 43.

Several <u>International</u> technical informational exchanges are adding to track knowledge. Figure 44 is an example of "panel" track laying as practiced in the Soviet Union.

Through exchanges of delegations composed of industry/government representatives, we are learning how the Soviets can move almost three times the freight tonnage we do with equivalent technology.

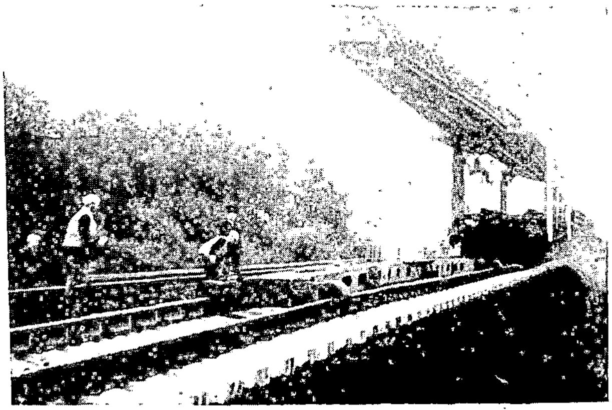


FIGURE 44. SOVIET "PANEL" TRACK

Track stiffness, strength and stress measurement research is being accomplished as a prerequisite for defining relevant track performance specifications. Figure 45 is one means by which track stiffness (or impedance as some prefer to call it) is being measured and assessed.

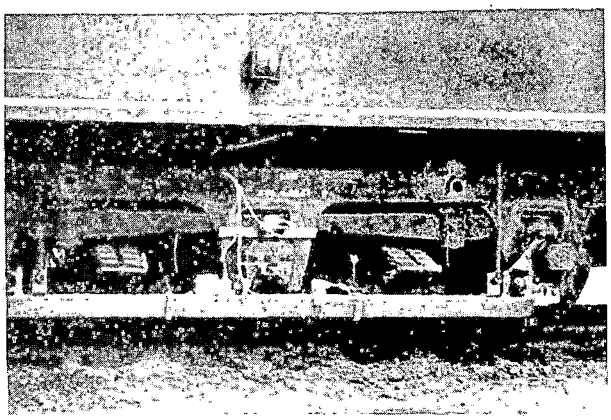


FIGURE 45. TRACK IMPEDANCE MEASUREMENT SYSTEM

A lot of jointed track will continue to exist in the U.S. for some time. Since joint maintenance is a definite safety concern, tests of extending acceptable joint performance life are in order. Figure 46 is a scene from a testing program on the method of "sleeve expansion--bolt hole hardening".

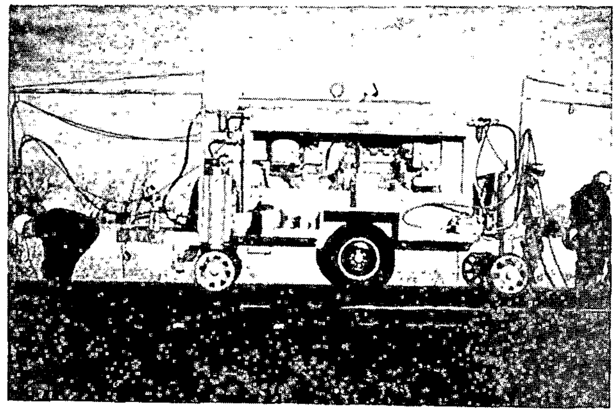


FIGURE 46. BOLT-HOLE HARDENING

Several track facilities to measure vehicle performance under specific (or severe) controlled conditions are in developmental stages. In order to ensure that newly introduced vehicles are compatible with a range of track conditions, the feasibility of a "perturbated" testing track at TTC is being investigated. Such an "obstacle course", together with accelerated time testing, may be critical to ensure "safe life" performance.

The effectiveness of track maintenance machinery such as the Ballast Consolidator of figure 47 has also been tested in cooperation with several railroads.

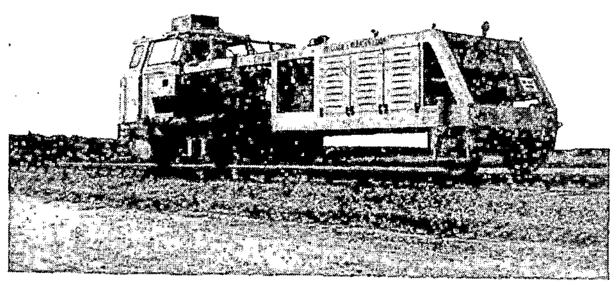


FIGURE 47. BALLAST CONSOLIDATOR

Data generated from track research

activities has been and will continue to be furnished in direct support of Northeast Corridor decisions, as well as the other industry/government interests.

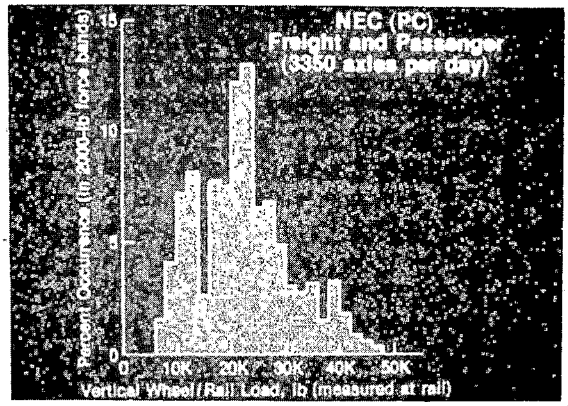


FIGURE 48. WHEEL/RAIL LOADING

Displays of experienced wheel/rail loads as in figure 48 are an example of the type of information which was all to often lacking in the past, but is now becoming an integral part of making sounder decisions.

IMPROVED INSPECTION, DETECTION AND TESTING

The Inspection Division functions for the obvious reason of assisting through research both industry and, government in monitoring the safety acceptability of various components of the rail transport system. The major categories of R&D emphasis are:

- . Track Inspection
- . Vehicle Inspection
- . Safety Life Cycle Methodology
- . Automated Track Inspection (Primarily for the FRA Office of Safety)

Figure 49 delineates the subprograms under each of the major programs of the Inspection Divisions.

The so-called "T" cars (figure 50) have been one of the most visible products of the Inspection Division's efforts--in the past.

Currently, there are seven such cars. T1/T3 was the early track geometry car combination with T2/T4 being recently converted into essentially the same track geometry measurement capability. T6 is a prototype of the eventual production cars; while T5 and T7 are both data acquisition cars used to obtain dynamic data measurements in train consists.

The T-cars now inspect about 30,000 miles of track annually--both for Office of Safety surveillance purposes--and to develop ways in which such measurements can assist the railroads--in such things as maintenance of way planning. Increased capabilities are being realized in both application areas.

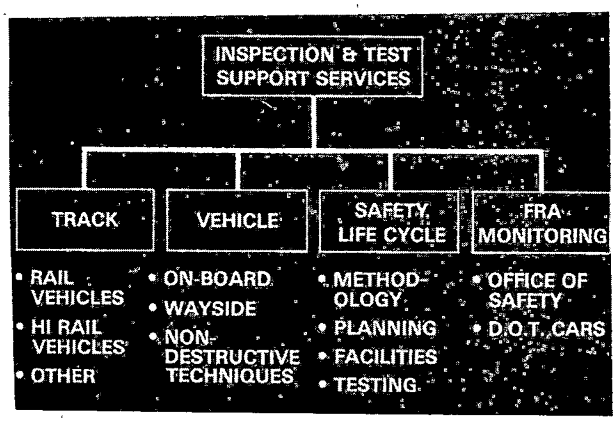


FIGURE 49. INSPECTION DIVISION

FRA Test Cars

Test Car	Current Instrumentation	Time Completed
T3/T1	Track Geometry/Support Car	Mar 1973
T2/T4	Track Geometry/Support Car	Nov 1976
T6	Track Geometry and Rail Flaw	Oct 1977
T5	Data Collection, Freight	Sep 1975
T7	Data Collection, Passenger	Jan 1977

FIGURE 50. TEST CARS

T-6, (figure 51) the newest car, incorporates track geometry measurement and rail flaw detection features.

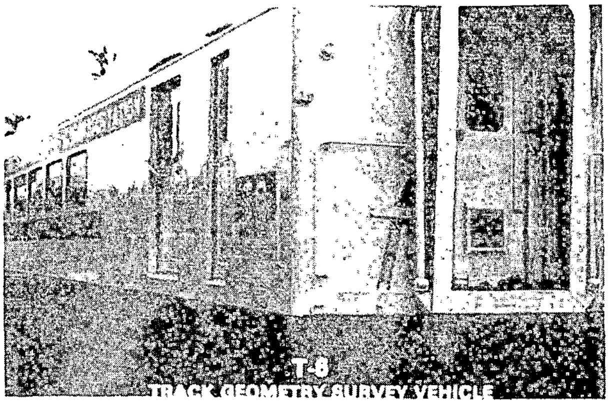


FIGURE 51. T-6, PROTOTYPE PRODUCTION VEHICLE

It is intended to be the forerunner of a fleet of cars to be operated by Office of Safety personnel. The car formerly was a U.S. Army Hospital vehicle.

Figure 52 depicts T-6 during testing and "shakedown" exercises at the Transportation Test Center.

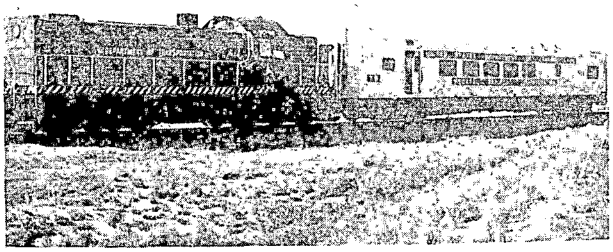


FIGURE 52. T-6, SHAKEDOWN TESTS

While there are still some "bugs" to work out, operations should commence early in 1978.

Figure 53 illustrates the important line and surface relationships which are measured--together with the method employed.

While the extreme right column may not mean much at first glance--it represents considerable developmental work.

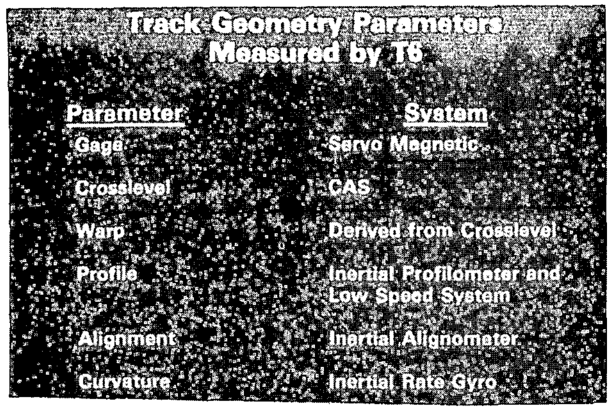


FIGURE 53. T-6, TRACK GEOMETRY CAPABILITIES

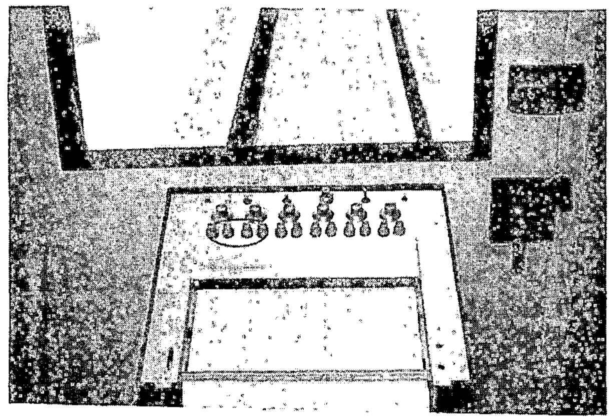


FIGURE 54. DATA COLLECTION AND TRACK OBSERVATION

Figure 54 confirms that the car has been designed so that measured results can be displayed--while observers view physical conditions. Rail flaw measurements are accomplished through use of ultrasonic probes located within two tandem, "membrane" type wheels--on each rail (figure 55).

The addition of the wheel pairs permits "pitch and catch" paths--making large improvement in performance possible.

Figure 56 is how the "hardware" looks as installed under T-6.

The two middle wheels on each side are the ultrasonic sensors. The entire carriage is lowered for rail flaw detection activation.

The contents of figure 57 indicate

the major software improvements which have been installed.

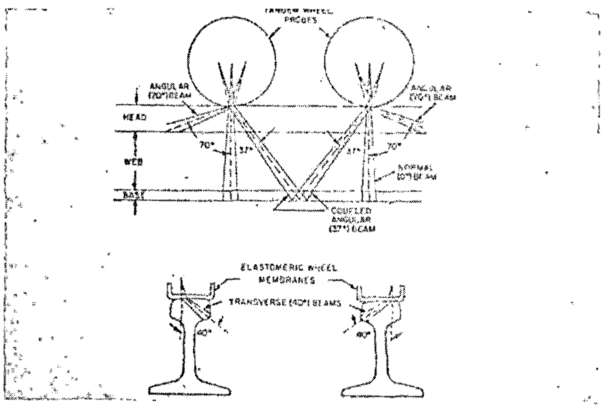


FIGURE 55. RAIL FLAW SYSTEM, T-6

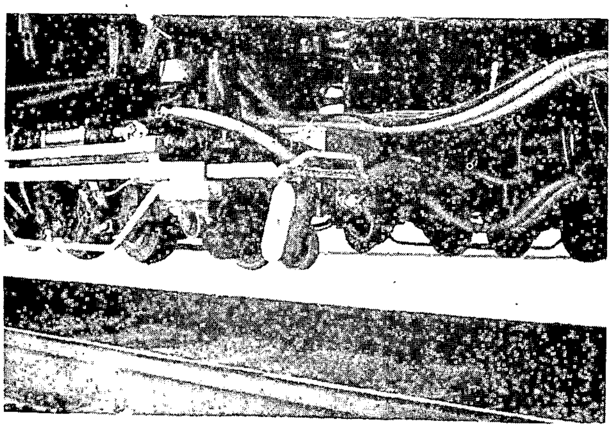


FIGURE 56. RAIL FLAW CARRIAGE

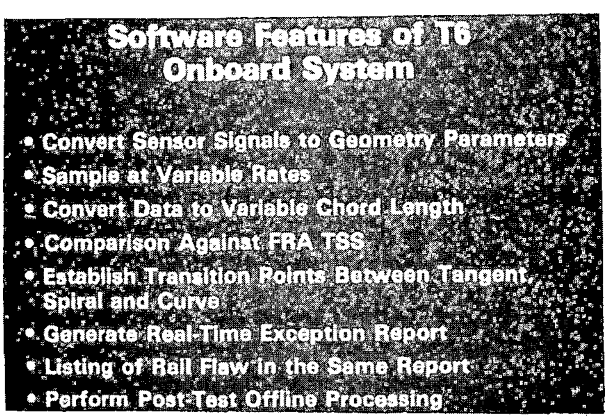


FIGURE 57. ON-BOARD COMPUTER FEATURES

Among the most notable qualities are:

Maximum utility for track standard measurements

and

. Flexibility to adapt easily to a range of output options in the field.

A different level of application may be in order for other than main lines. For Branch Lines, high-rail vehicle measurement of selected track geometry parmeters and rail flaw geometry parameters and rail flaw detection development is important. Figure 58 is a high-railer under test.

FIGURE 58. HIGH-RAIL VEHICLE

The State of Iowa has a contract with us to test one version of such a car and Office of Safety personnel are being familiarized with the operation of a similar vehicle.

In summary, substantial progress has been made in obtaining higher speed rail flaw detection--and parallel track geometry data processing--all in real time (figure 59).

"Instant" display of results and onboard processing is now almost routine.

After concept development, the practicality of "on-board" monitoring systems for selected applications to freight cars is being tested. Two safety concerns are being concentrated on for automated early detection of;

. excessive bearing temperatures;
. imminent derailment conditions

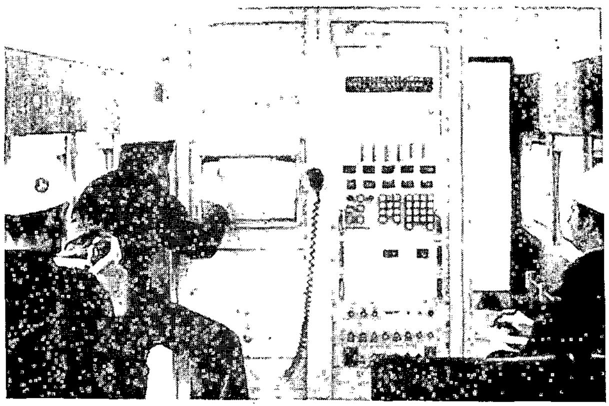


FIGURE 59. DATA PROCESSING SYSTEM

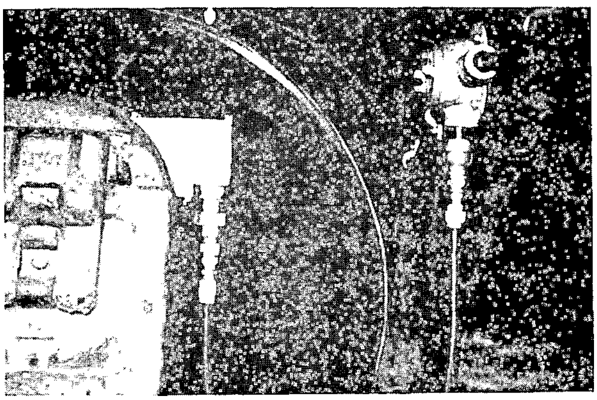


FIGURE 60. OVERHEATED BEARING DETECTOR

Figure 60 is a scene from actual field tests to appraise operating problems and the advantages of continuous monitoring and "triggered" train stop action--in cases where emergency states are encountered.

Conceptual studies are in progress to evaluate the potential benefits and feasibility of automated rolling stock performance monitoring via an INTEGRATED MODULAR WAYSIDE APPROACH which essentially utilizes existing devices.

Preliminary component (as in figure 61) testing for inclusion in the wayside "package" is underway for a variety of in-track sensor systems. The overall concept is meant for railroads to incorporate various wayside capabilities into a basic unit--as needed. Figure 62 is an artist's version of a full system complete with a communications link.

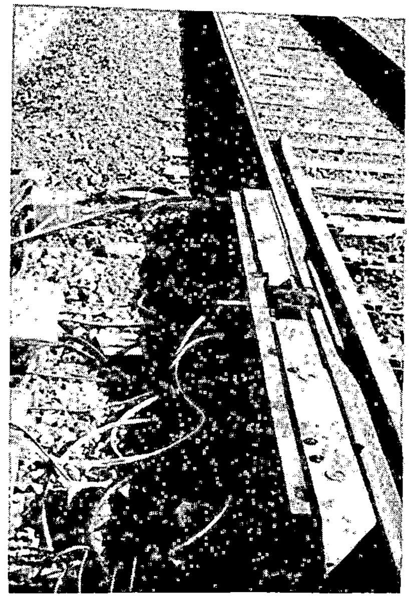


FIGURE 61. WAYSIDE INSPECTION COMPONENTS

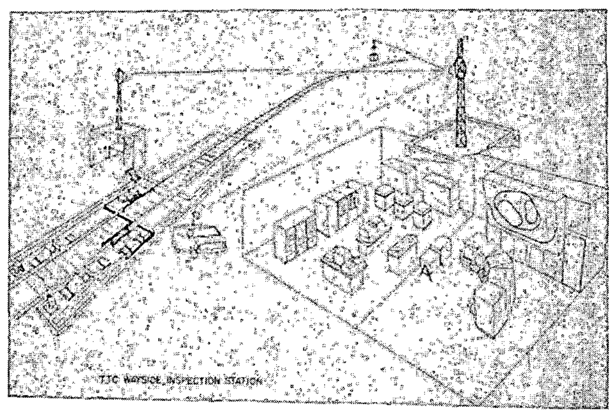


FIGURE 62. WAYSIDE INSPECTION SYSTEM

Modularity is the essential key. The desire is to be able to maintain flexibility, i.e., add a device or feature on an as needed basis--with an objective of maximum "portability". The approaches for an overall Safety Life Cycle Methodology definition are being investigated. The fundamental underlying reason is that rail system must not only behave satisfactorily when new--but also over entire life periods where a host of "wear" states and fatigue conditions may be encountered. We are searching to produce before-the-fact guidelines on:

- . Acceptance Testing
- . Identification of Potential Safety Problem Areas
- . Life Cycle or Time-Compressed Testing, and
- . In-Service Limitations/Monitoring

Facilities such as the FRA "T" cars are a necessary element of acceptance tests to document the dynamic performance of locomotives and cars in typical trains.

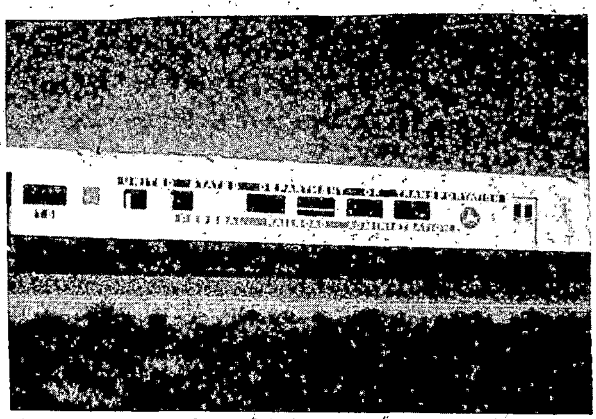


FIGURE 63. T-5, DYNAMIC MEASUREMENT VEHICLES

T-5 (figure 63) and T-7 continue to serve as the means for in-train data acquisition, with T-7 being utilized primarily for the characteristic higher speed runs of passenger train consists.

Conclusions

You have just been exposed to a pictorial "panorama" of the kind of Railroad R&D activities in which we are engaged. The ultimate goal of this government/ industry research is results reflected in favorable effects on safety trends (figure 64).

We view our business primarily in an applied sense. That is to guarantee that Research and Development is aimed at the real world of railroading with emphasis on the near future! We realize that research by itself cannot make a complete turn around in the safety record, but it can certainly be an assistance tool. We need continuing inputs from both the research "insiders" and the "outsiders" on how best to read the statistics and recognize the critical safety problems which R&D can help solve.

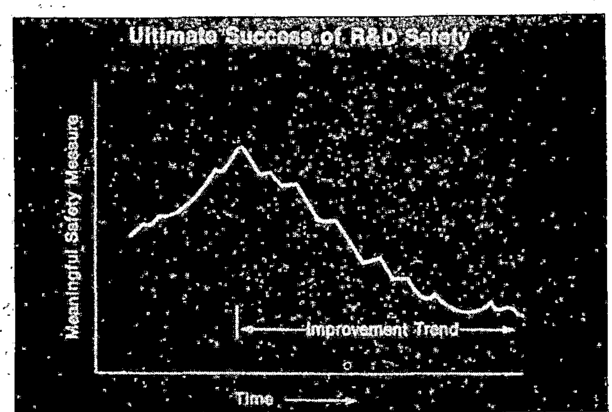


FIGURE 64. SAFETY IMPROVEMENT THROUGH R&D

Our work is intended to be open and responsive to legitimate safety concerns. As a result, the progress in railroad safety which is being achieved, is rightfully being conducted by and should be credited to the cooperative endeavors of the participants--railroads, supply industry, unions and government. We feel that this has been the case to date and that R&D will continue to assist in eventually making noteworthy improvement in safety statistics a fact. Hopefully, at a conference of this type in the near future, an update of safety statistics will substantiate significant progress.

The presentations which will follow in this session will dwell in greater detail on several of the more important projects which have only been briefly discussed.

ADVANCES IN RAIL FLAW DETECTION

by

H. L. CECCON

In 1974 the Federal Railroad Administration initiated a project to improve rail flaw detection capabilities through the improvement of current equipment and/or techniques and the development of new equipment in certain critical areas. The major emphasis was placed on automating inspection operations to improve speed and detectability and reduce the dependence on human operators on the quality of the inspection results.

This paper describes some of the equipment developed under the project and the integration of these new developments with conventional equipment into field operational systems.

INTRODUCTION

The Federal Government, through the Federal Railroad Administration, has initiated a program to reduce the occurrence of track-related accidents through the enforcement of safety standards. Since the beginning of the program in 1970, FRA Track Inspectors have monitored the railroad's safety conditions by checking inspection records and performing visual and manual inspections. In 1974, a project was initiated to provide the inspectors with automated rail inspection equipment so a wider and more thorough coverage could be made. The responsibility for implementing selected portions of the project was assigned to the U.S. Department of Transportation, Transportation Systems Center in Cambridge, Mass.

The objective of this paper is to provide a technical description of rail flaw detection equipment developed under the project and the results of the integration of this equipment into operational rail flaw detection systems.

The project began with an extensive inspection systems analysis and survey of state-of-the-art inspection systems. The survey revealed that the average inspection speed of rail flaw detection systems in the U.S. is approximately 5 mph, limited by the need to stop to hand verify defect indications detected by the mobile system. Although this performance has been accepted by the railroad industry, it was determined that improvements were needed to provide the type of performance and utility needed to support the FRA Track Inspectors in performing their regulatory duties. A survey of foreign technology showed that little could be applied to U.S. requirements basically because rail conditions are quite different and the post-processing of data used by the high speed systems is an undesirable feature.

To support decision-making early in the project, commercially available rail inspection equipment was ordered and tested at the TSC facility. The most significant of this equipment was a hy-rail flaw detection system. The system, is shown (figure 1) undergoing test on the TSC test track, served as a baseline for future development of equipment. Based on the analysis of survey information, and an evaluation of nondestructive testing techniques and equipment, the ultrasonic technique was selected as the primary inspection method around which the improvement project would be centered.

The overall information generated in the early phases of the project provided the FRA with information on which to base long range goals. These goals covered the development of two types of inspection systems: 1) a hy-rail system with a maximum inspection

---

Harry L. Ceccon received his B.S. and M.S. in Physics and Mathematics from Lowell Technological Institute. Since 1971 he has worked at the Transportation Systems Center in Cambridge, Mass., in the area of railroad component inspection techniques.

speed of 25 mph, and 2) a railbound system with maximum inspection speed of 40 to 50 mph.

FIGURE 1. HY-RAIL FLAW DETECTION VEHICLE

INSPECTION EQUIPMENT DEVELOPMENT

The major emphasis on improving the rail inspection capability was directed in three basic areas: 1) Improve sensor performance, 2) Develop automatic controls and 3) Improve the inspection data processing technology.

Ultrasonic Sensor Development

Early in the project the decision was made to develop wheel probes as opposed to sleds. This decision was based on the performance of each probe on poor rail surfaces and the lower couplant consumption (approximately 2 to 1 at high speeds) of the wheels. Figure 2 shows the wheel probes delivered with the FRA hy-rail system and currently in use by several railroads.

This basic three-transducer configuration was evaluated and found to have certain deficiencies. Based on this information, work was started to develop an improved ultrasonic wheel probe.

Figure 3 shows the tandem wheel probe which resulted from this work. This eight-transducer tandem wheel configuration can provide more than 10 channels of information per rail and was designed specifically to improve the detection of rail weld breaks and vertical split heads located off the center of the web area; shortcomings of the three and six transducer wheels.

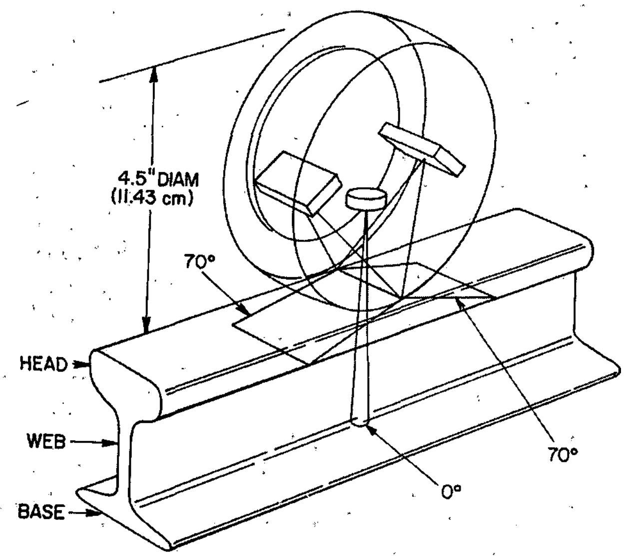


FIGURE 2. SINGLE WHEEL PROBE CONFIGURATION

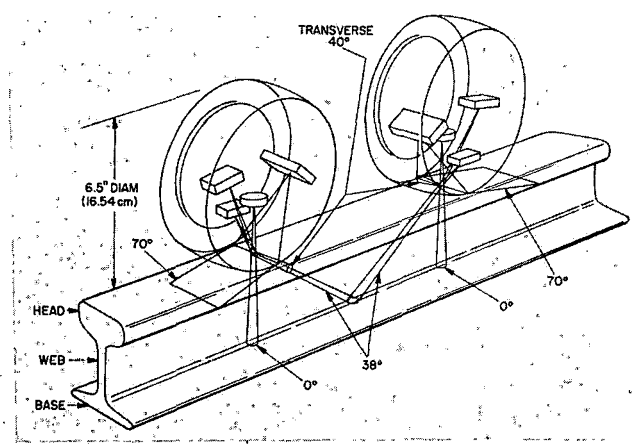


FIGURE 3. TANDEM WHEEL PROBE CONFIGURATION

This tandem wheel probe is being used on both hy-rail and rail-bound inspection systems. At least two railroads are presently planning to install these sensors in their equipment before the end of the year.

Automatic Controls

The main thrust of the work related to automatic controls was directed toward automating those functions which equipment operators must perform in order to ensure reliable inspection results. From an operational analysis of current systems and observing the operation of ultrasonic inspection equipment in the field on railroad lines, it was determined that, aside from inspection data interpretation, the next most important functions the operator must perform are: 1) maintain alignment of sensors, 2) control of amplifier gain and adjustment of signal gates. These are listed in order of their importance.

Figure 4 shows a block diagram of an automatic sensor alignment device which was developed for ultrasonic rail flaw detection systems. This hydraulic servo system is controlled by two position detector transducers which receive acoustic signals reflected from the base of the rail. The signals are input to a comparator circuit which can sense an unbalance in the signals amplitudes. Should an unbalance occur, the hydraulic servo system is commanded to adjust the position of the sensor until a balance occurs. This sytem is currently in operation on two FRA systems and is being installed on systems owned and operated by several railroads.

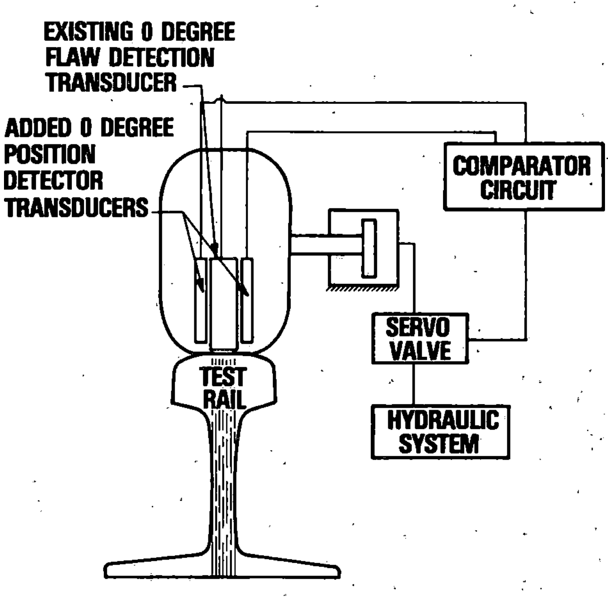


FIGURE 4. AUTOMATIC SENSOR ALIGNMENT DEVICE

Automatic amplifier gain control and automatic gate positioning devices were developed so ultrasonic inspection systems could quickly compensate for variations in the attenuation of acoustic signals due to the metallurgical properties of rail steels and wheel-to-rail coupling conditions and for variations in rail heights encountered over lengths of trackage. Both devices use the same methods for acquiring information on which automatic adjustments are based.

The devices sample consecutive acoustic pulses generated and received by the 0° transducer over a given length of rail (4 ft.). The pulses travel through the head and web of the rail and are reflected at the base of the rail back to the transducer. For the gain control, the return pulses are received over a four-foot length and the amplitudes are summed and divided by the number of pulses for an average amplitude. This amplitude is compared with a preset value. If the average amplitude is different from the preset value, it is adjusted to coincide with the preset value. The gate positioning circuit also samples pulses received over a one-foot section, the device automatically adjusts the gates to the new position. Both of the systems are currently operational in the field on FRA systems.

Data Processing and Display Systems

The data processing and display systems (DPDS) under development for the ultrasonic rail flaw detection systems for the hy-rail and rail-bound vehicles are significantly different. The DPDS for the hy-rail operation is being designed to operate at a maximum speed of 25 mph and uses a combination of data screening techniques and human operator interpretation. The DPDS for the rail-bound vehicle is being designed for operation at speeds up to 50 mph, with minimum operator interaction required. Both systems will be coupled to the tandem wheel probes described earlier in the paper.

The two basic components of the DPDS being developed for hy-rail operation are a data screening device and a consolidated B-scan display.

The data screen discards regularly occurring features, such as nondefective bolt hole patterns, from the data. The screened data is displayed on a TV monitor for operator interpretation in a consolidated B-scan format. A photograph of a TV monitor displaying a rail joint in a B-scan format with two bolt hole cracks and a head and web separation is shown in figure 5.

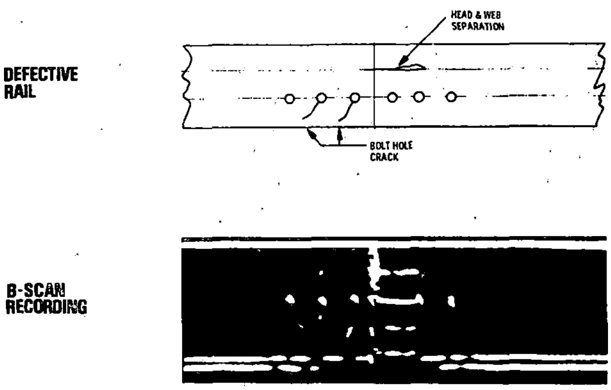


FIGURE 5. SCAN DISPLAY OF A DEFECTIVE RAIL

Several projects are underway, directed toward developing high speed data processing systems for rail flaw detection, which will be found compatible with track geometry measuring systems similar to those currently in operation by the FRA. In order for dual system operation to be cost effective, the operational characteristics of the two subsystems should be similar. Geometry measuring systems typically operate at speeds up to 70 mph, and therefore, if the rail flaw detection system is to be compatible in this dual operation, the operating speed must be at least in the vicinity of 50 mph. This is the driving force which has set the high speed requirement on systems housed in the rail-bound vehicles. Since most of the projects are still in the early research phases, they will not be discussed here. A rail-bound system which is currently operational in the field at 25 mph, is discussed in the next section of this paper.

OPERATIONAL SYSTEMS

Rail-Bound Vehicle

The FRA Track Inspection Vehicle (T-6) (figure 6) is currently operational in the field and undergoing testing at the Transportation Test Center. The system is used to make track geometry measurements at speeds up to about 85 mph and inspection of rail for flaws at speeds up to 25 mph.

FIGURE 6. TRACK INSPECTION VEHICLE (T-6)

The rail flaw detection system incorporates conventional ultrasonic electronics, the sensors and automatic controls described above to provide a high speed real-time inspection capability. The system characterizes the defects as to the general type and location within the rail and generates a real-time exception report detailing the anomalies along with locations in the track. This is the first rail flaw detection system to be developed having both high speed and real-time reporting capabilities.

Hy-Rail Vehicle

The hy-rail ultrasonic rail flaw detection system (figure 1) is currently stationed at the Transportation Test Center where it is supporting the FAST project and undergoing field evaluation in the Colorado area. This system uses the three transducer wheel probe and records three channels of inspection data. The inspection data is processed through a basic gate and threshold system typically used in the U.S. railroad industry. The binary data is displayed directly on chart paper for operator evaluation. Aside from some minor data screening by the system, all defect identification and characterization is made by the operator.

The automatic sensor positioning device has been installed on the system and has significantly improved the speed and detectability. The automatic gain and gate positioning devices are scheduled for installation in the near future.

ACKNOWLEDGEMENT

This project is sponsored through FRA Office of Rail Safety Research; Improved Inspection, Detection and Testing Research Division; and John C. Mould, Technical Coordinator.

The author wishes to acknowledge E. Howerter, ENSCO, Inc., for his work in developing data processing techniques; W. Kaiser, Battelle Columbus Laboratories for the design and construction of the automatic sensor positioning device and; D. Pagano, DAPCO Industries, for his contribution in the area of ultrasonic sensor development.

REFERENCES

1. Doyle, G. W., Kaiser, J., Hadden, F., "Automatic Carriage Control System", Report No. DOT-TSC-1164-3, Battelle-Columbus Laboratories, August 1977.

2. Howerter, E., Ceccon, H., Mould, J., "An Automated Approach to Ultrasonic Rail Flaw Detection", Southeast Conference, Williamsburg, VA., April 1977.

3. Kaiser, W., et al, "Rail Inspection Systems Analysis and Technology Survey" 1975-1976, Report No. DOT-TSC-979-1, Battelle-Columbus Laboratories, March 1977.

4. Pagano, D., Hajdu, F., "Detectability Evaluation of AAR Magnetic and DAPCO Ultrasonic Rail Inspection Systems" Report No. DOT-TSC-995, DAPCO Industries, July 1976.

5. "Induction-Ultrasonic Method for Testing Rails in Track", Sperry Rail Service, Division of Automation Industries, Inc., Danbury, Conn., 1975.

SIGNIFICANT DEVELOPMENTS IN FRA TEST CARS, AS EXEMPLIFIED BY T-6

BY

TA-LUN YANG

The Federal Railroad Administration has sponsored the development of track geometry measuring cars since 1966. Several generations of measurement instrumentation and data processing capabilities have since evolved. Track measuring consists T2/T4 and T6 were completed in FY 1977, these and the T1/T3 consist completed in 1973 are being used in conducting track inspection for the Office of Safety and in supporting other Government and industry research activities.

The latest track geometry car T6 has completed its final stages of testing and started track inspection operations in late 1977. The T6 track geometry system includes the most recent developments in instrumentation plus considerable advancement in on-board data processing capability. The track geometry system in T6 includes measurements of gage, crosslevel, rail profile by an inertial system and by a chord system, rail alignment, track curvature, speed and track location. An exception report is generated in accordance with the FRA Track Safety Standards in real time by the on-board computer. Track geometry inspection can be made at track speeds up to 120 mph.

In addition to track geometry measurement, T6 is also equipped with an ultrasonic rail flaw detection system which employs the state-of-the-art sensor technology and an automatic flaw detection and classification scheme implemented in microcomputers. Detected flaws are tabulated in the real-time exception report. Designed maximum test speed of the rail flaw system is 25 mph.

THE FRA TEST CARS

Seven rail test cars have been assembled by the FRA since 1966 for supporting track inspection and rail research activities. These test cars and their application are summarized in Table 1.

Since 1973 the track geometry cars T1/T3 have served as a track inspection consist as well as a test bed for exploring new instrumentation concepts.

In FY 1977, two new track inspection consists, T2/T4 and T6, were completed, providing a significant expansion to the track inspection capability.

T2/T4 incorporates some of the improvements on track geometry instrumentation while T6 includes all of the

TEST CAR	CURRENT INSTRUMENTATION	TIME COMPLETED
T1/T3	Track Geometry/Support Car	Mar 1973
T2/T4	Track Geometry/Support Car	Nov 1976
T6	Track Geometry and Rail Flaw	Oct 1977
T5	Data Collection, Freight	Sep 1975
T7	Data Collection, Passenger	Jan 1977

TABLE 1. FRA TEST CARS

Ta-Lun Yang is the Chief Engineer of the Transportation Group of ENSCO, Inc. He received his B.S. degree in Civil Engineering from National Taiwan University (1960); his M.S. in Structural Mechanics from the University of California at Berkley (1963) and his Ph.D in Mechanical Engineering (1967).

latest track geometry instrumentation development as well as the latest rail flaw detection instrumentation. Table 2 summarizes the advancements in track measurement and analysis techniques and their implementation on the FRA test cars.

Advancements in Measurement & Analysis Techniques

Parameter	Previous		New		
	Method	Disadvantages	Method	Advantages	Implementation
Gage	Fixed Capacitive Sensors	Affected by Rain and Snow, Accuracy Degrades on Curves of Wide Gage	Servo-Controlled Magnetic Sensors	All Weather Improved Accuracy	T2/T3/T6
Profile	Capacitive Sensors Mounted on 14.5 Beams	Affected by Rain and Snow, Reference Beam Is Too Short, Cannot Be Converted to Long Chord	Inertial Profilometer	All Weather Convertible to All Chord Length	T2/T3/T6
			Multi-Chord W/R Carbody		T6
Alignment	Same as Above	Same as Above	Combination of Inertial Reference and Servomag Gage	Same as Above	T6
Crosslevel	Vertical Gyro	Accumulates Errors In Curves	Compensated Accelerometer System	Continuous Compensation	T2, T3, T6
Rail Flaw	Not Measured	Not Measured	Ultrasonic Automated Analysis	Increased Capability	T6
Analysis	Post-Test Offline Analysis	Delay in Exception Report	Real-Time On-Line Analysis	Report Available to Test Completion	T6

TABLE 2. ADVANCEMENTS IN MEASUREMENT AND ANALYSIS TECHNIQUES

RE-INSTRUMENTATION OF T2/T4

T2/T4 served as the track geometry measuring consist prior to the completion of instrumentation in 1973. Track geometry instrumentation was removed from T2/T4 in 1973 and the consist was converted for providing general-purpose test support. A Raytheon 704 Computer was installed in the car for field data collection.

In July 1975, a program was initiated to convert T2/T4 back to a track geometry measuring consist by duplicating the latest instrumentation then existing on T3/T1. This effort was initiated simultaneously with the development of T6. The consist was completed in 1976 and placed into active service in November. A new model Raytheon Computer (RDS-500) was used in T2 and is the same as the one used in T6 which provides the option to implement to T6 software package when it is completed.

In December 1976, the T2/T4 consist suffered a minor derailment while being transferred to a test site. Some truck components and sensors installed on the truck were damaged. Repair of the consist was started in June 1977; track inspection operation resumed in October of 1977.

OTHER FRA TEST SUPPORT CARS, T5 AND T7

Both T5 and T7 were converted Army hospital cars. T5 was completed in 1975. It is equipped with signal conditioning electronics and a Raytheon 704 Computer for data collection. T5 has been using the Transportation Test Center in Pueblo, Colorado as its operating base, supporting research efforts primarily related to freight services.

T7 was completed in January 1977; it has signal conditioning equipment and a HP-2100 computer for data collection. Modifications have been made in the trucks (including 1:40 taper, hydraulic snubbers and softer springs) of T7 to make it suitable for sustained high-speed operation. Features were added to the carbody so that it can be coupled with Amtrak's Amfleet cars. T7 has been used to support passenger car and locomotive testing.

THE LATEST TRACK INSPECTION VEHICLE - T6

The FRA obtained several retired Army hospital cars in 1975; it was decided that one of these cars would be used as the base to implement a track geometry vehicle which contains the most up-to-date instrumentation in both track geometry and rail flaw measurement. The measurement systems and the data processing capabilities will be tailored specifically for track inspection in accordance with the FRA Track Safety Standards.

Instrumentation of T6 was initiated in July 1975. Considerable effort went into the design to incorporate the best features in existing systems and to add new features for overcoming weaknesses inherent in the existing systems. In a number of areas, the system design involved the implementation of a concept at experimental stages into an operating system. These areas included the rail flaw system, the alignment system, the low-speed profile system and new software algorithms.

T6 is undergoing its final phases of testing, it is expected to join the other FRA track inspection cars and raise the track measurement technology to a new plateau.

Brief descriptions of the T6 instrumentation are provided in this section.

VEHICLE LAYOUT

The vehicle was built by St. Louis Car Company in 1953. It is similar in construction to a conventional towed passenger coach. Some general data are shown below:

Total Length:	85 feet
Width:	10 feet
Height:	13-½ feet
Weight:	80 tons (20 tons/axle)
Truck Spacing:	59-½ feet
Axle Spacing:	8 feet
Wheel Diameter:	36 inches

The exterior finish of T6 is shown in figure 1.

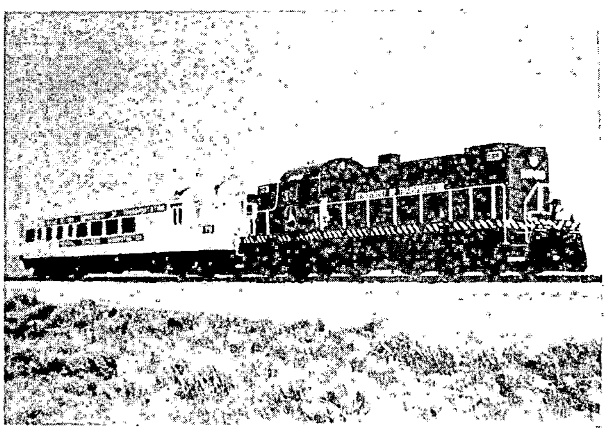


FIGURE 1. EXTERIOR VIEW OF T-6

An effort was made to minimize modification to existing facilities inside the hospital car. The kitchen area, roomettes and most storage cabinets were preserved (see figure 2 of general interior layout). The center portion was turned into the instrumentation and control center (see figure 3). One end of the car (nominal trailing end) was modified into an observation room (see figure 4). A chart recorder with an extended display table is placed in the center of the room for monitoring of track geometry data during a test. Large windows and tiered seats were installed so that track inspection personnel can visually observe the track and the data simultaneously.

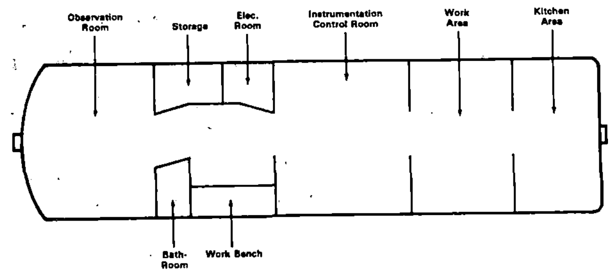


FIGURE 2. INTERIOR LAYOUT OF T-6

An intercom system is installed to maintain contact among the rear observation room, the control room and the forward observer located in the locomotive.

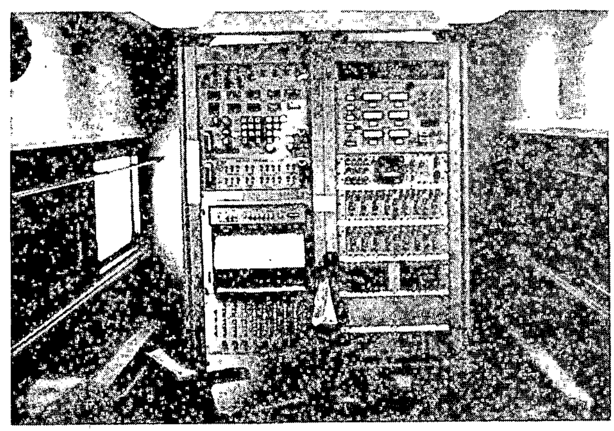


FIGURE 3. T-6 INSTRUMENTATION CONTROL CENTER

Support facilities include a full kitchen, a Microphor toilet, a shower, one private sleeping room, a pull bunk and convertible beds which can accommodate up to 3 people. The vehicle is equipped with two 55 Kw diesel generators to power the instrumentation, heating and air conditioning systems.

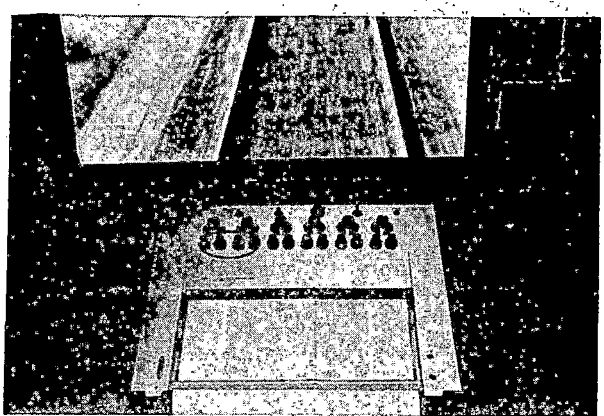


FIGURE 4. T-6 END-OF-CAR OBSERVATION ROOM

TRACK GEOMETRY INSTRUMENTATION

Track geometry and reference parameters measured by the track geometry system include:

- Gage
- Crosslevel
- Profile of each rail
- Alignment of each rail
- Track curvature
- Speed and distance
- Track location

A brief description is given for each of the systems.

Gage

This system uses the same basic operating principle as the magnetic systems installed in T2 and T3; however, the design contains substantial improvements in calibration and protection features. The T6 gage system is installed on a beam suspended across the truck from the bearing housings of an axle. Mounted on each end of the beam is a 0.5" diameter magnetic sensor to measure the distance between the sensor and the rail (see figure 5). Each sensor is mounted on an arm that is movable laterally by a drive screw. The drive screw is connected to a tachometer which monitors the angular position of the drive screw (see schematic in figure 6). A servo-mechanism is used to control the movement of the drive screw such that the sensor is maintained, at a fixed distance (0.5" nominal) from the rail.

FIGURE 5. BEAM SUPPORTING THE GAGE SYSTEM

The measured left and right sensor-to-rail gaps and the positions of the screw drivers are then combined to obtain the total distance between the rail heads.

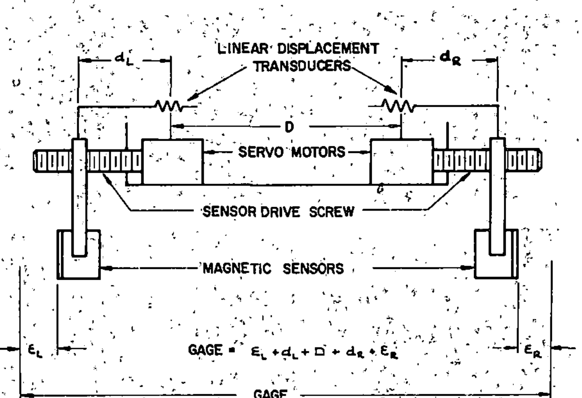


FIGURE 6. SERVO/MAGNETIC GAGE

Microprocessors are used to perform the servo-control functions and gage computation. With the use of microprocessors, it is possible to incorporate an automatic calibration feature. When a calibration command is given from the control room, the servo-motor will drive the magnetic sensor until it touches the rail. The motor then backs the sensor away from the rail through preset known distance steps while recording the outputs of the magnetic sensor in a storage array. The stored distance vs output array is used as the calibration for subsequent tests until another auto-calibration is performed. As long as the magnetic sensor is set up within its range of operation (even if it is in the non-linear range), the auto-calibration feature eliminates the need to bring the sensor response into its linear range and the need to get off the car for manual measurements.

Safety provisions are also included to reduce the vulnerability of the sensors to damage due to impacting obstructions. Sensors can be retracted into a protected position behind the wheel flange or be lifted above the track by manual commands. There are also automatic raise and retrace features triggered by a light impact of the sensor or a track switch sensed by a magnetic metal detector running in front of the gage sensor.

Crosslevel System

A Compensated Accelerometer System (CAS) is used to determine the roll angle of the carbody with respect to the gravitation direction. It is the same in design as the systems currently used on T3 and T2. The CAS contains a laterally mounted inclinometer which measured the component of the gravitation acceleration, g Sin θ as the carbody rolls by an angle θ (see figure 7). The inclinometer also responds to a centrifugal acceleration V^2/R due to curving and any lateral inertial acceleration of the carbody; these would induce errors in the result if the output of the inclinometer is used directly to compute the roll angle θ. A compensation is made to the inclinometer to correct for the centrifugal acceleration. The compensation is derived from the yaw rate of the car measured by a gyro and the measured speed. The responses due to inertial carbody accelerations are eliminated by passing the output signal through a low-pass filter with a corner frequency substantially lower than the carbody natural frequencies. The roll angle computed from the compensated and filtered inclinometer output provides the low frequency activity of the roll angle θ.

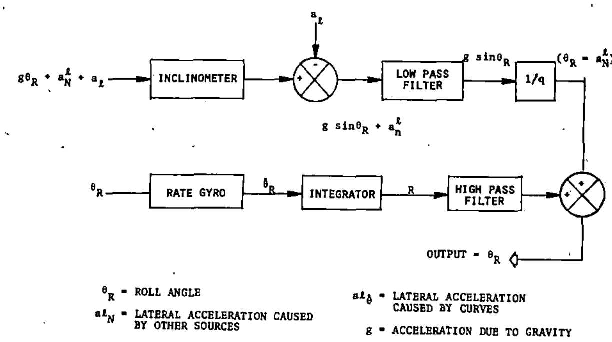


FIGURE 7. SCHEMATIC OF T-6 CROSSLEVEL SYSTEM

In order to recapture the high frequency component of the roll activity, the roll rate $\dot\theta$ of the carbody is measured by a gyro. The roll rate is integrated with respect to time and filtered by a high pass filter which is complimentary to the low pass filter used on the inclinometer output.

Crosslevel is computed by adding the carbody roll angle to a carbody-to-axle angle measured by two displacement transducers.

Profile of Each Rail

Two systems are used, the inertial profilometer and the low speed profile system. The inertial profilometer sensors are essentially the same as those used on T2 and T3; the processing of the sensor signals to obtain rail profile is changed considerably. A much larger portion of the processing is performed by computer software as opposed to analog hardware thereby improving the accuracy and stability.

Each sensor consists of a mass sliding on vertical guide roads. A spring-damper assembly isolates the mass from its mounting platform which is attached to the journal housing of an axle. The vertical motion of the wheel relative to the mass is measured by a displacement transducer. The inertial vertical motion of the mass is measured by a servo-accelerometer mounted on the mass. Figure 8 shows one of the two profilometer sensors installed on T6.

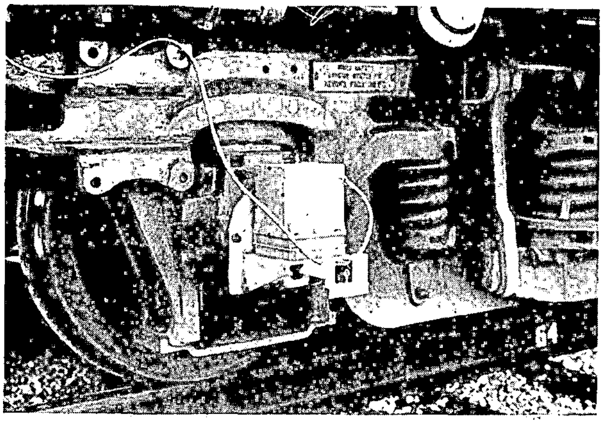


FIGURE 8. INERTIAL PROFILOMETER SENSOR

The outputs of the displacement transducer and the accelerometer are filtered and combined by software in the digital computer into an accurate mid-chord offset (MCO) of a short chord (two times the sample distance, nominally two feet). The short chord is then used to reconstruct MCO of other chord lengths, such as a 62 ft chord. Coefficients used in the software for short chord computation are adjusted automatically according to measuring speed. The effects of speed and phase distortion inherent in the accelerometer are compensated by this processing scheme. A correction for gravitational cross feed due to crosslevel, $g(1-\cos\theta)$, has also been introduced. The correction is derived from the crosslevel system output. The new processing technique lowers the minimum operating speed of the profilometer system to below 10 mph.

In order to maintain a capability to measure profile while standing still or traveling at very low speeds, a low speed profile system has been developed and installed in T6. This is the first system of this type installed in the FRA test cars. The carbody is used as a reference beam in this system to measure rail profile. Vertical motions of each of the eight wheels (relative to the carbody) are measured by displacement transducers (see figure 9). These quantities are then combined appropriately to form the MCO's of each rail in other chord lengths such as a 62 ft. chord.

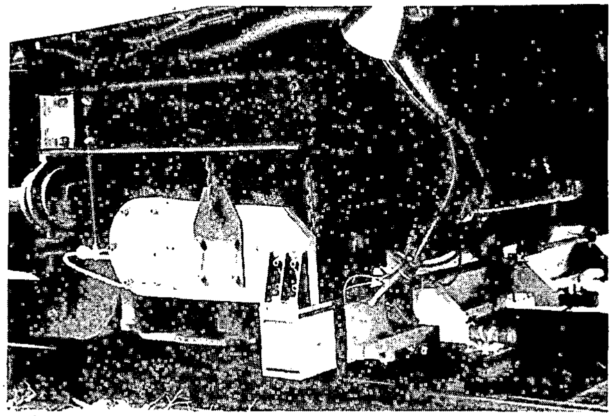


FIGURE 9. TRANSDUCER FOR LOW-SPEED PROFILE SENSOR

Alignment of Each Rail

An inertial-based alignment system is installed in T6. The concept of an inertial alignometer has been explored since the implementation of the inertial profile system on T3. Several design approaches have been unsuccessful. The system installed on T6 is the first successful implementation of the concept.

A servo-accelerometer is mounted laterally on the beam suspending the gage system (see figure 10). The accelerometer measures the inertial lateral movement of the truck. The path traveled by the truck is described by an accurate chord (a two-foot chord for a nominal sample distance of one foot) MCO derived from the accelerometer output. The distance from the truck path to each rail is measured by the gage sensors mounted on each end of the beam. The gage sensor outputs are also converted to short chord representations in the computer. These short MCO's are then combined to form alignment measurements for each rail in longer chord lengths such as a 62-foot chord.

In curved track, the lateral accelerometer output contains a gravitational acceleration component $g \sin\theta$ due to crosslevel. This component would introduce an error in the computed track path. Another error exists due to truck roll acceleration since the accelerometer is located above the rail head. Corrections are derived from the crosslevel and curvature system outputs to cancel both of the errors mentioned above.

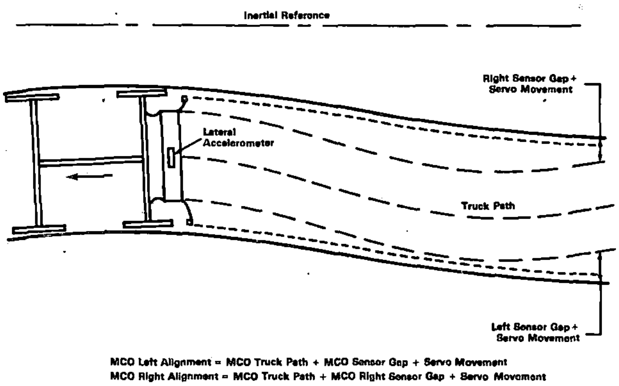


FIGURE 10. CONCEPT OF THE INERTIAL ALIGNOMETER

Track Curvature

The curvature system installed on T6 is essentially the same as those in T2 and T3.

A yaw rate gyro is used to measure the rate-of-turn of the carbody. Knowing the speed of the car allows the conversion of the time rate-of-turn to turn angle per distance.

The curvature measurement is obtained by scaling the measured instantaneous rate-of-turn distance to a rate-of-turn per 100 feet.

Yaw rate caused by oscillatory yaw motions of the carbody relative to the trucks is removed from the gyro output by measuring the relative motion with two velocity transducers. The corrected output of the curvature system therefore represents a measurement derived from truck paths. Measurements made in two passes of the same track will not repeat exactly if the paths followed by the trucks are different.

Speed and Distance

An optical encoder is mounted on a journal housing and is driven by the axle. The encoder generates direction-coded pulse trains at a rate proportional to speed.

The pulses are divided down to provide the distance-based sample rate in the computer and the drive to the paper speed of the data display charts.

A relative time clock is used in conjunction with the encoder to compute the speed of the vehicle. Except for the type and the mounting of the encoder, the speed-and-distance system is the same as those used in T2 and T3.

Automatic Location Detection (ALD)

Magnetic-type sensors are used on T6 for location detection as opposed to capacitive type used on T3 and T2. The main reason for the use of magnetic sensors is the application of their output in the gage sensor protection logic. Magnetic sensors are sensitive only to metallic targets, and insensitive to non-metallic objects such as ballast, grass and road crossings which are sensed by capacitive sensors. The use of magnetic sensors provides a more positive identification of track switches and reduces the chances of a false alarm.

Three ALD sensors are installed on the truck overlooking the left and right sides and the center of the track between the two rails. The center one is used to detect location targets for distance reference, the other two are used to trigger the gage system sensor protection logic.

COMPUTER AND SOFTWARE

A Raytheon model RDS-500 computer is installed. It is equipped with 65K of directly addressable memory, a 385k fixed-head disc drive, two 9-track, 800 BPI tape drives and an electro static printer/plotter.

The date collection and processing functions of the computer are controlled through a centralized control console which also controls the track geometry sensor operation.

The software performs several key

functions during a test, which includes:

- o Interface operator commands with computer and sensor system operation.
- o Sample track geometry sensor output at a selected distance interval (nominally at one foot).
- o Process sensor outputs into track geometry parameters as defined in FRA TSS.
- o Process sensor outputs into track geometry parameters as defined in FRA TSS.
- o Record processed data on magnetic tape.
- o Display data on distance-based strip charts. Generate exception report in accordance with the thresholds defined in the FRA TSS.
- o Interface with the rail flaw detection system.

Major features in the T6 software that are not included in previous test cars include:

- o variable sample rate,
- o software conversion of sensor signals to track geometry parameters,
- o real-time determination of transitions between tangents, spirals and curves, and
- o generation of on-line exception report.

A sample of the on-line exception report is given in figure 11.

The recorded data tapes can be processed by an off-line software package to reproduce additional strip charts or a more detailed exception report.

RAIL FLAW DETECTION SYSTEM

T6 is equipped with a rail flaw detection system which incorporated the results from the most recent research and development efforts sponsored by FRA.

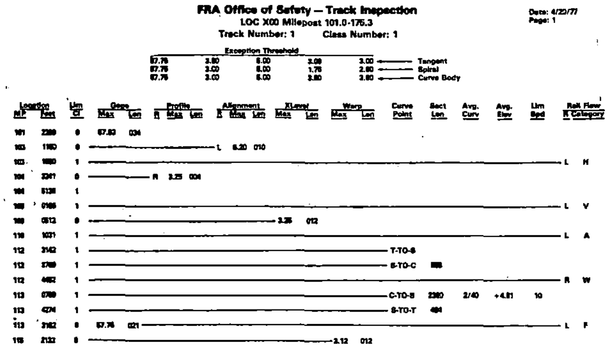


FIGURE 11. SAMPLE OF T-6 ON-LINE EXCEPTION REPORT

The system is designed to perform automatic rail flaw detection and classification up to 25 mph. The rail flaw and track geometry systems can be operated separately or simultaneously. The detected flaws in the rails will be indicated as an integral part of the track geometry exception report.

Sensor Carriage

The rail flaw sensors are mounted on a carriage under the center portion of the T6. The carriage can be raised in a stow position when not in use. Figure 12 shows the carriage in its operating position.

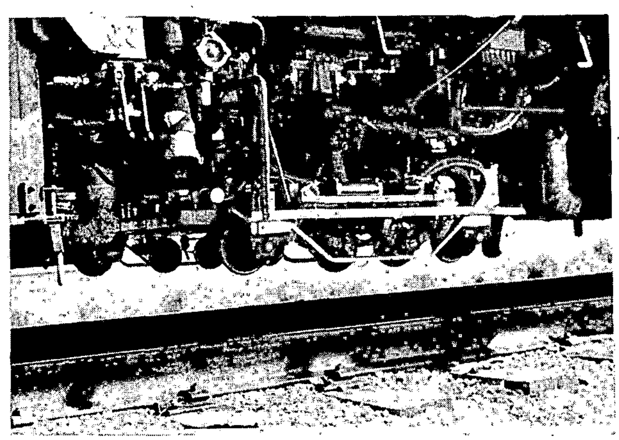


FIGURE 12. RAIL FLAW SENSOR CARRIAGE

Two wheel probes are installed on each side of the carriage. The lateral position of the wheels are guides by flanged mechanical wheels with fine adjustments for centering the sensor wheels with respect to the web of the rail, provided by a servo-hydraulic system. The distance between each pair of sensor wheels is also regulated by a servo-hydraulic scheme to compensate for changes in rail height.

The probe wheels consist of stationary yokes encased in liquid-filled elastomeric tires. Ultrasonic transducers are mounted on the stationary yokes immersed in the fluid. Figure 13 shows the aiming of the transducers in a wheel pair.

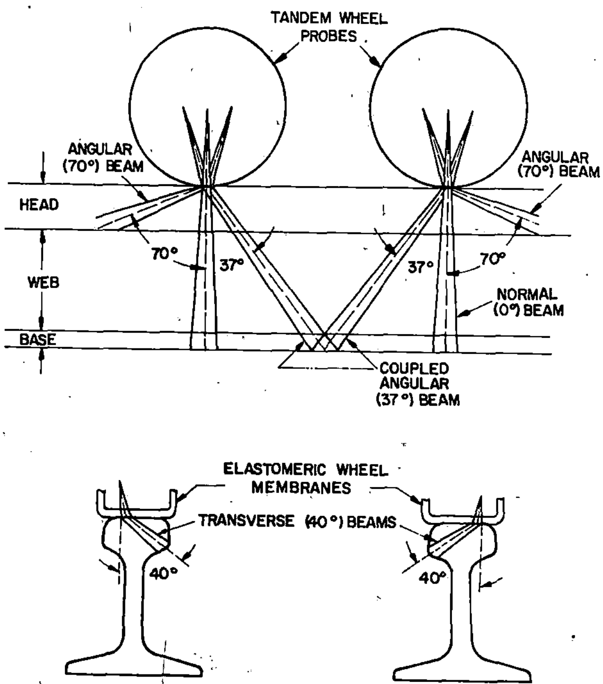


FIGURE 13. ULTRASONIC BEAMS FROM TRANSDUCERS IN A WHEEL PAIR

Sequenced pulsing of the transducers is controlled by a tachometer driven by a separate contact wheel. Typical pulse rate is 1/6" to 1/4" per pulse. To maintain proper coupling between rail and probe wheel, a water sprayer is used over each rail to keep the contact area wet. Return echos from the rail are monitored by transducers in the probe wheel. A gate and threshold analyzer is used to study each echo. Each echo is divided in time regions representing different depths in the rail. The echo strength in each region is examined and compared with anticipated returns from a perfect rail. Thresholds are set in each of the regions to detect an abnormally strong or weak return in that region. The output of a gate/threshold analyzer is therefore a sequence of binary pulses indicating whether there is or isn't abnormality in the return.

The rail flaw detection system has also been equipped with an automatic gain control to compensate for loss of signal strength due to variations in rail dimension, metallurgical properties and/or loss of coupling efficiency at the higher test speeds. Automatic gate control has also been provided to adjust critical gate position as a function of rail height.

Successive returns from the sensors contain sufficient information to categorize the locations and lengths of defects which may exist in the rail.

Joint Analysis of the Gate and Threshold Analyzer Outputs by Microprocessors

There are eleven binary output chains for each rail coming from the gate and threshold analyzers. The eleven channels are examined jointly and over successive pulses by a set of microprocessors operating in parallel. The parallel processing is duplicated for the other rail.

Programs in the microprocessors categorize the flaws into five different designations. Identifiable defects include horizontal head defects, transverse head and weld defects and angular web defects. The logic also has the capability to recognize bolt hole patterns and distinguish them from flaw patterns and to recognize coupling loss and sensor misalignment. Figure 14 shows the data flow of the rail flaw analysis system.

The information on the location and designation of a detected flaw is then transferred to the main on-board computer for incorporation into the exception report.

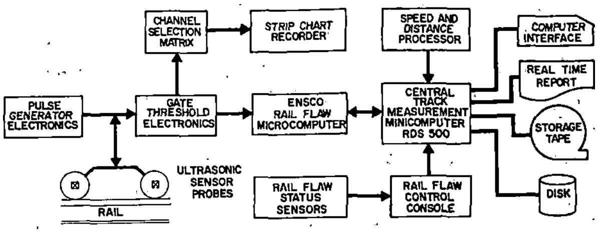


FIGURE 14. DATA FLOW OF THE RAIL FLAW ANALYSIS SYSTEM

A separate rail flaw system control console is located next to the track geometry control console. An operator can monitor and control the rail flaw detection equipment separately from the track geometry system. Return echos from any sensor can be displayed on a scope for monitoring purposes. The binary train from each gate and threshold analyzer can be displayed on a strip chart. Simultaneous examination of parallel pulse trains can verify the proper functioning of the flaw categorization logic in the microprocessors.

Parameters	Desirable Improvement	Possible Technique
Gage	Measurement Through Switches Reduce Sensor Vulnerability	Focused Optical Techniques
Alignment	Measurement Through Switches Performance at Low Speed	Focused Optical Techniques Better Inertial Sensor and Improved Processing Scheme
Combinations of TG Parameters	Detection of Dangerous Combinations or Segmented Defects in the Track	Expanded Computer Software Capability
Rail Flaw	Increase Test Speed, Improve Resolution of Detection Eliminate or Reduce Blind Spots	Improved Ultrasonic Probe Configuration, Simultaneous Pulsing of Multi-Sensors, Add Magnetic or Other Techniques, Expanded Computational Capacity
Railend Batter and Mismatch	Add New Capability	MCO of a Short Chord (<12 inches)
Track Stiffness	Add New Capability	Joint Processing of Profile (or Alignment) Measurements by Two Different Systems
Tie Condition	Add New Capability	Sonic or Ultrasonic

TABLE 3. POTENTIALS FOR FUTURE IMPROVEMENT

POTENTIALS FOR FUTURE IMPROVEMENT

The brief account of T6 instrumentation given in this article illustrates the significant advancements in the FRA test cars. These advancements make T6 one of the most fully equipped track inspection vehicles, however, there are areas in which further improvements are desirable. Several such areas are identified in Table 3. Possible methods for achieving these improvements are also suggested in the table.

Some of the potential future improvements such as detection of combinations of defects and measurement of track stiffness deal with parameters not included in the current FRA Track Safety Standards.

EXPERIENCE AND APPLICATION OF A HIGH-RAIL VEHICLE FOR TRACK MEASUREMENT BY A STATE D.O.T.

BY

M. SHERFY

Data processing, storage and collection systems on-board the Iowa Track Geometry Car and the operational format of the inspection vehicle are described herein. Programs using the track geometry data by the Iowa Department of Transportation, including a Track Geometry Rating System and a Track Geometry Priority Inspection System, are covered. This paper also reports on current research studies, under the sponsorship of the Federal Railroad Administration, to expand the application and interpretation of track geometry data.

INTRODUCTION

State participation in railroad planning represents a new endeavor for state governments. Traditionally the functions of maintenance planning and system planning have been performed solely by the railroad companies. Existing rail configurations, levels of service and maintenance policies have all resulted from the efforts of individual railroad companies, each seeking to obtain a strong economic position for itself.

A much broader perspective than the concerns held by private rail companies is taken on by state involvement in railroad planning and inspection. All actions involved in meeting the environmental, economic, and energy needs of the citizens within a state must be balanced with the development of a coordinated, intermodal system. This strong intermodal approach to transportation planning ensures a balanced relationship between transportation and the social well-being of a state's citizens.

Until recently, state involvement in railroad operations has been regulatory, accepting the existing railroad system and leaving all planning and maintenance decisions up to the railroads. With the passage of the 4R Act the state role in rail planning was expanded. Now, a state must reach decisions on which rail line must be retained within the transportation system and what methods of retaining service will be used. In Iowa these types of decisions are based on the results of a complex computer assigned freight flow analysis. Regardless of how the decisions are made it is apparent that the judgments must be based on detailed information on the quality of individual rail lines.

ADVANTAGES OF THE HIGH RAIL VEHICLE

Iowa ranks sixth in the nation in rail mileage. The present rail system, as shown in Table 1, is comprised of approximately 7,400 miles of roadway. Most of these rail lines were built before 1900.

Branch lines represent the backbone for rail service to rural agricultural states like Iowa where bulk shipments of farm products are assembled. Unfortunately the 4,300 miles of Iowa branch lines have suffered heavily from the deferred maintenance practices employed by most railroad companies.

The Iowa DOT quickly recognized the importance of a track inspection program for branch line trackage. As Iowa averages about 380 train derailments per year we concluded that the risk of derailment to a train type geo-

M. Sherfy has been Project Manager for the Iowa Department of Transportation, Office of Transportation Research since June 1977. Sherfy is a 1976 graduate from Iowa State University, with a B.S. in Industrial Administration.

metry car would be too high for its selection as a branch line inspection vehicle. The large train type geometry cars could not, in our opinion, be operated as safely as a high-rail vehicle.

IOWA TRACK GEOMETRY RESEARCH PROJECT

The need for detailed track information along with a concern over increasing derailments led the Iowa Department of Transportation to design and purchase a high rail track geometry measurement vehicle. To further improve the operation of this vehicle the Iowa DOT is working in conjunction with the Federal Railroad Administration (FRA) to expand the capabilities of this unique state-owned inspection vehicle. Under a federally sponsored research project the Iowa DOT is investigating high-rail operational questions such as:

TABLE 1
IOWA RAILROAD MILEAGE

RAILROAD COMPANY*	IOWA MILEAGE
Chicago & North Western	2,254
Chicago, Rock Island & Pacific	1,607
Chicago, Milwaukee & St. Paul & Pacific	1,579
Burlington Northern	838
Illinois Gulf Central	685
Norfolk & Western	168
Atchinson, Topeka & Santa Fe	20
Union Pacific	2
Class II Railroads	283
Total	7,436

*Companies referred to throughout this report.

We found that a high-rail vehicle would provide ready access to remote trackage with a low risk of derailment. Due to its mobility a high-rail vehicle is well suited to the inspection of branch line trackage without large amounts of nontesting travel. Highway travel can reduce back tracking, speed up between site travel and be more responsive to the scheduling problems encountered in inspecting trackage for several railroad companies.

Economics also indicated the use of a high rail vehicle as an inspection tool. Our vehicle had an initial purchase price of $125,000. New equipment for on-board data processing and storage cost is an additional $20,000. We estimate that a train type inspection vehicle would cost nearly 10 times as much to purchase. A high-rail vehicle can also be operated at a cost savings. A full sized rail inspection car needs a complete train crew and an operating crew while a high-rail car can operate with just one or two people. Other comparisons of the high-rail vehicle and the train type vehicle are shown in Table 2.

- o How do measurements collected by heavy vehicles vary from those made by light inspection vehicles?
- o Does the speed of inspection change the results of the inspection?
- o How do seasonal variations in roadbed conditions influence the inspection data?

Under the terms of the contract, operational issues regarding vehicle scheduling, access agreements, sufficiency rating systems, and safety rules are to be reported in full. One important subtask of our research contract is a study on the ability of measured geometric defects to predict non-geometric defects. If a correlation between measured geometric defects and the occurrence of non-geometric deviations such as bad ballast, bad ties, and broken angle bars is found, we will be able to expand the applications for the Iowa TGC. This correlation would allow development of rehabilitation cost estimating equations and lead to the eventual development of objective needs studies.

prove safety and operational service with a relatively small investment. A final report on this contract will be published in May 1978.

IOWA'S TRACK GEOMETRY CAR

Iowa's inspection vehicle is a light weight high-rail truck capable of measuring and recording gauge and cross levels. Figure 1 shows the Iowa Track Geometry Car (Iowa TGC) positioned on a railroad track. Gauge is measured using an under body carriage which uses a system of low-wear contact wheels. Measurements are taken 0.625 inches below the surface of the rail head. A superelevation gyro pendulum is used to correct centrifugal force

TABLE 2

VEHICLE COMPARISON*

COMPARISON	HIGH-RAIL VEHICLE	TRAIN TYPE VEHICLE
Length	20.5'	85'
Width	7.7'	10'
Weight	6 Tons	55 Tons
Weight/Axle	2 Tons	13.75 Tons
Max. Measurement Speed	30 MPH.	150 MPH.
Data Sample Interval	4.59'	2.42'
Track Geometry Parameters Measured	Gauge, Cross Level	Gauge, Cross Level, Profile, Alignment, Curvature

*Iowa Track Geometry Car represents a high-rail vehicle while one of the Federal Research Test Cars (designated T-1, T-2, T-3, and T-4) represents a train type vehicle.

The research contract with the FRA to examine both the technical and operational aspects of the Iowa TGC will aid in establishing the future role of light-weight high-rail inspection vehicles. These cars may be able to imand vehicle body lean found in measuring cross levels. Speed and distance is measured by an optical incremental encoder. This provides a recorder chart drive which is proportional to the distance traveled and a mile per hour readout. The entire measurement system and accompanying support system is housed in an environmentally protected and maintenance accessable vehicle. The system was designed to work as the vehicle moves over the track at speeds of 5 to 30 mph. However, we seldom operate the TGC below 10 mph and on Class II and III trackage rarely exceed 25 mph.

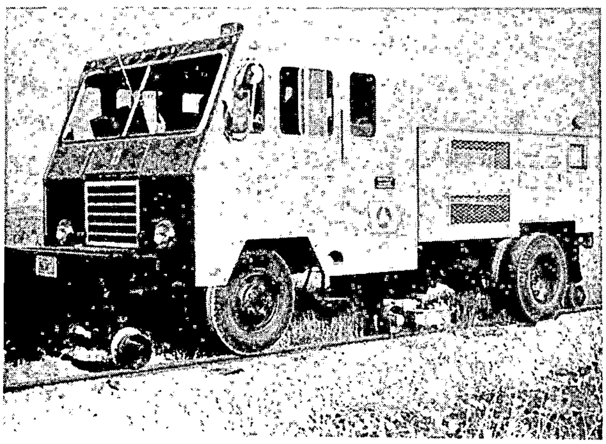


FIGURE 1. IOWA TRACK GEOMETRY CAR

Efficiency, safety, courtesy, and goodwill point toward cooperation between the Iowa DOT and railroads operating in Iowa. To obtain this cooperation we have made it a policy to obtain inspection permission by working with railroad officials to resolve any differences before scheduling a measurement test.

This unique state-owned inspection vehicle has been operating in Iowa for approximately two years. During this time our staff has emphasized the field use of the car with most of the development work in terms of reports and procedures aimed at roadmasters and lower levels of rail management. To this end we have been successful and find both the car and its reports to be well received by railroads operating in Iowa.

DATA COLLECTION & DISTRIBUTION

Data collected by the Iowa TGC is recorded on analog strip charts and magnetic tape. The strip chart has two traces for measured data and three traces for marked events -- manually inputed on-the-ground location markers such as bridges, crossroads, switches, etc. Since only one recorder unit (figure 2) is used on the Iowa TGC all strip charts must be returned to our central office to be copied before being distributed to the division office of the tested railroad. The rail carriers are asked to evaluate the charts and set priorities for their work forces. To correct critical areas the on-board rail representative is asked to keep a log so that such areas can be immediately repaired.

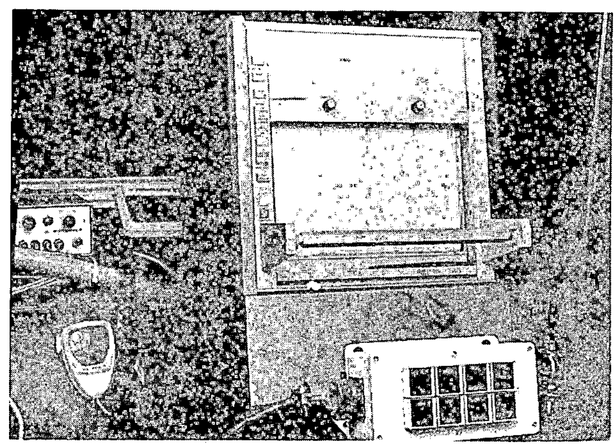


FIGURE 2. ANALOG RECORDER UNIT

Deviations in track measurements can be quickly and easily located on strip charts. We have found this format to be inexpensive and relatively easy to comprehend. However, the bulkness of the material, 31.68 inches of chart to one mile, and the non numerical form of the data together limits the application of the information to statewide analysis of the railroad network.

Equipment recently installed as part of the FRA research contract (figures 3 and 4) gives the Iowa TGC the capacity to be interfaced with our data processing center. Numerical data can now be recorded on magnetic tape for measurements collected at increments of 4.5 feet while the vehicle is operating at 25 miles per hour. This numerical data provides a base for calculating the geometry ratings discussed elsewhere in this report.

On-board data analysis is performed on a programmable calculator which functions as a minicomputer. The calculator was found to be more dependable than a minicomputer when subjected to the conditions encountered on track vehicles. Through special software programs we can use the calculator in a parallel operation thereby expanding its capacity without losing inputs when it is operating in a calc mode. This onboard processing examines each reading for deviations exceeding FRA track safety standards. Upon detection of a reading greater than standards the violation and its location is identified on the calculator output tape in the format shown in figure 2. This deviation listing is turned over to the railroad representative at the end of each testing. Hopefully this listing will replace the use of strip charts and inspection logs in locating deviations for the railroads.

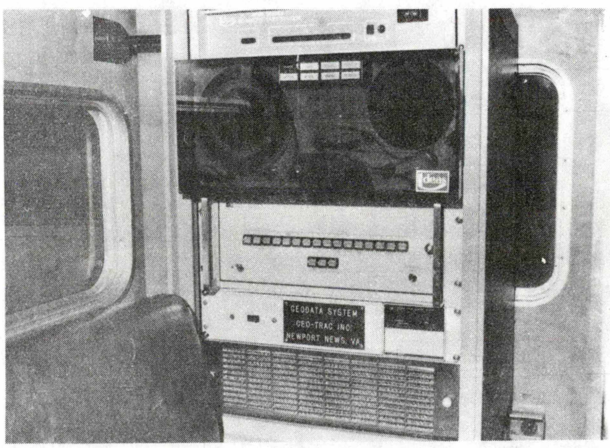


FIGURE 3. MAGNETIC TAPE SYSTEM

CREW REQUIREMENTS

One of our primary concerns is the safe and efficient operation of the Iowa TGC. This concern is best resolved when the vehicle is staffed by a two-

FIGURE 4. PROGRAMMABLE CALCULATOR

man crew. One man is needed to control the operation of the vehicle and one man handles the instrumentation. We have found the best operating procedure to be when a railroad representative and a state TGC operator work together.

The railroad representative drives the vehicle while it's on the track. This provides for safe on-track operation of the vehicle since the on-board rail representative is usually familiar with the territory being tested. This individual knows the operating rules for high-rail vehicles and is put in a position to ensure they are followed.

By precedent the cost of providing a railroad representative is absorbed by the railroads. To eliminate the liability exposure to the railroads when their employees drive the TGC the Iowa DOT has purchased a special liability insurance package. The rail representative is also requested to assist in placing the TGC on and off the track and to handle any necessary rail communications.

Presumably the railroads will continue to provide representatives who are involved in track maintenance. We believe it is best to have a representative at or above the track supervisor level so that any information developed by the Iowa TGC will be better understood on the spot. However, the prime function of the representative is to assist in the safe operation of the Iowa TGC through knowledge of the territory and the rank of the rail representative remains at the discretion of the railroad.

FIGURE 5.
SAMPLE ON-BOARD OUTPUT TAPE

DATE
22 NOV 76

RAILROAD
Ø2 C&NW

RR DIVISION
Ø3 IOWA

RR SUBDIVISION
Ø6 ANKENY

FRA TRACK CLASS
2

START MILEPOST
MP
39Ø.ØØ

STOP MILEPOST
MP
391.ØØ

TGC OPERATOR
DALE CHRISTOPHER

RR OBSERVER
REX HALL

GAGE****** -Ø.63
MP 39Ø.25

GAGE****** 1.83
MP 39Ø.76

XLEVEL**** 3.2Ø
MP 39Ø.46

XLEVEL**** -4.1Ø
MP 39Ø.97

FRA CLASS 2 VIOLATIONS

A state TGC operator drives the Iowa TGC to and from the testing sites. The operation of the instrumentation and minor in the field maintenance duties require that the state TGC operation be well versed in the technical components of the Iowa TGC. For this reason it would be difficult to loan out the Iowa TGC to individual railroads for private testing. Freeing our man from driving the vehicle while on track allows him to ensure proper testing procedures are followed and to mark the strip charts in terms of on the ground location markers such as bridges, road crossings, etc.

SCHEDULING PRACTICES

The practices adopted for scheduling affect the overall efficiency of vehicle operation and our relationship with the railroad companies serving Iowa. Poor scheduling practices cause trip cancellations, an improper ratio of idle to testing time, substantial deadheading and shortfalls in achieving production objectives. Currently the Iowa TGC is performing inspections nearly 80% of the time in the standard workday. The remaining 20% of the time is consumed by between site travel, scheduling delays and light maintenance. To obtain this high productivity the vehicle receives its normal maintenance work on weekends so that the car is ready for testing during the standard work week. A tentative testing schedule is prepared by the state TGC operator to meet the production objectives of the Iowa DOT -- approximately 900 miles per month. Approval of the schedule by the divisional railroad engineers of the track segments being inspected is then requested. When a full month's schedule if firmly established it is sent in written form to those division engineers affected by it. This work is to be completed 30 days prior to the first scheduled measurement trip. The railroad to be inspected is reminded of the upcoming inspection one week in advance of the scheduled date at the same time arrangements for meeting the rail representatives are completed. If a segment of track scheduled for inspection is cancelled we reschedule the segment at a later date. With sufficient advance notice of a cancellation we attempt to find a substitute segment of track for testing. However, lines already scheduled are not readjusted to cover the cancellation. If it is impossible to fill the open time with a test segment we use the time for light vehicle maintenance.

Our experience with a track geometry measurement car has identified several scheduling principles that have broad application. Perhaps the foremost principle is to make the trip as unobstrusive to normal operations as possible. This is followed closely by providing sufficient advance notice to the railroads for trip requests. Scheduling practices such as never scheduling more than three days in a given roadmaster's territory during a single quarter were then developed from experience.

ASSISTANCE TO A VISUAL INSPECTION PROGRAM

The basic purpose of the track geometry inspection program is to rapidly identify track segments containing geometric configurations which may be associated with train derailments. With the Iowa TGC we can annually inspect all of the rail lines within Iowa for deviations in gauge and cross levels as compared to Federal Safety Standards. Identification of such deviations is then expected to result in remedial actions by the appropriate railroad companies.

Operation of any track geometry vehicle over a track segment will not guarantee identification of all rail defects since geometry measurements only address a few of the defects that contribute to derailments. This is especially true of the Iowa TGC since it fails to measure all of the more common geometric measures such as alignment, profile, and curvature. Visual track inspections can consider all known safety standards however, they are more time consuming, are subject to human error, and cannot uniformly be used on an entire rail system. A joint program of visual and geometry vehicle type inspections may provide the maximum return for the dollars invested in an inspection program.

The Iowa DOT has developed a TGC priority ranking system in an effort to focus the inspection effort on those tracks most likely to incur track problems. The criterion for this ranking includes; 1) Service life for components of track; 2) Segment derailment history; 3) Segment deviations from FRA standards; 4) Elapsed time from last TGC inspection; and 5) Elapsed time from last visual inspection. These criteria are weighed on their relative importance and for track class to compute individual track segment rankings. The thrust of the TGC inspection priority ranking system is to provide a guide to the Railroad Division of the Iowa DOT in their routine scheduling of visual inspections. Division personnel incorporate as many of the high ranking track segments into the next quarter's operating schedule as is consistant with geographic locations, climate, special order runs and other constraints on scheduling.

GEOMETRY MEASUREMENTS USED FOR PLANNING

A wide variety of techniques, models, and simulations exist for transportation planning, however, few of them have been adapted to statewide rail planning. This analytical void has resulted of the absence of rail data needed for planning purposes. The Iowa DOT has been able to use the Iowa TGC to collect the data necessary to start adapting these techniques to railroad planning.

The information collected by the Iowa TGC has been used to develop a track geometry rating system. This system provides a numerical rating which represents track conditions for each mile of rail trackage in Iowa. The purpose behind this system is to provide administrators with a guide for programming construction funds so as to maximize benefits from project investments. This is a problem area that the Iowa DOT currently faces with its Branch Line Assistance Program and expects to encounter when working with the Federal Rail Service Continuation Assistance Program. The track geometry rating system can also be used to monitor maintenance practices. A year to year comparison of ratings will indicate the rate of progress, or lack of progress, being made in railroad maintenance.

The ratings are computed on a 30 point base, 15 points each for gauge and crosslevels in relationship to a predetermined set standard indicating overall track condition. A rating of 30 points indicates trackage where gauge and crosslevels are well within FRA standards for a particular class of track.

The track geometry ratings are summarized on colored maps and in a track geometry log. Colors used on the maps indicate whether the track geometry is good, fair, or critical. Rating distribution for each track class is shown in Table 3. Detailed descriptions of track geometry conditions are as follows:

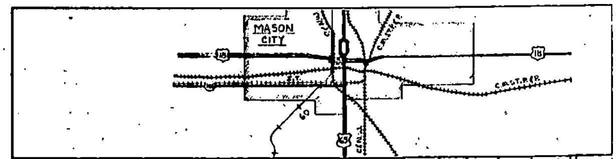
Good: Gauge and cross levels are typically well within FRA limits; occasional deviations may be found.

Fair: Gauge and cross levels frequently approach FRA limits; temporary slow orders may be common.

Critical: Deviations in FRA standards for gauge and cross levels are likely to be so frequent that significant maintenance effort may be required to maintain the present FRA track class; slow orders are likely to be in effect for extended periods of time.

TABLE 3. RATING DISTRIBUTION

	Condition		
FRA TRACK CLASS	Good	Fair	Critical
IV (High Speed)	28.0-30.0	23.0-27.9	0.0-22.9
III	27.0-30.0	20.0-26.9	0.0-19.9
II	25.0-30.0	18.0-24.9	0.0-17.9
I (Low Speed)	21.0-30.0	15.0-20.9	0.0-14.9



Route Number	Rail Class	Railroad	Milepost	Description	Section Length	Double Tr. Side	Gage	Cross Level	Rating	Weighted Map Rating
52B	I	C&NW	61		1.0		7.5	13.2	20.7	22.0
			60		1.0		8.3	13.2	21.5	
			59		1.0		12.1	13.8	25.9	
			58	Mason City CMStP&P Jct.	0.7		—	—	—	—

FIGURE 6. TRACK GEOMETRY RATING LOG

Figure 6 is a page from the track geometry log. Entries in the log include locational information, a partial rating for gauge and cross level, a total rating for each one-mile track segment and weighted rating for longer segments.

The results of the track geometry ratings are summarized in Table 4. Almost 900 miles of track have not been rated. The critical mileage in Table 4 would include 221.1 miles of track that could not be rated because the track was either out of service or in such bad condition that the Iowa TGC was unable to operate.

A track geometry rating system must not be considered as an overall sufficiency rating. Other factors such as rail, tie, and roadbed condition, ton miles, service frequency, access to alternate modes and energy usage must eventually be worked into a final rating. The conceptual design for such a rating system capable for comparing a given segment of track to other segments being developed as part of the FRA sponsored research contract with the Iowa DOT. When completed this objective rating would be of great assistance in planning the programming of public funds going into track rehabilitation.

TABLE 4
SUMMARY OF IOWA TRACK GEOMETRY RATINGS

FRA Track Class	Good		Fair		Critical			
	Miles	%	Miles	%	Miles	%	Miles	%
IV	479.4	70.0	205.5	30.0	0.0	0.0	684.9	100.0
III	1853.0	54.5	1480.6	43.6	65.9	1.9	3399.5	100.0
II	451.1	68.7	184.9	28.1	21.0	3.2	657.0	100.0
I	1170.6	64.8	434.0	24.1	200.7	11.1	1805.3	100.0
Total	3954.1	60.4	2305.0	35.2	287.6	4.4	6546.7	100.0

CONCLUSION

I would like to close, by expressing my belief that track geometry vehicles will assist in solving many of the problems found on statewide rail networks. Whether used in an enforcement or advisory mode of inspection the track geometry vehicle will assist railroad companies in developing maintenance of way programs that maximize limited railroad funding. As a uniform data collection technique the track geometry vehicle will supplement the rail planning effort of individual states. This dual need for track inspection and data collection will be an incentive for other states to purchase and operate this type of equipment. By sponsorship of studies like the Iowa Track Geometry Car Research Project the Federal Railroad Administration is expanding the application and interpretation of track geometry data, so that, appropriate systems for the analysis of railroad track are developed.

FEASIBILITY OF ROLLING STOCK PERFORMANCE DETECTION VIA AN INTEGRATED MODULAR WAYSIDE APPROACH

BY

J.D. FERGUSON

INTRODUCTION

The Federal Government has initiated R&D projects to help find solutions which will aid in reducing the occurrence of rail-vehicle caused accidents. FRA efforts to support research and development on automated trackside performance detection equipment and techniques are underway to increase fault detection efficiency and reliability, and to utilize a larger data sample for decision making. The results of these efforts will be made available to industry to aid them in their performance detection activities.

One part of FRA's effort is to sponsor research and development studies on rolling stock performance detection via an integrated modular wayside approach. This approach involves integrating various types of detection devices and equipments to sense, analyze, and display degraded performance of rail vehicle and rail vehicle components. Essential to this approach is the establishment of a research capability. This capability is being established at TTC and will provide FRA with a tool for studying various approaches to integrating and evaluating new and existing types of detection devices. The sensors and equipment utilized in these studies will be those which will have the greatest positive impact on accident statistics and economic considerations.

ACCIDENT STATISTICS AND ECONOMIC CONSIDERATIONS

Accident statistics from FRA and AAR data were analyzed for the years 1968 to 1972(1) as well as for the year 1974 (Table 1). The 1968-1972 five-year averages minimize periodic fluctuations from the data and presents a more realistic analysis of the distribution (percentages) and severity of failures. The magnitude of this contribution is listed by component and in the order of severity based on the formula:

$$\text{Severity Factor} = \frac{\text{failures/component group}}{\text{total component failures}} \times \frac{\text{accident cost/component group}}{\text{total cost of component accidents}}$$

It was found that equipment failures account for an average of 1,856 accidents per year at a cost of $41.7 million (1973 dollars) per year in damage to rail vehicles and roadbeds. The three major failure groups account for more than 81% of equipment related accidents and 84% of the accident cost. Total dollar damage data is given in Table 2. Additional losses are incurred due to damage to lading (goods), liability payments, train delays and train re-routing and are not accounted for in Table 2.

The accident statistics and economic considerations make it imperative that

(1) Bray, D.E., "Railway Accidents and Nondestructive Inspection," Paper No. 74-WA/RT-4, presented at the Winter Annual Meeting, American Society of Mechanical Engineers, New York, Nov. 1974, 1975 Rail Transportation Proceedings, American Society of Mechanical Engineers, New York, ASME 1976.

John D. Ferguson has served as Project Manager for FRA's Office of Rail Safety Research since 1976. Ferguson is a graduate of Lincoln University. Prior to his present position, he was with the Navy as a Project Engineer.

Component Failure Group	Percent of Failures	Severity Factor	Percent of Accident Cost
Wheels and Axles	39.2	2410	58.2
Truck Equipment	20.9	256	15.6
Couplers, Draft Gear and Related Parts	21.1	185	10.2
Brakes (hand brakes; air brakes, brake rigging)	10.8	35	
Car Structure Locomotive Others	8.0	49	16.0

TABLE 1. RAILED VEHICLE COMPONENTS FAILURE, SEVERITY AND COST

YEAR	DOLLAR DAMAGE*	YEAR	DOLLAR DAMAGE*	YEAR	DOLLAR DAMAGE*
1975	44,853,000	1972	34,613,000	1969	47,778,000
1974	53,562,000	1971	36,867,000	1968	41,118,000
1973	43,580,000	1970	35,786,000	1967	37,320,000

* Note: Equipment caused derailments total dollar damage reported. Damage includes track plus equipment losses from derailment.

TABLE 2. DOLLAR DAMAGES ATTRIBUTED TO ACCIDENTS

reliable detection approaches and techniques be developed to inspect components and component assemblies throughout their useful life. Economic benefits will be realized by minimizing premature replacement of components and assemblies and by timely replacement of critically defective components.

CURRENT USAGE OF PRESENT DAY DETECTION DEVICES

The current usage of detection devices is characterized by the fact that (1) each device produces individual, isolated outputs, (2) detection is based on sensing a critical condition in a single parameter before an alert or alarm status exist, and (3) many detection devices include human judgment in the assessment of a critical condition(s). Detection devices are presently used by railroads to detect hot boxes, overheated wheels, loose wheels, broken flanges, wheel surface defects, wheel flatspot, dragging equipment, high and wide load, shifted load, and L/V. The visual observer is also used to detect many of these abnormalities. Other detection devices which have potential for usage in component performance detection include a braking inspection system, infrared imaging, closed circuit T.V., and acoustic detection of cracked wheel plate. While many of these devices are used by some railroads, many other railroads fail to use these devices because of the initial investment, operating cost, and because of the none existence of data to support the effectiveness of these devices in detecting faulty performance of rolling stock. Presently, hot box detection systems are the most widely used detection devices by the railroad industry.

In view of today's increasing component related accidents, a need exists for research of potential improvements in the integration and application of new and existing types of rolling stock performance detection devices as identified by industry and the government.

AN FRA WAYSIDE DETECTION RESEARCH AND DEVELOPMENT EFFORT

In an effort to help establish a data base to support technical and economic decisions, FRA is supporting research using various types of detection devices which are currently being marketed or in some usable laboratory state of development. Emphasis must be placed on types of detection devices, rather than on the manufacturers of these devices.

This Wayside Detection Facility is composed of a group of detection systems with sensors located both in-track and on transportable stands along side the track (figure 1). The transportable sensor system and a van containing processing, recording and display electronics comprise that portion of the facility that can be moved to conduct measurements at other critical track locations. Consists will be monitored during passage over an instrumented rail section in order to detect vehicle performance characteristics and component defects such as: car dynamics, skewed axles and trucks, sticking or inoperative brakes, overheated bearings, cracked or broken wheel plates, flanges and riding surfaces, loose wheels, wheel flatspots and dragging equipment. Additional

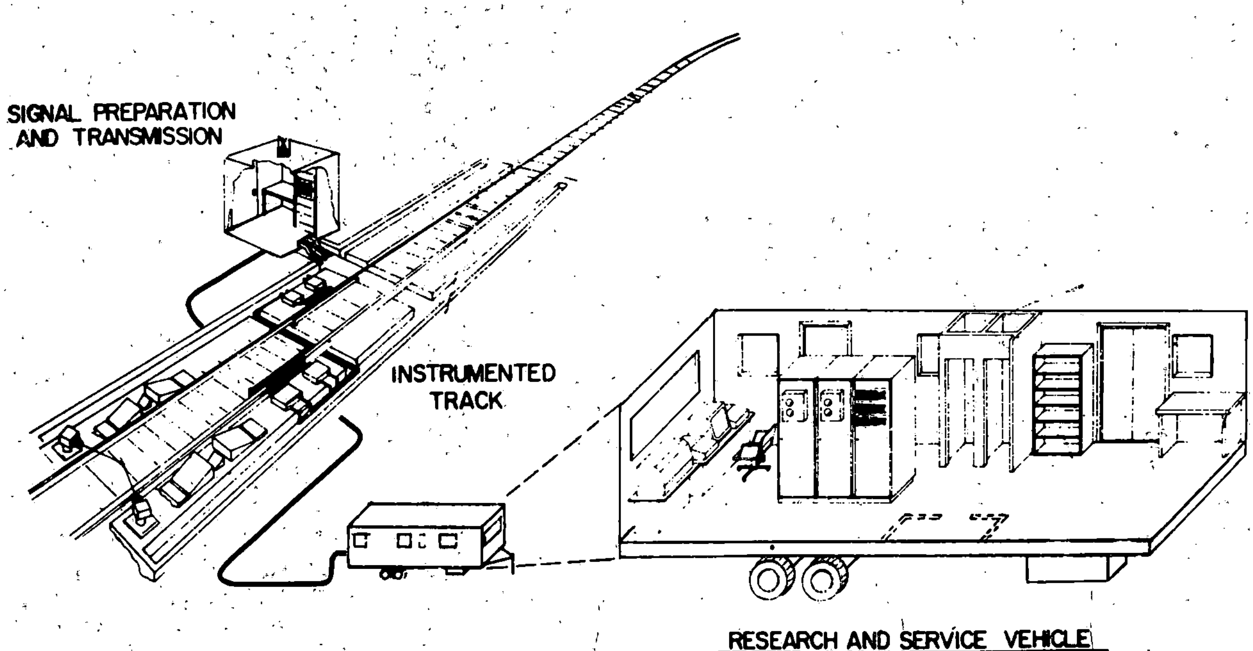


FIGURE 1. INTEGRATED MODULAR WAYSIDE DETECTION RESEARCH FACILITY

measurements to be made include weigh-in-motion, high and wide load, L/V ratio and shifted load.

INTEGRATED MODULAR WAYSIDE DETECTION RESEARCH FACILITY

One FRA sponsored research project is the development and utilization of an Integrated Modular Wayside Detection Research Facility being established at TTC to automatically detect and identify defective equipment during normal transit of consists.

Facility Usage Objectives

The Integrated Modular Wayside Detection Facility provides the nucleus of the Wayside effort. This Facility will be used as an R&D tool by the

FRA to:

- 0 Study the safety and economic benefits of simultaneous usage of detection systems to monitor hazardous vehicle performance and vehicle-component conditions.

- 0 Test, evaluate and demonstrate various types of detection systems and devices.

- 0 Aid in the development of detection systems for high speed consists.

- 0 Generate design, performance and economic data for industry's use in constructing automated wayside detection facilities.

- 0 Establish a data bank of component defects as a function of output signature for various detection systems and devices.

- 0 Detect and classify defects in vehicles components on consists.

- 0 Study data management (processing, storage, retrieval, display) and analysis techniques to improved utilization of wayside systems.

Anticipated Benefits

The following benefits are expected to result from modular integration of sensor system responses, via an automatic data analysis processing system:

- 0 Alarm Hardening - combination of sensor system responses and automatic data/signal analysis is expected to yield a significant reduction in the false alarm and failure-to-alarm rates associated with the individual sensors, as well as an improvement in the capability to identify and validate the nature of the defect giving rise to the alarm.

- 0 Improved Reporting - automatic analysis of the responses of a combination of sensors will result in near real-time reporting in a format which not only identifies alarm with its source, but identifies each source in an operationally efficient sequence, (e.g., car, truck, axle, wheel, etc.)

- 0 Configuration Flexibility - modular integration, employing a programmable analysis and report processor, will permit configuration of each facility to economically satisfy individual site requirements, including future modifications to upgrade or augment a facility.

- 0 Configuration Economics - integration and automation of facility equipments and operations are expected to result in significant procurement and operating economics by eliminating redundant sensor system components (e.g., power supplies, housings, signal processing, and specific sensor elements such as IR detectors), by obsoleting the need for specific costly sensor system components (e.g., strip chart recorders), and by virtually eliminating the requirement for manual operation and data analysis.

In addition to the immediate benefits associated with an Automated Wayside Detection Facility, there are numerous ancillary but no less significant advantages:

- 0 Standardization - deployment of these facilities is expected to result in the practical standardization of (a) equipment interfaces; (b) alarm definition, interpretation, validation (thereby eliminating most of the judgmental factors extant), and improved inspection practices, procedures and requirements.

- 0 Tracking - the inherent capability of the facility processing equipment will permit automatic transmission of vehicle ID, location, and associated fault data to a central data bank, thereby providing trend data and failure history information which may be utilized for individual car tracking as well as for large-scale failure analysis.

0 Sensor Evaluation - the data collection, reduction and display capability of the facility processing equipment will permit onsite test and evaluation of new and additional sensor types to determine suitability of such equipment in the proposed railroad environment.

Capabilities

The Wayside Detection Facility, through the combination of sensor systems and processing equipment, will be capable of monitoring the following car components and parameters, as a minimum:

0 broken and overheated bearings (journals and roller);

0 loose wheels, broken wheel flanges, worn flanges, cracked wheel plates and surfaces, wheel flatspots;

0 sticking and inoperative brakes;

0 brake rigging coming down, dragging chains, brake hoses and cables, etc.;

Detection Devices	Wheels and Axles	Trucks	Couplers	Braking System	Car Structure
Hot Box	Bearings				
Sticking Brake	Wheel Rim			Brakes	
Cracked Wheel Plate (Acoustic)	Wheel Plate				
Dragging Equipment		Truck	Couplers	Brakes	Structure
High and Wide Load					Structure/Load
Closed Circuit TV		Truck	Couplers	Brake	Structures
Thermal Vision (Infrared TV)	Wheels & Bearings			Brakes	
L/V		Truck		Brakes	
Braking Inspection				Braking	
Loose Wheel-Broken Flange	Wheel Flange				
Cracked Wheel Tread (Ultrasonic)	Wheel Tread				
Cracked Wheel (Magnetic)	Wheel Tread Wheel Flange				
Wheel Flatspot	Wheel Tread				
Weight-In-Motion	Wheel Axles	Truck		Braking	

TABLE 3. DETECTION DEVICES AND DETECTED PERFORMANCE AREA

0 car dynamics, skewed axles and/ or trucks, shifted or excessive loads;

Table 3 shows the performance areas each of the detection devices will sense.

Configuration

The basic Wayside Facility configuration will include the following sensors and sensory systems:

0 Hotbox - Sticking Brake

0 Cracked wheel surface (ultrasonic)

0 Cracked wheel plate (acoustic)

0 Braking inspection

Sensors not incorporated in the basic facility configuration due to cost or schedule constraints, as well as any newly developed or additional sensors, will be incorporated as funds and sensors become available. A list of those sensor systems is given below.

0 Loose wheel - broken flange

0 Cracked wheel surface (magnetic)

0 Wheel flatspot

0 Dragging equipment

0 Weigh-in-motion

0 L/V

0 High and wide load

0 Shifted load

0 Displacement/Velocity/Acceleration

0 Car counter and identification

0 Still camera (with time reference)

0 Motion picture camera/CCTV/ Videotape

0 Thermal images

PRIMARY DESIGN GOALS AND RESEARCH ACTIVITIES SCHEDULE

Design Goals

This facility will be used for conducting research on integrating the electronic signal outputs from various types of detection devices into a meaningful, functional, and economic systems approach to rolling stock performance detection.

The primary design goal for the wayside facility is automatic monitoring of rolling stock and identification of malfunctions, defects, failures and key performance parameters. The achievement of this goal will permit timely corrective action, without impeding normal rail transit during the monitoring operation. This goal will be accomplished by integrating the equipment, instrumentation and responses of various acoustic, ultrasonic, magnetic, infra-red, mechanical and optical sensor systems, whose outputs will be analyzed by a computerized processing unit, (see figure 2). The processing unit minicomputer will produce a near real-time diagnostic report, on both hardcopy record and CRT display. This report will identify the nature and magnitude of defects and key parameters, and the car and component with which each defect or parameter is identified. The wayside facility is intended, primarily, for installation at switching and transfer points, and will, therefore, be operational with consists traveling at speeds between 5 mph and 50 mph. A number of the component sensor systems however, are capable of individual operation with consists traveling at much higher speeds.

The wayside facility design concept represents a major attempt to advance the capability of railroad sensor sytems to provide meaningful, reliable diagnostic information in an efficient and timely manner. Integration of sensor system responses will provide more accurate identification of the nature, magnitude and location of defects than is currently obtainable from individual sensor responses, and will significantly reduce the incidence of false alarms.

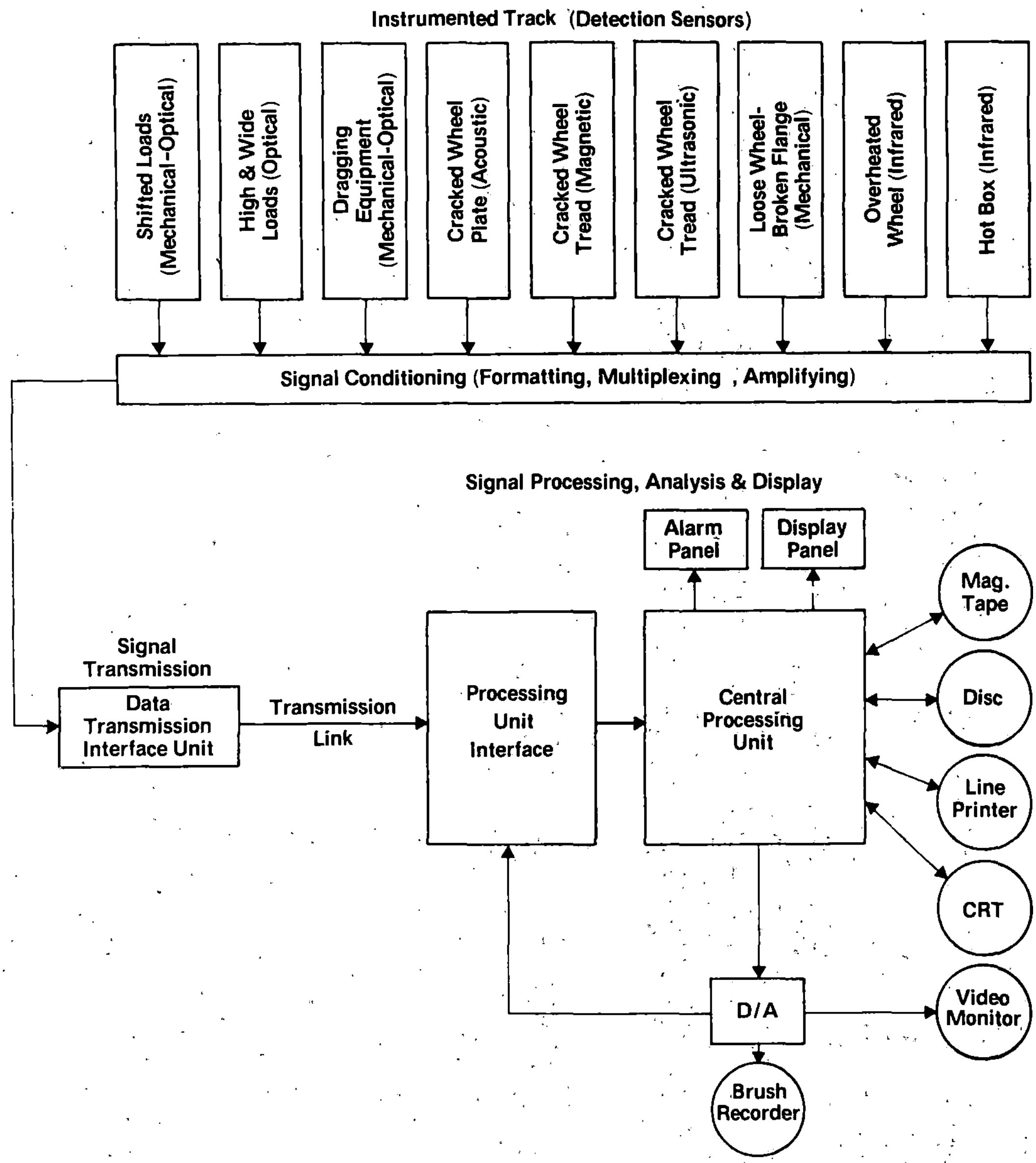


FIGURE 2. INSTRUMENTATION SCHEMATIC

Automation of the analysis and reporting processes will permit near real-time diagnosis, as compared with delayed manual data reduction currently required to analyze the responses of several of the newer sensor systems. Modular design of the facility systems will permit complete flexibility in the selection of sensor systems to be incorporated in the track instrumentation, including subsequent additions and deletions based upon operational needs and/or new sensor developments.

Research Activities

The research activities of this facility will be inclusive of, but not limited to the (1) integration of:

0 various electronic detection devices (modular approach);

0 electronic signal outputs from various devices;

0 output displays from various devices;

- 0 techniques utilized to interpret basic data measurements, and

(2) conduct of studies on:

- 0 the standardization of signal output levels and signal formats for various detection devices;
- 0 the use of a single, signal processing and integrating systems;
- 0 data processing, storage and retrieval;
- 0 transmission and display of raw data;
- 0 the reduction and analysis of basic data into statistical and functional parameters or variables.

Additional research activities will be included as the need and resources become available.

Schedule

The wayside facility development effort will be executed in three successive stages. The design and basic facility implementation stage is scheduled for completion in June, 1978. The basic R&D facility will consist of an instrumented bypass track at TTC, to be completed in April 1978, and an instrumented van which will be the facility's operations control center, containing the interfaces to the sensor systems and the data processing and display units. Track instrumentation for the basic R&D facility will include four sensor systems (brake inspection, hot bearings, cracked wheel plate, cracked wheel surface). Candidate additional sensors for the basic facility include weigh-in-motion, high and wide load, dragging equipment, L/V, shifted load, wheel flatspot and loose wheel-broken flange. Those sensors selected for the wayside facility, but not available for installation during the stage-one time frame, will be procured and installed during stage-two. The basic facility sensor systems will be individually checked out by running specially prepared consists over the instrumented track to determine the sensor response to known defects. The van processing and display equipment and software will also be validated to determine proper identification of sensor responses to consist car and component (e.g., wheel). The test and evaluation plan for the second stage will also be developed during stage-one.

During the second stage of this effort, the basic R&D facility will be augmented by integration of those additional sensors selected during stage-one, but not available for inclusion in the basic system; the track instrumentation for stage-two will ultimately consist of approximately 10-12 different sensor systems. Testing during stage-two will be an on-going process, utilizing specially prepared consists to verify sensor system and facility processing and display operation. Integration of sensor systems with the van processing unit, and augmentation of the processing minicomputer software to increase the automatic diagnostic and display capabilities of the facility, will be accomplished in a controlled succession based upon sensor installation and the on-going test results obtained.

During the final stage of this effort, beyond October, 1978, the wayside R&D facility will be used to evaluate advanced sensor systems, and to investigate the operating ranges and environment of sensor systems on an individual basis. The proximity of the facility to a high-speed track, adjacent to the wayside bypass, will permit use of the facility for evaluation of sensor systems operating with consists traveling faster than 50 mph, as well as sensors which are not necessarily intended for use in a wayside facility configuration. In addition, it is anticipated that normal FAST traffic will be diverted through the wayside bypass to provide diagnostic data to TTC, thereby enhancing their total capability to perform accelerated testing of rolling stock.

These capabilities, together with the necessary research effort as identified by the railroad industry and FRA, will provide the means for identifying improvements in the utilization and effectiveness of integrated modular vehicle performance detection devices.

EVOLUTION OF THE CONCEPT & POTENTIAL OF A RESEARCH LOCOMOTIVE AND TRAIN HANDLING EVALUATION

BY

JOHN T. WILSON
DSL DYNAMIC SCIENCES LIMITED

Significant new information leading to improved safety of rail operations will result from the early utilization of a Research Locomotive and Train Handling Evaluator. The program providing for development of the final Evaluator system concepts is well advanced. This paper describes what the program sponsors can expect to achieve with the Research Evaluator. Related work on locomotive simulator systems in the Rail Industry is reviewed to demonstrate that the expectations are well founded. For information, the basic concepts to be incorporated in the Evaluator system are reviewed. An attempt is then made to indicate the potential capability of the system in productive new areas of research. The latter will lead to a better understanding of the locomotive engineman's role and the impact of equipment innovations or operating procedures which may affect the safety and efficiency of train operations.

INTRODUCTION

Many unanswered questions persist relating to engineman performance and the operation of freight and passenger trains. The complexity of the dynamic man-machine system composed of engineman and train goes far beyond the level of simple control manipulation or behavioral response to unambiguous inputs. The engineman must retain some image of his train and the unseen terrain over which he will travel if he is to initiate control changes that will result in safe, yet effective, train operation. The difference between excellent and mediocre train handling can be attributed directly to the quality of the thought process referred to above.

The mental conditioning, aptitude and alertness of the engineman are all reflected in his performance at the controls. The evaluation of engineman performance can only be accomplished objectively in a controlled environment with a facility that has been specifically designed to progress research.

EVOLUTION OF RESEARCH EVALUATOR CONCEPTS

Historically, the study of "engineman-train" interaction has progressed from field observation on-board a real locomotive, to observation in locomotive simulators. While the simulator provided a step upward in terms of effectiveness for observation and monitoring of the engineman, it must be recognized that the evaluation of the student engineman's performance is still subjective. The instructor provides immediate interaction with the student, providing guidance as required to encourage rapid learning of the basic handling skills. In the controlled simulator environment, the instructor can "rerun" a sequence of events to demonstrate better handling philosophies. The simulator has at this point in time demonstrated the valuable role it can play in both initial training and retraining of enginemen.

The research evaluator system can be considered as the next major step in the logical progression of technological development, as it relates to the study of engineman train interaction.

The research evaluator will not replace training simulators in any way. However, the evaluator will provide a unique capability which in scope goes far beyond that realized with the simulator.

Dr. John T. Wilson is President, DSL Dynamic Sciences Limited in Montreal, Canada. Wilson received his Mechanical Engineering Degree (1962) and later obtained his Ph.D in Electrical Egineering, specializing in rail vehicle dynamics and optimal control theory.

Figure 1 is included to graphically portray the relationship of field observation, training simulator, and research evaluator concepts. The research evaluator concepts encompass all those which have evolved from field work and simulator application. In addition, new capabilities have been conceptually defined. Most significant is the fact that the evaluator will offer a uniform objective evaluation of the engineman's performance. Alternate control procedures may be assessed without incurring bias which may follow naturally from the conditioning of experienced operating personnel. While the precise nature of in-train longitudinal dynamics has only recently been dealt with in real-time, one of the evaluator concepts is to have a real-time system, with the flexibility to accommodate any of the documented train models which have resulted from extensive research programs in the industry.

EVOLUTION OF CAPABILITIES

FIELD – SIMULATOR – EVALUATOR

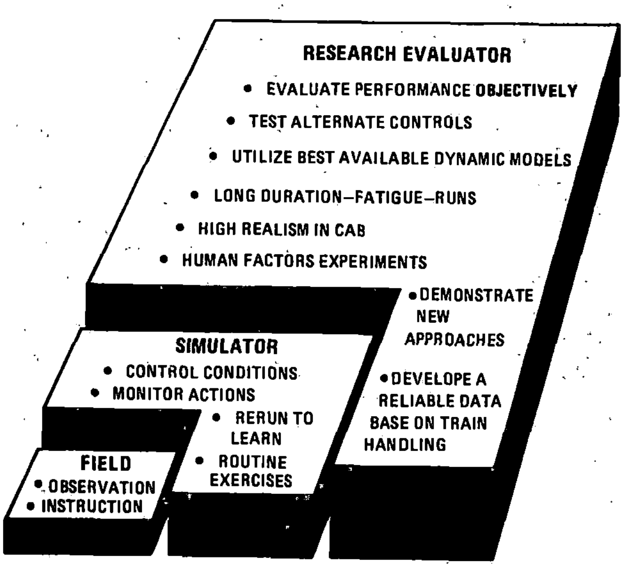


FIGURE 1. EVOLUTION OF SYSTEMS CAPABILITY

The evaluator facility will be equipped to support all manner of basic human factors experiments. Two main areas of research may be denoted. The first relates to the performance of the engineman under several different sets of conditions presented in normal or difficult road operation. The loss of engineman effectiveness with fatigue or as a result of undesirable environmental conditions is a matter of concern. The second research area is involved with the evaluation of design innovations. The ergonomic aspects of design changes will be studied, permitting some optimization of new controls or displays from the viewpoint of engineman interaction or comprehension.

REVIEW OF TRAINING SIMULATOR SYSTEMS

Training simulators were first introduced to the rail industry in 1970. Two major operations, The Southern Pacific Transportation Company and The Atchison, Topeka and Santa Fe Railway both embarked on simulator based programs in that year (1,2). The Canadian National Railways followed, initiating simulator training early in 1973, and expanding its training facility (3) with more advanced simulators in 1974 and again in 1977. While all of these systems shared some basic design concepts, each of the five simulator realizations has been unique.

In each of the training simulators built to date a minicomputer has been included to calculate the progression of the locomotive over the terrain. The same machine has also coordinated the visual, sound, motion and display systems which give a student the sensation of riding and controlling the train. The degree of realism achieved with the implementation of each of the major subsystems has been significantly different from unit to unit. To indicate the variations in design some examples will be given.

Two of the five simulator units do not replicate full sized locomotive cabs, but still provide the complete engineman's work space with dimensional accuracy. Of the full sized locomotive cabs, one is a light-weight mockup of a real cab, while the other two are actual locomotive cabs produced on the assembly lines of two different locomotive manufacturers. In every case the wiring, piping and support structures have been custom designed, to facilitate the interaction of the engineman with computer interface devices which simulate the real-life locomotive and train.

The visual systems implemented have been based on both 16 mm and 35 mm filming and projection. Three of the five simulators were designed to operate with 35 mm film which offers superior resolution and wider field of view. While two simulators provide for rear projection on plain screens, one provides a rear projection system with a "lens" shaped window to obtain a "virtual image" display. Two simulators are equipped with spherical screen segments on which images are front projected via aspheric mirrors.

As far as motion systems are concerned, all five units have different capabilities. The amplitude of motion provided for in each of the principal directions and the amount of cab roll which may be induced is the same in only two of the five cases.

Figures 2 and 3 are included to demonstrate the level of realism obtained in the latest Training Simulator, "OSCAR IV", operated by CN Railways. As figure 2 indicates, the instructor's module has been integrated with the full size locomotive cab to provide the characteristic appearance of a real unit, complete even to steps and handrails. The photograph of the student's operating station in figure 3 shows the forward view provided by the curved screen visual system. A computer generated track profile display and in-cab signal unit are also shown forward of the student. These provide unique training cues which may be activated at the instructor's discretion.

In figure 4, the full complement of instructor's display and control devices is shown. While extensive CRT prompting is provided, the instructor can interact with the controlling computer through a numeric key pad and an array of 40 malfunction switches on the left side of the module. The controls at the right side of the module provide for mode selection, signal display and projector controls. It may be seen that the instructor has a clear view of the student through a sliding window, and can monitor both track profile and down track visual displays.

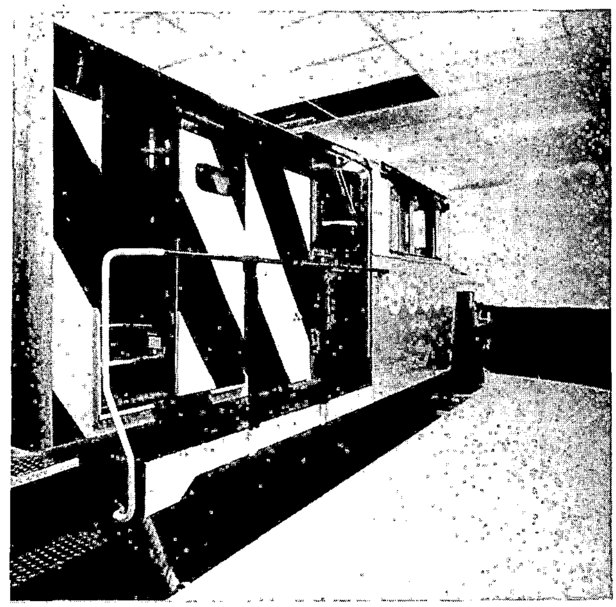


FIGURE 2. A MODERN TRAINING SIMULATOR "OSCAR IV"

FIGURE 3. ENGINEMAN'S CAB--OSCAR IV SIMULATOR

The various training simulator installations referred to include significantly different computer systems. Only the two newest units have provision for realistic computation of the longitudinal dynamic motions within the train. The real-time computation of longitudinal dynamics with independent motion of all cars in the train was achieved with a hybrid computer system introduced by CN Railways in 1976. The impact of this development has not yet been fully assessed in training since the first production hybrid has to date been utilized extensively for engineering research activities.

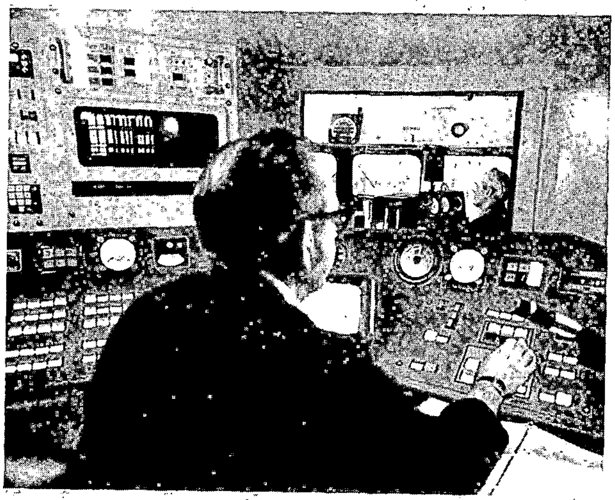


FIGURE 4. INSTRUCTOR'S MODULES FORWARD VIEW

All three railways which have utilized simulators agree on the importance of the total training facility concept. Classroom instruction and preparatory training on equipment mockups take a significant amount of a student engineman's time. This peripheral instruction permits most effective utilization of time on the simulator units.

DESCRIPTION OF THE RESEARCH EVALUATION FACILITY

The research evaluator facility would be configured to provide a highly realistic locomotive cab environment much like that afforded by the newest of the simulators described in the previous section. However, the construction of the cab system would be modular to permit reconfiguration of controls and displays for experiment purposes. To demonstrate the total complement of systems, a schematic of the research evaluator concept is included in figure 5.

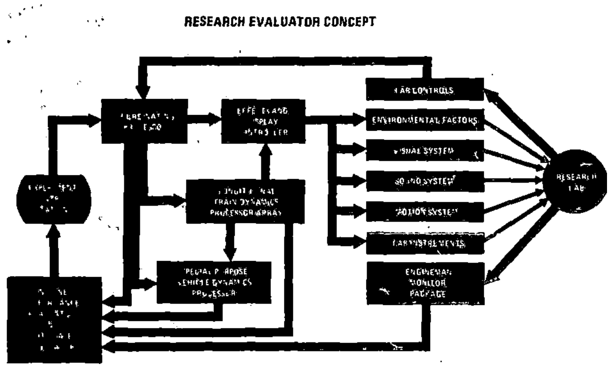


FIGURE 5. SCHEMATIC OF THE RESEARCH EVALUATOR CONCEPT

A coordinating processor under the control of personnel in the experiment control station would direct the total complement of subsystems in the following manner. An effects and display controller could be dedicated to synchronize all the significant perceptual cues for the engineman. Environmental factors including temperature and humidity would be carefully controlled. A full-fledged visual system featuring multiple projectors and spherical screen would be required to generate a high resolution forward display with selection of events on-line. The sound system, cab motion system, and cab instruments must all be driven to fully replicate the real life circumstances in all modes of operation. Cab controls must be continually monitored as feedback to the coordinating processor so that the total system is responsive to the engineman's action.

While the systems described in the foregoing could well be included in an advanced simulator, the unique features of the research evaluator will become more evident as the remaining elements in figure 5 are discussed. As additional observations or feedback to the experiment group, a comprehensive selection of biotechnical variables will be monitored to disclose the condition of the engineman. Telemetry links and observer cameras will monitor the engineman and provide the data for on-line evaluation.

Aside from monitoring the engineman and evaluating his physical condition, it would also be necessary to ascertain the state of his locomotive and train system. Both the longitudinal dynamics of the train and the characteristic dynamic responses of selected vehicles in the train must be faithfully generated. This requirement can be met if dedicated processors are employed. As indicated in an earlier paper (4) by the author, solution of the longitudinal train dynamics equations has not been accomplished in real-time with an all digital computer system. In fact, with a large computer such as the IBM 360/75, experience has shown that computer time taken to simulate the real phenomena would exceed the time duration of the actual phenomena, by a factor of 6 to 10. Nonetheless, recent developments have shown that it is possible to achieve real-time solution of the train dynamic equations with an array of dedicated processors. The objective for the research evaluator train action hardware design, is to permit the representation of a very general model, which will allow inclusion of a variety of draft gear performance characteristics. Complete flexibility in the definition of individual cars and their braking equipment must also be provided for. The dimension of the longitudinal train dynamics problem is formidable as it stands. It would not be practical to consider all the degrees of freedom or all the vehicles in the train. However, it would be practical to incorporate special purpose vehicle dynamics processors to produce the response of selected vehicles in a train. Thus, vehicles which are of special interest would be modeled in complete detail. Lateral loads on track caused by specific locomotives or cars would be estimated.

The final block included in figure 5 is labeled "On-line Performance Evaluator and Data Base Generator". All of the significant inputs from engineman monitor package, longitudinal train dynamics processor array, and special purpose vehicle dynamics processor hardware feed into the on-line performance evaluation equipment. Speed, cab control information and other significant factors also feed forward from the coordinating processor. Through the execution of advanced algorithms on the processor based evaluator system, an objective evaluation of the performance of engineman and train will be implemented. This feature of the research evaluator is of primary importance.

UTILIZATION OF THE EVALUATOR

When completed, the evaluator facility will offer the industry a new means of problem solving and investigation. In many of the problems encountered in road operation the capability of the engineman, or the adequacy of information made available to him, must be taken into account. For management to deal effectively with such problems, sound technical data is required.

All data relating to the human element involves statistical variation. In order to generate data of statistical significance, closely controlled experiments will be required. Even then, careful planning of experiments and screening of subject enginemen will be required to ensure that results are not biased by any extraneous factors. Once the nature of possible investigations has been indicated, it should be more apparent that the evaluator offers the only practical experimental approach.

To demonstrate the nature of investigations, several suggested experimental objectives are set out below:

a) Assess the basic nature of the "engineman-train" interaction. Determine quantitatively what control procedures are actually common.

b) Assess the degree of variation in train handling as demonstrated by a significant number of enginemen in critical train control situations.

c) Determine which control procedures are most effective in handling over difficult terrain. This experiment has several dimensions since length and consist of train can be varied as well as the severity and type of track undulation.

d) Determine which control procedures are most reliable under adverse operating conditions. A selection of the best handling procedures identified in experiment (c) above would serve as a basis for the start of this experiment. The reason for this study is to select from the good policies those which provide the greatest margin of safety.

e) Assess the relative difficulty of a variety of normal train-handling tasks. Indicate which tasks require greatest concentration or precision. In later experiments, it will then be possible to pay special attention to performance of the most difficult tasks. It could be conjectured that the performance of these would be the first to degrade when fatigue and loss of alertness or concentration are experienced. Determine the most common types of human error experienced under demanding situations. Rank the significance of the error from the viewpoint of impact on safety of operations.

f) Determine the most common types of human error experienced under demanding situations. Rank the significance of the error from the viewpoint of impact on safety of operations.

g) Determine which tasks are downgraded or ignored by the engine-man when he is subjected to high stress, long work hours, or environmental upsets.

h) Assess the relative importance of human states which affect task performance. For example, is the state resulting from physical fatigue equivalent to the state resulting from boredom. Contrast hours on-the-job versus hours under heavy workload. Determine which combinations of train, terrain, and speed create most monotonous conditions from the engineman's viewpoint.

i) Determine the effect of work-rest cycles, the effect of variation in hours worked per week, and in the case of two-man crews, determine the best task rotation schedule.

j) Determine the relative proficiency of selected populations of engineers who have been assigned to different types of service for extended periods. For example, show how skills differ between enginemen from high speed short duration runs and enginemen from long duration drag runs.

k) Demonstrate the effectiveness of alternate training or retraining schemes. Examine the performance of individuals trained under a variety of different schemes.

l) Assess the merits of alternate design concepts for new cab displays, controls, and communications or signal equipment.

While the list of objectives is far from complete, it includes many experimental objectives which could not be considered without the evaluator. It is apparent that much attention will be given to performance of the engineman at times when his task is most demanding. These would be exactly the times when it would be most inappropriate to employ any instrumentation or observers in a real-life operation. The potentially hazardous situations which will be investigated on the evaluator, could not be set up with live equipment because of the risks involved. Indeed, some advantage may be taken of the evaluator's capability to simulate in detail the events which could lead up to an accident. When utilized to re-enact events preceding a known accident, the evaluator may provide information which would show how the accident could be avoided.

CONCLUSION

An effort has been made to describe how the locomotive and train handling research evaluator will be configured and utilized. While there are similarities between the evaluator unit and the newest of the engineman training simulators, the evaluator will clearly be more complex. Nonetheless, past experience with training units supports the conclusion that much beneficial research can be performed with the evaluator and the need for objective evaluation of performance.

It is the author's conviction that much can be gained through the early development of the Research Evaluator. A better understanding of the engineman's task and normal performance will result. This, in turn, will yield improved safety of operations, better use of human resources and, hopefully, new economics in operation.

BIBLIOGRAPHY

1. Adams, G. B., The Dynamics of the Locomotive Simulator; AAR Track/Train Dynamics Conference, Dec. 15, 1971.

2. Chastain, H. N., Resume of Locomotive Simulator Training and Equipment; AAR Track/Train Dynamics Conference, Dec. 15, 1971.

3. Wilson, J. T., Development of an Advanced Simulator for Training of Locomotive Crews; Fourth International Symposium on Railroad Cybernetics, Washington, April, 1974.

4. Wilson, J. T., Train Action on Undulating Profiles; AAR Track/Train Dynamics Conference, Dec. 15, 1971.

QUESTIONS SESSION III

Session Chairman -- L. A. Peterson

Attendee: G. Platt, Phillips Petroleum Company

Attendees's Question: At the present accelerated level of investment in the upgrading of rails and roadbeds, what is the estimate in years to obtain near completion of this task? The second part of the question, what is the expected life of the roadbeds and rails that have been upgraded to date?

R. Parsons: George, its premature at this time to answer your question. As part of FRA studies required by section 504, of the 4R Act, we are working with the industry to determine the actual amount of deferred maintenance. The answer to both your questions hinges on this rather elusive figure. Once the magnitude is agreed upon within the agency, we'll be in a better position to answer meaningful questions like yours.

Attendee: G. Platt, Phillips Petroleum Company

Attendee's Question: With regard to the third part of my question, is there any movement in the Federal Railroad Administration to fund rail and roadbed upgrading.

R. Parsons: The answer is a definite yes. Sections 505 and 511 of the Act provide preference share financing and loan guarantees to the railroads for this purpose. In addition, the majority of the $1.6 million of the public and private funds for the Northeast Corridor are for this purpose. In the last year we in research have started to investigate cost effective maintenance of way procedures to help AMTRAK and CONRAIL and other railroads.

Attendee: E. Dailey, Koppers Company

Attendee's Question: Do you attribute any grade crossing accidents to the crossings' surface itself?

L. Peterson: At the present time the collection of accident data is improving both the type information gathered and its usefulness in determining safety trends. Information this detailed, while it would be useful, is considered a secondary issue. The expense and time involved with collecting this kind of data currently offsets its benefits.

Attendee: G. Holabeck, Bethlehem Steel Corporation

Attendee's Question: Forty-two percent of the accidents are attributed to track conditions. Could you break this down by geographical location; northeast, southwest, west, and so forth?

L. Peterson: I don't happen to have those figures at my fingertips but the FRA Accident/Incident bulletins break down the accidents not only by geographical districts but also by railroad and by state. The 1975 and 1976 bulletins are currently available from the FRA Office of Safety.

Attendee: A. Gillispie, Transportation Safety Institute

Attendee's Question: Are other rail flaw detection techniques under development or consideration; in addition do you foresee the demise of the magnetic particle technique for rapid inspection? What means are now being used for surface flaw detection at high speed where visual inspection is impractical?

H. Ceccon: The FRA is beginning an investigation into the area of magnetic detection techniques. We have a current contract that's looking at data processing techniques for systems presently in the field. The effort is centered around the AAR residual magnetic system and we are also looking at some new magnetic techniques. These are problems, however, at high speed with magnetic inspections sytems. Two new techniques being investigated are an eddy current technique and the use of Hall Effect Probes. This work will be finished in a few months and based upon the results, we may build a prototype and test it in the field. Regarding the rest of your question; no, I don't think magnetics are on the way out. With the T-6 vehicle, we determined something is needed to back up our ultrasonic system. Certain defects, on engine burns for instance, are giving our systems a problem. A magnetic system can give a no test indication from engine burn and this would be valuable information. One of the problems with magnetic systems however, is that you don't get good penetrations in the rail. A lot of people talk about the Russian system as an answer. Two years ago, I was in Russia and rode on that system for 50 kilometers. I think the reason we didn't find any defects is because there weren't any defects within the first 10 millimeters of the rail head. As operational speeds increase with the magnetic system, the skim effect increases and the penetration is limited. In conclusion, I think that the system is a very good complimentary system for ultrasonics.

Attendee: L. W. Lemmon, Pacific Car & Foundry Company

Attendee's Questions: You mentioned some flaws are not deteriable, can you specify more fully what kind you cannot pick up?

H. Ceccon: Did you mean the single wheel that I talked about earlier?

L. W. Lemmon: Either one, yes.

H. Ceccon: I mentioned earlier the development of the tandem wheel probe. There are two defects we had in mind as we started improving the sensors. One is the vertical split head located off the center of the rail. The other is engine burn fracture. One of the things we're looking for in our sensor development program, is a sensor which will give us an indication of a defect in more than one channel. We want to ensure that our false alarm rate is low, therefore the more channels in which a defect appears, the higher the probability that a defect actually exists. So, we're attempting to see a defect in at least two channels.

Attendee: K. Smith, Electro-Motive Division

Attendee's Question: Is the track geometry data generated by the T-6 track car used to study rail problem areas other than those you have discussed such as rate of change of gage, etc.?

T. Yang: T-6 has just been completed and is not yet operational. Therefore, we have not used T-6 in that respect but that is possible future use of the system. One new item being evaluated on T-6, is changing the normal sampling rate to one foot per sample as compared with 2.4 feet per sample used with T-2 and T-3. This will give much better coverage in shorter wavelengths. The alignment system now measures alignment on both rails. Many lateral hunting problems and derailment problems have been identified as closely related to rail alignment. Perhaps, with the T-6 alignment system, we may not have to look at track gage change and other similar problems. We may be able to go directly to the problem in the future.

Attendee: E. Lombardi, National Railroad Passenger Corp. (AMTRAK)

Attendee's Question: As a result of the FRA study of locomotive operating cabs, is it the administration's intent to promulgate rules or standards for cab design?

L. Peterson: Speaking as a research type, we would prefer voluntary adoption of beneficial changes. Because of the cooperative work that's been done the industry, the locomotive manufacturers and the railroads have incorporated, voluntarily, some 22 changes in the cab to make them safer. This is the way we prefer to operate.

Attendee: J. Robinson, Boeing Vertol

Attendee's Question: Does the human factors program entail modification of locomotive systems such as braking, fault detection, diagnostics and propulsion control?

J. Wilson: One of the purposes of the evaluator will be to evaluate existing and proposed control systems. The evaluator will provide an objective means of evaluating these systems and certainly will tend to encourage new ideas.

Attendee: K. Cappel, Wyle Laboratories

Attendee's Question: The concept given in the presentation has been developed and implemented by the aircraft industry, the military, and the airlines over the past 20 years. The most complex and costly component has been the visual display where discrepancies are most readily perceived by the subject. How will this problem be addressed in the proposed research evaluator with particular regard to the interactive aspects?

J. Wilson: You're right, it's a very difficult problem. I'd like to point out first that the requirement for the flight simulator or space vehicle simulator, would be completely different from the problem we're faced with in either the locomotive simulator, or more particularly, in this research evaluator where great detail is important. For the most part, in aviation systems it has been possible to generate simple displays-computer generated displays are often used. It's my feeling, at this time, that a visual system based on film will be necessary to provide the clarity of view and the realism required in terms of interactive aspects. However, CGI (Computer Generated Images) might have some limited use in the evaluator.

L. Peterson: You're also saying John, I think, that the displays proposed for the research evaluator will be greatly improved over existing displays.

Attendee: H. Law, Clemson University

Attendee's Question: What types of models and analysis procedures are envisioned for the longitudinal train dynamics? For example, how many degrees of freedom per car will be provided? Will equations of motion be integrated using numerical, analog, or hybrid techniques? What braking models will be used? What non-linears will be included? Will coupling to lateral dynamics and vice versa be considered?

L. Peterson: Before you answer John; Harry, I would like to say that at this stage we're close to coming out with an RFP which will attempt to deal with these problems. John can give some of his opinions on how these might be handled but, I hope you don't take it as a way that the evaluator will be constructed.

J. Wilson: Well, I'll just make some suggestions. The idea in the schematic which I showed is one which I believe is logical. One, we treat the longitudinal dynamics of the train as one package, then the detailed dynamics of some individual car or many separate individual cars as a problem for a sub-system or sub-processor systems. All models would have to be non-linear, in my estimation, to be useful. For the longitudinal model of the train, I would attempt to do the same thing that I've done previously with digital computers in less than real time. That would be to consider only the longitudinal degree of freedom for each and every vehicle in the train. It would be a programmable hardware system rather than a hybrid system. As you know, I built a hybrid system which has solved this problem in real time. I think it's a very cost effective solution to the longitudinal train dynamics problem. The only thing it lacks is flexibility. In terms of braking models, we do have a number of good empirical braking models and I believe that these are sufficient, or that some small improvement of them would be good, for this research evaluator application. In other words, we don't need to have a full blown dynamic model of a braking system. To me,that is a lot of fun, but it's a research project on its own.

Attendee: R. A. Caruth, Maxson Corporation

Attendee's Question: You stated that 25,000 miles of track are being surveyed annually. When was this type effort started, also when will a completed cycle be performed?

L. Peterson: I'm sure you're speaking about the DOT cars, the first inspections were performed in 1973. At the present time, we're inspecting upwards of 25,000 miles of track this year, while our goal is 30,000 miles of track per car a year. I'd like to make it clear at this stage that we don't consider these cars are being developed exclusively for use by the government. We hope that the techniques developed can be used by the industry in a more routine way, preferrably as part of the railroad business by the railroads themselves. That brings us to the second question, what is the planned re-cycling period for going over trackage surveyed? Again frequency probably will be dependent upon several considerations including the past safety record, the class, gross tons, type of traffic, speed, etc. However, what I think the question alludes to, is what are the Government's intentions concerning track inspections? That depends on how well the railroads advance on their own. I think the Government is going to do whatever is necessary in the inspection area to continue progress to ensure safe rail operations.

Attendee: W. Shannon, Trailer Train

Attendee's Question: What spacing or distance is contemplated between wayside inspection locations in order to detect potential failures of freight car components before they progress to the point of causing derailments?

L. Peterson: John, before you answer that, I'd like to preface this by saying John's paper was entitled, "A Concept" and he's going to investigate a number of ideas in a pilot project at TTC.

J. Ferguson: Due to the present limited scope of the project, FRA has not initiated any deployment studies. However, it is evident that station spacing will be a function of the effectiveness of the wayside instrumentation group at any given location and on the propagation rate(s) of various defects.

Attendee: K. Sammoul, MTS Systems Corporation

Attendee's Question: A significant number of derailments are caused by suspension component failures. Have you considered any form of wayside test facility to test and evaluate suspension characteristics and perhaps detect failed springs or shock absorbers that could lead to derailment?

L. Peterson: Initial work indicates an indirect way; that is to measure what kind of forces, both vertical, lateral and the L/V rate behavior is as a vehicle goes over an instrumented section of track. Based upon a correlation of the deviation in relationships or signatures, we can infer, hopefully, that certain behavior corresponds to identified suspension malfunctions such as a broken spring. As far as any direct method, I would like John to respond.

J. Ferguson: Our test approach will be just as Leavitt described. Since we don't know all of the forces and other measurable anomolies expected from defective components, such as broken springs, we will generate a data bank of performance characteristics using a dedicated consist containing known defects. Controlled experimentation will allow us to relate the defects to various sensor responses.

Attendee: F. King, Canadian National Railways

Attendee's Question: Have you considered using the sensors to monitor the speed of trains passing a test site? This can be done with a very modest increase in cost.

J. Ferguson: Yes, we have considered this as a possible cost effective approach.

Attendee: F. Danahy, Association of American Railroads

Attendee's Question: Your cost of accidents caused by equipment; are they normalized to constant dollars over the last nine years and what part of this is a result of wheel failure that could be detected by the proposed system?

J. Ferguson: Yes, this has taken inflation into account and, therefore, we feel these dollars are realistic. The portion that's attributed to wheel and axle failure is something like 40 to 50 percent of the related accidents and approximately 58 percent of the related cost.

L. Peterson: I think you have something in mind when you asked what part of this is a result of wheel failures that could be detected by the proposed system. Are you referring to anything in particular? Plate cracks?

F. Danahy: I was referring to things that, perhaps, couldn't be detected by this sytem such as a thin flange that meets a switch in the yard. Plate cracks, broken wheels, the 40 to 50 percent surprises me, I, like you, haven't read the FRA accident report for so many years that I don't recall it either.

J. Ferguson: What you mentioned could be detected. We've looked at what can be detected and what is detected and, with present day technology, close to 70 to 80 percent of the accidents that occurred as a result of what you mentioned could be detected if present day technology were utilized to its maximum capability. With the systems we have now, I can't tell you what that percentage is and that's part of this program, to try to improve what can be done with just the systems we presently have in our hands.

L. Peterson: Are there any systems that detect thin flange now? Are there any systems that could be utilized for that? I recall that the Wheel Checkers system did isolate thin flanges as some switches do.

J. Ferguson: No, not the thin flange per se.

Attendee: M. W. Giesking, Wheel Checkers, Inc.

Attendee's Response: The Wheel Checker Loose Wheel - Broken Flange Detection will detect thin flanges.

Attendee: J. Read, B&LERR

Attendee's Question: Is any attempt being made to develop engineering track standards? How about the ability to detect sun kinks before they appear?

L. Peterson: There's been a lot of interest and activity on our part to develop engineered track standards. Many current problems in derailment explanations would benefit from such an approach; but it has some formidable obstacles.

R. Parsons: I don't want to talk all night because we're tired, but the first part of your question, that's probably the most frustrating part of my job since I've been in this seat. A lot of it is the fault, I think, of the private sector not really coming forward and helping us in the public sector, in a coordinated approach, to come up with "engineering standards for track." We've got a law on the book which you've seen in this presentation and we're pursuing with vigor to enforce it in a much more efficient manner. It places much, much more importance on how realistic are our present track standards. Attempts by my office to date to get active industry participation in this program have not had much success. So, we (FRA) alone right now, have been spending 4 to 5 million dollars on a multi-faceted track program which was not presented today.

L. Peterson: The second part then John, about the ability to detect sun kinks before they appear. We have some research going on which is looking both at the causes of buckled track and at detection and measurement of longitudinal stresses. Unless somebody else out there knows something I don't, we don't have anything very reliable at this time except the track inspectors and the people who look at the track condition and the location and effectiveness of rail anchors.

Attendee: B. Pinnes, Department of Transportation

Attendee's Question: It is my understanding that the accident rates in practically all categories are continuing to deteriorate. You showed us a conceptual plot indicating the reversal of this trend. I have two questions. First, when can we expect to see such a reversal, second, what are the most promising mechanisms for achieving the goal?

L. Peterson: Before I answer, Bob, I want to put something up front.We're having a hard time getting a broad base agreement as to what those measures should be. It certainly isn't the actual number of accidents or fatalities that occur. It has to be comparative number, such as the fatalities per revenue ton mile or per train mile or something like that. We and others are struggling to agree on a good measure. Given a good index, when could you expect such a reversal? I think I'd be very foolish to stand there and say that research alone is going to make that reversal. The major requisite for any dramatic turn around is that the track, equipment etc., are going to have to receive the care and maintenance they need. I don't think the key in the reversal is research alone, although research work can help you decide where to put the priorities and how to best spend your money. However, it would be very presumptuous to say that research work is going to do it all. It's got to be a combination with the 4R Act and so forth leading. Second, as we saw in our recent Soviet visit; it's the railroads doing their job and getting the track, equipment and the operations into a state that is both economical and safe. As for the "most promising mechanism", research can only assist in accomplishment of unfavorable trends reversal through strategies, tactics and open interactions which are focused on down-to-earth real problems. Of course, establishment of priorities toward achievement in the areas of maximum potential improvement is crucial in making the comparatively limited R&D resources in the railroad industry.

SESSION IV REPORT ON ASSOCIATION OF AMERICAN RAILROAD'S R&D ACTIVITIES

Session Chairman. . . . . *George H. Way*
Assistant Vice President, AAR

REMARKS TO FRA ENGINEERING CONFERENCE

BY

G. WAY

Executive Summary

The AAR's Research Program is in a real sense a misnomer. There really is no such thing as an AAR Research Program. We cannot claim pride of authorship over most of what we do nor are we unhappy over this fact. It is by intention that we cannot, for reasons I will refer to.

NIH in this case refers to, "Not Invented Here". NIH has been used to describe the common sociological phenomenon of prejudice against or mistrust of an idea developed outside of our own institution or organization.

I do not want to condemn NIH, even though it has been responsible for delaying the acceptance of many good technological advances. Nor am I here to extol its virtues, even though I recognize that often a solution developed on one property is not necessarily appropriate to the peculiar circumstances which may prevail on another. Or that a widget invented in an ivory tower may not function as intended in the cold practical world of reality.

On the surface NIH is simply a human frailty engineers and researchers had better learn to deal with effectively. But on careful reflection I think we can all agree that NIH exists in part for justifiable reasons. We learned at Chessie's Research Department that if Operating or Engineering Department people with a problem participated in its investigation the eventual solution was easier to sell and more rapidly implemented. Not simply because the users had pride of authorship, although that should not be quickly dismissed, but most importantly because when the users of technology are a part of its development it is invariably better technology.

The rail industry is frequently criticized as being outmoded, backward in its thinking, slow to adopt new ideas, and overly cautious. Some of this criticism is simply unjustified. Most stems from an admittedly poor public image in which many opinionmakers sense on one hand, that railroads offer significant social advantage in terms of energy usage and minimal adverse environmental effect; but on the other that railroad managements lack the skill or desire to bring these advantages to fruition. We at the AAR are trying to correct this mistaken view. However, some of the criticism is in fact justified. Like many other mature industries we in the Railroad Business have been conservative.

We have in many instances acted in a reactionary fashion rather one of anticipation. There are situations when we have reacted too slowly. Like government, mature industries, railroads included, are bureaucracies. And all bureaucracies exhibit a considerable degree of inertia.

There should not be an AAR research program and there is not. There is little to be gained by proprietary research programs regardless of their sponsors. Just as rail failures of freight design problems can't claim institutional cause, neither should their investigation be institutional. These are industry problems and they demand industry-wide approaches to their solution.

FRACTURE PROPERTIES OF AAR CAST STEELS

BY

W. S. Pellini
D. H. Stone

Association of American Railroads
Research and Test Department

Executive Summary

Fracture test data for AAR cast steels were developed by the Case Western Reserve University from samples provided by RPI foundries. The data involved Dynamic Tear (DT), Drop Weight Test (NDT) and Charpy V properties. Steels of Grades B, C (N&T), C (Q&T) and E, from various sources, were included in the study.

This paper presents interpretative fracture mechanics analyses of the data. The analyses clearly define the railroad service temperatures for which these steels feature brittle, semi-ductile and ductile fracture properties.

The fracture properties were also examined in relationship to metallurgical variables, such as composition and microstructures. It is shown that the fracture properties of Q&T steels of C and E grades are seriously degraded by the presence of pearlitic microstructures. The importance of adequate alloy content (hardenability) and effective quenching of castings is emphasized by these data.

It is recommended that fracture properties be specified in terms of the DT test. Minimum fracture-energy values at 0°F are suggested for the C (Q&T) and E steels, and at 120° for the B and C (N&T) steels.

The single most important <u>practical</u> question, is to establish if fracture-safe couplers can be guaranteed for railroad service temperatures. Everything else simply contributes to this objective, i.e., are matters of detail.

The most important conclusions are, that couplers can be produced economically which -

(1) completely eliminates fracture problems to lowest service temperatures for the Q&T steels
(2) have high commercial-production reproducibility of fracture properties
(3) are specified in modern engineering terms for fracture properties
(4) have improved weldability for repairs, etc.
(5) will satisfy all concerned that casting properties can be reasonably guaranteed, as well as test-block properties.

ON THE EFFECT OF TRACK GEOMETRY ON VEHICLE RESPONSE IN CURVE NEGOTIATION

BY

E. H. Chang
and
D. R. Sutliff

Executive Summary

Spiral lengths and the amount of superelevation for simple and reverse curves have been a subject of concern to railroad engineers and maintenance personnel for many years. This concern is primarily motivated by the desire to improve track geometry, to decrease track maintenance, to improve train handling and, of utmost importance, to ensure the safe operation of the railroad. The previous study on "Minimum Tangent Length between Reverse Curves for Slow Speed Operation" indicates the usefulness of the Quasi-Static Lateral Train Stability (QLTS) model in studying track geometries for various types of railway vehicle operation. It is the intent of this paper to study the effects of spiral length and superelevation on vehicle response when they are introduced to simple curves and to reverse curves with minimum tangent length. The selection of critical vehicle consists are based on the population, weight and length of vehicles. The parameters which are analyzed include the lateral over vertical force ratio (L/V), the lateral coupler angle and the overturning speed of the vehicle. The analysis is performed for the empty consists because the L/V ratios are more critical for the light car than for the loaded car in curve negotiation. The methodology used in selecting "optimum" spiral length for a given consist and track configuration is presented. The "optimum" spiral lengths for a given amount of superelevation are determined and compared with those based on existing standards or formulas.

In this study, no attempt has been made to cover all the possible car combinations, loading conditions, or track configurations. Since the L/V ratios for the light cars are more critical than the loaded cars in curve negotiation, the analysis would be solely based on the empty consists. Only those consists considered to be representative and critical are analyzed. Due to the fact that the weight and the height of the center-of-gravity of the loaded cars would be different from the light cars, the results and recommendations presented in this paper must be applied with care in practice for the partially or fully loaded consists.

FREIGHT CAR HUNTING MODELS AND THEIR VALIDATION

BY

Y. H. Tse
V. K. Garg
D. R. Sutliff

Executive Summary

The operating speed of a freight car is often limited by 'truck hunting', a self-excited vibration phenomenon in which a truck oscillates violently in the lateral direction of travel. The paper describes two mathematical models developed at the Association of American Railroads under the Track Train Dynamics Program to investigate the hunting behavior of a freight car. One of the models is a linear model and is based on the eigenvalue-eigenvector solution of the equations of motion. The other is a non-linear model which provides a time domain solution for the system. The field test conducted at the Union Pacific Railroad to validate the hunting models is briefly discussed. Results of the models are compared with the field test data.

STRUCTURAL DYNAMIC ANALYSIS & FATIGUE LIFE PREDICTION OF A FLAT CAR

BY

V. K. Garg
B. Prasad
A. M. Zarembski

Executive Summary

Dynamic characteristics of a trailer-on-flat car (TOFC) are investigated using finite-element techniques. Three different finite-element models of the flat car were developed. These models were validated by comparing the predicted vibration mode shapes and frequencies with test results. Further validation of the models is carried here by comparing the analytical transfer function with test results. The values of the transfer functions are computed, using NASTRAN, at four different locations along the center line of the flat car. Experimental values at these locations are found to be in good agreement with the computed results. Using the space beam model of the flat car fatigue life of an arbitrarily selected member of the flat car is then carried here to demonstrate the application of finite-element structural dynamics analysis to fatigue life prediction. The fatigue life values obtained using this approach, are then compared with the so called ad hoc approach which uses a nominal stress value obtained from a pseudo-static analysis.

SIMULATION COST MODELING FOR THE DETERMINATION OF FREIGHT CAR COMPONENT OPERATING COSTS

BY

Keith L. Hawthorne
Allan I. Krauter
Rajendra Saroop

Executive Summary

This paper describes a technique developed to determine freight car component operating costs. Early in the planning of the Phase II Track Train Dynamics Program, the need for a methodology to determine freight car component operating costs was established. The proposed creation of performance specifications necessitated the establishment of a methodology to determine the economic impact of design changes. After a review of alternative methods of cost determination, a technique known as Simulation Cost Modeling (SCM) was selected. Simulation Cost Modeling has three elements: the schematic diagram which describes component usage; the computer program which implements the diagram; and the input data set. Development of the Simulation Cost Modeling technique and its relationship to Track Train Dynamic component cost evaluations are described. A preliminary input data set and corresponding approximate results are given for the freight car wheel. This reference case approximates the current usage of freight car wheels and the associated costs of acquiring, operating, and maintaining freight car wheels.

REMARKS TO FRA ENGINEERING CONFERENCE

BY

G. WAY

Good Morning, I am supposed to make a few brief introductory remarks today about AAR's research program. I will confine myself to philosophy and let those that follow address technology and details of specific programs.

The AAR's Research Program is in a real sense a misnomer. There really is no such thing as an AAR Research Program. We cannot claim pride of authorship over most of what we do nor are we unhappy over this fact. It is by intention that we cannot, for reasons I will refer to, but let me digress.

When I am in Washington I drive to the office by way of New York Avenue. At its eastern extremity, New York Avenue parallels the rail line constructed between 1903 and 1907. To bring the then Pennsylvania, Baltimore, and Ohio Railroad's trackage into Washington's new Union Station. Across from the Hecht Company Warehouse is the Ivy City Team Track Yard and Ivy City Engine House.

I first came to know Ivy City when, as a kid, my parents took me by train from our home in New Jersey to visit their families in West Virginia. In the late thirties there were always elegant President Class Pacifics being serviced at Ivy City to be admired as the train slowed coming into Washington, twenty years later as a brand-new ignorant and naive Assistant Supervisor on the Pennsylvania, many of the GG-1's I rode -- to check the safety performance of track gangs -- called Ivy City their home. The GG-1's except for one, aren't as handsome now as they were then and they share Ivy City with E-60's and SDP-40's instead of P-7's, but there is usually something interesting to see over to the right when New York Avenue traffic backs up. For the past several weeks there has been a new attraction, the United Aircraft Turbo trains, their red paint faded to pink, looking abandoned just as Train X did in Indianapolis in 1960. As a railroad Engineer and technologist, I worry about such failures just as I take pride in things like the success story of the GG-1, even though I had no part in either. There they sit GG-1's and Turbo trains side by side at Ivy City to remind us how good or how fruitless railroad technology can be.

In thinking about this problem, one of the factors which keeps surfacing in my mind is the NIH syndrome; NIH in this case refers to, "Not Invented Here". NIH has been used to describe the common sociological phenomenon of prejudice against or mistrust of an idea developed outside of our own institution or organization.

I do not want to condemn NIH, even though it has been responsible for delaying the acceptance of many good technological advances. Nor am I here to extol its virtues, even though I recognize that often a solution developed on one property is not necessarily appropriate to the peculiar circumstances which may prevail on another. Or that a widget invented in an ivory tower may not function as intended in the cold practical world of reality.

On the surface NIH is simply a human frailty engineers and researchers had better learn to deal with effectively. But on careful reflection I think we can all agree that NIH exists in part for justifiable reasons. We learned at Chessie's Research Department that if Operating or Engineering Department people with a problem participated in its investigation the eventual solution was easier to sell and more rapidly implemented. Not simply

George H. Way, Jr. is Assistant Vice President of the Research and Test Department, AAR. Way received his B.S.E. from Princeton University, with honors, (1952).

because the users had pride of authorship, although that should not be quickly dismissed, but most importantly because when the users of technology are a part of its development it is invariably better technology. Scientists and engineers isolated in drafting rooms and laboratories away from real-world railroad environments cannot be expected to deal with those environments successfully. They need to rub shoulders with the mechanics and craftsmen who will be using the products of their research. They need to get their hands dirty struggling with real problems, if their research is to be on target.

We accept this principle at the AAR as gospel. That is why I said at the beginning that we do not have an AAR Research Program. We participate in a research program and we attempt to give that program focus. But, the program is the Railroad Industry's program and it is just as much the property of the other participants as it is ours. The other participants in this program have allegiance to a host of supply firms, railroad companies, universities, government agencies and trade associations. But underneath these formal differences is a common interest in railroad problems. This is the basis of cooperative research. From an administrative standpoint it tends to be a nightmare. Lines of authority are difficult to define and more difficult to enforce; its contractural arrangements are at times close to impossible. Politically it is tedious. From the standpoint of gross cost it can be expensive. Frequently it is time consuming and inefficient. Yet with all of these problems, all of us at the AAR feel that it is a bargain. Since cooperative research produces workable solutions, and in the long run not only the best solutions, but frequently the only acceptable solutions. It allows the best people to work on a problem without having to be pirated from their jobs. It permits cost sharing in terms of dollars, equipment or manpower, and, it allows the users to participate and direct programs aimed at solving their problems.

The railroad industry is frequently criticized as being outmoded, backward in its thinking, slow to adopt new ideas, and overly cautious. Some of this criticism is simply unjustified. Most stems from an admittedly poor public image in which many opinion-makers sense on one hand, that railroads offer significant social advantage in terms of energy usage and minimal adverse environmental effect; but on the other that railroad managements lack the skill or desire to bring these advantages to fruition. We at the AAR are trying to correct this mistaken view. However, some of the criticism is in fact justified. Like many other mature industries we in the Railroad Business have been conservative. As Bob Gallamore said last night, "We operate in an environment of constraints". We have in many instances acted in a reactionary fashion rather one of anticipation. There are situations when we have reacted too slowly. Like government, mature industries, railroads included, are bureaucracies. And all bureaucracies exhibit a considerable degree of inertia. Like a 20,000-ton unit coal train, it takes an enormous amount of energy to set them in motion, and equal amount to slow or stop them, and constant attention to see that they get to where they ought to go. At the same time they are singularly efficient mechanisms for handling large volumes of homogenous commodities or work. Engineering research, however, does not constitute large volumes of homogenous investigation. Therefore, cumbersome, bureaucracies are not ideally suited to the research function. Ad hocracies as first defined by Alvin Toffler, on the other hand, are more easily assembled, disbanded and reshaped. In other words, they are more flexible. Consequently, they are better able to react quickly to a changing environment. They may even be able to act in an anticipatory fashion. Cooperative research attempts to operate on this principle. With little or no permanent staff, cooperative project teams do not suffer the stifling bureaucratic tendency to self-preservation and expansion and are, therefore, free to devote their engergies to the task at hand.

So I say again, there should not be an AAR research program and there is not. There is little to be gained by proprietary research programs regardless of their sponsors. Just as rail failures or freight design problems can't claim institutional cause, neither should their investigation be institutional. These are industry problems and they demand industry-wide approaches to their solution.

FRACTURE PROPERTIES OF AAR CAST STEELS

BY

W.S. PELLINI

D.H. STONE

ASSOCIATION OF AMERICAN RAILROADS
RESEARCH AND TEST DEPARTMENT

Fracture test data for AAR cast steels were developed by the Case Western Reserve University from samples provided by RPI foundries. The data involved Dynamic Tear (DT), Drop Weight Test (NDT) and Charpy V properties. Steels of Grades B, C (N&T), C (Q&T) and E, from various sources were included in the study.

This paper presents interpretative fracture mechanics analyses of the data. The analyses clearly define the railroad service temperature for which these steels feature brittle, semi-ductile and ductile fracture properties. The fracture properties were also examined in relationship to metallurgical variable, such as composition and microstructures. It is shown that the fracture properties of Q&T steels of C and E grades are seriously degraded by the presence of pearlitic microstructures. The importance of adequate alloy content (hardenability) and effective quenching of castings is emphasized by these data.

Recommendations are presented for (1) optimization of fracture properties and (2) for revised AAR specifications based on DT test reference to fracture mechanics criteria.

INTRODUCTION

A survey of fracture properties of commercially produced AAR cast steels was conducted as part of the Railroad Coupler Safety Research and Test Project (RPI-AAR Program). All aspects of the preparation were intended to be equivalent to commercial production. The test samples were cast and heat-treated at RPI member foundries as tapered plates of 1.5 to 0.75 inch thickness (figure 1). The fracture tests were performed at Case Western Reserve University (CWRU) under the direction of Prof. J.F. Wallace. A report was prepared which presents and summarizes the data.[1] The CWRU data was provided to the AAR Metallurgy Division by the Project Director, for purposes of fracture mechanics and metallurgical analyses. The fracture test data includes grades B, C (N&T), C (Q&T) and E steels.

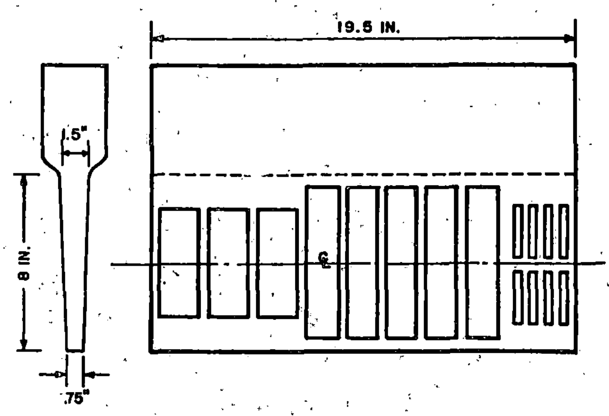


FIGURE 1. TAPERED PLATE CASTING FOR TEST SAMPLES

The first objective of the analyses was to characterize the fracture properties of these steels in modern fracture mechanics terms. The analyses also include detailed examination of the

---

W. S. Pellini, a graduate of Carnegie Institute of Technology, is in private consulting practice having retired as Superintendent of the Metallurgy Division of the Naval Research Laboratory. Mr. Pellini has received numerous awards for his scientific and engineering achievements, including recognition from the Department of Defense and the U.S. Navy.

D. H. Stone is Director-Metallurgy at the Association of American Railroads Technical Center in Chicago, Ill. Mr. Stone holds a B.S. in Metallurgical Engineering from the University of Pittsburgh and received his Masters degree from UCLA in 1966.

effects of metallurgical variables on fracture properties. These analyses indicate desirable directions for statistical improvements of fracture properties for the C (Q&T) and E steels.

Because of the pending revisions of the AAR specifications involving M-201, M-210 and M-211, these analyses are also of interest in establishing appropriate fracture toughness specification requirements. Since the pay-off of the fracture test studies is, in fact, improved steels covered by improved specifications, emphasis is placed on these aspects.

The use of modern fracture mechanics principles for fracture properties characterization, analysis in context with metallurgical factors, and in specifications is essential at this time. To do otherwise is neglecting modern technological practices, now widely used in the general engineering field.

The subject of fracture mechanics principles and methods requires general introduction. The report provides this introduction, reduced to the most essential aspects.

The essence of fracture mechanics methods is exact prediction of the service performance of the steels as brittle, semi-ductile, or fully ductile. These predictions can be made with high confidence for specific service temperatures. Extensive documentation, including failure analyses, exists for the case of rolled or forged steels. Thus, the comparisons take advantage of this existing information and serve as engineering documentation for the validity of the findings developed for the cast steels.

AAR use of fracture mechanics principles is not new. For example, the TC-128 tank car steels were characterized and analyzed statistically in fracture mechanics terms. Predictions of service performance, in the case of railroad accidents, were fully validated. This is presented in a report developed for the Railroad Tank Car Safety Research and Test Project (RPI-AAR).[2] Rail steels are another example of the use of modern fracture mechanics principles for characterization and analysis of properties. It is to be expected that fracture mechanics will be applied to all fracture-related engineering problems in the railroad industry. Such application obviously involves use in AAR specifications.

FRACTURE MECHANICS PRINCIPLES

The fracture transition of steels evolves over a range of temperatures as indicated in figure 2. The critical temperature reference points are defined as L (plane strain limit) and YC (yield criterion).

At temperatures below L, the steel is highly brittle (defined as plane strain). Fractures may be initiated from relatively small crack-like defects. The fractures then may extend through regions of the structural component, featuring nominal engineering stresses as low as $0.2\sigma_{ys}$. (0.2 of the yield stress).

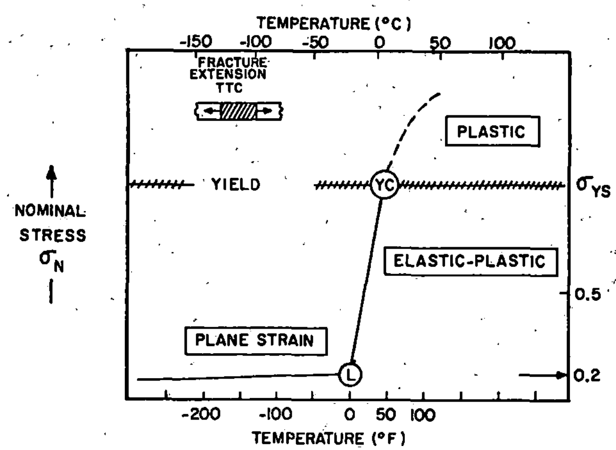


FIGURE 2. ENGINEERING SIGNIFICANCE OF L-YC TRANSITION FROM PLANE STRAIN BRITTLE TO PLASTIC (DUCTILE) FRACTURE

At temperatures between L and YC there is a sharp increase in fracture extension resistance. The nominal engineering stress (σ_N) increases from approximately 0.2 σ_{ys} at L to 1.0 σ_{ys} (yield level) at YC. This is the origin of the term "yield criterion" (YC). Above the YC temperature, the fracture can be extended only by plastic overload stresses, i.e., it is a plastic fracture.

The L to YC temperature range is called the elastic-plastic (E-P) transition, i.e., brittle to plastic fracture transition.

All steels of the strength level of interest to this report have a specific L to YC transition temperature range. This temperature range can be moved up or down the temperature scale by changing the microstructural quality features of the steel.

By defining the L to YC transition temperature range, the steel quality is defined with respect to service temperatures.

The correspondence to performance in service is direct. Service temperatures below L signify that brittle fractures can develop. Service temperatures above YC signify that only ductile fracture (by overload) can develop.

For most structures, a fracture value of 0.5 YC (halfway between L and YC) is generally sufficient to prevent fracture. This is due to the fact most structures are designed to nominal stresses less than $0.5\sigma_{ys}$.

Because of possibilities for impact overloads in railroad service, it is judged that fracture-prone components, such as couplers, should have fracture properties at least equal to the range of 0.5 YC to YC (high elastic-plastic). Preferably, the high end of the range (close to YC) should be used as the minimum fracture toughness for the lowest service temperature of interest.

The practical way to define the L to YC temperature range of a steel is to conduct a Dynamic Tear (DT) test. Figure 3 illustrates the DT test specimen used in these studies. It is an ASTM defined fracture test specimen.[3]

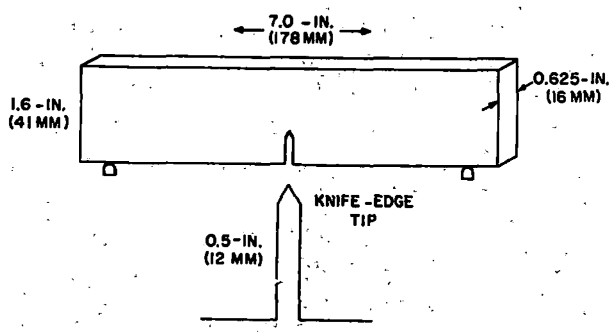


FIGURE 3. STANDARD (ASTM) DT TEST SPECIMEN

A series of DT test specimens are broken using anvil impact and the energy transition is recorded as illustrated schematically in figure 4. Other features of the Transition include:

* change of fracture mode from brittle to ductile.
* change of the anvil-tup force-time curves as shown in the inserts of figure 4. (Force-time data are used only in research).

It should be noted the NDT temperature obtained by the ASTM Drop Weight Test corresponds to the point of initial rise of the DT energy curve, i.e., the L point.

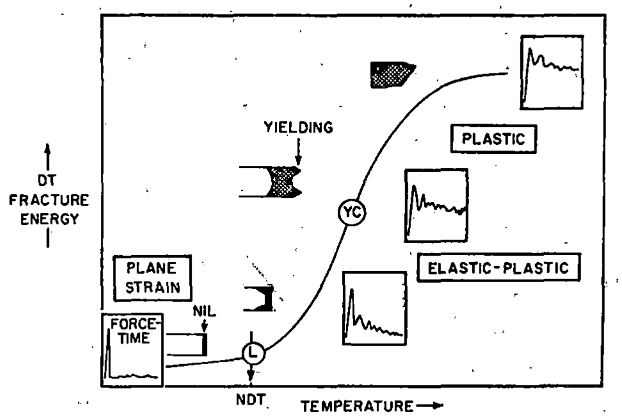


FIGURE 4. FEATURES OF DT TEST TRANSITION

Figure 5 illustrates the method of indexing L and YC temperatures to the DT test energy curve, as follows:

(1) The L point corresponds to the NDT temperature and to the temperature of initial rise of the DT energy curve. For this report both reference methods are used, and the corresponding L temperatures are averaged.

(2) The YC point corresponds to the 50% energy and to the 50% shear fracture temperatures. Either are reliable and conservative indications that YC properties are attained. For this report both reference methods are used and the corresponding YC temperatures are averaged.

The top of figure 5 illustrates the simplified engineering method of plotting the L to YC temperature range as a simple straight line.

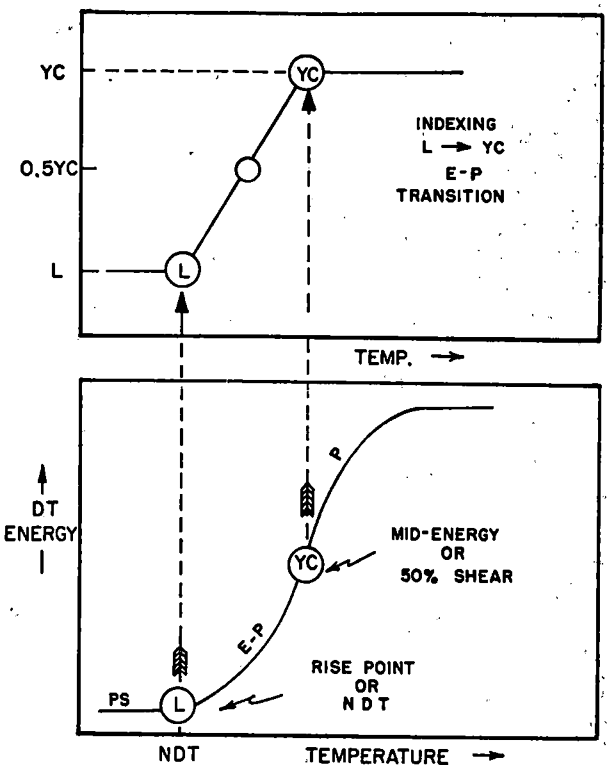


FIGURE 5. INDEXING OF L-YC TRANSITION TO DT TEST CURVES AND TO NDT

All DT and NDT data developed in the project was plotted and analyzed by the use of this simple straight-line reference for the L to YC transition temperature range of the steels.

DATA ANALYSIS

Figure 6 presents the bands for grades B and C (N&T) steels. Note these steels have exactly the same bands of L to YC fracture properties. Note also these steels are indicated to be susceptible to brittle fracture at temperatures as high as 40° to 60°F. This fact is deduced by the high-end of the band, as referenced to the L to 0.5 YC portion, i.e., the lower part of the L to YC transition. Service experience has indicated such susceptibility to brittle fracture to temperatures as high as 60° to 80°F. For this reason, we draw a L to YC (hatched) line to indicate the estimated high end of the band for service castings of these steels. Most importantly, the L to YC analysis for these project steels are in exact agreement with service experience. The narrower aspects of the test steels band is simply a reflection of limited data. The 20°F lower limit of the "high end" is to be expected for the simple casting shape of the test samples, compared to production castings.

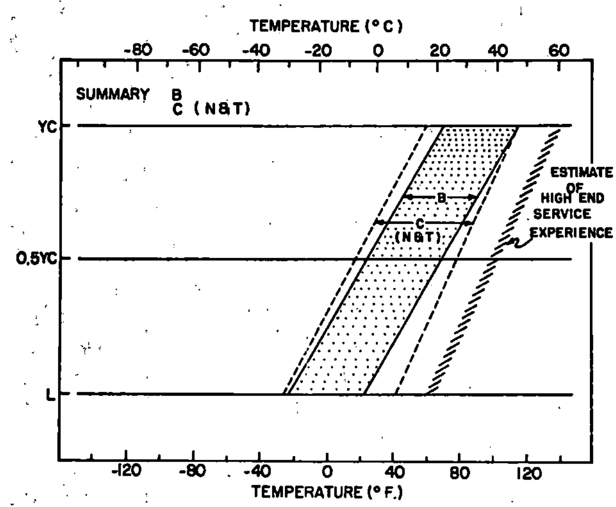


FIGURE 6. BANDS FOR BAND C (N&T) STEELS

Figure 7 presents the bands for the C (Q&T) and E steels. The E steel is represented by two bands because the EH and EL samples were cited as experimental. These two samples establish a very wide band. The remainder of the steels fall in a narrower band, as noted.

An important feature of the bands is the two grades of Q&T steels overlap considerably, indicating equal potential for developing good (low temperature) fracture properties.

Figure 8 presents the statistical bands for all steels. The bold arrow designates the gap between the Q&T steels and the B, combined with C (N&T) band for the pearlitic steels. For the purpose of statistical analysis, the three Q&T steels noted in the figure as exceptions, are not considered. These appear to be outside of the statistical expectancy grouping. However, their specific properties will be considered in the section on metallurgical variables.

TABLE 1.

COMPOSITION AND PROPERTIES OF AAR CAST STEELS
EVALUATED IN CWRU STUDY

GRADE B (NORMALIZED) STEEL

	Chemical Composition Element (weight percent)						Mechanical Properties					
							Yield Strength		L		YC	
SOURCE	C	Mn	Ni	Cr	Mo	V	Ksi	MPA	F	C	F	C
A	.30	.75	.08	.14	.03	.01	48	331	5	-15	70	21
B	.27	.76	.06	.09	.03		46	317	10	-12	80	27
C	.27	.72	.10	.12	.08	.01	46	317	5	-15	75	24
D	.21	.66	.15	.18	.09		42	290	-25	-32	70	21
AT	.29	.71	.20	.28	.03	.04	50	345	20	-7	115	46
E	X	X	X	X	X	X	48	331	15	-9	80	27

GRADE C (NORMALIZED & TEMPERED)

	Chemical Composition Element (weight percent)						Mechanical Properties									
							Tempering Temperature	(Time)		Yield Strength		L		YC	
SOURCE	C	Mn	Ni	Cr	Mo	V	F	(Hrs.)	C	Ksi	Mpa	F	C	F	C
A	.28	1.62	.10	.10	.03	-	1020	(2)	549	66	455	20	-7	105	41
C	.29	1.46	.10	.11	.13	-	1220	(4)	660	61	421	40	4	115	46
D	.29	1.30	.26	.22	.22	-	1150	(3)	621	65	448	-30	-34	60	16
E	.29	1.34	.09	.12	.02	-	900	(2)	482	62	427	10	-12	95	35
F	.27	1.04	.72	.52	.17	-	1200	(2.5)	649	64	442	0	-18	95	35

GRADE C (QUENCH & TEMPERED) STEEL

	Chemical Composition Element (weight percent)						Mechanical Properties								
							Tempering Temperature	(Time)		Yield Strength		L		YC	
SOURCE	C	Mn	Ni	Cr	Mo	V	F	(Hrs.)	C	Ksi	MPa	F	C	F	C
B	.27	.76	.06	.09	.03	-	1020	(3)	549	66	647	0	-18	50	+10
A(C)	.33	1.65	.08	.10	.04	-	1225	(2)	663	82	804	-95	-71	-20	-29
E	.24	1.24	.10	.11	.03	-	1100	(2)	593	81	794	-35	-37	45	7
C	.29	1.46	.10	.11	.13	.07	1300	(4.5)	704	66	647	-95	-71	-40	-40
A(Q)	.28	1.37	.08	.34	.17	-	1300	(2)	704	83	814	-95	-71	-35	-37
D	.29	1.30	.26	.22	.22	-	1200	(3)	649	86	843	-95	-71	-10	-23
F	.22	1.28	.90	.51	.28	-	1250	(4.5)	677	71	696	-110	-79	-35	-37

GRADE E (QUENCH & TEMPERED) STEEL

	Chemical Composition Element (weight percent)						Mechanical Properties								
							Tempering Temperature	(Time)		Yield Strength		L		YC	
SOURCE	C	Mn	Ni	Cr	Mo	V	F	(Hrs.)	C	Ksi	MPA	F	C	F	C
A	.28	1.38	.06	.28	.17	-	1040	(2)	616	127	1245	-70		15	
C	.32	1.47	.12	.16	.24	.02	1100	(4.5)	593	122	1196	-75		-5	
D	.25	1.25	.19	.30	.27	-	1200	(3)	649	105	1029	-55		-10	
E	.29	1.50	.12	.45	.23	-	1100	(2)	593	128	1225	-95		-35	
EL	.22	1.22	.16	.40	.20	-	1100	(2)	593	111	1089	-5		-50	
EH	.30	1.46	.19	.52	.30	-	1200	(2)	649	115	1128	-135		-90	
F	.22	1.28	.90	.51	.28	-	1025	(3.5)	550	115	1128	-95		-40	

X Data Not Available

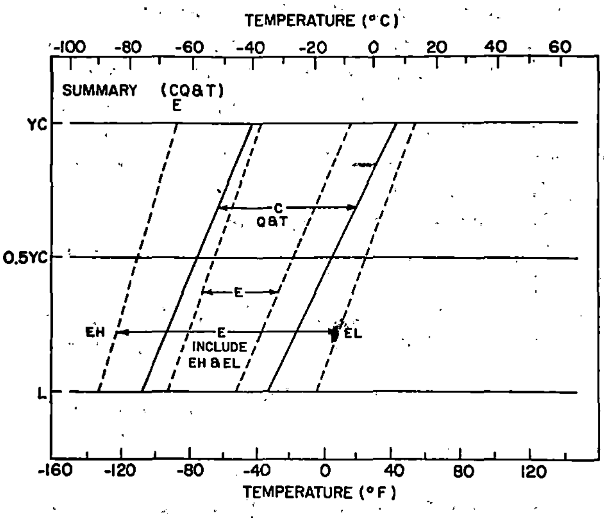


FIGURE 7. BANDS FOR C (Q&T) AND E STEELS

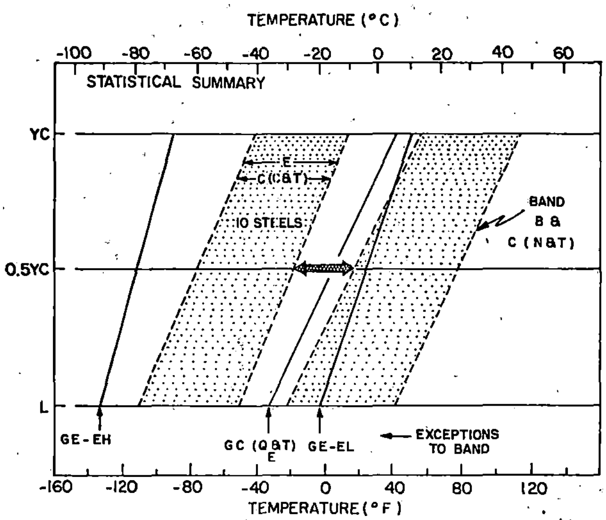


FIGURE 8. COMPARISON OF BANDS FOR Q&T AND FOR NORMALIZED STEEL

Figure 9 presents statistical expectancy bands based on adjusted figure 8 bands. The high end of the B and C (N&T) steels band is placed at the approximate location of the dashed line of figure 6. Figure 9 represents the best estimates which can be made presently for the bands.

The high ends of these bands are used to designate a tentative specification limit for the steels (TSL). The TSL line location for the Q&T steels may appear somewhat optimistic because it is at the high end of the statistical band. However, the following metallurgical analyses suggest it can be met.

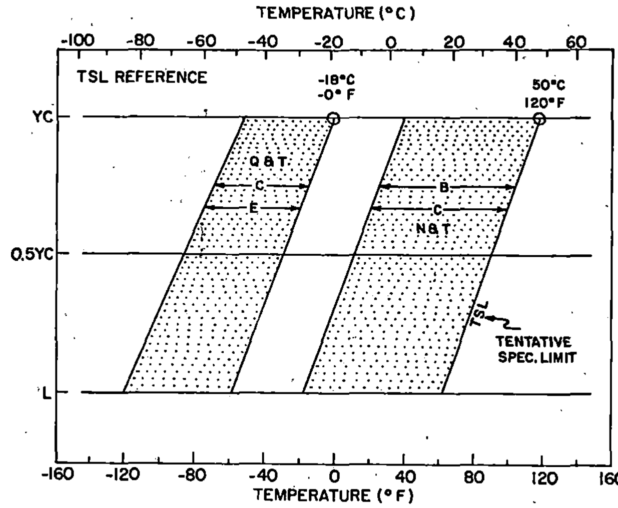


FIGURE 9. LOCATION OF TENTATIVE SPECIFICATION LIMIT (TSL) LINE FOR Q&T AND FOR NORMALIZED STEELS

If the TSL line is used for specification purposes, then YC properties no higher than 0°F could be guaranteed statistically for the Q&T steels. That is, both C (Q&T) and E steels (bainite-martensite steels).

Similarly, YC properties no higher than 120° could be guaranteed for the N&T steels (pearlitic steels).

The guarantee (YC = 0°F) serves a very useful purpose for the Q&T steels in assuring fracture prevention to service temperatures as low as 30° to 0°F. This is the high end of the band for the 0.5 YC to YC temperature range.

The same guarantee (YC = 120°F) serves only a limited purpose for the N&T (pearlitic) steels. This is to eliminate brittle metals that are abnormal to the population of production material.

There is no way the pearlitic steels can be improved to the extent of being equal to fracture properties of the Q&T (bainite-martensite) steels. The gap between the two could be closed, i.e., the TSL can be lowered by approximately 30°F, with shift of the total pearlitic steel band to lower temperatures. This was shown possible for pearlitic rolled plate steels. However, such attainment requires exacting metallurgical factor adjustments, which would not be economical for the castings.

For these reasons the N&T (pearlitic) steels will not be discussed further. Emphasis will be placed on metallurgical factors analyses for the Q&T steels of bainite-martensite microstructure.

FRACTURE PROPERTIES RELATIONSHIPS TO METALLURGICAL FACTORS

The analyses were conducted in three steps:

- Step 1 — Steels from specific sources.
- Step 2 — All sources combined.
- Step 3 — Tempering curves and microstructural factors.

The data in the figures for specific sources do not provide sufficient information for generalizatons. Their primary purpose is to key chemical composition, tempering conditions and yield strength to the L-YC fracture data curves.

It is understood all producers have used fine grain (Aluminum etc.) deoxidation practices, and temperatures of 1650° to 1700°F for austenitization.

The factor for which there is inadequate information is that of quenching effectiveness. The importance of this factor is disclosed by microstructural examinations, or the presence or absence of pearlitic microstructures for specific samples of Q&T steels.

The combined analyses for all courses (prior to micro-structural considerations) are presented in figures 10 and 11.

C (Q&T) Steels

Figure 10 illustrates the aggregrate of L-YC curves below the TSL line have one major distinguishing feature. These steels are high-tempered (1200°F or higher). A great variation of alloy content and yield strength is included in the aggregrate curves.

With good hardenability for the section sizes of interest, there should be no problem in achieving excellent fracture properties consistently in practical castings for the C (Q&T) steel, at strength levels of either 60 ksi (min.) or 70 to 75 ksi (min.).

It is noted that the only C (Q&T) project steels of poor fracture properties are:

* a low alloy composition resulting in a partially pearlitic microstructure;
* a Grade B steel composition.

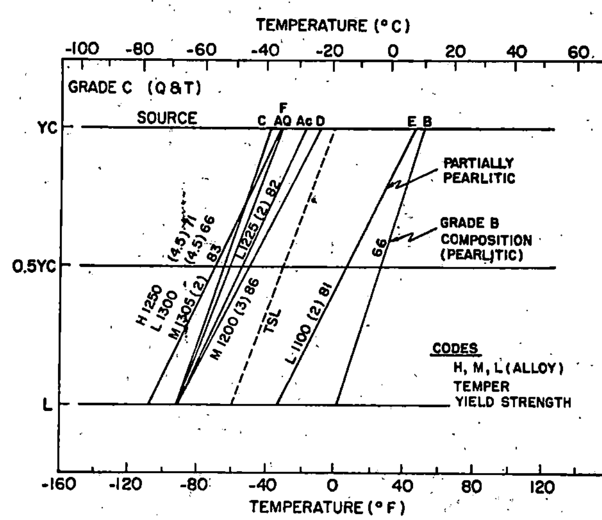


FIGURE 10. SUMMARY OF GRADE C (Q&T) STEELS, REFERENCED TO METALLURGICAL FACTORS

E Steels

Figure 11 illustrates the aggregrate of L to YC curves show a general tendency of the individual curves to shift from low to higher temperatures (with reference to the TSL line) with:

* decreased hardenability, combined with;
* decreased tempering temperature.

It is noted the only E steel, of poor fracture properties (significantly above the TSL line) features a partially pearlitic microstructure.

For both C (Q&T) and E steels, it is apparent the development of pearlitic microstructures results in a shift of the L - YC curves in the higher-temperature direction of the band for the normally pearlitic steels. The importance of ensuring fully bainitic-martensitic microstructures for the Q&T steels is evident from this data.

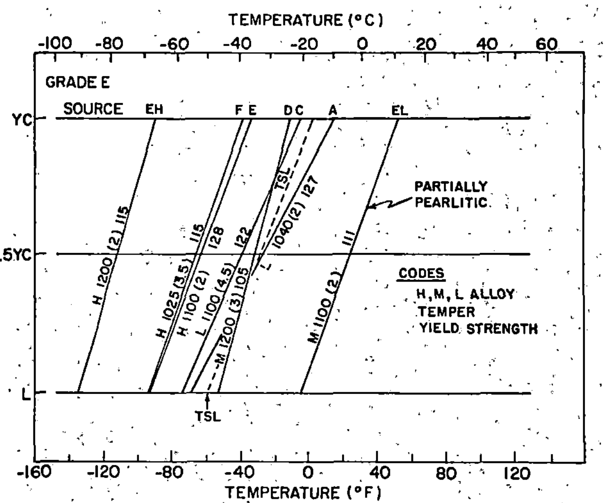


FIGURE 11. SUMMARY OF GRADE E STEELS, REFERENCED TO METALLURGICAL FACTORS

TEMPERING CURVES AND MICROSTRUCTURAL FACTORS

In order to understand the tempering response of the quenched steels a summary was made as shown in figure 12. The codes are as follows:

(A) = Source
(2) = 2 (hours tempered)
.22-H = Carbon and high alloy
.22-M, L, etc. = Carbon and medium or low alloy

It is apparent the tempering curve is steep in the 1175°F to 1225°F range.

It appears the E steels could be tempered in the range noted as "aim", possibly as high as 1200°F, in order to obtain best fracture properties. It is probable that carbon could be lowered to 0.25% (aim) for improved weldability. To do this may require a medium-high alloy content and probably vanadium additions.

Such combinations for optimizing fracture toughness and weldability should provide 110-120 ksi yield strength levels, on the average.

The C (Q&T) steel could be optimized in the same way. Since the primary alloy-level requirement derives from hardenability reasons, it appears the use of the same composition for the C (Q&T) alloy is expected to be in the indicated aim zone of figure 12.

Obtaining the same fracture properties for the two Q&T steels in production castings requires transformations to bainite-martensite. Pearlite must be avoided in the castings as well as in test coupons.

Variable casting properties may be expected if the alloy is too lean and split transformations develop.

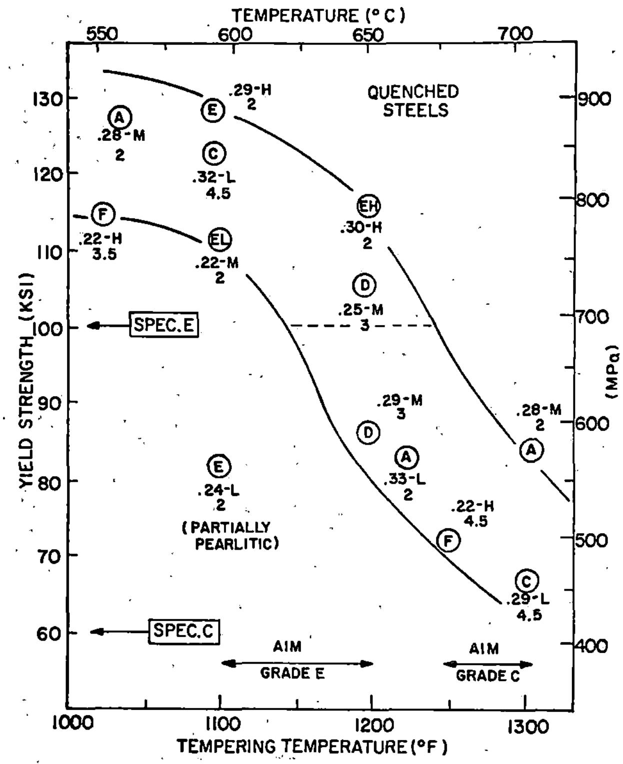


FIGURE 12. TEMPERING CURVE FOR Q&T STEELS

Microstructural Considerations

The microstructural data presented in the CWRU report are used as reference.

The high temperature location of the L-YC bands for the B and C (N&T) steels are in line with expectations for the pearlitic microstructures, noted by CWRU examination. No further discussion is necessary.

The C (Q&T) steel (E source) that features L - YC significantly above the TSL line (figure 10), is noted to have acicular (upper bainites) and/or fine pearlite. Following tempering, it is difficult to discriminate between these two types of microstructures. The relatively high L - YC indicates the structure is probably fine pearlite.

The other steel (B source) above the TSL line (figure 10) is actually a Grade B steel and is noted to result primarily in pearlite structures as expected.

All of the C (Q&T) steels which are below the TSL temperature line (figure 10) are cited to be of martensite-bainite transformation type. Steel is cited to possibly contain some fine pearlite or upper bainite. The good fracture properties of D indicate the presence of some acicular bainite and not pearlite.

The only E steel which definitely contains pearlite is the EL material. This is the only steel of this grade that lies above the TSL line (figure 11). The pearlite explains the poor fracture properties of this steel.

All other E (Q&T) steels appear to be of the appropriate martensite-bainite microstructure. This explains their position below the TSL line having good fracture properties.

It is obvious the development of pearlite as part of the quenched and tempered microstructures is highly detrimental to fracture properties. This is a well-established fact and the present data is in-line with this general information.

It is known that a relatively small percent of pearlite (say 10% - 15%) is highly detrimental in microstructures that otherwise are of martensite- bainite type. These mixed microstructures must be avoided if meeting of TSL fracture quality is to be expected for the Q&T steels.

The casting used for the samples is of simple geometry that provides for fast cooling rates in quenching. More complicated shapes may be expected to develop slower quench-cooling rates. Thus, mixed structures are likely for the low alloy compositions (low hardenability) and poor fracture properties may result. The answer is obviously to use sufficient alloy to ensure good quenching response, that is, to eliminate mixed microstructures in the production castings.

The experimental E grade steel EL requires additional discussion. The alloy content of this steel is compared to experimental steel EH:

	C	Mn	Ni	Cr	Mo
EL	0.22	1.22	.16	.40	.20
EH	0.30	1.46	.19	.52	.30

The differences in alloy do not explain adequately the presence of large amounts of pearlite in steel EL as compared to the largely martensitic structure of steel EH. Both alloys should be expected to avoid significant pearlite transformation, for a good quench of the simple plate sample.

If these alloy differences are sufficient to result in mixed structures then we are in real trouble with castings for all compositions. However, it is not reasonable that the alloy effect is this critical.

A more reasonable conclusion is steel EL did not receive an effective quench. The quench variable, if coupled with alloy compositions on the low side, may be expected to pose problems.

Normalizing of C (Q&T) Compositions

We now turn our attention to a C (N&T) of high alloy composition.

	C	Mn	Ni	Cr	Mo
F	0.27	1.04	.72	.52	.17

This steel falls in the center of the C (N&T) fracture properties band (figure 6). It is normal to the statistical population of this grade.

The CWRU microstructural examination cites 50% pearlite and 50% ferrite. This is also normal to the microstructure reported for the other C (N&T) steels. Most importantly, there are no mixed microstructures reported, i.e., martensites and bainites with the pearlite.

It is important to know the specific cooling conditions actually used for this steel in normalizing. If these are normal, as for the other C (N&T) steels, then we have to conclude that the low Mn (1.04% compared to the 1.30% - 1.65% range used for C (Q&T) steels) has a very potent effect on preventing mixed structures from developing in normalization for alloy steels.

At this point, it cannot be concluded that a C (Q&T) alloy composition can be normalized generally, with results fully equal to the fracture properties of C (N&T) steel compositions. Metallurgical logic is opposite to such a conclusion.

It is possible to do this by adjusting normalizing cooling rates with composition, or composition with usual cooling rates. No novel metallurgy is involved. However, the real issues of railroad reclaiming practices are not resolved by this single experiment.

The data for this steel cannot be resolved without specific information as to the normalization cooling rates.

METHOD OF DT SPECIFICATION

In developing a DT specification for fracture properties, a decision is first made as to the desired fracture mechanics criteria. In this case, it is desired to completely exclude brittle fracture for components having a history of such brittle fracture as couplers, for example.

From service experience, it is well-established that both rolled and cast steels are immune to brittle fracture (due to small crack-like defects and/or usual service stress levels) if the fracture properties are in the 0.5YC to YC temperature range. It is the below L, and to a lesser extent, the L to 0.5YC temperature range that provides for such fractures.

The service stress levels at corner or hole-geometry points of couplers may be assumed to approach yield stress levels as a limiting condition. The stress intensification at these points is conservatively estimated to be in the order of three to five times that of points of smooth geometry. In comparison, it should be noted that the best of pressure vessel designs feature 3x stress intensification at nozzles in comparison to hoop stresses.

The regions of stress intensification (geometry change points) of couplers are also the points of most probable location of casting or fatigue crack-like defects. These are the locations for which fracture initiation is most likely to develop.

Thus, the criteria of YC or "close to YC" levels of fracture properties are desirable for assurance of fracture prevention for couplers. This level of fracture properties must be assured statistically for the lowest temperatures of anticipated service.

The circled point noted as YC in figure 13 indicated for E steels:

* statistical YC-level fracture properties can be assured to 0°F;
* close to YC level fracture properties can be assured at 30° to 40°F.

If the subject point (150 ft-lb at 0°F) is used as the DT test specification value, a guarantee of close to YC-level fracture properties is provided to the lowest anticipated temperatures of service (-30° to -40°F).

Such a specification is conservative and should essentially guarantee elimination of fracture possibilities for couplers. In other words it should guarantee a fracture-safe coupler.

The minimum of 150 ft-lb at 0°F is not met (figure 13) by steels that are partially pearlitic, as indicated by the EL steel curve. It is expected all E steels that are quenched to fully bainite-martensite microstructures should meet this specification. This was explained in other sections of this report, and is the basis for the proposed DT specification value and temperature of testing.

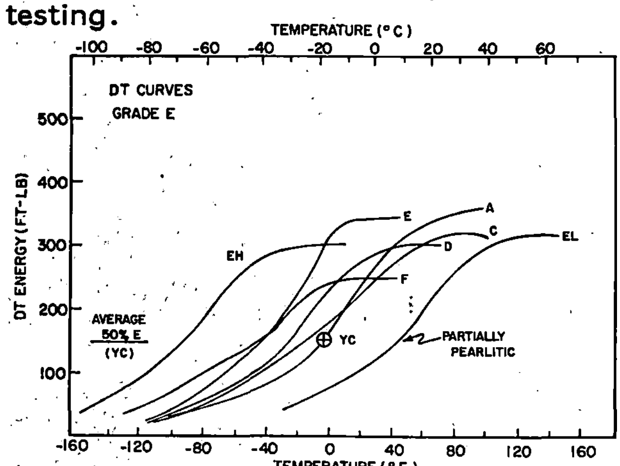


FIGURE 13. SUMMARY OF DT CURVES FOR GRADE E STEELS. THE CIRCLED POINT IS SUGGESTED AS A SPECIFICATION VALUE.

For the C (Q&T) steels (figure 14) the shelf levels of the DT curve are generally higher and YC-equivalent point is closer to 200 ft-lb.

However, the use of the same 150 ft-lb value as for the E steel is a reasonable approximation for close to YC-level fracture properties. For practical reasons of using the same specification value for both E and C (Q&T) steels, it is suggested the specification point shown in figure 14 be used. This is a minimum of 150 ft-lb at 0°F.

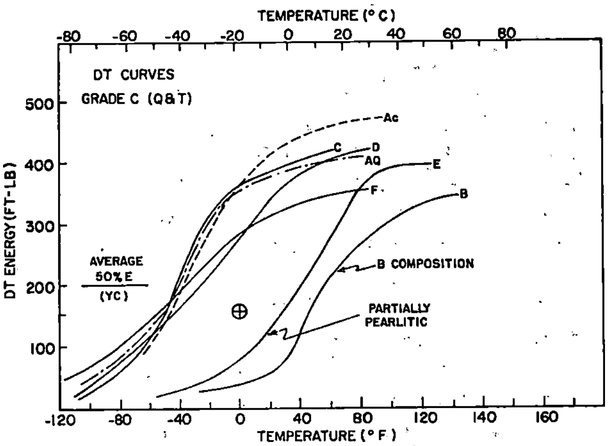


FIGURE 14. SUMMARY OF DT CURVES FOR GRADE C (Q&T) STEELS.
THE CIRCLED POINT IS SUGGESTED AS A SPECIFICATION VALUE.

This minimum value is not met by C (Q&T) steels of partially pearlitic composition, as indicated by the curves for steels E and B. It is met easily by the remainder of the steels which have bainite-martensite microstructures. The lower strength level of the C (Q&T) steels favors the development of good fracture properties at low temperatures.

For B and C (N&T) steels the YC-level point can be set only at 100° and 120°F, as indicated in figures 15 and 16. It is suggested that 120° should be used as the test temperature, for reasons that are evident from the service experience curve of figure 6. The only purpose of this DT specification is to eliminate unusually poor steels.

There is no way to improve B and C (N&T) pearlitic steels to provide fracture insensitivity to lowest service temperatures.

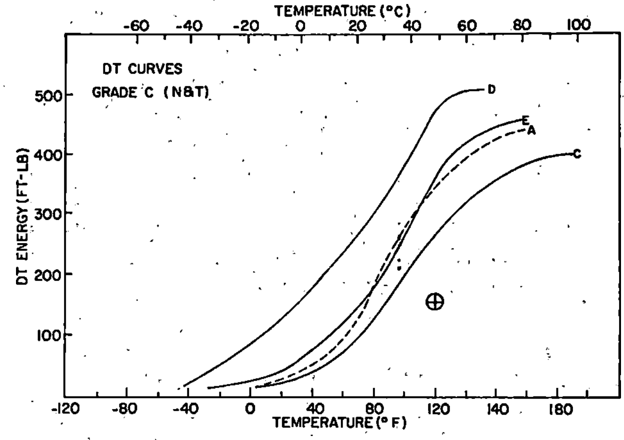


FIGURE 15. SUMMARY OF DT CURVES FOR GRADE C (N&T) STEELS.

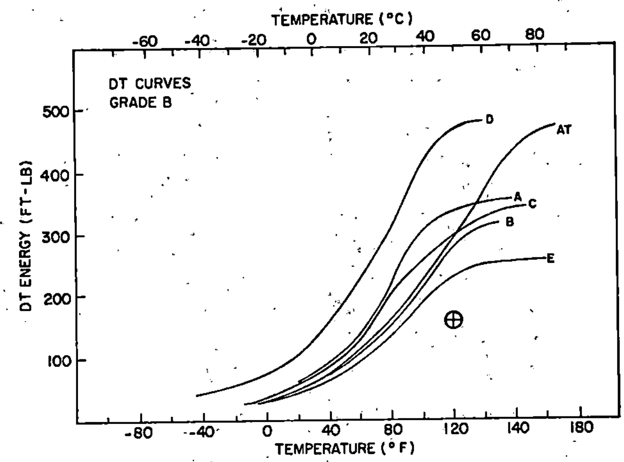


FIGURE 16. SUMMARY OF DT CURVES FOR GRADE B STEELS.

The described DT test procedures include:

* testing at only one temperature;
* meeting a minimum foot-pound value at the test temperatures;
* because of low scatter, two test specimens are sufficent.

The specified test-temperature properties can be verified as statistically attainable in production, by sampling (1) keel block from production heats, and (2) samples from castings. Both types of information are of interest.

SUMMARY

It is recommended that fracture properties be specified in terms of the DT test. Minimum fracture-energy values at 0°F are suggested for the C (Q&T) and E steels, and at 120°F for the B and C (N&T) steels.

The single most important practical question, is to establish if fracture-safe couplers can be guaranteed for railroad service temperatures. Everything else simply contributes to this objective, i.e., are matters of detail.

The most important conclusions are, that couplers can be produced economically which -

(1) completely eliminates fracture problems to lowest service temperatures - for the Q&T steels

(2) have high commercial-production reproducibility of fracture properties

(3) are specified in modern engineering terms for fracture properties

(4) have improved weldability for repairs, etc.

(5) will satisfy all concerned that casting properties can be reasonably guaranteed, as well as test-block properties.

REFERENCES

1. "Evaluation of the Toughness Properties of Various Cast Steels," by P. C. Sigala and J. F. Wallace, Case Western Reserve University, June, 1977.

2. "Fracture Properties of Tank Car Steels - Characterization and Analysis," RA-03-4-32 AAR R-192, August 20, 1975.

3. "Dynamic Tear Test of Metallic Materials," ASTM STANDARDS, Part 10, pp-826-833, 1975.

ON THE EFFECT OF TRACK GEOMETRY ON VEHICLE RESPONSE IN CURVE NEGOTIATION

BY

E.H. CHANG
AND
D.R. SUTLIFF

Spiral lengths and the amount of superelevation for simple and reverse curves have been a subject of concern to railroad engineers and maintenance personnel for many years. This concern is primarily motivated by the desire to improve track geometry, to decrease track maintenance, to improve train handling and, of utmost importance, to ensure the safe operation of the railroad. The previous study on "Minimum Tangent Length between Reverse Curves for Slow Speed Operation" indicates the usefulness of the Quasi-Static Lateral Train Stability (QLTS) model in studying track geometries for various types of railway vehicle operation. It is the intent of this paper to study the effects of spiral length and superelevation on vehicle response when they are introduced to simple curves and to reverse curves with minimum tangent length. The selection of critical vehicle consists are based on the population, weight and length of vehicles. The parameters which are analyzed include the lateral over vertical force ratio (L/V), the lateral coupler angle and the overturning speed of the vehicle. The analysis is performed for the empty consists because the L/V ratios are more critical for the light car than for the loaded car in curve negotiation. The methodology used in selecting "optimum" spiral length for a given consist and track configuration is presented. The "optimum" spiral lengths for a given amount of superelevation are determined and compared with those based on existing standards or formulas.

INTRODUCTION

In this study, no attempt has been made to cover all the possible car combinations, loading conditions, or track configurations. Since the L/V ratios for the light cars are more critical than the loaded cars in curve negotiation, the analysis would be solely based on the empty consists. Only those consists considered to be representative and critical are analyzed. Due to the fact that the weight and the height of the center-of-gravity of the loaded cars would be different from the light cars, the results and recommendations presented in this paper must be applied with care in practice for the partially or fully loaded consists.

CRITICAL CAR CONSISTS

The potential car candidates used in this study were selected from the AAR Standard Car Library (4). The car dimensions and properties are those stored in the data files of the car library. According to the car length, the potential candidates were grouped into four categories:

1. Long car (approximately 90')
2. Medium long car (approximately 70')
3. Medium short car (approximately 45')
4. Short car (under 40')

From these four categories a total of six cars were chosen based on their weights, height of the center-of-gravity, truck center distance, popula-

E. H. Chang is Research Engineer for the AAR. He received his B.S. in Civil Engineering from the Illinois Institute of Technology and his M.S. in Civil Engineering from Carnegie-Mellon University.

D. R. Sutliff is Director - Track/Train Dynamics Phase II, for the AAR. He holds a B.S. and M.S. in Civil Engineering from the University of Tennessee and a Ph.D in Engineering Mechanics from the same University.

tion of the car, and maximum attainable coupler angle. Two cars each were selected from the long and medium long categories (95' auto parts, 89' piggyback, 70' gondola, and 68' insulated box) and one each from the medium short and short groups (44' box and 32' gondola). Specifications of the six test cars are summarized in Table 1. It has to be emphasized that the heights of the center-of-gravity of the test cars are for empty cars only. When the cars are loaded, their c.g. heights would vary considerably. For example, the c.g. height of the 95' auto parts may be as high as 90" to 98" in loaded condition. The effective coupler length and the maximum attainable coupler angle can affect substantially the results and the analysis in this study. The detailed dimensions and properties of various types of coupler can be found in reference (5). It is worth mentioning that couplers of equal length may have different maximum attainable coupler angles. Their limits depend on the coupler yoke and striker used. It was assumed that the selected cars have met all design specifications recommended by the AAR (5).

(1) CAR SPECIFICATION NAME	(2) BOLSTER CENTER DISTANCE (INCHES)	(3) DISTANCE BETWEEN BOLSTER CENTER & COUPLER PIN (INCHES)	(4) CAR LENGTH OF COUPLER (INCHES)	(5) MAXIMUM ATTAINABLE COUPLER ANGLE (DEGREES)	(6) LIGHT WEIGHT (TON)	(7) OVERHANG (A/B) RATIO	(8) ESTIMATED EMPTY CAR C.G. HEIGHT
AUTO PARTS LA9 95'	792.0	114.0	60.0 (F) or (G)+	15°	65.0	0.776	70.0
PIGGYBACK/ CONTAINER LFC2 89'	780.0	101.0	43.0 (C) or (E)	13°	34.0	0.794	34.5
GONDOLA EG-6 70'	660.0	61.0	29.0 (D)	13°	30.0	0.884	50.0
INSULATED BOX LRB-6 68'	492.0	119.0	43.0 (C) or (E)	13°	43.0	0.674	50.0
BOX CAR XBM1 44'	348.0	57.0	33.0 (B)	8°	22.0	0.753	46.0
GONDOLA LG-3 32'	228.0	49.0	29.0 (A)	7°	23.0	0.699	40.0

+ Coupler Type	Effective Length (INCHES)
(A) B-E60B-HT, B-Y40, AAR, PL 532-B Striker	28.46
(B) E67AHT, B-Y41, AAR PL 542 Striker	33.28
(C) E68AHT series, 6, B-Y45, 13° Striker	43.00
(D) F70BHT, B-Y45, S16B Striker AAR PL-538	29.25
(E) F79BHT Series, 7, B-Y45, 13° Striker	43.00
(F) E69HTE Series, 4, B-Y45, 15° Striker	60.00
(G) F73HT Series, 5, B-Y45, 15° Striker	60.00

TABLE 1. CAR DATA USED IN THE QLTS STUDY

The "Minimum Tangent Length Study" (3) has shown that it is adequate to use a six-car consist to simulate the typical portion of the long train in operation. Any number of cars greater than six would produce negligible effects on the results as presented here. A seventh car is added to the consist as a "dummy" car in order to obtain complete response information about the sixth car. (Refer to model limitation (1)). Based on common train make-up practice, the six cars selected for investigation were coupled together to form five unique car consists. They are given as:

- 3 long cars (95' auto parts) - 3 long cars (89' piggyback)
- 3 long cars (95' auto parts) - 3 medium long cars (70' gondola)
- 3 long cars (95' auto parts) - 3 medium long car (68' insulated box)
- 3 long cars (95' auto parts) - 3 medium short car (44' box)
- 3 long cars (95' auto parts) - 3 short cars (32' gondola)

QUASI-STATIC LATERAL TRAIN STABILITY (QLTS) MODEL

The Quasi-Static Lateral Train Stability (QLTS) Model simulates a train on a prescribed track in a buff or draft mode. The detailed theoretical development of the model can be found in the report by Thomas and coworkers (1). A typical usage of the model is demonstrated in the user's manual (2). The input to the model includes: (a) car geometry, (b) car characteristics, (c) track geometry data, and (d) steady-state curving forces. The output from the model consists of: (a) derailment quotients for wheel climb and rail rollover, (b) lateral coupler angle and displacement, and (c) displacements and forces of vehicle components. The capabilities and limitations of the model are discussed in great detail in reference (1).

In calculating the lateral angle encountered by the coupler during curve negotiation, the QLTS model assumes the maximum coupler swing limits for all the couplers to be 20 degrees. In reality, the maximum attainable limits for the couplers in the consist differ from each other depending on the type of coupler used. In practice, force or moment redistributions take place once the maximum attainable limit has been reached, but the redistributions are not accounted for in the model. Consequently, due to this limitation, the calculated L/V ratios would not reflect the actual condition whenever the maximum attainable limit of the coupler has been exceeded.

In addition to the above output variables, the QLTS model is capable of calculating the 'critical speeds'. Based on the given operating conditions and track geometry, the 'critical speeds' (speeds at which the resultant force acts on points at distance of one-half gauge and one-sixth gauge) can be evaluated simply by varying the operating speed. This option can provide some insights to the stability of the consist and the tendency of a car to overturn. The critical speed based on the one-half gauge criteria can be physically interpreted as the overturning speed, beyond which the safety of the consist may be endangered. The critical speed based on the one-sixth gauge criteria can be viewed as the general practice for acceptable curve negotiation with an adequate safety factor against overturning. In the case of passenger system, it is one of the means in achieving good ride quality. It is worth mentioning that the critical speeds calculated in this study are for the light cars consists only. Different values of the critical speeds would be obtained in the case of the loaded consists.

DISCUSSION OF RESULTS OF THE QLTS STUDY

SIMPLE CURVES NEGOTIATION

Due to the topographical barriers or the functional requirements of the track, curves are usually introduced between portions of tangent track to by-pass obstacles, provide longer and easier gradients, and route the line through traffic centers. Normally, spirals are introduced at the approaches to the curves to provide gradual transition from tangent to curved track. This permits an increase in operating speed through the curve, improves ride comfort, and lessens in-train and track structure forces. Many standards and formulas for calculating the required spiral for a given track geometry exist. They are developed primarily based on the considerations of the lateral acceleration in the carbody, the car geometry and its rigidity, and the rate of the lateral accelerations (lateral jerk) in the carbody. However, little study has been made in comparing the relative merits of the various methods used in calculating the spiral length. The existing standards or formulas are, quite often, being modified to suit the intended purposes of each individual railroad. Consequently, the "optimum" spiral and superelevation remain to be evaluated. It is the intent of this section to investigate the effects of spiral length in curve negotiation and possibly to gain some insights on the "optimum" spiral and superelevation. The selection of the "optimum" spiral for a given track configuration is primarily based on the considerations of (a) L/V ratios, (b) lateral coupler angle, (c) the overturning speed, (d) the lateral acceleration in the carbody (reflected in acceptable operation speed), (e) overhang ratio (car geometry), and (f) the operation conditions (buff forces). In general, trade-off study has to be performed on the above mentioned parameters. The spiral length at which the most favorable condition is achieved for most (if not all) of the parameters would be considered as "optimum". Of course, in some cases, no "optimum" spiral may be expected, then the 'best' or the most appropriate spiral would be selected.

(A) Simple Curves with no Spiral

Comparisons between the results for the simple curves with no spiral indicate that if the consist is operated at 3" unbalance speed and at constant buff, the maximum L/V ratios generally decrease as the degree of curvature decreases. The largest reductions in the maximum L/V ratios are 0.14 and 0.16 for the wheel-climb and rail-rollover, respectively, as the curvature decreases from 16° to 6°. Figure 1 summarizes the variations in the maximum coupler angle with respect to the degree of curvature. In order to maintain the maximum coupler angles below the maximum attainable limits, the highest operational curvatures are 13°, 15.3°, 16°, 13.5° and 10.3° for the 95'-89', the 95'-70', the 95'-68', the 95'-44' and the 95'-32' consists respectively.

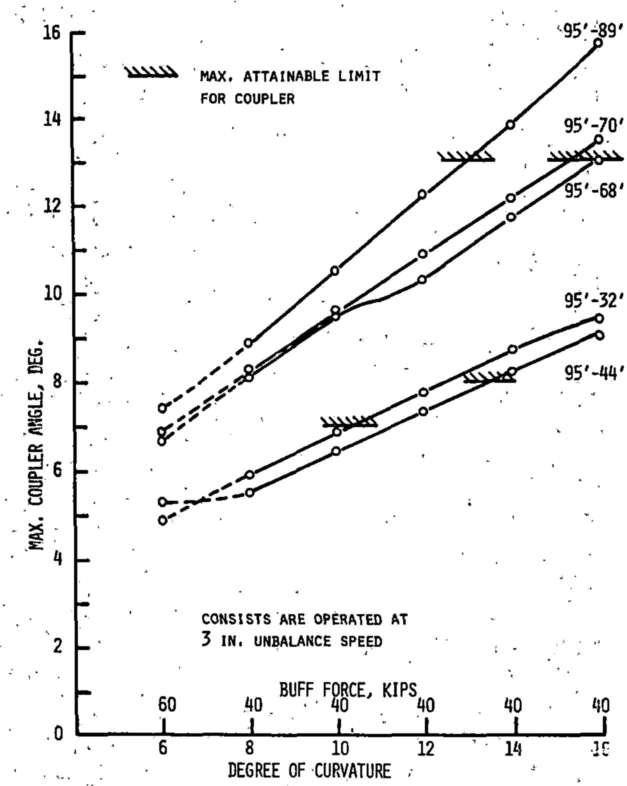


FIGURE 1. MAXIMUM COUPLER ANGLES FOR 3" UNBALANCE SPEED OPERATION AND DESIGNATED BUFF LEVEL VS. CURVATURE

The calculations of the 'critical speed' are based on the criteria that the resultant of forces (centrifugal, gravitational and coupler forces) acting on carbody passes through points of distances (a) one-half of gauge (G/2) and (b) one-sixth of gauge (G/6) away from either side of track center. It is obvious that the overturning speed is a function of the height of center of gravity of the vehicle, the weight of vehicle, the forces exerted on vehicle and the track geometry. Wellington (6) gives overturning speed as equal to $185.1/\sqrt{D}$ for a vehicle with center-of-gravity 6' (72 in.) above rail and running on a D degree of curvature. This may be representative for empty car of modern design. It is common practice with modern equipment to assume the center of gravity as 7' (84 in.) above the rails, leading to overturning speed of $170.0/\sqrt{D}$, a more conservative and safer value (6). For a car with 98" c.g. height (for example, loaded 100 ton hopper car), the overturning speed would be $x/\sqrt{D}$, where x is a constant. These expressions are only valid for a single vehicle running on curves and with center-of-gravity in the neighborhood of 6' to 7'. Since there is a considerable variation in the c.g. height of railway freight cars as well as the coupler forces resulting from operation conditions, the application of these formulas would be inappropriate for the train-consist in general. In this study, the effects of center-of-gravity and the coupler forces are considered in the overturning speed and acceptable operation (G/6 criteria) speed calculation. Six curvatures are analyzed. The selected buff levels are 60, 40, 40, 40, 40 and 40 kips corresponding to 6, 8, 10, 12, 14 and 16 degree of curvatures, respectively.

As can be seen from figure 2, at constant buff level, the critical speed decreases rather rapidly as the curvature increases. For the 95'-89' consist, the 95' car controls the overturning speed while the 89' car governs the acceptable operation speed. The overturning speed expressions (discussed previously) and the maximum allowable operating speeds suggested by the FRA are also illustrated for demonstration purposes. The plot of the acceptable operation speed suggests that the 95'-89' consist has to be operated at relatively low speed (below 10 mph) on 11° or higher curve in order to achieve the G/6 goal. It is interesting to note that the controlling overturning speed almost coincides with the speed given by the $170.0/\sqrt{D}$ expression. The overturning speed for the 89' car is substantially higher than those given by either one of the expressions, it is mainly due to the low center-of-gravity of the flat car. Since the buff level has been increased from 40 to 60 kips for the 6° curve, no correlation can be made with those obtained for other curvatures. However, the results are plotted for reference purposes. Similar studies have been performed for the other selected consists; however due to space limitations, the results are not presented.

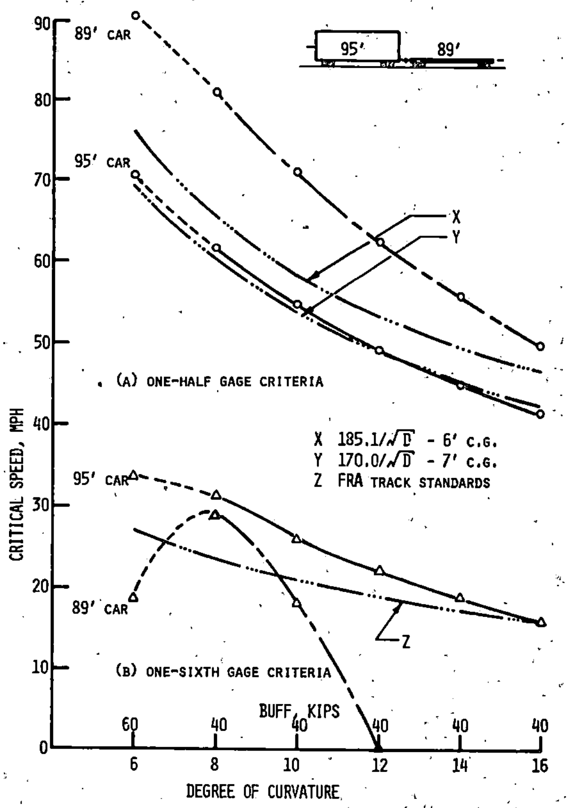


FIGURE 2. CRITICAL SPEED VS. CURVATURE

(B) Simple Curves with Spiral and no Superelevation

Figure 3 shows the plots of maximum L/V ratios and maximum coupler lateral angle versus spiral lengths for the 95'-89' consist negotiating at 14° simple curve with no superelevation at 17 mph and 40 kips buff. Two sets of simulation data have been presented. The solid line curves represent the critical state pertaining to the B cars (or a consist of similar cars) in the A-B consist and the dotted line curves indicate the critical value for the Car B at the interface of the A-B consist only. The former can be interpreted as the behavior of the portion of the typical consist of a long train or as the average behavior of unit train. The latter can be viewed as the behavior at the interface of cars located in the intermediate section of a typical consist. With no spiral, the maximum coupler lateral angle of the consist exceeds the maximum attainable limit. As the spiral increases, both the maximum L/V ratios and the maximum coupler angle decreases. At the point beyond which no further improvements in the maximum L/V ratios and the maximum coupler angle can be observed, the "optimum" spiral length based on the L/V ratios and coupler angling criteria would be achieved. The most appropriate spiral length for the 14 degree simple curve with no superelevation would be 125'. It is noticed that a minimum of 35' of spiral is required in order to keep the maximum coupler angle of the 89' car below ts maximum attainable limit in negotiating the 14° simple curve. It is interesting to note that if a spiral of length longer than the "optimum" value is introduced to the curve, minor adverse effects in maximum L/V ratios and maximum coupler angle would result. The observation may be due to the fact that no alignment control is provided for the consist in the simulation. Other considerations in selecting the "optimum" spiral are the overturning speed the comfortable speed, the overhang ratio and the operation conditions. They would be discussed in the latter sections.

The percent reduction in the maximum L/V ratios and the maximum coupler angle for the 14° simple curve with 125' of spiral (with reference to the no spiral case) is given in Table 2 column (2). The overall reduction ranges from 8.87% to 13.11% for wheel-climb, from 11.76% to 18.64% for the rail rollover, and from 8.93% to 36.86% for the coupler angle. Thus, the advantage of introducing a spiral into the curve is obvious.

The effects of spiral length on the overturning speed and the acceptable operation speed are demonstrated in figure 4 for the 95'-89' consist. In general, the critical speed increases with respect to spiral length. The 95' car controls the overturning speed and

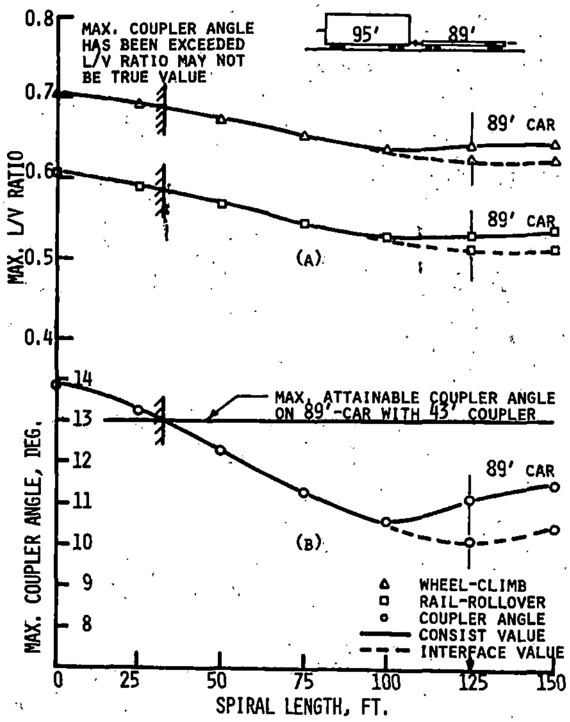


FIGURE 3. MAXIMUM L/V RATIO (A) FOR MAX. COUPLER LAT. ANGLE VS. SPIRAL LENGTH FOR 14 DEG. SIMPLE CURVE WITH NO SUPERELEVATION AT 17 MPH 40 KIPS BUFF

	(1)			(2)			(3)			(4)			(5)			(6)		
DEGREE OF CURVATURE (NO SUPERELEVATION)	16°			14°			12°			10°			8°			6°		
SPEED, MPH	16			17			19			21			23			27		
SPIRAL LENGTH, FT.	150			125			100			80			100			125		
VARIABLES / CAR CONSISTS	WC	RR	CA	WC	RR	CA	WC	RR	CA	WC	RR	CA	WC	RR	CA	WC	RR	CA
95' - 89'	8.04	9.18	17.83	9.22	12.84	20.71	9.19	12.06	23.55	7.14	9.91	20.15	10.00	29.15	22.90	15.15	19.38	28.65
+	18.27	13.22	34.52	11.15	15.35	24.37	9.19	12.06	23.55	4.77	9.04	21.81	11.31	29.64	33.14	16.67	21.19	18.65
95' - 78'	8.90	13.87	15.30	9.38	13.56	16.53	10.82	15.04	19.21	12.42	16.26	35.42	13.39	19.80	38.55	15.37	7.37	40.58
+	16.55	20.16	32.39	13.49	18.64	36.04	13.03	18.58	35.03	12.42	16.26	35.42	13.39	19.80	38.55	15.37	7.37	40.58
95' - 60'	10.16	12.08	13.08	8.87	11.74	15.38	8.86	9.18	19.23	7.35	9.47	20.74	12.30	17.98	37.65	16.00	23.23	43.28
+	12.76	16.73	31.88	8.87	11.76	24.57	8.86	9.18	21.63	7.35	9.47	23.12	12.30	17.98	37.65	16.00	23.23	45.97
95' - 44'	13.28	16.00	10.67	13.11	16.03	13.41	11.71	15.51	15.75	2.17	3.67	25.00	7.90	10.32	26.36	11.51	15.83	34.33
+	13.28	16.00	36.44	13.11	16.03	35.36	11.71	15.51	29.45	2.17	3.67	25.00	7.90	10.32	36.36	11.51	15.83	34.33
95' - 32'	10.17	11.67	4.44	10.49	14.09	8.93	10.29	13.79	8.03	1.16	1.41	9.82	1.37	2.13	14.50	12.16	16.45	18.56
+	10.17	11.67	22.34	10.49	14.09	24.40	10.29	13.79	19.69	1.16	1.41	14.37	1.37	2.13	16.98	12.16	16.45	18.56

NOTE : WC - WHEELCLIMB RR - RAIL ROLLOVER CA - COUPLER ANGLE
+ FOR INTERFACE OF THE CONSIST ONLY

TABLE 2. PERCENT REDUCTION IN MAXIMUM L/V RATIO AND MAXIMUM COUPLER ANGLE DUE TO THE INTRODUCTION OF SPIRAL INTO THE SIMPLE CURVE

the 89' car governs the limit for acceptable operation. For the 14° curve with 125' spiral length, the overturning speed is approximately at 46 mph and the acceptable operation speed is at 16.7 mph. It is interesting to note that the acceptable operation speed with a 125' spiral being introduced is approximately equivalent to the allowable speed with 3" unbalance superelevation. (i.e., the 3" unbalance speed is equal to the acceptable operational speed (G/6)).

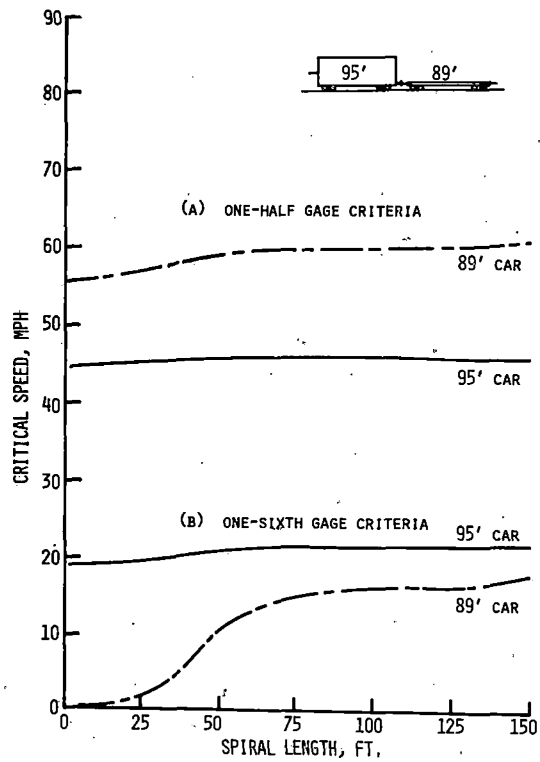


FIGURE 4. AVERAGE CRITICAL SPEEDS VS. SPIRAL LENGTH FOR 14 DEG. SIMPLE CURVE WITH NO SUPERELEVATION AT 40 KIPS BUFF.

The effects of buff force on the maximum L/V ratios and the maximum coupler angle for the 95'-89' consist is shown in figure 5 for the 14° curve with 125' spiral. The plot indicates the upper limit on buff level, which should not be exceeded during train operation, in order to keep the maximum L/V ratios and/or the maximum coupler angle under the desired limits.

The methodology for studying the "optimum" spiral for curves with spiral and no superelevation has been presented. Six different curvatures have been analyzed. They range from 6 degrees to 16 degrees. The "optimum" spiral length and the corresponding reduction in the maximum L/V ratios and the maximum coupler angle is summarized in Table 2.

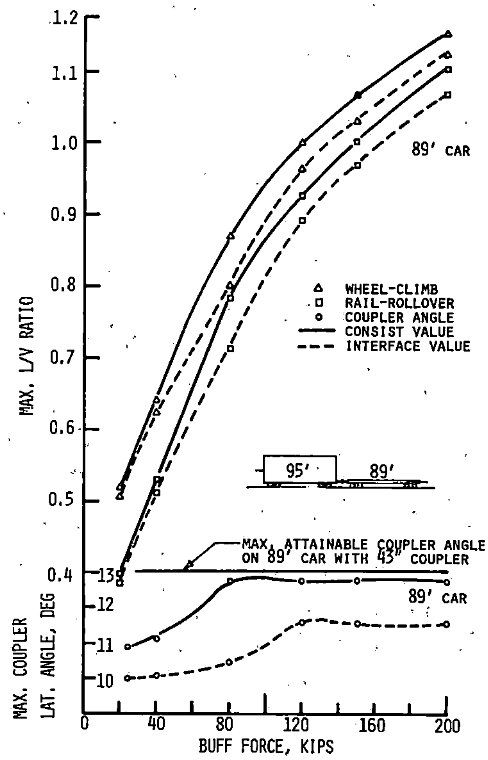


FIGURE 5. MAXIMUM L/V RATIO AND MAX. COUPLER LAT. ANGLE VS. BUFF FORCE FOR 14 DEG. SIMPLE CURVE WITH 125 FT. OF SPIRAL BUT NO SUPERELEVATION AT 17 MPH.

(C) Simple Curves with Spirals and Superelevation

The plots of maximum L/V ratios and maximum coupler lateral angle versus spiral lengths for the selected consists running on a 10° curve with 4" superelevation at 32 mph and 40 kips buff are shown in figure 6. Since the consists are operated at 3" unbalance speed on the curves with and without superelevation, it is not too surprising to find that the plots of maximum L/V ratios and maximum coupler angle for the curves with and without superelevation are identical. It has to be pointed out that the superelevated curves with zero spiral do not exist in reality.

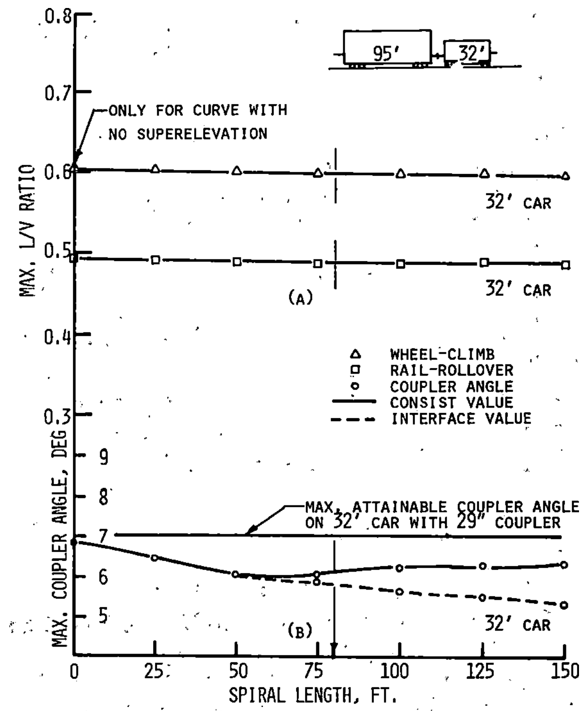


FIGURE 6. MAX. L/V RATIO (A) AND MAX. COUPLER LAT. ANGLE (B) VS. SPIRAL LENGTH FOR 10 DEG. SIMPLE CURVE WITH NO SUPERELEVATION AT 21 MPH AND 40 KIPS BUFF ALSO FOR 10 DEG. SIMPLE CURVE WITH 4" SUPERELEVATION AT 32 MPH AND 40 KIPS BUFF

By increasing the spiral length, the maximum L/V ratios for rail-rollover and wheel-climb, and the maximum lateral coupler angle, generally, decreases or remains constant. As observed, the maximum coupler angle tends to decrease more rapidly than the maximum L/V ratio as the spiral increases. At a relatively short spiral length (say 25'), the maximum L/V ratios for wheel-climb and rail-rollover are approximately 0.6 and 0.5 respectively. In no case has the maximum attainable limit for the coupler angle been exceeded.

As the spiral length continues to increase, the reduction in the maximum L/V ratios and the maximum coupler angle tend to diminish. The point, beyond which further increase in spiral would produce no effects or even adverse effects with respect to the maximum L/V ratios and the maximum coupler angle, would be the appropriate cut-off point of the spiral. In the case of 10° curve with 4" superelevation, the cut-off point for the spiral is 80'.

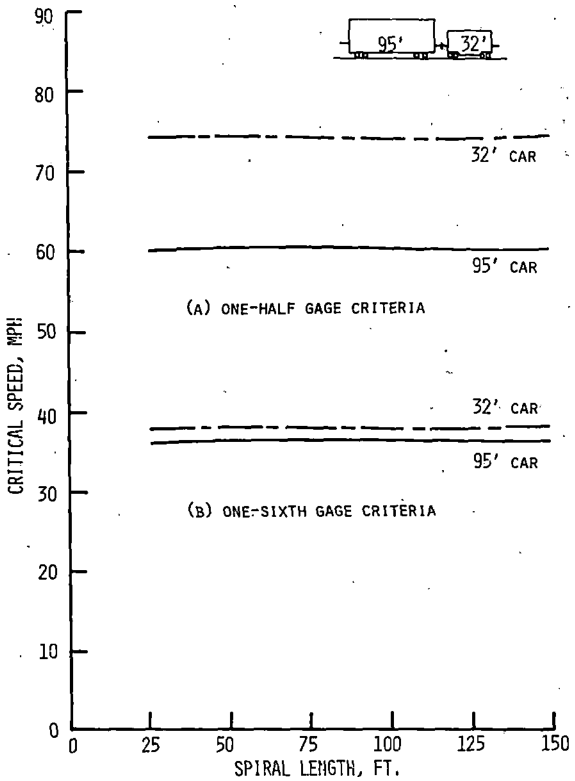


FIGURE 7. AVERAGE CRITICAL SPEEDS VS. SPIRAL LENGTH FOR 10 DEG. SIMPLE CURVE WITH 4" SUPERELEVATION AT 40 KIPS BUFF

The effects of spiral length on the critical speed of the selected consists are shown in figure 7. The results indicate that the acceptable operation speed and the overturning speed are insensitive to the spiral length. This is basically due to the fact that the curve section of test track is critical as far as the acceptable operation speed and overturning speed are concerned. For the consist studied, the 95' car governs the acceptable operation speed and the overturning speed. For the 10 degree superelevated curve equipped with 80' of spiral, the overturning speed is approximately at 60 mph. The acceptable operation speed is 37 mph for the 95'-32' consist. The results suggest that the speed limit would be 37 mph if it is desirable to keep the resulting force acting on carbody falling within the center third of the track gauge.

(D) <u>Effects of Superelevation on Spiral Length and Critical Speeds</u>

The 4° simple curve with superelevation and spiral was chosen, and the selected consists are operated at constant speed of 50 mph with 80 kips buff condition. The superelevation is varied from 4 in. to 6 in. The spiral length considered ranged from 100 ft. to 200 ft. The methodology presented previously is used to evaluate the spiral length needed. Then the rate of change of superelevation may be calculated.

		(1)	(2)	(3)
Curvature, Deg.		4°	4°	4°
Superelevation, in.		4	5	6
Speed, mph		50	50	50
Buff Force, Kips		80	80	80
Spiral, ft.		150	175	175
Runoff ft/in.		37.7	35	29.2
Rate of change of superelevation in/sec.		1.96	2.10	2.52
Consists 95'-89'	A	97.0	99.0	100.5
	B	61.0	63.2	66.5
95'-70'	A	97.0	98.0	100.0
	B	59.0	62.3	64.8
95'-68'	A	97.0	98.6	100.4
	B	61.0	63.5	66.0
95'-44'	A	95.5	98.0	99.5
	B	47.6	44.0	53.0
95'-32'	A	95.0	97.0	99.5
	B	40.7	44.4	50.0

NOTE: A - Overturning Speed (G/2 criteria)
B - Acceptable Operation Speed (G/6 criteria)

TABLE 3. EFFECTS OF SUPERELEVATION ON SPIRAL LENGTH AND CRITICAL SPEEDS

Comparisons between the results for 4", 5" and 6", superelevation indicate that the maximum L/V ratios, the maximum coupler angle and the critical speed behave in similar fashion. Table 3 summarized the results for the 4° curve with 4", 5" and 6", superelevation. The required spiral lengths are 150', 175' and 175' for the 4", 5", 6", superelevation respectively. The runoff decreases from 37.7 ft/in to 29.2 ft/in., as the superelevation increases from 4" to 6". Basically, this is due to fact that the unbalanced superelevation is decreased from 3" to 1". The critical speeds for both overturning and acceptable operation generally increase as the superelevation increased from 4" to 6". Finally, the comparison indicates that the maximum L/V ratios decrease slightly and the maximum coupler angle is maintained in same level as the superelevation is increased from 4" to 6".

(E) Comparison with Existing Standards

A literature survey indicates that three methods exist for determining the length of spiral for curve track. They are given as:

I. Based on superelevation only
$SL = 62\ E_a$ to $104\ E_a$

II. Based on unbalanced elevation and speed
$SL = 1.22\ E_u\ V$ to $1.63\ E_u V$

III. Based on speed and time
SL = 3/4" superelevation per sec to 1¼" superelevation per sec

where

SL = length of spiral
E_a = actual superelevation in inches
E_u = unbalanced superelevation in inches
V = speed in mph

The first and the second method correspond to the formulae suggested by the AREA. They are based on the considerations of the torsional rigidity and the lateral accelerations of the carbody.

In Table 4, the spiral lengths obtained from the QLTS study are compared with the values calculated from the existing formulae or standards. It can be observed that the spiral lengths from the QLTS study are slightly lower than the minimum spiral suggested by the AREA. Comparison with the FRA standards indicates that the spiral length for 10°, 8°, and 6° curves from the QLTS study is close to the values suggested by the FRA standards for the class 4 track. The spiral length for 4° and 2° curves falls between the values for the class 5 and the class 6 recommended by the FRA. The runoff based on the QLTS study is approximately 2¼" superelevation per second travel time.

Curvature (DEG)		10	8	6	4	2
Superelevation (IN)		4	5	6	4	6
Buff Force (KIPS)		40	40	60	80	80
Velocity (MPH)		32	38	46	50	80
QLTS STUDY -Spiral Length		80	100	125	150	275
AREA	Min $1.22\ E_u\ V$	117	139	168	183	293
	Max $1.63\ E_u\ V$	156	186	225	245	391
FRA	CLASS 3	62	78	93	62	93
	CLASS 4	83 +	103 +	124 +	83	124
	CLASS 5	124	155	186	124	186
	CLASS 6	248	310	372	248 +	372 +
Based on Superelevation only	$62\ E_a$	248	310	372	248	372
	$104\ E_a$	416	520	624	416	624
Based on runoff	3/4"/sec	250	372	540	391	939
	1"/sec	188	279	405	293	704
	1¼"/sec	150	223	324	235	563
	2"/sec	94	139	202	147 +	352
	2¼"/sec	75 +	111 +	162 +	117	282 +

TABLE 4. COMPARISON OF SPIRAL LENGTH WITH EXISTING STANDARDS

REVERSE CURVES NEGOTIATION

Reverse curves without superelevation and without spirals are found quite frequently in yards where a high degree of curve was used to maximize car capacity of each track, to permit access to existing facilities, or in rearranging trackage on existing right-of-way to effect improved operations. A number of yards require the movement of long trains into and out of yards through a series of cross-overs where little or no tangent exists between curves.

The previous study (3) on track geometry is devoted to the determination of the minimum length of tangent necessary between reverse curves (with no superelevation and no spirals) to assure a safe operation of all car consists. The first part of the discussion will demonstrate the methodology used to evaluate the minimum tangent between reverse curves. The effects of adding spirals to the reverse curves with minimum tangent length will be presented in the second part of discussion. Basically, it is the extension of the previous minimum tangent length study.

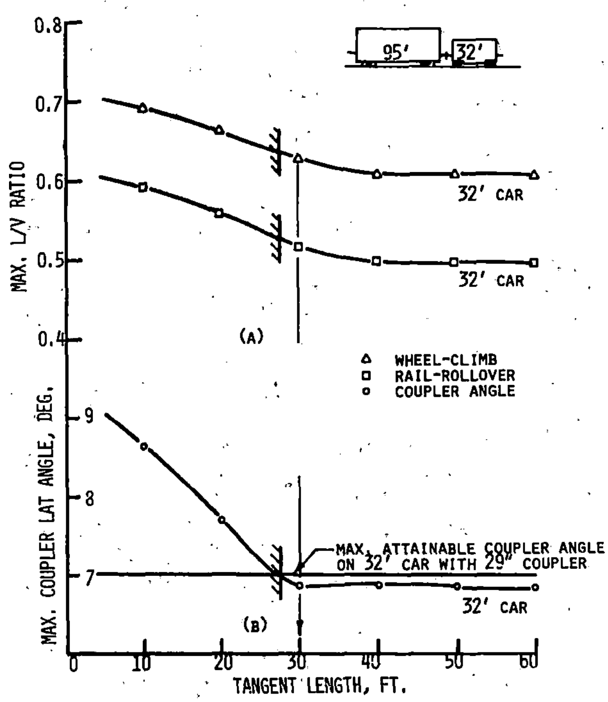


FIGURE 8. MAX L/V RATIO (A) AND MAX. COUPLER LAT. ANGLE (B) VS. TANGENT LENGTH FOR 10 DEG. REVERSE CURVES WITH NO SPIRAL AND NO SUPERELEVATION AT 21 MPH AND 40 KIPS BUFF

(A) Minimum Tangent Length (MTL) Study

The maximum L/V ratios and the maximum lateral coupler angle for the 95'-32' (long car-short car) consist for the 10 degree reverse curves with various tangent lengths are shown in figure 8. The consist is operated at the speed equivalent to 3" unbalance. The tangent length is increased from zero to fifty feet in 10' increments. The plot of maximum lateral coupler angle shows a considerable decrease from 8.6 degrees at 10' to 6.8 degrees at 30'. At tangent lengths greater than 30', no improvement is observed. The plots of the L/V ratios for rail-rollover and wheel-climb reveals a similar behavior. Since the maximum attainable coupler angle for the 32' gondola equipped with a 29" coupler shank is 7 degrees, a minimum tangent length of 30' would be necessary for 10 degree reverse curves.

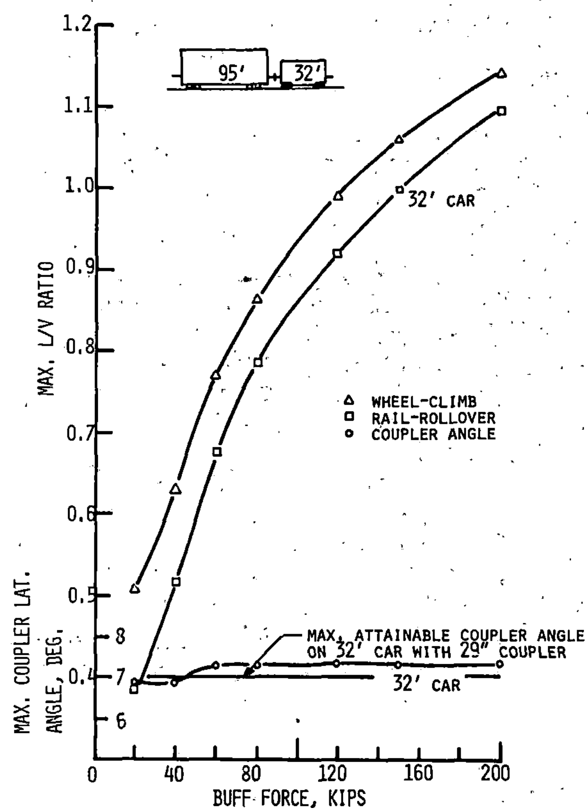


FIGURE 9. MAX. L/V RATIO AND MAX. COUPLER LAT. ANGLE VS. BUFF FORCE FOR 10 DEG. REVERSE CURVES WITH 30 FT. TANGENT BUT NO SPIRAL AND NO SUPERELEVATION AT 21 MPH.

The maximum L/V ratio for wheel-climb and rail-rollover are greatly influenced by the buff level at which the train is operated. For the 10 degree reverse curves, a buff level of 40 kips was used to evaluate the minimum tangent length. Once the minimum tangent length has been located at a selected buff level, other buff loads from 20 kips to 200 kips were investigated. In figure 9, the relationships between the L/V ratios and buff load are shown for the 10 degree reverse curves with 30' of included tangent section.

In figure 10, the plot of the critical speed for the 95'-32' consist is presented. It is interesting to note that the acceptable operation speed increases as the tangent length increases. At 30' of tangent, the acceptable operation speed corresponds approximately to 3" unbalance speed.

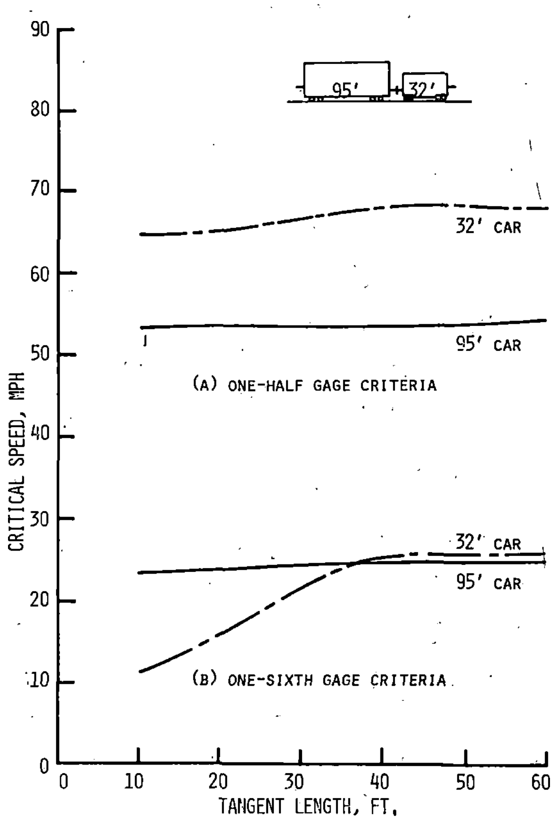


FIGURE 10. AVERAGE CRITICAL SPEEDS VS. TANGENT LENGTH FOR 10 DEG. REVERSE CURVES AT 40 KIPS BUFF

Based on the same methodology, the minimum tangent lengths for reverse curves (from 6° to 16°) are evaluated. The results are summarized in Table 5 (column 3). The results obtained from the previous AAR study and the Southern Railway study are also tabulated. Excellent agreement is found between the MTL study and the earlier studies.

Degree of Reverse Curves DEG.	A.A.R. Study	Southern Railway Study	A.A.R.-"MTL" Study Recommended for yard operation
Under 6	0'	0'	0'
6	0'	20'	0'
7	0'	30'	10'
8	10'	40'	20'
9	20'	60'	25'
10	30'	70'	30'
11	40'	70'	40'
12	50'	70'	50'
13	60'	70'	60'
14	70'	70'	See Note A
15	70	70'	See Note A
16	-	-	See Note A

Note A : Based on the "MTL" study and the car combinations used, reverse curves 14° and larger cannot be negotiated without exceeding the maximum coupler angle regardless of tangent length. If these curves must be used, spiral lengths should be provided between the tangent and the curve.

TABLE 5. MINIMUM TANGENT LENGTHS BETWEEN REVERSE CURVES

(B) Reverse Curves with Minimum Tangent Length and with Spirals

In the "MTL" study, it is found that reverse curves of 14° and larger cannot be negotiated (for the selected consists used in the study) without exceeding the maximum attainable coupler angle regardless of tangent length. The introduction of spiral length into the reverse curves of 14° or higher is recommended if such curves cannot be avoided. In this section, the effects of the spiral length on the vehicle response during the reverse curves negotiation are investigated.

Figure 11 shows the maximum L/V ratio and the maximum lateral coupler angle for the 95'-32' consist during the 14 degree reverse curves traversal. The consist is operated at the speed of 17 mph with 40 kips buff. The reverse curves include a 70' tangent section. Based on the decreasing pattern of the maximum L/V ratios and the maximum coupler angle, the minimum spiral length required is 90' to maintain the coupler angle within its maximum attainable limit. Since there is no significant improvement in the maximum coupler angle for the spiral of length beyond 125', it would be the appropriate cut-off point. Therefore, a spiral length of 125' is recommended for acceptable operation in 14 degree reverse curves with 70' tangent. The percent reduction in the maximum L/V ratios and the maximum coupler angle for the 14 degrees reverse curves with 70' tangent and 125' spiral is tabulated in Table 6 column (1). Figure 12 illustrates the relationships between the maximum L/V ratio, the maximum coupler angle and the overhang (A/B) ratio. The plot also demonstrates that, the overhang (A/B) ratio is one of the important parameters to be considered in classifying vehicle types as well as in designing the vehicle and its components. For example, from the design aspect, the maximum L/V ratios can be kept approximately at a desired level by varying the properties of vehicle and similarly, by proper choice of coupler type, the maximum coupler angle can be kept within its maximum attainable limit. Consequently, the corresponding selected spiral length for the specific reverse curves would be "optimum" as soon as the lowest level of the maximum L/V ratios and the largest "marginal" safety with respect to the maximum coupler angle are achieved.

The results of the study show that for reverse curves (with minimum tangent length and no spiral) smaller than 12 degrees, the maximum attainable coupler angle would not be exceeded. These findings are more conservative than those obtained from the MTL study. The discrepancy is due to the variations in the assumed c.g. heights for the selected cars in both studies. The introduction of the spiral into these reverse curves would only reduce the maximum coupler angle and/or the maximum L/V ratios. Figure 13 illustrates the decreasing pattern of the coupler angle for the 95'-32' consist for the 10 reverse curves. The reduction in maximum coupler angle tends to diminish as the spiral length increases.

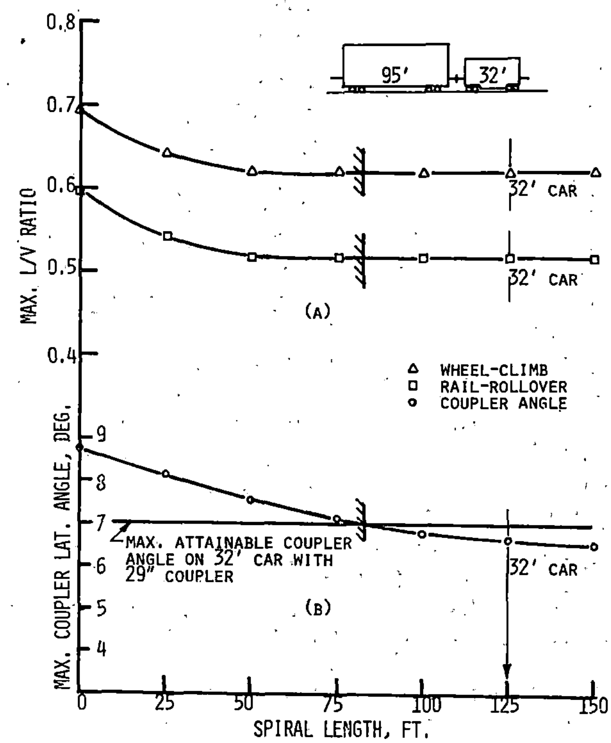


FIGURE 11. MAX. L/V RATIO (A) AND MAX. COUPLER LAT. ANGLE (B) VS. SPIRAL LENGTH FOR 14 DEG. REVERSE CURVES WITH 70 FT. TANGENT LENGTH AND NO SUPERELEVATION AT 17 MPH AND 40 KIPS BUFF

The recommended spiral length is 80' beyond which no substantial improvement is observed.

Table 6 summarizes the percent reduction in maximum L/V ratio and maximum coupler angle due to the introduction of the spiral into the reverse curves which range from 6 degrees to 14 degrees. In Table 7 the recommended spiral lengths (for acceptable operation) for the reverse curves with minimum connecting tangent length are presented.

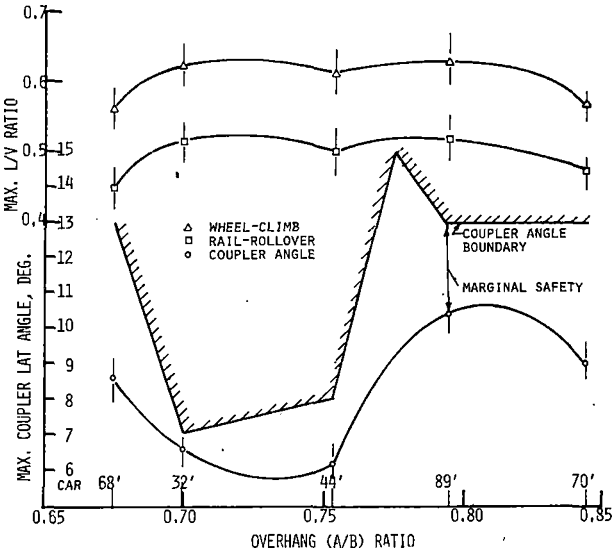


FIGURE 12. MAX. L/V RATIO AND MAX. COUPLER LAT. ANGLE VS. OVERHANG (A/B) RATIO FOR 14 DEG. REVERSE CURVES WITH 70 FT. OF TANGENT AND 125 FT. OF SPIRAL LENGTH AND NO SUPERELEVATION AT 17 MPH AND 40 KIPS BUFF

	(1)			(2)			(3)			(4)			(5)		
Reverse Curves	14°			12°			10°			8°			6°		
Tangent Length	70'			50'			30'			20'			0.0'		
Spiral Length	125'			100'			80'			70'			50'		
Variables / Car Consists	WC	RR	CA	WC	RR	CA	WC	RR	CA	WC	RR	CA	WC	RR	CA
95'-89'	11.1	14.2	26.6	9.2	12.3	25.3	10.1	13.6	29.6	10.9	14.2	26.7	6.6	9.9	24.5
95'-70'	15.8	20.0	26.0	14.7	18.6	21.9	13.3	17.4	24.8	12.2	16.5	38.9	6.6	7.8	28.6
95'-68'	10.0	12.6	29.4	10.5	12.0	24.1	6.7	9.5	25.5	5.4	6.7	38.9	4.9	6.6	24.7
95'-44'	12.9	16.7	28.7	12.2	15.5	31.3	11.4	14.4	32.8	12.5	16.3	28.8	15.2	19.7	39.2
95'-32'	11.0	14.2	24.1	10.7	13.8	19.5	4.0	5.2	14.7	5.7	9.8	17.3	12.3	15.7	23.4

Note: WC - Wheel-Climb
RR - Rail-Rollover
CA - Coupler Angle

TABLE 6. PERCENT REDUCTION IN MAXIMUM L/V RATIO AND MAXIMUM COUPLER ANGLE DUE TO THE INTRODUCTION OF SPIRAL INTO THE REVERSE CURVES

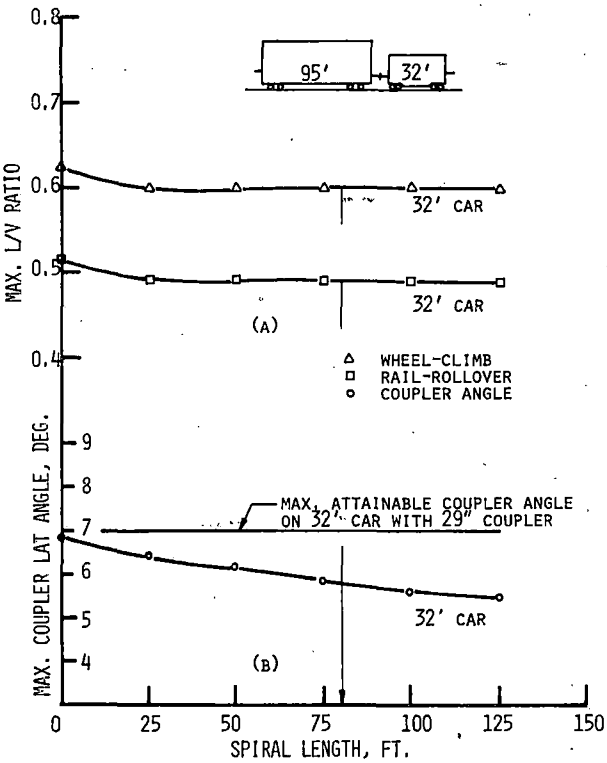


FIGURE 13. MAX. L/V RATIO (A) AND MAX. COUPLER LAT. ANGLE (B) VS. SPIRAL LENGTH FOR 10 DEG. REVERSE CURVES WITH 30 FT. TANGENT LENGTH AND NO SUPERELEVATION AT 21 MPH AND 40 KIPS BUFF

Degree of Reverse Curves	Recommended Tangent Length **	Recommended Spiral Length fc Acceptable Opera
Under 6°	0'	50'
6°	0'	50'
7°	10'	60'+
8°	20'	70'
9°	25'	75'+
10°	30'	80'
11°	40'	90'+
12°	50'	100'
13°	60'	115'+
14°	70'*	125'
15°	70'*	150'+
16°	70'*	175'+

* Tangent length not recommended without the introduction of the spiral into reverse curves (Ref.3)

** Based on the study of the Minimim Tangent Length between Reverse Curves (Ref.3)

\+ Estimated value

TABLE 7. SUMMARY OF THE STUDY OF REVERSE CURVES WITH MAXIMUM TANGENT LENGTH AND SPIRAL LENGTH

SUMMARY AND CONCLUSIONS

It has been shown that the QLTS model is an efficient tool in correlating the track geometry to the vehicle response. The response parameters considered include the maximum L/V ratios for wheel-climb and rail-rollover and the maximum coupler angle. The overturning speed and the acceptable operation speed are also studied. An indication has been found that the QLTS model can also be used as an effective tool in classifying and designing railway vehicles and their components as well as the track structure. In the following section, the conclusions reached from the analyses presented in previous sections are given.

SIMPLE CURVE NEGOTIATION

(A) Simple Curve with no Spiral

For the consists operating at 3" unbalance speed and at constant buff level, the maximum L/V ratios generally decrease as the degree of curvature decreases. The largest reductions in the maximum L/V ratios are 0.14 and 0.16 for the wheel-climb and rail-rollover, respectively, as the curvature decrease from 16° to 6°. Figure 1 summarizes the variations in the maximum coupler angle with respect to the degree of curvature. In order to maintain the maximum coupler angles below the maximum attainable limits, the highest operational curvatures are 13°, 15.3°, 16°, 13.5°, and 10.3° for the 95'-89', the 95'-70', the 95'-68', the 95'-44' and the 95'-32' consist, respectively. The overturning speed and the acceptable operation speed for the selected consists are tabulated in Table 8. In figure 14, the average critical speeds for simple curve entries (with no spiral and no superelevation) are presented.

(B) Simple Curves with Spiral and no Superelevation

The introduction of the spiral to the simple curve with no superelevation has great effects in reducing the maximum L/V ratios as well as the

BUFF FORCE (kips)		60	40	40	40	40	40
CONSIST	CURVATURE	6°	8°	10°	12°	14°	16°
95'- 89'	A	70.5	61.4	54.5	49.0	44.8	41.5
	B	18.6	28.7	18.0	-*-	-*-	-*-
95'- 70'	A	70.0	61.7	54.3	49.0	44.5	40.8
	B	8.7	26.2	13.3	2.3	-*-	-*-
95'- 68'	A	70.6	61.8	54.3	49.0	44.7	41.0
	B	28.0	29.8	23.2	19.2	15.0	9.4
95'- 44'	A	70.0	61.3	55.2	48.8	44.5	41.0
	B	29.0	30.0	24.1	12.5	10.0	7.4
95'- 32'	A	70.0	61.5	54.0	49.0	44.4	41.0
	B	5.2	30.1	25.2	21.3	18.3	2.0

A - Overturning Speed, mph
B - Acceptable Operation Speed, mph
-*- The operation of the consist on the designated curvature is not recommended without the introduction of the spiral

TABLE 8. SUMMARY OF OVERTURNING SPEED AND ACCEPTABLE OPERATION SPEED FOR SIMPLE CURVES WITH NO SUPERELEVATION

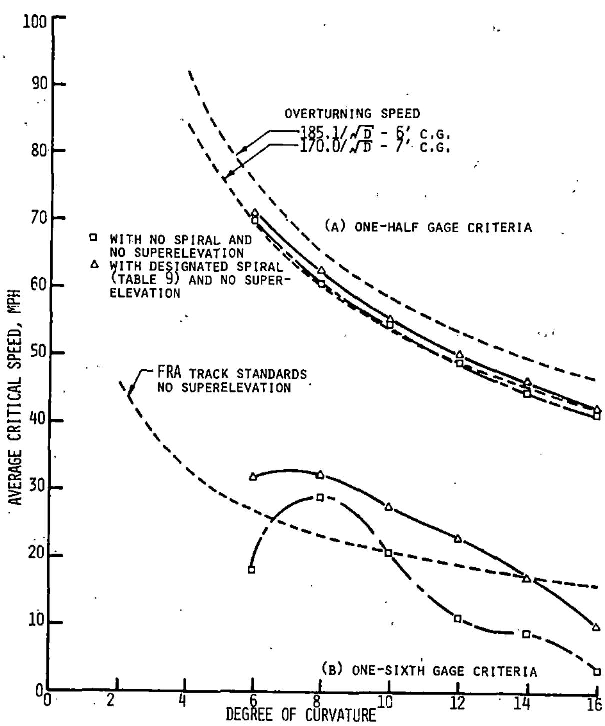


FIGURE 14. AVERAGE CRITICAL SPEEDS FOR THE NEGOTIATION OF CURVES WITH NO SPIRAL AND NO SUPERELEVATION, AND CURVES WITH SPIRAL AND NO SUPERELEVATION

maximum coupler angle. The critical speeds are increased due to proper choice of spiral. Based on the analysis performed, the 95'-32' consists is the most critical one with respect to the maximum coupler angle for all the simple curves with designated spiral lengths. The overturning speed and the acceptable operation speed analyses do provide a guideline on train operation at the given condition. Figure 14 summarizes the improvement on the average overturning and acceptable operation speed due to the introduction of the spiral. The maximum allowable operating speeds suggested by the FRA are also shown for comparison purposes. The overhang (A/B) ratio can be considered as one of the important parameters in classifying the railway vehicle as well as in designing the vehicle and its components. The recommended spiral lengths for simple curves are tabulated in Table 9 and the corresponding percent reduction in the maximum L/V ratio and the maximum coupler angle is given in Table 2.

Degree of Curvature	Speed mph	Recommended Spiral Length for Acceptable Operation, ft. *
Under 6		125
6	27	125
7		115+
8	23	100
9		90+
10	21	80
11		90+
12	19	100
13		115+
14	17	125
15		140+
16	16	150

+ Estimated Value

* Based on one-sixth gauge criteria

TABLE 9. SUMMARY OF THE STUDY OF SIMPLE CURVES WITH SPIRAL LENGTH

(C) Simple Curves with Spirals and Superelevation

By introducing the appropriate spiral to the simple curves with selected superelevation, the maximum L/V ratios as well as the maximum coupler angle have been reduced. The improvements tend to diminish as the curvature decreases. Based on the analysis performed, limited insights have been obtained with respect to the suitable runoff which is to be used for each particular superelevated curve. The overturning speed and the acceptable operation speed analyses do provide a guideline on train operation at the given condition. In figure 15, the average critical speeds for curve negotiation are given. The maximum allowable operating speeds as suggested by the FRA are also presented for comparison purposes. The overhang (A/B) ratio is one of the important parameters in classifying as well as in designing the railway vehicles and their components. The recommended spiral length for the simple curve with designated superelevation is given in Table 10.

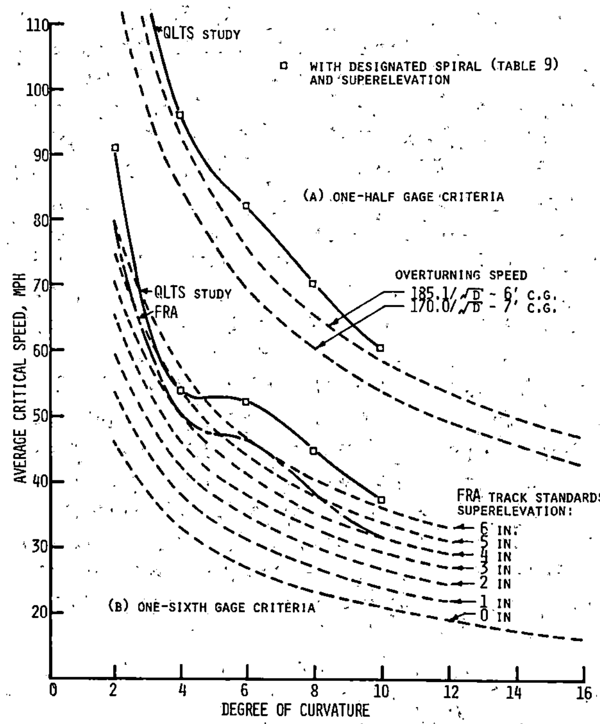


FIGURE 15. AVERAGE CRITICAL SPEEDS FOR THE NEGOTIATION OF CURVES WITH SPIRAL AND SUPERELEVATION

The suggested run-off for the spiral is also indicated. Finally, the overturning speed and the acceptable operation speed for the superelevated curves considered are summarized in Table 11.

CURVATURE DEG.	10°	8°	6°	4°	2°
SPEED, MPH	32	38	46	50	80
SUPERELEVATION INCHES	4	5	6	4	6
SPIRAL LENGTH FEET	80	100	125	150	275
RUNOFF FT/IN SUPER.	20.0	20.0	20.8	37.7	45.8
RATE OF CHANGE IN SUPERELEVATION IN/SEC.	2.35	2.79	3.24	1.96	2.56

TABLE 10. SUMMARY OF CURVE ENTRIES WITH SPIRAL AND SUPERELEVATION

Curvature, Deg.		10°	8°	6°	4°	2°
Spiral Length, Ft.		80	100	125	150	275
Superelevation, in		4	5	6	4	6
Speed, MPH		32	38	46	50	80
Consist \ Buff Force, Kips		40	40	60	80	80
95' - 89'	A	60.8	70.4	82.0	97.0	145.3
	B	38.0	45.5	54.0	61.0	97.5
95' - 70'	A	60.5	69.5	82.0	97.0	144.0
	B	36.0	44.7	52.6	59.0	95.0
95' - 68'	A	60.8	70.0	82.3	97.0	144.0
	B	37.5	45.0	53.4	61.0	97.0
95' - 44'	A	60.0	69.9	81.6	95.5	144.3
	B	36.6	44.8	50.5	47.6	68.0
95' - 32'	A	60.4	69.4	81.0	95.0	144.2
	B	37.0	42.8	39.1	40.7	96.6

NOTE:

A - Overturning Speed, mph
B - Acceptabel Operation Speed, mph

TABLE 11. SUMMARY OF OVERTURNING SPEED AND ACCEPTABLE OPERATION SPEED FOR SIMPLE CURVE WITH DESIGNATED SUPERELEVATION AND SPIRAL

REVERSE CURVES NEGOTIATION

(A) Minimum Tangent Length Study

The MTL study (1) indicated that there appears to be no need for tangent between curves of 6 degrees or less. For reverse curves over 6 degrees but under 14 degrees tangents should be provided between the curves at least as long as those indicated in Table 5. For reverse curves 14 degrees or larger, the maximum coupler angle is exceeded regardless of the length of tangent between curves. Unless spiral lengths are provided between the tangent and the curve, reverse curves of 14 degrees or larger should, therefore, be avoided.

(B) Reverse Curves with Minimum Tangent Length and with Spirals

The introduction of the spiral to the reverse curves with minimum tangent length definitely has made a great contribution to the reduction of the maximum L/V ratios and the maximum coupler angle. Based on the analysis performed, the 95'-32' consist is the most critical with respect to the maximum coupler angle for all the reverse curves with designated tangent lengths and spiral lengths. It is apparently due to the short coupler length equipped on the 32' gondola and its short truck center distances. The overhang (A/B) ratio is an important non-dimensional quantity in classifying the railway vehicle. The recommended spiral length for acceptable operation on reverse curves is summarized in Table 7, and the percent reduction in the maximum L/V ratio and the maximum coupler angle due to the introduction of the recommended spiral length can be found in Table 6.

ACKNOWLEDGEMENTS

The research reported herein was supported by the FRA under Contract No. DOT-FR-64228. The authors wish also to express their appreciation for the help given them by Mr. C. L. Gatton, Assistant Chief Engineer, M. of W., L&N Railroad Co.

REFERENCES

1) Thomas, L.R., MacMillian, R.D., Martin, G.C., "Quasi-Static Lateral Train Stability Model-Technical Documentation", Research Rpt R-209, Track-Train Dynamics.

2) Thomas, L.R., MacMillian, R.D.,

Martin, G.C., "Quasi-Static Lateral Train Stability Model-User's Manual", Research Rpt R-207, Track-Train Dynamics.

3) Gatton, C.L., "Minimum Tangent Length Between Reverse Curves for Slow Speed Operation", Track-Train Dynamics, Research Rpt R-228, AAR, Chicago, Ill.

4) "AAR Standard Car Library" AAR, Technical Center, Chicago, Ill., August 1976.

5) "Specifications for Design, Fabrication and Construction of Freight Cars", Association of American Railroads, Operations and Maintenance Department, Mechanical Division, September 1964.

6) Hay, W.W., Railroad Engineering-Volume One, John Wiley & Sons, Inc., New York, 1953.

FREIGHT CAR HUNTING MODELS AND THEIR VALIDATION

BY

Y. H. TSE

V. K. GARG

D. R. SUTLIFF

The operating speed of a freight car is often limited by 'truck hunting', a self-excited vibration phenomenon in which a truck oscillates violently in the lateral direction of travel. The paper describes two mathematical models developed at the Association of American Railroads under the Track Train Dynamics Program to investigate the hunting behavior of a freight car. One of the models is a linear model and is based on the eigenvalue-eigenvector solution of the equations of motion. The other is a non-linear model which provides a time domain solution for the system. The field test conducted at the Union Pacific Railroad to validate the hunting models is briefly discussed. Results of the models are compared with the field test data.

INTRODUCTION

One of the many problems faced by U.S. railroads today is freight car hunting. Hunting is a phenomenon which occurs predominatly on tangent track due to lateral dynamic instability of the freight car/truck system. The hunting motion is a major constraint to satisfactory operation of the freight car on tangent track.

Hunting, a coupled lateral and yaw motion of wheel-axle sets and/or side frames, is mainly due to a combined interaction of creep forces and wheel/rail contact geometry. At low operating speed with small lateral displacement, the wheel conicity provides guidance to the wheel-axle set within its clearances. But at high speeds, the conicity of the wheel may induce lateral dynamic instability. To provide self-centering action and for a satisfactory response of the freight car to track inputs, a large wheel conicity is often desirable. However, to achieve satisfactory dynamic stability on tangent track at high speed, conicity should be small. Thus, the conflicting requirements for guidance and stability pose a challenge to the freight car/truck designer.

Two mathematical models have been developed under the auspices of the Task III activities of the Track Train Dynamics Phase II Program to analyze the hunting behavior of freight cars. One is a linear model based on the eigenvalue-eigenvector formulation. The model assumes the freight car as a linear dynamic system and predicts the critical speed and critical frequency for the onset of truck hunting in a freight car/truck design. One of the uses of this model therefore, is as a design tool. By varying individual design parameter for the suspension system, it is possible to study the influence of each on the critical speed of hunting.

The other freight car hunting model is a nonlinear model and provides a time-domain solution. Under a specified set of initial conditions, time histories for all the degrees of freedom

Yan H. Tse is an Engineer Analyst for the AAR. He received both his B.S. and M.S. degrees in Mechanical Engineering from the Illinois Institute of Technology.

V. K. Garg is Manager of Dynamics Research for the AAR. He received his B.S. from Banaras University, India in 1960, his M.S. from the University of California in 1966 and his Ph.D from the Illinois Institute of Technology in 1973.

D. R. Sutliff is Director - Track/Train Dynamics Phase II, for the AAR. He holds a B.S. and M.S. in Civil Engineering from the University of Tennessee and a Ph.D in Engineering Mechanics from the same University.

are generated. The resulting forces and displacement are computed.

To validate above models, a comprehensive field test program was conducted on the Union Pacific Railroad. In this paper, these hunting models, along with field test results are described briefly. A comparison between the model results and the test data is also presented.

WHEEL-RAIL INTERACTION MODEL

Basically, hunting is induced by wheel/rail interaction. On a tangent track, wheel conicity offers guidance to the wheel-axle set for small displacements and low operating speeds. A slight lateral offset of the wheelset will induce a difference in the rolling radii between two wheels of an axle. The axle will try to steer itself back to its equilibrium position, and will make an angle with the rails. As the axle rolls forward, the wheelset lateral and yaw motion induce the hunting condition. As the wheelset moves laterally, the geometry changes and symmetrical equilibrium between two wheels of an axle is disturbed. A net lateral restoring force trying to push the wheelset back to its original position is formed. For new wheels with constant conicity, this force is proportional to the offset and the proportionality constant is often termed "lateral gravitational stiffness". Without damping, the wheelset will continue to oscillate with a lateral displacement amplitude equal to that of the initial offset. The creep forces between wheel and rail try to act to restore the wheelset to an equilibrium position. Creep force depends on the relative velocity between wheel and rail and the loading on the rail. The damping term given by the creep force is simplified by Reynolds [1] as:

$$\frac{\text{Speed } (-W \times \text{geometry} + S^2 \times \text{inertia})}{\text{Creep Coefficient}}$$

W, weight S, speed

When speed is small, the damping term is negative indicating real damping. Oscillations thus will be damped out. Above a certain speed, the second term becomes predominant and there is a negative damping. Any perturbation will induce the axle lateral and yaw displacement to build up until flange contacts the rail. At that instance, the truck oscillates with a large constant amplitude. In system control theory, this is referred to as limit cycle oscillation, meaning that a closed loop or path is formed on a state plane trajectory.

Thus, it is evident that a complete analysis of the lateral stability of a freight car should include a suitable model for the wheel-rail interaction. The linear and nonlinear hunting models have different approaches in modeling this interaction.

LINEAR MODEL

A linear wheel-rail interaction model consists of two parts:
(a) creep-force and (b) gravitational-force. In order to use the eigenvalue-eigenvector formulation, these forces should be linear. The rationale for developing these linear equations is not given here but can be found in reference [2].

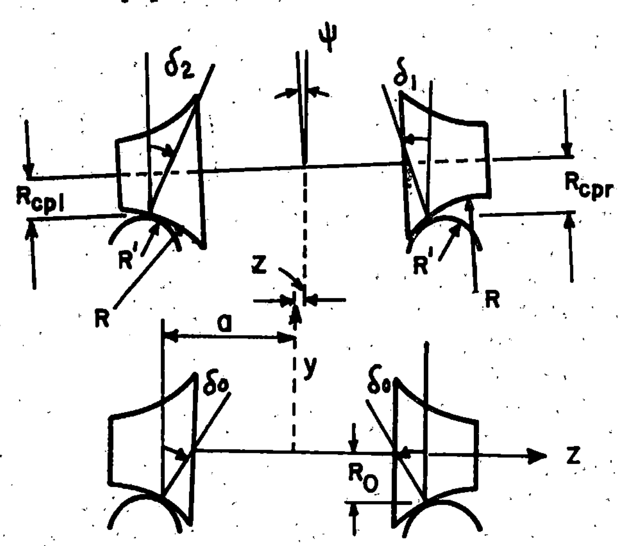


FIGURE 1. LINEAR WHEEL-RAIL INTERACTION MODEL

Considering a single wheelset, the net lateral creep force is given by (2)

$$F_{cz} = 2f_{22}\frac{(\dot{Z}-V\phi)}{V} + 2f_{23}\frac{\dot{\phi}}{V} - \frac{2f_{23}}{R_0}\frac{(\delta_1-\delta_2)}{2}$$

and the net creep moment is:

$$M_e = -2f_{23}\frac{(\dot{Z}-V\phi)}{V} + 2f_{33}\frac{\dot{\phi}}{V} - \frac{2f_{33}}{R_0}\frac{(\delta_1-\delta_2)}{2} + 2f_{11}\left(\frac{a^2\dot{\phi}}{V} + \frac{\lambda a Z}{R_0}\right) \qquad (3)$$

where

$$\frac{\delta_1-\delta_2}{2} = \frac{Z}{b}\left[\frac{b}{R-R'} + \frac{R\delta_0}{R-R'}\right]$$

and refer to figure 1 for other notations.

The lateral gravitational force is

$$F_G = \frac{W}{R-R'}\left[1+\frac{2R\alpha_0+2R_0\alpha_0}{a}\right]Z \qquad (4)$$

and the gravitational moment is

$$M_G = W\alpha_0 a\phi \qquad (5)$$

It may be noted that the independent variables are Z, $\dot{Z}$, ϕ, and $\dot{\phi}$ and all terms in the equations are linear.

NONLINEAR MODEL

As mentioned earlier, the linear hunting model assumes a constant creep coefficient. This implies the creep forces increase whenever the creepage is increased. In reality, the creep force cannot be higher than the pure sliding force. Thus above a certain creepage, the limiting coefficient of friction is the constant sliding coefficient. A literature survey reveals that there is a variety of opinions on the relationship between the equivalent coefficient of friction and the creepage.

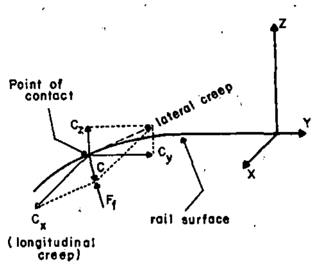


FIGURE 2. CREEP COMPONENTS

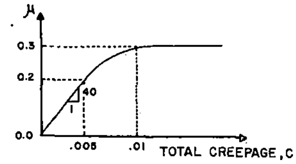


FIGURE 3. TYPICAL CREEPAGE-COEFFICIENT OF FRICTION RELATIONSHIP

Figure 3 shows a typical representation of this relationship for the nominal condition. Creepage is defined as the relative velocity between the wheel and rail divided by the forward velocity. C_x, C_y and C_z are the creep velocities in respective direction as shown in figure 2.

$$C_x = \dot{X} + (Z_{cp}-Z)\dot{\phi}\operatorname{Cos}\phi - (\dot{X}/R_0)R_{cp}\operatorname{Cos}\phi \qquad (6)$$

$$C_z = \dot{Z} - (Z_{cp}-Z)\dot{\phi}\operatorname{Sin}\phi + (\dot{X}/R_0)R_{cp}\operatorname{Sin}\phi$$

$$C_y = \delta C_z$$

Then the total creepage,

$$C = \sqrt{Cx^2+Cy^2+Cz^2}/\dot{X} \qquad (7)$$

From figure 3, the equivalent coefficient of friction can be obtained using C, equation (7). The creep force is then given as

$$F_{cf} = \mu F_N \qquad (8)$$

The three components of the creep force will be

$$F_{cfx} = F_{cf}\frac{C_x}{C\ \text{Total}} \qquad (9)$$

$$F_{cfy} = F_{cf}\frac{C_x}{C\ \text{Total}}$$

$$F_{cfz} = F_{cf}\frac{C_x}{C\ \text{Total}}$$

where C total $= \sqrt{C_x^2 + C_y^2 + C_z^2}$

Zcp, Rcp and δ in equations (6) and (7) depend upon the contact point between wheel and rail. In linear analysis, these are the linear functions of wheelset lateral displacement and wheel conicity. In the nonlinear model, a scheme has been developed to establish the contact point between wheel and rail for a given configuration. The scheme is initiated by moving a search point across the railhead. At each search point, the distance of the "closest approach" is defined here as the path the wheel and rail will move along if the point is the correct contact point. Among all "closest approaches" at these search points, the shortest distance is selected as the actual path of approach and the appropriate search point will be the actual contact point. Figure 4 shows the flow chart of this scheme. In the model, the wheel configuration is considered to be a combination of the wheel vertical displacement, lateral displacement, yaw angle and the roll angle. This scheme enables one to establish the contact geometrical characteristics of any combination of wheel and rail profiles. The scheme has been verified for a new wheel and new rail combination. Further development will extend it to an arbitrary combination.

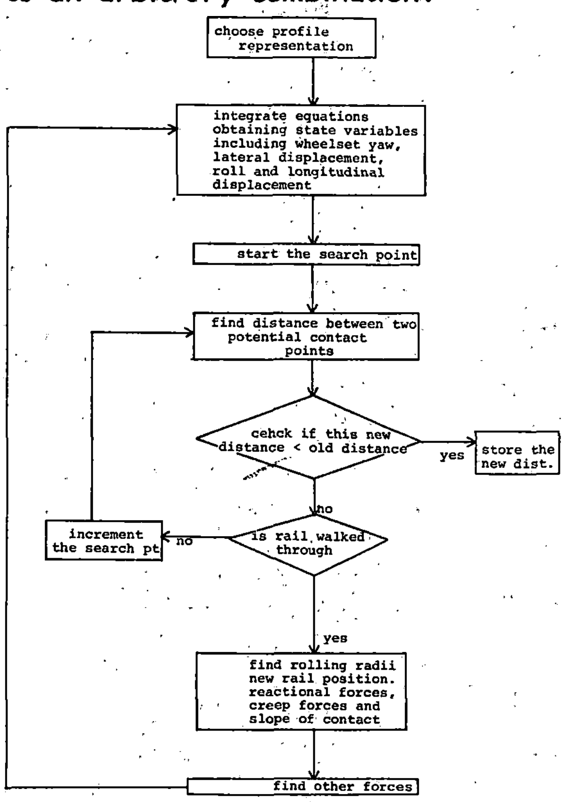


FIGURE 4. FLOWCHART OF WHEEL-RAIL INTERACTION MODEL

VEHICLE MODEL

By modeling, the complex freight car mechanical system is transformed into a network of rigid bodies connected with spring and damper elements. In the linear analysis, spring and damping forces are linear functions of displacement and velocity, respectively. In the nonlinear analysis, these functions may be bilinear, piecewise continuous or step.

LINEAR MODEL

The linear model for a four axle freight car system has twenty-five degrees of freedom. Each wheel-axle set is assigned two degrees of freedom corresponding to lateral and yaw motions. Each truck bolster has degrees of freedom in the lateral, yaw and roll directions. The truck side frames are allowed degrees of freedom in the lateral and longitudinal directions. The carbody is provided with three degrees of freedom corresponding to lateral, yaw and roll motion (figure 5).

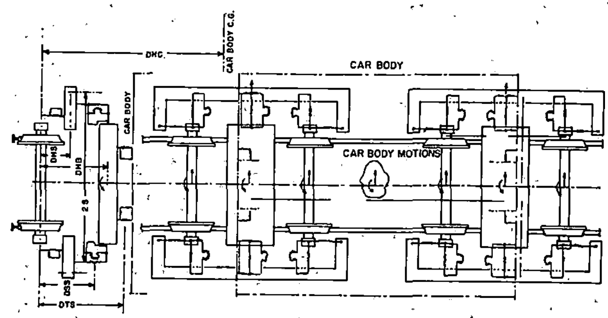


FIGURE 5. LINEAR TRUCK HUNTING MODEL

In the model, the axles are assumed to run freely under the journal bearing without bearing friction. Lateral clearances between wheel-axle sets and truck frames have been neglected. All displacements are assumed to be small. Nonlinearities arising from suspension stops, wheel flange contact, dry friction in suspension elements etc., have been neglected.

Initially, equations of motion for the system are developed to relate the motion of one component to another. These components are conceived as connected by spring-like stiffnesses and dashpot-like dampers. By using Newton's Law of Motion, $\Sigma F = M\ddot{q}$, 25 equations are developed.

NONLINEAR MODEL

Figure 6 shows one-half of the nonlinear car model. The other half is identical. Table 1 indicates the degrees of freedom for the half-car and full-car model. For the half-car model, one end of the carbody is assumed to be connected to a ground reference. The full car model has a torsional stiffness to simulate the carbody flexibility in the "twist" mode, which plays an important role for certain cars during hunting [3].

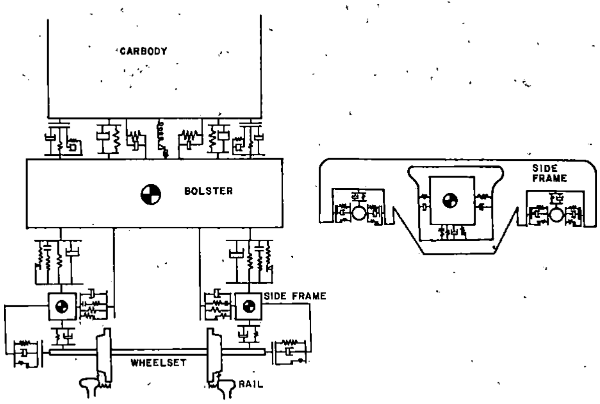


FIGURE 6. NONLINEAR FREIGHT CAR HUNTING MODEL

A detailed derivation of the equations of motion is not within the scope of this paper and thus is not included here. However, a general description of the interactive force formulation at the joint is briefly discussed. Figure 7 shows a general representation of the elements which simulate stiffness and damping at a joint. At some locations, these elements may not be all present. For example, in the vertical connection between side frame and axle, f and K_1 should be taken as zero.

K_o is a constant contract stiffness

$$F_{k_0} = K_0(x_1-x_2) \qquad (10)$$

c is a viscous damping

$$F_c = c(\dot{x}_1-\dot{x}_2) \qquad (11)$$

K_1 is a stiffness with a clearance

$$F_{k_1} = \begin{cases} 0 & (x_1-x_2)<\delta' \\ K_1(x_1-x_2-\delta') & (x_1-x_2)>\delta' \end{cases}$$

K_f and f correspond to the slip and stick coulomb friction

$$F_f = \begin{cases} K_f(x_1-x_2)^* & K_f(x_1-x_2)^*<f \\ f & K_f(x_1-x_2)^*>f \end{cases} \qquad (13)$$

Where $(x_1x_2)^*$ in equation (13) does not refer to the origin but to the last lock-up point. At low amplitude oscillations, the friction element may be locked up. K_f, which represents the components elasticity and is connected to f in series, allows the components to have relative displacements.

The total forces

$$= F_{k_0} + F_c + F_{k_1} + F_f \qquad (14)$$

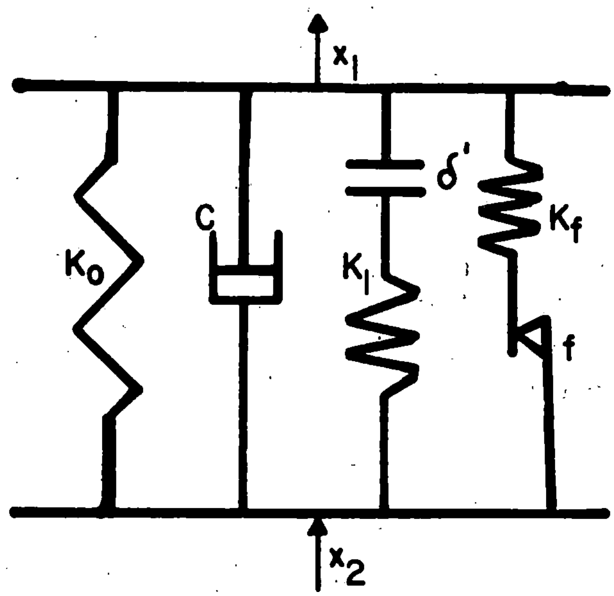


FIGURE 7. GENERAL REPRESENTATION OF JOINT ELEMENTS

A similar method is applied to find the total moments at the joint. This general model was suggested by Abbott, Morosow and MacPherson [4]. A description of the typical data or the joint parameters is included in [4].

METHOD OF ANALYSIS

Linear Model

The equations of motion for the linear model can be represented in matrix form as

$$[M]\ \{\ddot{q}\} + [C']\ \{\dot{q}\} + [K']\ \{q\} = [Q] \tag{15}$$

In the linear analysis, [Q] depends upon $\{\dot{q}\}$ $\{q\}$ and the speed of the vehicle and is related to Equations (2) to (5). Equation (15) can be reduced to the form:

$$[M]\ \{\ddot{q}\} + [C]\ \{\dot{q}\} + [K]\ \{q\} = [0] \tag{16}$$

Equation (16) represents a set of 25 homogeneous equations without any external forcing functions. Matrices [C] and [K] are no longer symmetric. The stability of the system (16) can be investigated by solving the eigenvalue problem.

$$\{p\} = [D]\ \{\dot{p}\} \tag{17}$$

where $\{p\} = \dfrac{\{\dot{q}\}}{\{q\}}$

$$[D] = \left[\begin{array}{c|c} [o] & [I] \\ \hline -[K]^{-1}[M] & -[K]^{-1}[C] \end{array}\right]$$

in which [o] and [I] denote the null and unit matrices, respectively. [D] is the dynamical matrix. Solution of the eigenvalue problem. Equation (17) results in 50 different eigenvalues and their associated eigenvectors. The complete solution of [17] would be:

$$\{q(t)\} = \sum_{k=1}^{50} g_k \{\phi_k\} e^{\lambda_k t} \tag{18}$$

where λ_k and $\{\phi_k\}$ are the kth eigenvalue and eigenvector and g_k are arbitrary constants.

The resulting eigenvalues will be complex conjugate and will be of the form

$$\lambda_k = \alpha_k + i\beta_k \tag{19}$$

As explained earlier, the mode of oscillation is stable when α_k is negative indicating an exponentially decaying oscillation. Zero implies pure oscillations and positive implies an unstable mode oscillating with an exponential build-up. β_k is the damped frequency of the oscillation associated with the kth mode.

Nonlinear Model

The formulation of equations of motion for this model is similar to the linear mode. However, in this case the terms in [C'] and [K'] in equation (16) are no longer constants. They are usually step or piecewise continuous functions of {q} and $\{\dot{q}\}$. Thus, the method used to solve equation (16) cannot be applied. A step-by-step integration scheme, based on the Runge-Kutta formulation has been used to solve the equations of motion.

FIELD TEST DESCRIPTION

The objective of this test program was to measure and record the dynamic response of an 80-Ton Open Hopper Car in a railroad environment so that the data could be used to validate various analytical models. Besides being used by the AAR to validate freight car hunting models, data are also used by Clemson/Arizona State University in their Freight Car Dynamics Research Program.

The test vehicle was a 80-Ton Hopper Car (LN 184701) equipped with 70-ton, A-3 Ride Control trucks with truck center distance of 33'10". Special equipment on the car for this test included:

1) Hydraulic truck forcers used to rotate the truck to a warp configuration while running below the critical speed. Upon a quick release, decaying oscillations could be recorded.

2) Air bag side bearings which allowed pneumatic adjustment of side bearing loads.

3) Truck frame stiffener attached to both ends of each side frame in order to increase the truck resistance to warping forces (use of this device constituted one of eight test configurations).

There were a total of 90 transducers. A data acquisition system (a mini-computer system to condition, collect, format and store the data on a nine-track, 1600 bpi tape) was installed on board at the AAR Research Car, AAR 100. Each channel was sampled at the rate of 100 samples per second. For details of the measurement methodology and data acquisition, refer to [5]. The test train consist is shown in figure 8.

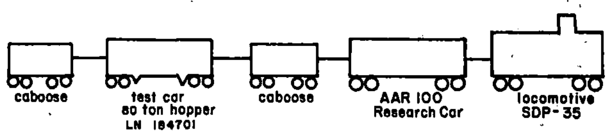


FIGURE 8. TEST CONSIST

The test configurations are listed in Table 2. The CN profiled wheels were used to simulate a worn wheel configuration. The CN profile, which resembles the nominal worn wheel profile, has been developed by the Canadian National Research Department. CN wheels were used because their profiles are better defined than the common worn wheels. The new wheel configuration used the 1/20 conicity AAR profile. The airbags, when side bearing load was specified as zero, were deflated and adjusted to nominal clearance specifications. In configuration 3 and 4, the air bag side bearings were inflated to give the specified pressure on the two side bearings. Nominal centerplate lubrication conditions was presented by a lubricated centerplate in accordance with Rule 47 of the Field Manual of AAR Interchange Rules. "Moly-Kote" lubricant was used. The torsional resistance of the lubricated condition is given in [6]. Field tests were conducted on Union Pacific track which conforms to FRA track class 4 or better. The test section is a continuous welded rail section suitable for speeds up to 79 mph.

RESULTS

The models have been used to simulate test results of configuration 6, empty car with the new wheels. Validation with other configurations are still in progress and will be reported subsequently at a later date. According to test procedure, the speed was first incremented until it was found visually that severe hunting oscillations occurred. For this configuration, the critical speed for severe hunting was around 60 mph. The onset hunting speed may be 2 to 3 mph below the critical speed. The test train then ran through the test zone repeatedly with target speeds 25 mph, 35 mph, 45 mph and 55 mph. For each speed, two types of operations were performed, namely forced and unforced runs. For the forced runs, hydraulic truck forcers were used to intentionally distort the truck to a warp configuration. Upon a quick release, decaying oscillations were recorded. Sample test outputs are shown in figures 9 to 12. Unforced runs were made without the forces being actuated. From figures 9 to 12, it is noted that the decaying rate after each activation of hydraulic forcer decreases as the speed is increased. It was mentioned that the stability depends on the sign of α_k in equation 19. α_k can be represented by $§\omega_k$ where ω_k is the natural frequency of the kth mode and is always positive. The sign of §, usually referred as damping ratio, is the determining factor of stability. The mode is stable when § is positive or vice-versa.

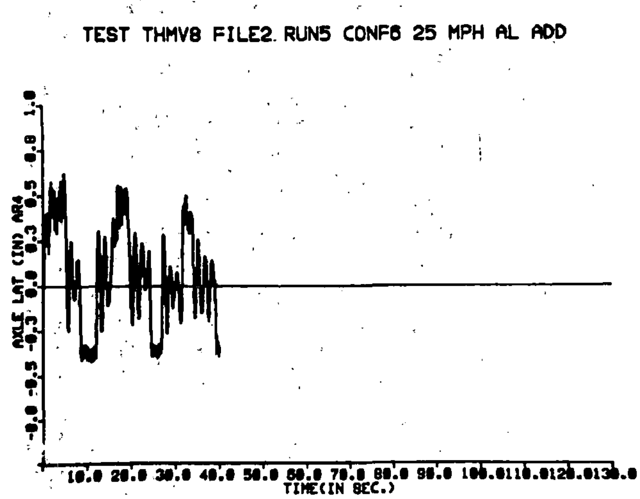


FIGURE 9. FORCER RUN OUTPUT AT 25 MPH

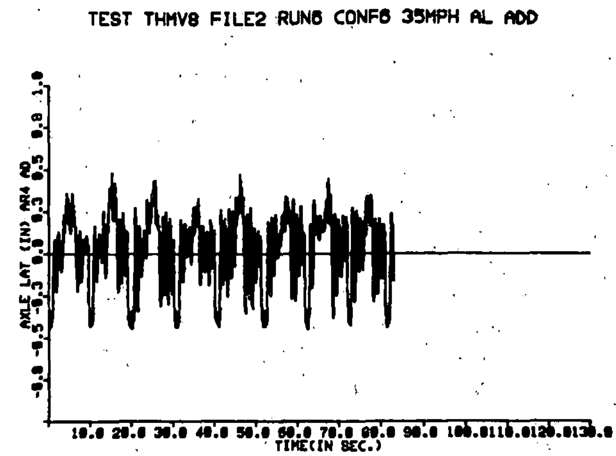


FIGURE 10. FORCER RUN OUTPUT AT 35 MPH

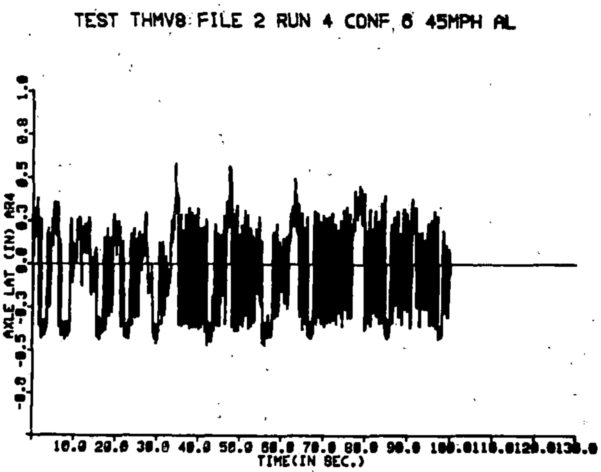


FIGURE 11. FORCER RUN OUTPUT AT 45 MPH

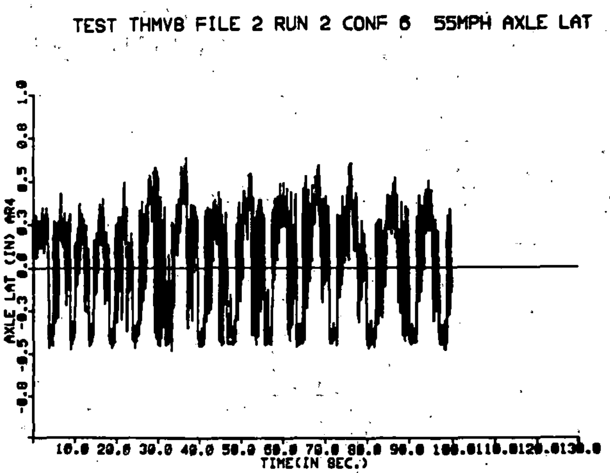


FIGURE 12. FORCER RUN OUTPUT AT 55 MPH

From logarithm decrement theorem and linear theory, § can be obtained by the formula

$$\ln (x_1/x_2) = 2\pi § / (1-§^2) \qquad (20)$$

where x_1 and x_2 are two consecutive high peaks.

From the test data, a damping ratio was calculated by equation (20) for every two peaks found. An average was taken over the damping ratios found for each speed. To illustrate the variation of the data, the $+\sigma$ and $-\sigma$ curves are plotted in figure 13. It shows that for low speed, the variation is small meaning that the car is more stable and the system behaves linearly. Conversely, the larger variation at high speed implies that the car tends to be more unstable and the response may not decay exponentially, a character of nonlinearities. Nevertheless, the mean should represent a damping ratio of an equivalent linear system. Figure 13 also shows that the damping ratio predicted by the model follows the same trend as that deduced from the test data. As speed goes up, the damping ratio decreases. The damping ratio is close to 0 at 55 mph in the test. This implies that the system will go unstable above that speed. It was observed during the test that the onset hunting speed occurred around 51 mph. The linear model predicts a zero damping ratio at 48 mph.

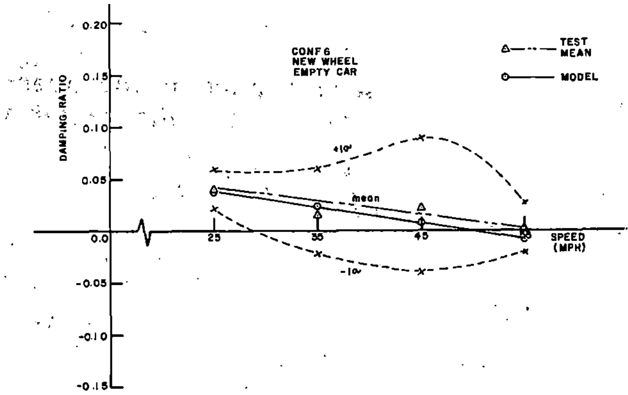


FIGURE 13. COMPARISON OF DAMPING RATIO BETWEEN LINEAR MODEL AND TEST

Figure 14 shows a comparison on the oscillating frequency of the axle lateral mode between the model prediction and the test results. When deducing the frequency from the test data, the time between two consecutive high peaks is taken as the period of oscillation at that point. Due to environmental factors and nonlinearities involved, the frequency may vary point to point in a test run. The +0 and 0 curves in figure 14 reveals that this variation is not as large as the damping ratio. The mean frequency varies with speed almost linearly. In general, the model prediction shows a very similar trend. As speed is increased, the frequency of oscillation is increased also. The frequency predicted by the model is consistently 8% higher than the actual test results at all speeds.

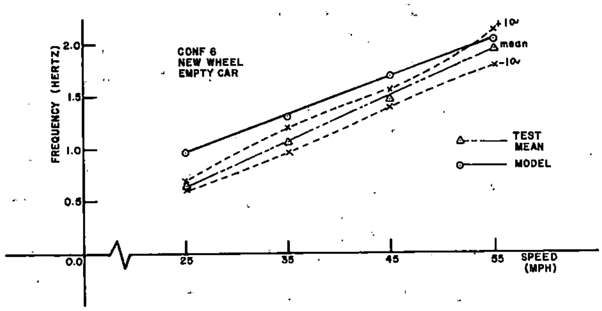


FIGURE 14. COMPARISON OF OSCILLATING FREQUENCY BETWEEN LINEAR MODEL AND TEST

The unstable mode shape shown by the linear model is a fishtailing mode; i.e., the front end of vehicle has a large lateral motion compared with the rear end. The phase difference between the lateral motions of the two trucks are about 145° out of phase. Figure 15 shows the lateral motion of front bolster and figure 16 shows the same motion of rear bolster in the test at hunting speed.

As seen from these two figures, the test vehicle shows a similar behavior as predicted by the model. The front bolster has a peak to peak magnitude of about 0.8 inch which is about the gage limit. Figures 17 and 18 show the lateral motions of the fourth axle (A-end) and first axle (B-end) The axles behave in the same manner as the bolsters although in the front end the relative displacement between bolster and axle may be larger.

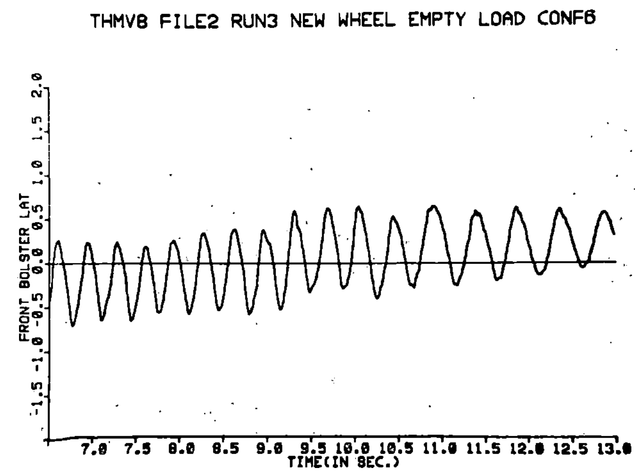


FIGURE 15. LATERAL MOTION OF FRONT BOLSTER AT HUNTING SPEED

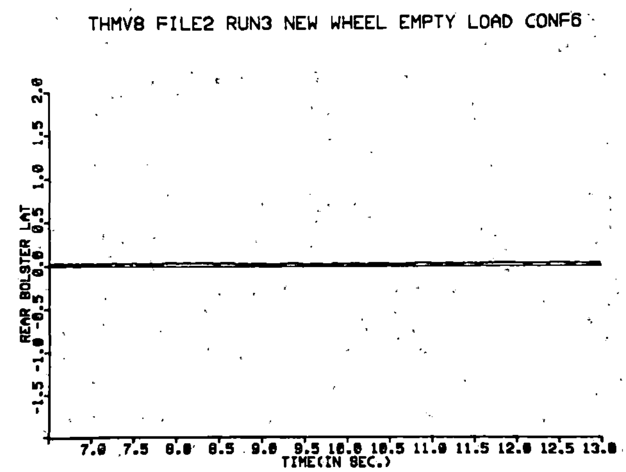


FIGURE 16. LATERAL MOTION OF REAR BOLSTER AT HUNTING SPEED

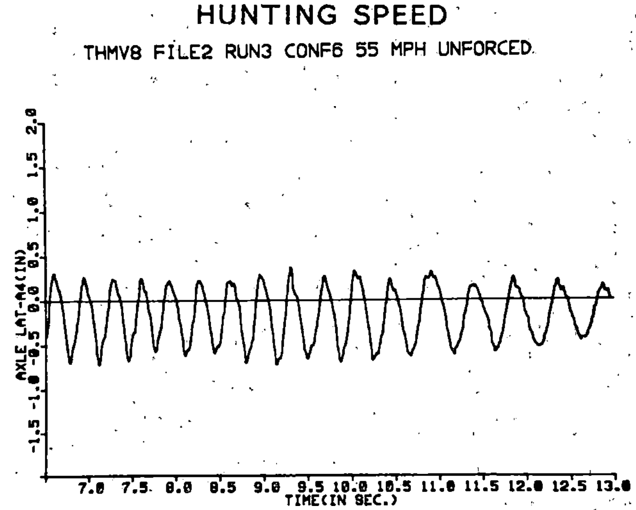


FIGURE 17. LATERAL MOTION OF FOURTH AXLE AT HUNTING SPEED

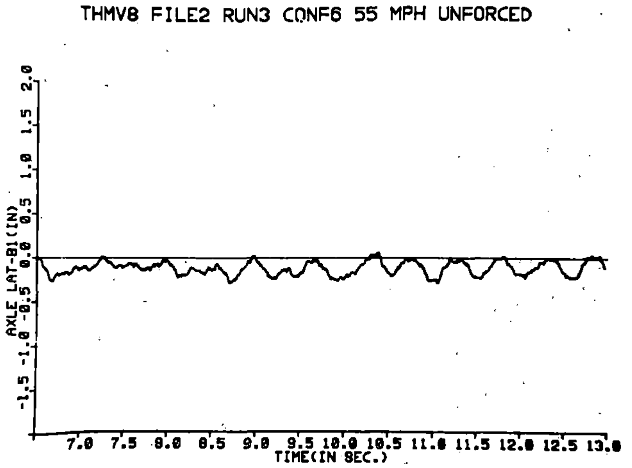


FIGURE 18. LATERAL MOTION OF FIRST AXLE AT HUNTING SPEED

Reynolds, reference 1, pointed out in his paper that the vehicle fishtailing mode can be due to non-symmetrical configurations between front and rear. Theoretically, the fishtailing mode can occur even if there are symmetrical configurations. This has been also found by Hull & Cooperrider reference 7 . Figure 19 shows a possible configuration when the carbody yaws to an extreme when both trucks have flange contact. As shown in the figure, the carbody tries to yaw back in an anti-clockwise direction. The front truck, having the larger rolling radii on the right, also tries to yaw in the same direction. Thus the front truck is in synchronization with the carbody yaw motion. The rear truck will behave in the opposite manner. It will try to yaw in the clockwise direction with the larger rolling radii on the left, acting against the body yaw motion. Thus, the motions between the front and rear are not symmetrical and form the basis for fishtailing mode.

For the case of nonlinear mode, runs were made with half-car option. In this option, one truck is assumed stationary in the lateral mode to simulate the fishtailing. The total number of degrees of freedom is reduced from 58 to 30 and thus computer time can be saved. For each simulation with a certain speed, a perturbation was introduced into the axle lateral mode. If the system was stable, the system would be damped or vice-versa. A large body motion was also experienced as in the field test. Figure 20 shows the axles lateral motion at the speed above critical speed. The axle lateral displacement builds up quite rapidly until flange contact, and enters a limit cycle as shown in figure 21.

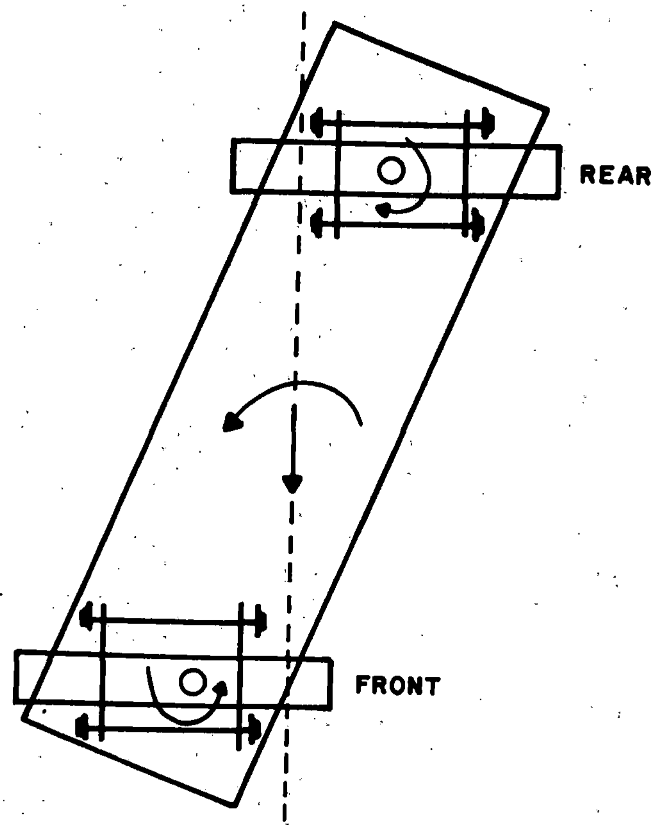


FIGURE 19. BODY YAW MOTION

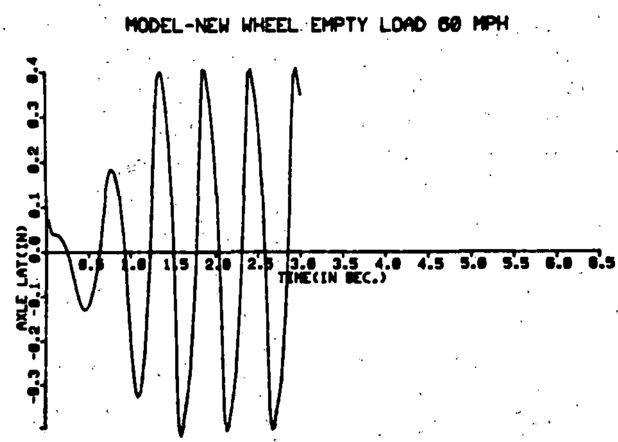


FIGURE 20. AXLE LATERAL RESPONSE OF NONLINEAR MODEL (ABOVE HUNTING SPEED)

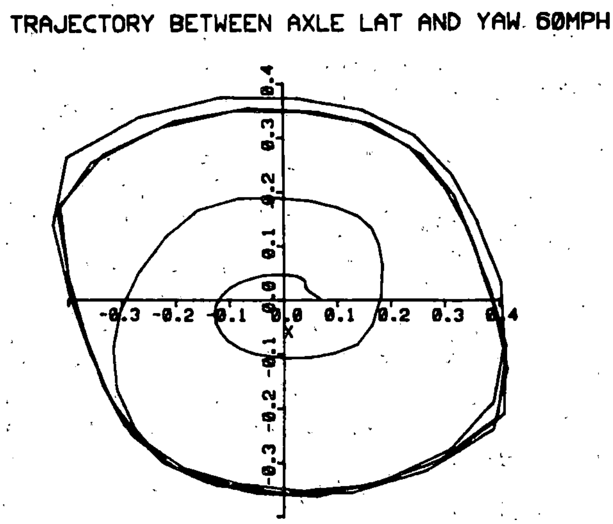


FIGURE 21. STATE TRAJECTORY BETWEEN AXLE LATERAL AND YAW DISPLACEMENTS (ABOVE HUNTING SPEED)

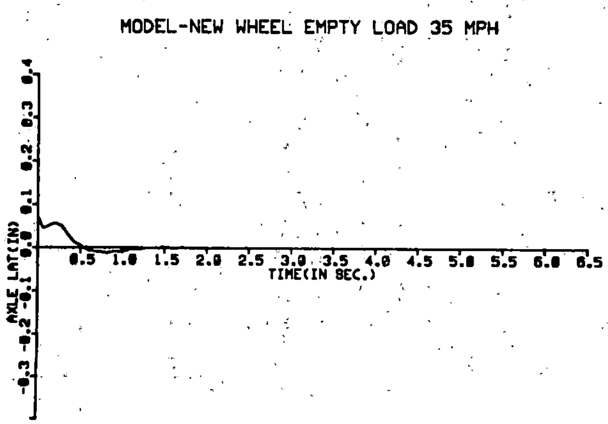


FIGURE 22. AXLE LATERAL RESPONSE OF NONLINEAR MODEL (BELOW HUNTING SPEED)

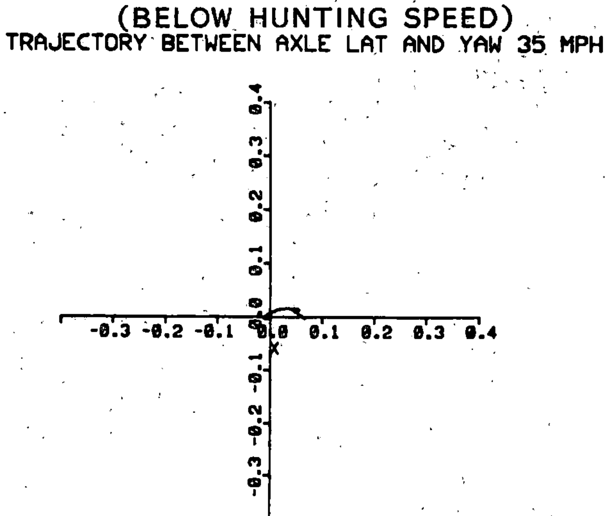


FIGURE 23. STATE TRAJECTORY BETWEEN AXLE LATERAL AND YAW DISPLACEMENTS (BELOW HUNTING SPEED)

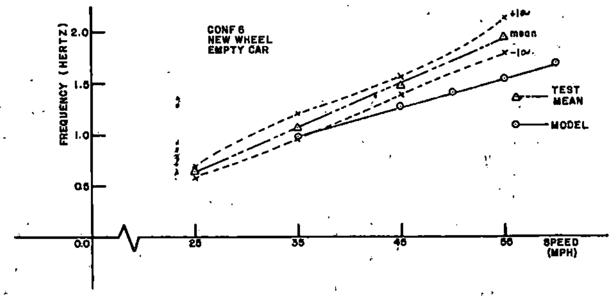


FIGURE 24. COMPARISON OF OSCILLATING FREQUENCY BETWEEN NONLINEAR MODEL AND TEST

Figures 22 and 23 show the same motion at 35 mph which is below the critical speed. Although the critical speed predicted by the model is quite close to the test onset hunting speed, the frequency predicted is generally below the test results. As shown in figure 24, the deviation of the model prediction from the test is larger as the speed is increased. This is probably due to the flexible rail simulated in the model. This aspect will be investigated in the future.

DISCUSSION

In this paper, two mathematical models were presented. One is a linear model and the other is nonlinear. In the linear model, the stability of the dynamic system is studied by finding the eigenvalues since all calculations are performed only once, the method requires a relatively small amount of computer time. For a mathematical model with nonlinearities, the eigenvalue-eigenvector formulation cannot be applied. The most common method used is a step-by-step numerical integration method. This method involves repeated computation for each time step and requires a large amount of computer time. Nevertheless, a nonlinear model does have other advantages besides giving a more representative theoretical representation. A railroad freight car system has numerous nonlinearities such as clearances, friction damping, torsional resistance etc. Sometimes is is difficult to justify transforming this system to be an equivalent linear freight car dynamic system. A nonlinear model generally accepts the data for nonlinearities directly without requiring a modification or conversion to linear element data. Quasi-linearization has a better approximation than pure linearization but still requires preliminary conversion.

The nonlinear system here is studied with a time-domain solution which, besides giving a clearer picture of how components react with each other, also has other merits. In dynamics analysis, loadings on components are of major concerns for a stress analysis. Engineers are often interested in knowing L/V ratio on the wheel in order to predict possible wheel-climb. Also, a time-domain model can be used to study the effect of external impact loadings.

CONCLUSIONS

Two mathematical models which were developed to study hunting behavior of freight cars are presented here. One model considered the freight car system as a linear dynamic system and the other considered it as a nonlinear system. The linear model which assumes a linear wheel/rail interaction is based on the eigenvalue-eigenvector solution. The model provides information about the critical velocity and critical frequency of the freight car system. It does not give any detail about the forces which may be generated during the hunting condition. The nonlinear model includes a nonlinear wheel/rail interaction, nonlinear friction creep relationship and other nonlinearities present in the suspension system. Since the simultaneous nonlinear differential equations are solved in time-domain using a step-by-step integration technique, the computational cost is significantly higher as compared to the linear model. The model provides information about the forces, displacements, etc., which may occur during the hunting condition. The performance of the model in comparison with the test data was found satisfactory to a certain level in the case of the new wheel/empty car configuration. The linear model can be successfully used in performing parametric study while the nonlinear model may be employed where a detailed analysis for hunting behavior is desired.

ACKNOWLEDGEMENT

The authors wish to express gratitude to a number of individuals whose efforts and co-operation have aided in this work. These include N. Darien, Test Director; J. Sims, B. Barker, and J. Valasina of the AAR Computer System Division. The cooperation from UP personnel during the testing phase is very much appreciated.

REFERENCES

1. Reynolds, D. J., "Hunting in Freight Cars". ASME paper # 74-RT-2, 1974.

2. "Technical Documentation - Linear Freight Car Hunting Model", AAR Publication under preparation.

3. Mims, W. E. & Yang, T. H., "Investigation of a Torsionally Flexible Freight Car", ASME paper # 75-WA/RT-13, 1975.

4. Abbott, P.W., Morosow, G., & MacPherson, J., "Truck and Carbody Characterization", Track Train Dynamics Publication, Report #R-186.

5. Darien, H., "Measurement, Methodology and Instrumentation Freight Car Dynamics Model Validation Field Test" Report to Track Train Dynamics, 1976.

6. "Harmonic Roll Series Vol. II", Track Train Dynamics Publication, 1974.

7. Hull, R., & Cooperrider, N. K., "Influence of Nonlinear Wheel/Rail Contact Geometry on Stability of Rail Vehicles", ASME paper # 76-WA/RT-2, 1976.

8. "Hunting Models Validation" Report under preparation.

NOMENCLATURE

a - half of rail gage

b - contact point spacing from axle center

C - total creepage

c - viscous damping coefficient

C_X - longitudinal creep velocity

C_Y - vertical creep velocity

C_Z - lateral creep velocity

F_c - viscous damping force

F_{cf} - total creep force

F_{cfx} - longitudinal creep force

F_{cfy} - vertical creep force

F_{cfz} - lateral creep force

F_{cz} - lateral creep force (linear model)

K_f - frictional force

F_G - lateral gravitational force

F_{Ko} - constant contact stiffness force

F_{Kl} - force of stiffness K_l

F_N - normal force on rail

f - dry friction constant

f_{11} - longitudinal creep coefficient

f_{22} - lateral creep coefficient

f_{23} - lateral/spin creep coefficient

f_{33} - spin creep coefficient

K_f - stiffness in series with friction

K_o - constant contact stiffness

K_l - stiffness with clearance

M_G - gravitational moment

R - concave transverse wheel tread radius

R' - convex transverse rail head radius

R_{cp} - rolling radius at contact point

R_o - initial reference rolling radius

V - vehicle velocity

W - weight on rail

X - longitudinal velocity of axle

Z - lateral velocity of axle

Z_{cp} - contact point spacing from track centerline

ϕ - yaw angle of axles

$\dot{\phi}$ - yaw velocity of axle

δ - slope of contact

δ' - clearance

δ_0 - initial slope of contact

δ_1 - slope of contact on right rail

δ_2 - slope of contact on left rail

λ - effective conicity

α_o - wheel coning angle

$\S$ - damping ratio

Configuration	Wheels	Load	Side Bearing Load	Truck Stifner	Centerplate Condition
1	Worn	Empty	Zero	Off	Dry
2	Worn	Empty	Zero	Off	Nominal Lubrication
3	Worn	Empty	2000#	Off	Nominal Lubrication
4	Worn	Empty	6000#	Off	Nominal Lubrication
5	Worn	Empty	Zero	Stiffen	Nominal Lubrication
6	New	Empty	Zero	Off	Nominal Lubrication
7	New	Loaded	Zero	Off	Nominal Lubrication
8	Worn	Loaded	Zero	Off	Nominal Lubrication

Table 2 Test Configurations

STRUCTURAL DYNAMIC ANALYSIS & FATIGUE LIFE PREDICTION OF A FLAT CAR

by

V. K. GARG
B. PRASAD
A. M. ZAREMBSKI

Dynamic characteristics of a trailer-on-flat car (TOFC) are investigated using finite-element techniques. Three different finite-element models of the flat car were developed. These models are validated in reference [1] by comparing the predicted vibration mode shapes and frequencies with test results. Further validation of the models is carried here by comparing the analytical transfer function with test results. The values of the transfer functions are computed, using NASTRAN, at four different locations along the center line of the flat car. Experimental values at these locations are found to be in good agreement with the computed results. Using the space beam model of the flatcar fatigue life prediction of an arbitrarily selected member of the flat car is carried here to demonstrate the application of finite-element structural dynamics analysis to fatigue life prediction. The fatigue life values obtained using this approach, are then compared with the so called ad hoc approach which uses a nominal stress value obtained from a pseudo-static analysis.

INTRODUCTION

Dynamic analysis of the railcar structure has been considered unmanageable in the past due to lack of proper analysis tools, namely solution techniques which utilize high speed digital computers. Consequently, the conventional method of dealing with dynamic loads in railcar design is to augment the static load solution with a selected dynamic load factor derived from an ad hoc approach based on experience. For relatively rigid structures, this usually ends up in overdesign, unnecessary cost and production of a much heavier car structure than required. For flexible railcar structures, the ad hoc approach may actually be unconservative.

With the advent of finite element solution techniques, extensive and successful dynamic analyses are now being carried out in most of the industries. The objective of this paper is to introduce dynamic structural analysis of the flat car structure and to illustrate the implementation of this analysis in the fatigue life determination.

The Trailer-on-Flat Car (TOFC), tested at the Rail Dynamics Laboratory (RDL) of Transportation Test Center (TTC) is used for which dynamic characteristics are available for the analytical simulation.

Three finite element models have been developed and reported in [1]. Dynamic characteristics were obtained using one or more commercially available finite element computer programs, namely SAP4 and NASTRAN.

The validation of these models was

V. K. Garg is Manager of Dynamics Research for the AAR. He received his B.S. from Banaras University, India in 1960, his M.S. from the University of California in 1966 and his Ph.D from the Illinois Institute of Technology in 1973.

B. Prasad serves as Senior Research Engineer for the AAR. He received his B.S. degree from Patna University, India in 1964, his M.S. from I.I.T. Kanpur, India in 1971, and his Degree of Engineer in 1975 from Stanford University.

Dr. A.M. Zarembski currently serves as Senior Research Engineer for the AAR. Zarembski received his B.S. in Aeronautics and Astronautics (1971) and his M.S. in Engineering Mechanics (1973) from New York University. In 1975 he received his PhD in Civil Engineering from Princeton University.

carried out by comparing the predicted vibration mode shapes and natural frequencies with those observed during the flat car test. A brief description of the models and the computed frequencies is given in the paper. A further validation of the models was performed here by comparing analytical transfer functions to the test data.

An application of the finite element dynamic analysis in predicting the fatigue life of a typical flat car component has also been performed in the paper in order to demonstrate its applicability. Additionally, the fatigue life of the component obtained using the ad hoc approach is compared to the one obtained using finite element structural dynamic analysis.

FINITE ELEMENT MODELS

In this section, three finite element models of the flat car are described briefly. Details regarding the geometric and material properties can be found in reference [1].

Simple Beam Model: The simplified beam model consists of 21 nodes, 20 beam elements, and 60 dynamic degrees of freedom. Three different beam cross-sections have been used in the model representing: (a) the deep cross-section of the flat car in the center, (b) the shallow cross-section of the flat car at the ends, and (c) a cross-section obtained by averaging sections in the transition region.

Space Beam Model: The complete flat car, excluding the trucks, was modeled using only beam elements. The model includes 158 nodes and 299 beam elements. Although, the deck plate is not physically modeled, the stiffening effect of the deck plate due to the torsional and flexural stiffness in the beam elements near the deck has been included. All the side sill stringers and cross members are located on the same horizontal plane, thus neglecting the effect of offset. The middle portion of the center sill is represented by an equivalent beam located at the centroid of the section.

Detailed Beam Plate Model: The detailed finite element model of the flat car encompasses the entire structure, figure 1. The model consists of 299 nodal points, 330 beam members, and 180 plate elements. The flanges of the center sill, stringers, side sills and cross members have been modeled as beams. The deck plate and center sill web, at the deep section, are represented by plate (membrance and bending) elements. The center sill at the deep section is modeled as a single vertical plate with beam elements running along its top and bottom edges. The combined beam and plate model was considered necessary for the proper representation of large member offset. Torsional stiffness of the center sill is assigned to the top beam. Since the plate elements representing the deck plate contribute to the axial stiffness and mass, a negligible value is assigned to the top beam cross sectional area. The bottom beam represents the center sill cover plate.

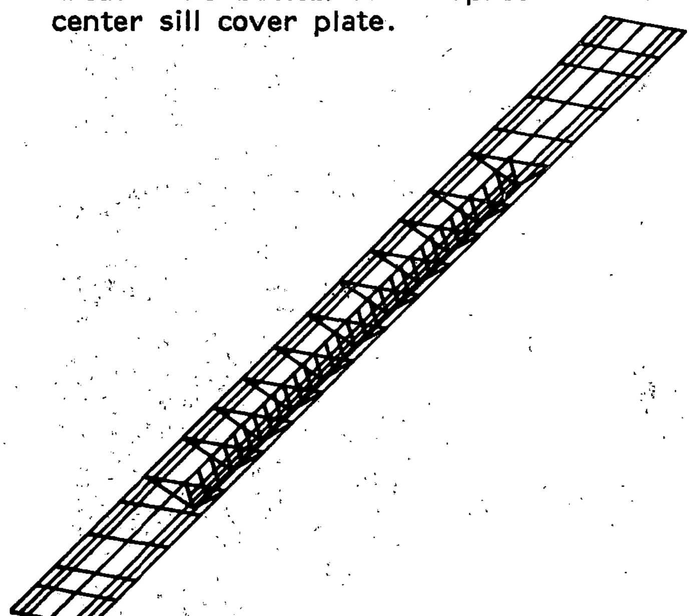


FIGURE 1. FINITE ELEMENT MODEL OF 89'-4" FLUSH DECK FLAT CAR

Vertical columns are added to represent center sill separations at cross bearer locations, figure 2. Nineteen different types of beam elements and two types of plate elements are used in the model.

FREE VIBRATION RESPONSE

In order to demonstrate the ability

of the finite element technique to predict dynamic behavior. the computed natural frequencies are compared with test results. The four flexural and three torsional natural frequencies recorded during the test along with analytical results obtained for the SAP4 and/or NASTRAN computer programs are given in Table I. A comparison of mode shapes with the test results can be found in reference [1] .

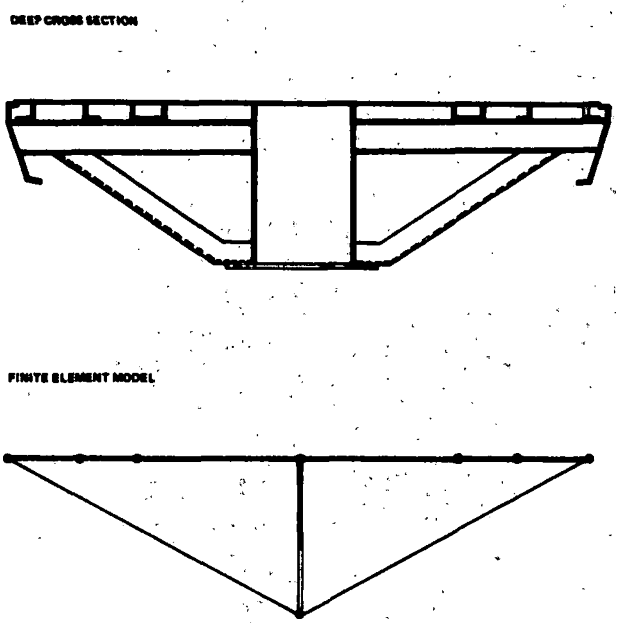


FIGURE 2. MODELING OF DEEP CROSS-SECTION

Although the models do not reproduce the TOFC test results for all modes, the overall agreement is adequate for all practical purposes. Selection of the type of model and level of refinement depend on the quality of results desired. A simple beam model may be considered adequate for a preliminary estimate of the fundamental frequency, but a detailed model is necessary to obtain the internal loads accurately. Simplifying assumptions about the boundary conditions in the analytical models and difficulty in isolating mechanical parameters during tests precluded a total agreement between the model and test results.

COMPARISON OF TRANSFER FUNCTIONS

In the dynamic study of a system, a more general concept relating the transformed response to the transformed excitation is often used. This concept is based on the "System Function" or "Transfer Function". For a n-dimensional dynamic system having a set of equations of motion of the form

$$[m]\{\ddot{x}\} + [c]\{\dot{x}\} + [k]\{x\} = \{p(t)\} \quad (1)$$

where $[m]$, $[c]$ and $[k]$ represent the mass, damping and stiffness matrices of the system; $\{\ddot{x}\}$ $\{\dot{x}\}$ and $\{x\}$ correspond to the acceleration,

Bending Frequencies (CPS)

Number of Bending Mode	Model I Simple Beam		Model II Space Beam		Model III Detailed	TOFC Test Results
	SAP4	NASTRAN	SAP4	NASTRAN	NASTRAN	
1	4.16	4.21	3.84	8.86	4.12	4.28
2	11.11	11.28	8.94	8.89	9.11	9.02
3	14.06	14.42	11.50	10.89	11.36	11.31
4	22.36	22.36	16.77	17.56	18.68	18.46

Torsional Frequencies (CPS)

Number of Torsional Mode	Model I Simple Beam		Model II Space Beam		Model III Detailed	TOFC Test Results
1	--	--	5.06	4.70	4.72	6.02
2	--	--	6.67	7.51	10.81	10.22
3	--	--	11.22	11.27	13.91	12.84

TABLE 1. COMPARISON OF BENDING AND TORSIONAL FREQUENCIES

velocity, and displacement vectors of the nodal points and $\{p(t)\}$ is the force vector, the transfer function is represented as (2)

$$\bar{G}(s) = \frac{\bar{x}(s)}{\bar{p}(s)} = \frac{1}{[m][s^2+2\,\xi_n\omega_n s + \omega_n^2]}$$

where s denotes the laplace transform parameter defined by (3)

$$\bar{x}(s) = \mathcal{L}[\{x(t)\}] = \int_0^\infty e^{-st}\,\{x(t)\}dt$$

In equation 2, ξ_n, ω_n are the viscous damping factor and undamped natural frequency of the system, respectively. Equation 2 can be written as (4)

$$\bar{x}(s) = \bar{G}(s) * \bar{P}(s)$$

Consequently, the transfer function can be regarded as an algebraic operator that operates on the transformed excitation to yield the transformed response. This concept is shown by the following block diagram:

Transformed -excitation $\bar{P}(s)$ → Transfer Function $\bar{G}(s)$ → Transformed Response $\bar{X}(s)$

A procedure similar to above has been used in NASTRAN.

The response of the flat car in terms of its transfer functions was computed for different input frequencies using plate and beam model. The results are shown in figure 3 (a) and figure 3 (b). Four sets of curves for the transfer function corresponding to four representative locations along the center and off-center line of the flat car are plotted. These locations represent the positions of the accelerometers 110, 119, 112, and 118 used in the test [3]. The results obtained from the test are also plotted. These are marked with (*). The computed transfer functions agree fairly well with the test results.

FORCED VIBRATION ANALYSIS

In order to investigate the forced vibration response of the car, we consider the damped equations of motion, equation 1.

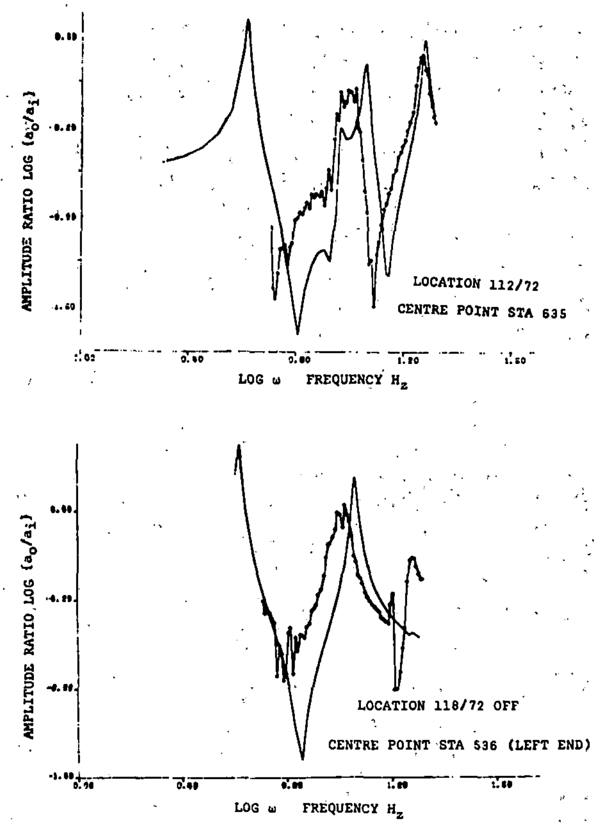


FIGURE 3(a). COMPARISON OF TRANSFER FUNCTION

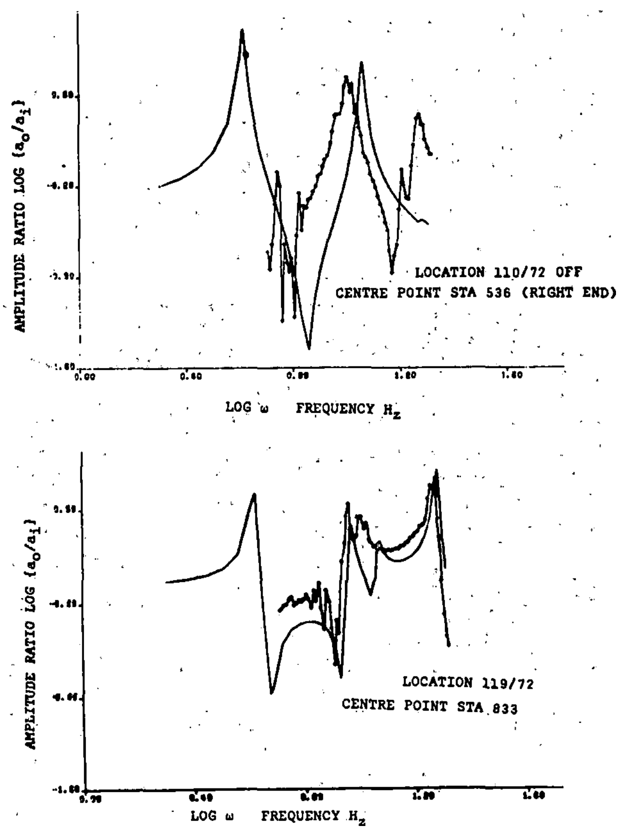


FIGURE 3(b). COMPARISON OF TRANSFER FUNCTION (CONTINUED)

The total displacement can be expressed as the sum of the modal components.

$$\{x\} = [\phi]\{U\} \tag{5}$$

where the mode shape matrix $[\phi]$ transforms the generalized coordinates $\{U\}$ to the geometric coordinates $\{x\}$.

It is assumed that the damping is a linear function of mass and stiffness matrices and is of the form

$$[c] = \alpha[m] + \beta[k] \tag{6}$$

where α and β are arbitrary proportionality factors.

Now, using equation 6 and the following orthogonality conditions:

$$\{\phi_m\}^T[m]\{\phi_n\} = 0 \tag{7}$$

$$\{\phi_m\}^T[k]\{\phi_n\} = 0 \;;\; m \neq n$$

$$\{\phi_m\}^T[c]\{\phi_n\} = 0$$

Equation 1 can be written as

$$M_n\ddot{U}_n + C_n\dot{U}_n + K_nU_n = P_n(t) \tag{8}$$

in which

$$M_n = \{\phi_n\}^T[m]\{\phi_n\} \tag{9}$$

$$C_n = \{\phi_n\}^T[c]\{\phi_n\} = 2\,\xi_n\,\omega_n\,M_n$$

$$K_n = \{\phi_n\}^T[k]\{\phi_n\} = \omega_n^2\,M_n$$

$$P_n(t) = \{\phi_n\}^T\{p(t)\}$$

$$\xi_n = (\alpha+\beta\omega_n^2)/2\omega_n$$

Equation 8 is a single degree of freedom equation of motion for the nth mode. M_n, C_n, K_n, and $P_n(t)$ are the normal coordinates generalized damping, generalized stiffness and generalized load for the nth mode, respectively. ξ_n is the damping ratio for the nth mode.

Equation 8 may be written in the form

$$U_n = \frac{P_n(t)}{M_n\omega_n^2} - \frac{2\xi_n}{\omega_n}\dot{U}_n - \frac{\ddot{U}_n}{\omega_n^2} \tag{10a}$$

$$U_n = U_{n,I} + U_{n,II} \tag{10b}$$

where we define

$$U_{n,I} = \frac{P_n(t)}{\omega_n^2 M_n}$$

and

$$U_{n,II} = -\frac{2\xi_n}{\omega_n}\dot{U}_n - \frac{\ddot{U}_n}{\omega_n^2}$$

The first term $U_{n,I}$ in equation 10b represents the response due to load $P_n(t)$ applied to the structure statically when $\dot{U}_n = \ddot{U}_n = 0$. The second term, $U_{n,II}$, represents the responses due to velocity $\dot{U}_n$ and acceleration $\ddot{U}_n$ of structure when it is vibrating in its nth mode. Instead of computing $U_{n,I}$ corresponding to each mode and adding the corresponding internal forces, the total exciting force $P(x,t)$ at time t can be applied to compute the total static internal forces, $P_I(x,t)$. The internal force $P_{II}(x,t)$ corresponding to $\{\dot{x}\}$ and $\{\ddot{x}\}$ can be obtained by computing internal force $P_n(x)$ corresponding to the nth normal mode $\{\phi_n(x)\}$ from inertial force associated with this mode and using $U_{n,II}$ as:

$$P_{II}(x,t) = \sum_{1}^{n} P_n(x)\;U_{n,II}(t) \tag{12}$$

The total internal force or stress is then given by

$$P(x,t) = P_I(x,t) + P_{II}(x,t) \tag{13}$$

$P_I(x,t)$ is completely accounted for irrespective of the number of mode considered in the analysis whereas the internal force $P_{II}(x,t)$ computed from equation 12 is dependent on the number of modes considered.

The scheme based on this procedure has been adapted in the SAP4 program [8] for computing stress time history of components.

FATIGUE LIFE PREDICTION

In order to utilize fatigue life analysis techniques in freight car design, the linear cumulative damage concept was introduced in the Interim AAR Guidelines for Fatigue Analysis of Freight Cars [5]. Though there are many theories available at present [4,10], the theory of Miner-Palmgren was selected because of its simplicity, ease of application, and overall agreement with reality. This theory, as it is utilized in the Interim AAR Guidelines, requires the input of the material properties of freight car component under investigation, together with an appropriate characterization of the load environment that the component experiences during the course of its service life. This load characterization should be defined in terms of maximum-minimum loading cycles, since the change in stress during a cycle can be as important as the maximum stress value. In order to develop this characterization, a Rainflow cycle counting 6 of the load or stress time histories (figure 4) is used to develop an environmental characterization spectrum of the form shown in figure 5. Though it is possible to obtain the stress time history directly from a road test through the strain gaging of specific components, a more desirable alternative is to obtain an acceleration time history, at the point of loading, i.e., at the bolster, and convert it to stresses. This allows for the determination of the fatigue life of any component of the freight car. However, this approach is dependent on the ability of an adequate stress analysis.

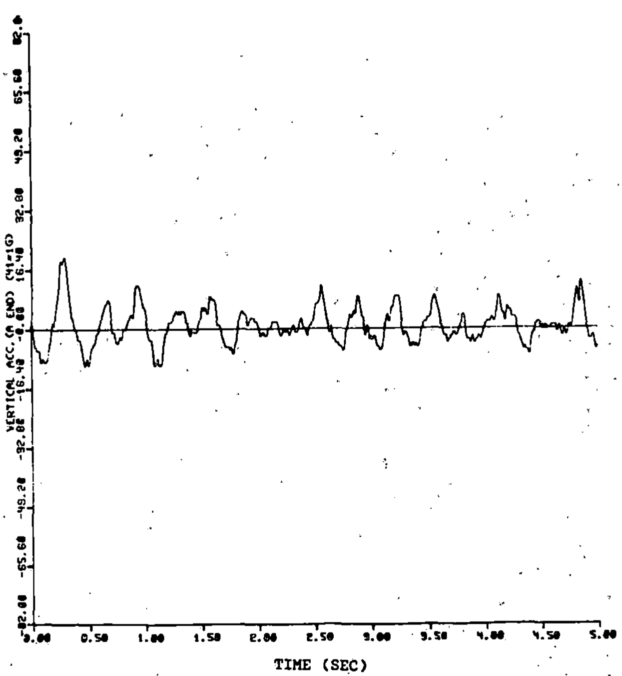


FIGURE 4. A TYPICAL ROAD ACCELERATION TIME HISTORY

There are, at present, two distinct approaches for the calculation of the stress history of a freight car component using a known acceleration time history. The first approach is quasi-static. A static stress analysis is used to calculate the static stress at the component under investigation. A simplified model of car structure is often used for this purpose. The static stress is then augmented by a dynamic stress component which is obtained from the product of the acceleration (in g's) and the static stress value. This approach will be referred to herein as the ad hoc approach. Since a linear relationship between stress and acceleration or load is used, the service environment is characterized by a load spectrum, such as that shown in figure 5, which is obtained from the Rainflow cycle counting of the acceleration time history (figure 4). It is then converted to the corresponding stress spectra for use in the fatigue analysis 5 .

MAXIMUM(VERTICAL AXES) VS. MINIMUM(HORIZONTAL AXES)

AXES VALUES ARE IN KIPS

200.00 / 180.00	0.00	0.84	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.84	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
180.00 / 160.00	0.00	0.00	0.00	0.00	0.00	0.68	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	9.00	0.00	0.00	1.68	
160.00 / 140.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00		
140.00 / 120.00	0.00	0.84	0.00	0.00	0.00	0.00	0.00	0.84	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
120.00 / 100.00	0.00	0.00	0.00	0.00	0.00	1.68	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.68	0.00	0.00				
100.00 / 80.00	0.00	0.00	0.00	0.00	0.00	0.00	3.36	3.36	1.68	0.00	0.00	0.00	0.00	0.00	0.00					
80.00 / 60.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.68	0.00	0.00	0.00	0.00	0.00						
60.00 / 40.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	3.36	1.68	6.72	1.68							
40.00 / 20.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.68	1.68	5.04	3.36								
20.00 / 0.00	0.00	0.00	0.00	0.00	0.00	1.68	0.00	0.00	0.00	16.81	1.68									
0.00 / -20.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	10.08	11.76										
-20.00 / -40.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	3.36	0.00											
-40.00 / -60.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	4.20												
-60.00 / -80.00	0.00	0.00	0.00	0.00	0.00	5.04	0.00													
-80.00 / -100.00	0.00	0.00	0.00	0.00	0.00	0.00														
-100.00 / -120.00	0.00	0.00	0.00	0.00	0.00															
-120.00 / -140.00	0.00	0.00	0.00	0.00																
-140.00 / -160.00	0.00	0.00	0.00																	
-160.00 / -180.00	0.00	0.00																		
-180.00 / -200.00	0.00																			

-200.00 -160.00 -120.00 -80.00 -40.00 0.00 40.00 80.00 120.00 160.00 200.00
-180.00 -140.00 -100.00 -60.00 -20.00 20.00 60.00 100.00 140.00 180.00

RAINFLOW PERCENTAGE FOLD

THE TOTAL NUMBER OF HALF CYCLES COUNTED IS: 119.

THE LARGEST DATA POINT IN ENGINEERING UNITS IN THE MATRIX IS: 188.49

FIGURE 5. ENVIRONMENTAL SPECTRUM

The second approach utilizes a dynamic stress analysis to convert acceleration time history into a stress time history at the component under investigation. This is done here by using the SAP4 finite element program and the space-beam model of the flat car. The resulting stress time history is then reduced, through Rainflow cycle counting, to the spectrum form which is illustrated in figure 5, and input directly into the fatigue analysis program. This approach will be referred to herein as the dynamic approach.

At present, the ad hoc approach is the one most commonly used in the freight car industry, and it is the approach presented in the Interim AAR Guidelines [5]. This is primarily due to the fact that the dynamic approach is an elaborate, time-consuming technique which often requires a finite element analysis capability. Additionally, it is expensive and consequently cannot be used for long duration time histories. Thus, even if the capability existed, it could only be applied to short segments of the load time histories, which may not be representative of the total service environment. Furthermore, it could not be used for the detailed analysis of a large number of components, but should be reserved for only the most fatigue critical ones. Finally, the ad hoc approach offers the advantage of being able to be performed quickly, and repeatedly, with limited or no computer support. However, in doing so it requires many simplifying assumptions, which may not be very valid. Thus, this may demand for a thorough testing in order to qualify the predicted fatigue life of the component under investigation.

The dynamic approach on the other hand offers the significant advantage

Space Beam Model			Area	Section Modulus			
Section No.	Member Description	Element No.	A in^2	Z_T in^3 Top	Z_B in^3 Bottom	Z_L in^3 Left	Z_R in^3 Right
14	Crossbearer (Deep Sec.)	263 & 264	8.25	29.1241	17.5802	7.2440	7.244

TABLE 2. PROPERTIES OF THE MEMBER CHOSEN FOR FATIGUE EVALUATION

Acceleration Scale Factor	Fatigue Life : Cycle's/Mile		* Material Prop. No.
	Adhoc Approach	Dynamic Approach	
1.00	8.7646897×10^7	3.52674×10^7	1
2.00	4.316198×10^6	9.951996×10^5	1
2.00	6.77034×10^7	1.321035×10^7	2
3.00	3.562526×10^7	7.884276×10^6	3

* 1 ; B = 8000., m = 1.0, K = .32
2 ; B =15000., m = 1.0, K =0.29
3 ; B =19000., m = .675,K =.145

TABLE 3. FATIGUE LIFE PREDICTION OF A TYPICAL FLAT CAR COMPONENT

of more accurately representing the transfer of the input accelerations to the component under investigation, while retaining the time and frequency dependencies. The resulting output is a well-defined stress time history for the component under investigation. Since both the dynamic and ad hoc approach utilize linear cumulative damage theory for fatigue life calculation, the validity of replacing the component dynamic stress time history with the quasi-static time independent stress spectrum obtained by the ad hoc approach, and the resulting implication for the fatigue life analysis should be investigated.

In order to perform a comparison between these two approaches, the unloaded flat car described earlier was analyzed using both the ad hoc and the dynamic approaches. A typical road acceleration time history shown in figure 4, was selected and the component to be analyzed was a deep cross bearer section at the car center. The geometric properties of this member are given in Table 2. The two analytical approaches were then used to predict the fatigue life of the component. These values are shown in Table 3. In order to preclude any bias due to differing time durations, the range of input acceleration time history was taken to be equal to that of the output stress time history.

As can be seen in Table 3, the ad hoc approach predicts a fatigue life for the freight car component that is non-conservative when compared with the life value obtained in the dynamic approach. However, the value obtained in the ad hoc approach is within an order of magnitude of that obtained from the more precise dynamic approach. Thus, it may be inferred that the careful use of the ad hoc approach is justified for all but the most fatigue critical components. In the later case, use of the more precise dynamic approach appears to be warranted.

CONCLUSION

Finite element models of the flat car developed in reference [1] is further validated here by comparing the transfer functions with the test results. A good agreement can be noted. One of the models was used for obtaining dynamic stresses of a typical component of the flat car under a given time history. Fatigue life prediction of the component was made using the ad hoc and the dynamic approach. The results of the two approaches were compared. Fatigue life predicted by the ad hoc approach appears to be unconservative. The use of the ad hoc approach thus may be limited to most but a few critical components where the dynamic approach should be used. Since the ad hoc approach does not take into consideration the frequency contents of the environmental history, a further analysis is needed to compare the two approaches. Careful evaluation of the ad hoc approach is suggested to establish its validity for frequency dependent environment.

ACKNOWLEDGEMENT

This work was performed under Task IX of the Track-Train Dynamics Phase II program. It is the result of a cooperative effort of many persons in several organizations, participating railroads, supply industries including Pullman-Standard and Trailer Train. Arrangements for the flat car tests at the DOT Transportation Test Center in Pueblo, Colorado, were made by Dr. David Sutliff, Director of Phase II of TTD Program, and Ms. Grace Fay, Manager, Freight System Research of the FRA. The actual tests were conducted by ENSCO, Inc. The reduction of data and presentation of experimental results was performed by Wyle Laboratories, Colorado Springs Division.

REFERENCES

1. Task IX - "Demonstration Test and Analysis of the Dynamic Characteristic of the Flat Car"; Volume I - Free Vibration Study, AAR Technical Center, Chicago, Ill. 60616.

2. Task IX - "Demonstration Test and Analysis of the Dynamic Characteristic of a Flat Car"; Volume II -

Forced Vibration Study and Fatigue Life Prediction, under preparation. AAR Technical Center, Chicago, Ill.

3. Task IX - AAR - Dynamic Analysis Test Program; Engineering Report, David W. Gibson, Wyle Laboratories, Colorado Springs, Colorado 80915, March 1977.

4. Wirsching, P.H. and Yao, J.T.P., "Statistical Methods in Structural Fatigue", Journal of the Structural Division, ASCE, Vol. 96, No. ST6, Proc. Paper 7377, June 1969, pp. 1201-1219.

5. Interim AAR Guidelines for Fatigue Analysis of Freight Cars, Association of American Railroads, Report R-245,, Chicago, Ill., July 1977.

6. "Users Guide to Rainflow Counting Program", Association of American Railroads, Report R-274, Chicago, Ill., July 1977.

7. "Specification for the Design, Fabrication and Construction of Freight Cars", AAR, Revised March 2, 1976.

8. SAP4 - "Structural Analysis Program for Static and Dynamic Response of Linear Systems" ed: Bathe, K. J., Wilson, E. L. and Peterson, F. E., Report No. EERC-73-11, June 1973, Berkeley, Calif.

9. "Users Guide to Fatigue Life Analysis Program (FLAP)", AAR Technical Center, Report R-273, July 1977.

10. Stallmeyer, J. E. and Walker, W. H. "Cumulative Damage Theories and Application", Journal of Structural Division, Proceedings of the ASCE, Volume 94, No. ST12, December 1968, pp. 2739-2751.

SIMULATION COST MODELING FOR THE DETERMINATION OF FREIGHT CAR COMPONENT OPERATING COSTS

BY

KEITH L. HAWTHORNE

ALLAN I. KRAUTER

RAJENDRA SAROOP

This paper describes a technique developed to determine freight car component operating costs. Early in the planning of the Phase II Track Train Dynamics Program, the need for a methodology to determine freight car component operating costs was established. The proposed creation of performance specifications necessitated the establishment of a methodology to determine the economic impact of design changes. After a review of alternative methods of cost determination, a technique known as Simulation Cost Modeling (SCM) was selected. Simulation Cost Modeling has three elements: the schematic diagram which describes component usage; the computer program which implements the diagram; and the input data set. Development of the Simulation Cost Modeling technique and its relationship to Track Train Dynamic component cost evaluations are described. A preliminary input data set and corresponding approximate results are given for the freight car wheel. This reference case approximates the current usage of freight car wheels and the associated costs of acquiring, operating, and maintaining freight car wheels.

INTRODUCTION

The fundamental purpose of the Phase II Track Train Dynamics Program is to develop recommended performance specifications for selected railroad equipment and track structures. During the planning of the Phase II program, it became evident that the implementation of such performance specifications would require the availability of design guidelines to enable both railroads and railroad suppliers to meet the new specifications. It was also evident that new specifications would require economic as well as technological evaluation. This paper describes the economic technique, simulation cost modeling, which was selected for the evaluation of performance specification alternatives.

The simulation cost modeling technique was selected after a review of other techniques, such as life cycle costing, indicated that their data requirements would be difficult to fulfill. Using the simulation cost modeling technique, the annual present and projected future costs associated with using specific components for the freight car can be predicted.

The original SCM technique was developed under a Department of Transportation (DOT) contract to evaluate the cost of operation and ownership of freight car roller bearings.1 Within the scope of another DOT contract, present value analysis and life cycle costing were considered as alternative tools for component cost evaluation.2 However, the limitations in the use of those tools for determining railroad

---

Keith Hawthorne is the Director of Safety Research and Applied Technology and Deputy Director of Track Train Dynamics Program with the AAR. He received his M.S. in Mechanical Engineering at Lamar State College of Technology in Beaumont, Texas.

Dr. K. L. Krauter is the Senior Mechanical Engineer for the Shaker Research Corporation. He received his B.S. in Mechanical Engineering from Stevens Institute of Technology (1963). In 1964 he received his M.S. and in 1968 his PhD from Stanford University. Dr. Krauter is a registered Professional Engineer in New York.

Rajendra Saroop serves as Project Manager, Improved Track Structures Research Division for the FRA. He received his B.S. in Mechanics from the Indian Institute of Technology, Canpur, India (1965). He received his M.S.I.E. from West Virginia University in 1968.

component costs led to further development of the cost modeling technique. This paper describes recent work to refine the technique, its current status, and its preliminary application to one of the railroad industry's most costly components--the freight car wheel.

REVIEW OF SIMULATION COST MODELING

The simulation approach to cost modeling is different from other commonly used cost modeling techniques. Another frequently used technique, life cycle costing, follows a particular component from its original purchase, through use, to its eventual retirement (in a statistical sense). All cost attributable to the component are included in the cost total. A life cycle cost model is useful in determining the actual overall cost of the component to the industry which is employing it. The life cycle cost model does not, however, lend itself to the determination of the yearly cost required to use the component. In addition, data requirements required for implementing a life cycle cost model can be difficult to meet.

The life cycle technique has other characteristics which make it unwieldy in the railroad environment. One of these is that the consideration of the entire lifetime of the component interferes with representing component usage at any one instant of time in a detailed manner. As an example, a life cycle cost model of say, wheels, might require combining many effectively separate inspection costs together. Another characteristic is that the method does not lend itself easily to the determination of overall cost changes due to changes in individual operating costs or operating practices. Finally, effects on the life cycle costs of rather uncertain individual costs and component reliability characteristics can be large. In such situations, the calculated and actually experienced life cycle costs can differ substantially.

Refinement of the simulation cost modeling technique has overcome most of the shortcomings of the life cycle costing method. In the simulation approach, the costs are obtained by following the system (e.g., railroad or railroad industry) in its use of the component. A representation or schematic diagram of the use of the component of interest by the railroad is developed. The model then computes the cost per year to operate that part of the system related to the component under consideration. This is accomplished through the use of a digital computer program which implements the schematic diagram and the associated input data. The emphasis is on the instantaneous behavior of the component flowing through the system. Consequently, the data requirements are relatively easy to satisfy. The technique allows sensitivity analyses and cost benefit studies to be performed easily and provides many opportunities to check the accuracy and reasonableness of the computed results. In addition, since the cost modeling is based on dynamic simulation procedures, the technique allows future costs and usage projections to be made. Such projections can include the effects of introducing a new or improved component, such as may result from a performance specification, into the existing system. The ability of the technique to evaluate the effects of changing either the specification or the manner in which the system employs the component makes it readily applicable to the needs of the Track Train Dynamics Program.

By following the flow of a component in use, the SCM technique calculates the cost per unit time (typically a year) required by a railroad or the railroad industry to operate a given component. The cost of component operation includes both maintenance costs and acquisition costs. In order to calculate the cost of component usage, it is first necessary to create a schematic diagram representing all the potential paths of component flow. The schematic diagram identifies the component related parts of the system, system interactions which involve the component, and the decisions which take place concerning the component. Continuing the example of the freight car wheel, such a schematic diagram is given in figure 1.

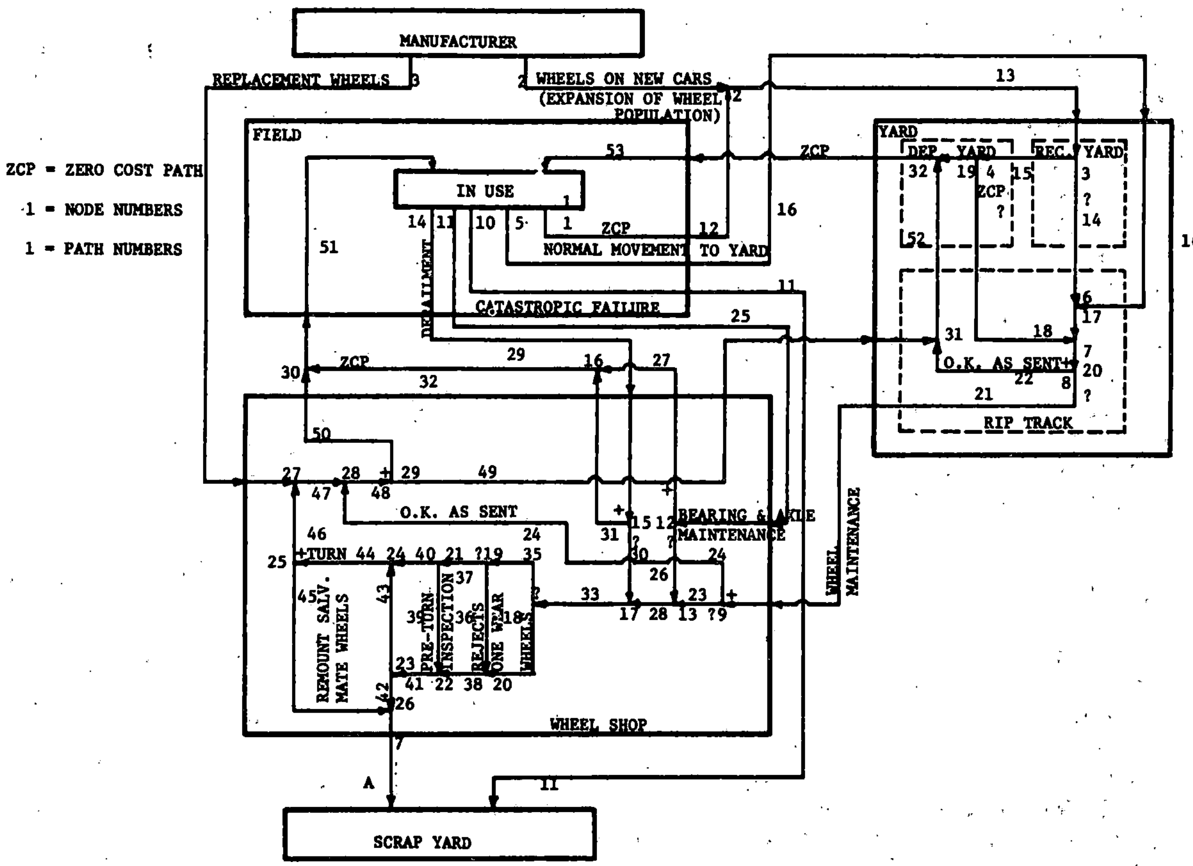


FIGURE 1. SCHEMATIC DIAGRAM FOR FREIGHT CAR WHEEL UTILIZATION

A review of the diagram given in figure 1 reveals the following features of the SCM technique:

- The parts of the railroad which affect the component (such as manufacturer, yard, and wheel shop) are major elements of the schematic diagram.

- The paths (uncircled numbers) represent the alternative directions in which a component may flow either within or among the major parts of the system. The movement along each path is characterized in terms of flow rate (e.g., the number of wheels per year which flow along a given path). Flow through most paths results in associated costs characteristic of the given path. Those paths which have no cost are identified as zero cost paths (ZCP).

- The intersections of paths or nodes are identified by circled numbers. Nodes at which path flows divide are branch points and nodes at which paths join are summation points. At each branch point a decision affecting the component occurs. This decision results in the proportion of arriving units which move along each of the two departing paths. As an example, the inspection of a wheel flange at a receiving yard results in a decision to either leave the wheelset on the car and thus return it to service (in use) or move the car to the rip track for removal of a wheelset bad ordered by the inspector.

Once the schematic diagram has been completed, each of the paths and nodes are incorporated in a digital computer program. This program implements the schematic diagram by performing the following tasks:

(1) The program computes the number of units which move along each of the paths. To do this the program uses known values of the size of the component's population, time, age of the population, and decision criteria at the branch points.

(2) The program computes the costs associated with the computed flow in each path. A summation of the computed costs for all paths gives the annual operating costs.

(3) The rate of change of the population size and the rate of change of population age are computed. These rates of change are used to predict the size and age (the state variables) at the next time of interest.

(4) Once the state variables at the next time of interest are determined, steps (1) and (2) are repeated. As the decision parameters and individual path costs vary with time and/or with population size and age, a new set of flows and costs are computed. Continued repetition of the process produces a dynamic simulation which predicts component usage and associated costs at future times.

In addition to producing the dynamic simulation of component usage and cost, the computer program can perform the production of a sensitivity analysis. This sensitivity analysis reflects the change in the annual system operating costs which result from a change in a decision criterion or an individual path cost. It is generally computed in step with a reference time selected for the simulation. The primary purpose of the sensitivity analysis is identification of decisions and cost elements which most affect the system operating costs. This identification is very helpful in determining which data values should be most accurately estimated.

THE REFINED SIMULATION COST MODEL

The refined simulation cost model is conceptually identical to that first developed for the Department of Transportation. Refinements developed for the Phase II Track Train Dynamics Program have, however, led to significant improvements in the accuracy of component usage representation, and in the manner in which the computer implements the schematic diagram and its associated data. These refinements are described below.

QUALITY REPRESENTATION

In refining the model to meet the needs of the Phase II Track Train Dynamics Program, it became necessary to better describe the flow through each path. To meet this need, the current model characterizes each path by three variables rather than by one variable. These three variables are: flow rate (e.g., wheels/year); mean component age; and component quality. The component quality is defined as the proportion of the components flowing through the path which are defective. For example, if 10,000 wheels per year move along a given path and if 3,700 are defective by the AAR rules, then the quality in that path is 0.37.

Inclusion of component quality in the characterization of a path allows a generalization in the type of decision which can occur at the branch point at the end of the path. Such decisions can be represented by non-linear functions of time, of the number of units in the arriving path, of the component's age in the path, and/or of the quality in the path. From this set of possible decisions, two are deemed to be representative of actual events which occur in the utilization of freight cars. Specifically, the decision made at certain nodes may not be dependent on the quality of the arriving stream, while at other nodes the decision is dependent on that quality.

For those nodes at which the decision to be made is not dependent on the quality of the components in the arriving path, a known proportion of the components departs on each of the outgoing paths. The value of this proportion can be obtained either directly or indirectly from available data and in turn presets the decision to be made at the node. In general, this proportion can be a known function of time.

In other cases, such as the inspection of wheels arriving at a receiving yard, the decision made is dependent on the quality of the arriving stream. Specifically, a proportion of the arriving defective wheels are correctly classified as defective and a proportion of the arriving good wheels are incorrectly identified as defective. In addition, a certain proportion of defective wheels are not detected at the receiving yard and continue in use. In order to determine the flow of wheels which continue in use, both defective and non-defective, the model determines from the quantity and quality of incoming wheels at the inspection point that proportion of wheels which are deemed defective. This enables data to be collected at one facility. The proportion of the components which branches toward the other departing path and the quality on that path are obtained from conservation of flow requirements at the branch points. This ability to evaluate the nodal decision on the basis of component quality significantly reduces data collection demands and further provides a technique for evaluating alternative inspection techniques.

RATE OF CHANGE OF QUALITY

The use of quality in the characterization of the population and of the flow in each path requires that the rate of change of population quality with time be treated. In order to evaluate the rate of change of population quality, the cause of this change must be known. The rate of change of population quality is, in part, governed by the scrapping of defective units and the addition to the population of new units. The rate of change of population quality is also dependent on the rate at which good units become defective as time progresses. In the model, this generation of defective from good units is represented by application of the Weibull quality distribution. In the model this technique allows the rate of change of quality of the population at any time to be determined from the age of the population, the Weibull slope, and the characteristic age at which a known percentage of the population becomes defective.

PATH COST REPRESENTATION

The refined simulation cost model allows a general representation of the cost of each path. Depending on the specific study being undertaken, the cost of a path can be represented as a non-linear function of time, of the number of units in the path, of the age in the path, and/or of the quality of the path. Typically, this allowable generality is not used because of the actual nature of the path cost in a given application or because of the availability of data. In the wheel example described in this paper, the path costs have been generally taken to be linearly dependent on the number of wheels in the path. That is, the costs are on a per wheel basis.

SENSITIVITY ANALYSIS

The current model produces a simulation containing the reference case (present time) and predicted future component usage and costs. In addition, the model can also provide a sensitivity analysis for the reference case. To do this, the reference case is automatically run repeatedly and for each run the values of all nodal decision proportions or path costs are varied slightly. This sensitivity analysis reflects the change in total operating costs which would be associated with a 1% increase in:

(1) The number of units branching to a path intended for the defective units, or

(2) The number of defective units correctly identified as defective, or

(3) The number of good units incorrectly identified as defective, or

(4) The path cost.

SAMPLE APPLICATION: THE FREIGHT CAR WHEEL

Under the Phase II Track Train Dynamics Program the refined model is being applied to evaluate costs associated with freight car wheels, roller bearings, side frames, and bolsters. Because wheel costs represent a major portion of freight equipment costs, this area was selected for preliminary analysis. The initial step, which has been completed, involves production of approximate values for decision and cost parameters in the running of the reference case and sensitivity analysis. Estimates were based on data provided by a number of sources including the AAR Mechanical Division and individual railroads. Sensitivity results obtained have been used to define the data collection effort necessary to further refine the data for the model so that the model may be used in its simulation mode to predict the effects of potential component changes. Upon completion of the data collection effort, the model will be used to evaluate cost versus benefit changes which result from changes in wheel system usage and in wheel specifications.

APPROXIMATE DATA FOR FREIGHT CAR WHEELS

Data used to determine approximate values for path costs in the wheel model were collected from a variety of sources. The AAR Mechanical Division was exceedingly helpful in providing approximate data. Table 1 presents the cost for each path involved in the wheel model. In addition, Table 1 provides a description of events which occur on the path and the rationale behind the wheel cost estimates given.

Approximate values for decision probabilities were primarily obtained through analysis of the AAR Car Repair Billing System. Tables 2A and 2B give the data used, the manner in which the data were applied to the schematic diagram, and the assumptions made. Table 2A presents path data. Table 2B presents decision data as well as the probability values for each branch point.

Data used in this preliminary study of wheel costs are intended to apply to the entire railroad industry on the basis of the car repair billing costs. Consequently, the results given in the wheel example for the reference case and for the sensitivity analysis represent approximate wheel usage and costs for a composite of the entire freight system.

REFERENCE CASE

The reference case, or present time result, based on the data exhibited in Tables 1 and 2, is shown in Tables 3 and 4. Table 3 presents the wheel quantity and quality flowing through each path. Table 4, in turn, presents the costs associated with each path and the total cost. This total cost of some \$396,000,000 is an estimate of the annual total cost required to maintain and acquire wheels for the entire United States railroad industry.

It can be noted from Table 3 that the flows in several paths are simply those given by the input data in Table 2A. This is necessarily true for the reference case. In use of the model to predict future usage, these flows will change as the population size, age, and quality change. It can also be noted from Table 4 that, of the total operating \$396,000,000, approximately \$286,000,000 is associated with the purchase of new wheels from the manufacturers.

SENSITIVITY ANALYSIS

The results for the sensitivity analysis for wheels are given in Table 5. The decision sensitivities, shown at the top of the table, contain either one or two per branch point. In those cases where two numbers are given, the decision is one which is dependent on the quality of the arriving system. The first number reflects the proportion of the arriving defective units correctly classified as defective and the second number reflects the number of arriving good units which are incorrectly identified as defective. In cases where only one number is given, either one of the above parameters is one or zero, or the branch is one at which the decision is not dependent on the quality in the path.

As an example of the use of decision sensitivity results, suppose it is desired to determine the change in present system operating costs produced by an increase in receiving yard inspection effectiveness. In particular, suppose the proportion of good wheels erroneously bad ordered is decreased by 15%. As indicated under node No. 3 in the table, a 1% change in the decision probability at this node results in a cost change of \$307,495.32. If this in turn is multiplied by -15 to reflect the percentage decrease, a savings of \$4,612,429.80 would result.

The cost sensitivities at the bottom of Table 5 can be interpreted similarly. For example, if the per wheel cost associated with path 14 is increased by 12%, the change in system operating cost is \$67,012.80% x 12% = \$804,153.60.

The sensitivity analyses conducted to date suggest that the most important usage factors affecting wheel costs are at the yard. Specifically, the frequency of car motion to the classification yard combined with the effectiveness of the yard inspection significantly affect the present system costs. The most important cost factor is the acquisition cost of the wheel.

TABLE 1 - WHEEL COST INPUT DATA

PATH	DESCRIPTION	PER WHEEL VALUE (DOLLARS)	SOURCES AND REMARKS
1	Wheel defect detector operation.*	$ 1×10^5	10% per year of total investment (100 wheel check @ $5,000, 10 @ $10,000, and 4 @ $100,000)
2	Installed new wheel	228.65	Job Code 3085 ($13.45 per 1/16 inch of tread), 17/16 inch new wheel tread (average)
3	New wheel and installation	228.65	See 2
7	Scrap Value	- 14.04	Job Code 3085
11	Cost of wheel caused derailment accidents	120,300.00	FRA "Journal Failure Report - 10/72", cost escalated by 3.3 to cost of private property
13	Receiving yard inspection	0.02	Inspection time @ $17.27 per hour for 20 seconds per 4 wheels
14	Delay associated with sending car to the rip track	10.00	Two day delay @ $5.00 per diem
15	Departure yard inspection	0.02	See 13
16	Cost associated with moving car from field to rip track	1,781.33	20 mile per hour speed decrease for 50 miles @ $688.53 per train hour. Also $10 for 2 days per diem and 10 car days lost time during slowdown for remainder of cars
18	Delay associated with sending car to the rip track and breaking the train	1,412.97	3 men, 30 min. each @ $17.27 plus 2 day delay @ $4.00 per diem plus 2 hours train delay @ $688.53/train hour
20	Wheel inspection at rip track	1.44	5 minutes @ $17.27 per hour
21	Remove wheel set, ship to wheel shop, out of service cost, wheel shop inspection cost	15.13	Removal: 1½ hours @ $17.27, shipping $2.50 (1000 pounds, 300 miles, $0.0162/ton/mile), wheelset delay: one week of wheelset life or ($228 × 2 + $200)/520 or $1.26, inspection: one minute each wheel @ $17.27, divide by 2 to get per wheel cost
25	Wheel shop inspection	0.29	One minute @ $17.27 per hour
29	Wheel shop inspection	0.29	See 25
33	Inspection for non-correctable defects	0	All inspections performed in paths 21. 25, and 29
35	Single/multiwear inspection	0	All inspections performed in paths 21, 25 and 29
37	Final pre-turn inspection	0	All inspections performed in paths 21, 25 and 29
41	Wheel demounting and inspection	1.44	5 minutes @ $17.27 per hour
43	Remounting of salvageable wheels	1.44	5 minutes @ $17.27 per hour
44	Wheel turning and inspection	17.42	Two hours @ $17.27 per hour plus one minute @ $17.27 per hour, divided by 2 to get per wheel cost
45	Demount wheels	1.44	5 minutes @ $17.27 per hour
49	Shipping of wheel-axle combination to rip track and installation	14.20	Shipping: $2.50 (1000 pounds, 300 miles, $0.0162/ton/mile), installation: $25.91 (1½ hours @ $17.27 per hour), divided by 2 to get per wheel cost
50	Shipping of wheel-axle combination to field and installation	2,754.12	½ day (4 hours) train delay @ $688.53 per hour
52	Movement from rip track to departure yard	0	Movement part of normal classification yard operation

* Total system cost, not a per wheel cost.

TABLE 2a - DECISION INPUT DATA

PATH	DESCRIPTION	WHEELS IN PATH*	QUALITY (PROPORTION DEFECTIVE)	SOURCES AND REMARKS
1	In use	13,800,000	0.01	Population Size: "Yearbook of Railroad Facts" (3) Used for freight car population, times 8 for wheel population. Population quality: 0.01 typical from AAR personnel
2	Expansion of wheel population	337,675	0	Quantity: Difference between wheels sold in 1975 as per AAR and quantity in path 7. Quality: New wheels assumed to have no defects
11	Wheel caused derailments	475	1	Quantity: 1975 FRA Derailment Data See (4). Quality: All failed wheels are defective
21	Sent to wheel shop for turning or scrapping	670,128	0.7785	Quantity: AAR CRB data: All removals in 1975 less Why Made Code 11 (Associated Repairs), See (5) for definitions of AAR Why Made Codes and Rules. Quality: Proportion of defective wheels given by (Quality in Path 21 less Why Made Code 90 (Mate Wheel)/(Quantity in Path 21)
25	Sent to wheel shop for bearing and axle maintenance	206,856	0.01	Quantity: AAR CRB data: Why Made Code 11 (Associated Repairs), assumed ½ of these not axle related so that wheelsets stay at rip track. Quality: Representative of in use population
29	Roller bearing involved in non-bearing caused derailments	196,164	0.01	Quantity: AAR CRB data, Rule 36, Why Made Code 33. Quality: Representative of in use population
34	Discarded because of obvious serious defect	684,132	0.55	Quantity: 3/4 of scrapped wheels (path 42) taken as discarded upon first inspection (node 18). Quality: Mate wheels, sometimes defective, also scrapped. 10% of mate wheels taken as defective.

PATH	DESCRIPTION	WHEELS IN PATH*	QUALITY (PROPORTION DEFECTIVE)	SOURCES AND REMARKS
39	Discarded because wheel not turnable	228,044	0.55	Quantity: ¼ of scrapped wheels (path 42) taken as discarded upon second inspection (node 21). Quality: Mate wheels, sometimes defective, also scrapped. 10% of mate wheels taken as defective.
42	Scrapped prior to turning	912,176	1	Quantity: 85% of wheels received at wheel shop are scrapped prior to turning (observations). Quality: All scrapped wheels are defective.
44	Wheels turned	160,972	9.2563×10^{-4} (after turning)	Quantity: 15% of wheels received at wheel shop are turned (observation). Quality: AAR CRB data, Why Made Code 88 (Subsurface defects uncovered during turning).

*AAR CRB data for Rule 41 unless otherwise noted. Quantities escalated by 2.4 to railroad industry rates.

TABLE 2b DECISION INPUT DATA AND DECISION PARAMETER VALUES

NODE	DECISION	C VALUE	D VALUE	E VALUE	SOURCES AND REMARKS
1	Normal movement to yard	52	-	-	Assumed car enters yard once per week
3	Bad ordered in receiving yard	-	7.2700×10^{-2}	2.0884×10^{-4}	Calculated from quantities and qualities in paths 13 and 14
4	Bad ordered in departure yard	0	-	-	CRB data for wheel maintenace not broken down by receiving versus departure yard bad order. Also, wheels typically inspected in detail only at receiving yard.
5	Direct movement from field to rip track	0	-	-	Data representative of national practice not available. Also, CRB Why Made Code items normally identified visually during receiving yard inspection.
8	Sent to wheel shop for turning or scrapping	1	-	-	Data representative of national practice not available. Also, normal practice is removal of wheelsets in rip track upon bad ordering in yard.
9	Wheel shop action required	1	-	-	Standard practice is turning or demounting of all wheels received at wheel shop
10	Wheel caused derailments	-	3.4420×10^{-3}	0	Calculated from quantities and qualities in paths 1 and 11
11	Sent to wheel shop for bearing and axle maintenance	1.4990×10^{-2}	-	-	Calculated from quantities and qualities in paths 1 and 25
12	Wheel shop action required	1	-	-	See 9
14	Roller bearing involved in non-bearing caused derailments	1.4215×10^{-2}	-	-	Calculated from quantities and qualities in paths 1 and 29
15	Wheel shop action required	1	-	-	See 9
18	Discarded because of obvious serious defect.	-	7.1572×10^{-2}	5.6238×10^{-1}	Calculated from quantities and qualities in paths 33 and 34.

NODE	DECISION	C VALUE	D VALUE	E VALUE	SOURCES AND REMARKS
19	Discarded because of insufficient tread for turning	0	-	-	Decisions at nodes 19 and 21 physically occur simultaneously
21	Discarded because wheel not turnable	-	8.3923×10^{-1}	4.2836×10^{-1}	Calculated from quantities and qualities in paths 37 and 39
23	Scrapped prior to turning	1	-	-	Standard practice is to discard all mate wheels
25	Discarded because of subsurface defects uncovered during turning	-	1	0	Typically all defective wheels are found and no defective wheels are discarded here
29	Returned directly to rip track	1	-	-	Standard practice is to return wheel sets to rip track for installation in trucks

TABLE 3.

BASE CASE RESULTS - QUANTITIES AND QUALITIES

Path No.			
1;	NUM =	13800000,	QUAL = 0.0100
2;	NUM/YR =	337675,	QUAL = 0.0000
3;	NUM/YR =	912324,	QUAL = 0.0000
4;	NUM/YR =	0,	QUAL = 0.0000
5;	NUM/YR =	0,	QUAL = 0.0000
6;	NUM/YR =	0,	QUAL = 0.0000
7;	NUM/YR =	912324,	QUAL = 0.5500
8;	NUM/YR =	0,	QUAL = 0.0000
9;	NUM/YR =	0,	QUAL = 0.0000
10;	NUM/YR =	0,	QUAL = 0.0000
11;	NUM/YR =	474,	QUAL = 1.0000
12;	NUM/YR =	717600000,	QUAL = 0.0100
13;	NUM/YR =	717937675,	QUAL = 0.0099
14;	NUM/YR =	670127,	QUAL = 0.7785
15;	NUM/YR =	717267547,	QUAL = 0.0092
16;	NUM/YR =	0,	QUAL = 0.0100
17;	NUM/YR =	670127,	QUAL = 0.7785
18;	NUM/YR =	0,	QUAL = 0.0092
19;	NUM/YR =	717267547,	QUAL = 0.0092
20;	NUM/YR =	670127,	QUAL = 0.7785
21;	NUM/YR =	670127,	QUAL = 0.7785
22;	NUM/YR =	0,	QUAL = 0.0000
23;	NUM/YR =	670127,	QUAL = 0.7785
24;	NUM/YR =	0,	QUAL = 0.0000
25;	NUM/YR =	206855,	QUAL = 0.0100
26;	NUM/YR =	206855,	QUAL = 0.0100
27;	NUM/YR =	0,	QUAL = 0.0000
28;	NUM/YR =	876983,	QUAL = 0.5972
29;	NUM/YR =	196163,	QUAL = 0.0100
30;	NUM/YR =	196163,	QUAL = 0.0100
31;	NUM/YR =	0,	QUAL = 0.0000
32;	NUM/YR =	0,	QUAL = 0.0000
33;	NUM/YR =	1073147,	QUAL = 0.4898
34;	NUM/YR =	684131,	QUAL = 0.5500
35;	NUM/YR =	389015,	QUAL = 0.3841
36;	NUM/YR =	0,	QUAL = 0.3841
37;	NUM/YR =	389015,	QUAL = 0.3841
38;	NUM/YR =	684131,	QUAL = 0.5500
39;	NUM/YR =	228044,	QUAL = 0.5500
40;	NUM/YR =	160971,	QUAL = 0.0009
41;	NUM/YR =	912175,	QUAL = 0.5500
42;	NUM/YR =	912175,	QUAL = 0.5500
43;	NUM/YR =	0,	QUAL = 0.0009
44;	NUM/YR =	160971,	QUAL = 0.0009
45;	NUM/YR =	148,	QUAL = 1.0000
46;	NUM/YR =	160822,	QUAL = 0.0000
47;	NUM/YR =	1073147,	QUAL = 0.0000
48;	NUM/YR =	1073147,	QUAL = 0.0000
49;	NUM/YR =	1073147,	QUAL = 0.0000
50;	NUM/YR =	0,	QUAL = 0.0000
51;	NUM/YR =	0,	QUAL = 0.0000
52;	NUM/YR =	1073147,	QUAL = 0.0000
53;	NUM/YR =	718340694,	QUAL = 0.0092

TABLE 4.

BASE CASE RESULTS - COSTS

Path No.		
1;	PATH COST =	$100000.00
2;	PATH COST =	$77209388.75
3;	PATH COST =	$208603110.93
4;	PATH COST =	$0.00
5;	PATH COST =	$0.00
6;	PATH COST =	$0.00
7;	PATH COST =	$-12809042.98
8;	PATH COST =	$0.00
9;	PATH COST =	$0.00
10;	PATH COST =	$0.00
11;	PATH COST =	$57142499.90
12;	PATH COST =	$0.00
13;	PATH COST =	$14358753.50
14;	PATH COST =	$6701279.99
15;	PATH COST =	$14345350.94
16;	PATH COST =	$0.00
17;	PATH COST =	$0.00
18;	PATH COST =	$0.00
19;	PATH COST =	$0.00
20;	PATH COST =	$964984.31
21;	PATH COST =	$10139036.63
22;	PATH COST =	$0.00
23;	PATH COST =	$0.00
24;	PATH COST =	$0.00
25;	PATH COST =	$59988.23
26;	PATH COST =	$0.00
27;	PATH COST =	$0.00
28;	PATH COST =	$0.00
29;	PATH COST =	$56887.55
30;	PATH COST =	$0.00
31;	PATH COST =	$0.00
32;	PATH COST =	$0.00
33;	PATH COST =	$0.00
34;	PATH COST =	$0.00
35;	PATH COST =	$0.00
36;	PATH COST =	$0.00
37;	PATH COST =	$0.00
38;	PATH COST =	$0.00
39;	PATH COST =	$0.00
40;	PATH COST =	$0.00
41;	PATH COST =	$1313533.43
42;	PATH COST =	$0.00
43;	PATH COST =	$0.00
44;	PATH COST =	$2804132.20
45;	PATH COST =	$214.55
46;	PATH COST =	$0.00
47;	PATH COST =	$0.00
48;	PATH COST =	$0.00
49;	PATH COST =	$15238701.55
50;	PATH COST =	$0.00
51;	PATH COST =	$0.00
52;	PATH COST =	$0.00
53;	PATH COST =	$0.00
	TOTAL COST =	$396228819.55

TABLE 5.

SENSITIVITY RESULTS - DECISIONS

Node No.		
1,	SENSITIVITY =	$1886787.74/%
3,	SENSITIVITY =	$1292398.50/%
3,	SENSITIVITY =	$307495.32/%
4,	SENSITIVITY =	$0.00/%
5,	SENSITIVITY =	$0.00/%
8,	SENSITIVITY =	$0.00/%
9,	SENSITIVITY =	$0.00/%
10,	SENSITIVITY =	$571424.99/%
11,	SENSITIVITY =	$375042.83/%
12,	SENSITIVITY =	$0.00/%
14,	SENSITIVITY =	$355657.57/%
15,	SENSITIVITY =	$0.00/%
18,	SENSITIVITY =	$120039.99/%
18,	SENSITIVITY =	$349205.87/%
19,	SENSITIVITY =	$0.00/%
21,	SENSITIVITY =	$248879.26/%
21,	SENSITIVITY =	$203628.48/%
23,	SENSITIVITY =	$0.00/%
25,	SENSITIVITY =	$0.00/%
29,	SENSITIVITY =	$0.00/%

SENSITIVITY RESULTS - COSTS

Path No.		
1,	SENSITIVITY =	$1000.00/%
2,	SENSITIVITY =	$772093.88/%
3,	SENSITIVITY =	$2086031.10/%
7,	SENSITIVITY =	$128090.42/%
11,	SENSITITITY =	$571424.99/%
13,	SENSITIVITY =	$143587.53/%
14,	SENSITIVITY =	$67012.80/%
15,	SENSITIVITY =	$143453.50/%
16,	SENSITIVITY =	$0.00/%
18,	SENSITIVITY =	$0.00/%
20,	SENSITIVITY =	$9649.84/%
21,	SENSITIVITY =	$101390.36/%
25,	SENSITIVITY =	$599.88/%
29,	SENSITIVITY =	$568.87/%
33,	SENSITIVITY =	$0.00/%
35,	SENSITIVITY =	$0.00/%
37,	SENSITIVITY =	$0.00/%
41,	SENSITIVITY =	$13135.33/%
43,	SENSITIVITY =	$0.00/%
44,	SENSITIVITY =	$28041.32/%
45,	SENSITIVITY =	$2.14/%
49,	SENSITIVITY =	$152387.01/%
50,	SENSITIVITY =	$0.00/%
52,	SENSITIVITY =	$0.00/%

REFERENCES

1. "Improvement of Railroad Roller Bearing Certification Test Procedures and Development of Roller Bearing Diagnostic Techniques", Volume I, Phase I, Draft Final Report by Shaker Research Corp., April 4, 1975, Contract DOT/TSC-917.

2. Krauter, A. I., and Waldron, W. D., "Application of Simulation Cost Modeling to Railroad Roller Bearing Usage", presented at the Joint ASME-IEEE Railroad Conference, Washington, D. C., March 30-April 1, 1977. ASME Paper Number 77-RT-7.

3. "Yearbook of Railroad Facts - 1975 Edition" Economics & Finance Department, Association of American Railroads, 1920 L St., NW, Washington, D.C., 20036.

4. "Wayside Derailment Requirements Study for Railroad Vehicle Equipment" Final Report for DOT/TSC-1029, Shaker Research Corporation, Revised December 1, 1976.

5. "Field Manual of the AAR Interchange Rules - 1976 Edition" Mechanical Division, Association of American Railroads, 1920 L St., NW, Washington, D.C. 20036.

QUESTIONS SESSION IV

Session Chairman -- George H. Way

Attendee: R. W. Radford, Canadian National Railways

Attendee's Question: Current practices in freight car operation require the use of center plate lubricants to reduce truck rotational torque values. To eliminate hunting use is then made of constant contact side bearings to increase track rotation torque values. Did the test indicate that the effective hunting control could be obtained with dry center plates?

D. Sutliff: To answer the question, the variations in test configurations, (nominal center plate lubricant, and heavily lubricated) and the various side bearing loads were intended to obtain different values of damping between the truck and the car bolster. I think it's safe to say that those data did tend to indicate that as the damping went up there was a beneficial effect on the truck hunting. However, one must also consider curve negotiation. I would be very reluctant, from our test data, to recommend that any such thing as a dry center plate unless considerable attention was paid to the wear characteristics of the components involved.

Attendee: F. Dean, Battelle Columbus Laboratories.

Attendee's Question: When you apply 1/6th track gage and 1/2 gage criteria, are these based on an effective gage 59.5 to 60 inches or the nominal gage 56.5 inches? And does the 1/6th gage criteria have an official status with the AAR?

E. Chang: In answering the question we used the nominal track gage.

G. Way: The second part of the question is, does the 1/6th gage criteria have an official status with the AAR?

E. Chang: No, it does not.

Attendee: R. Jeffcoat, Analytic Sciences Corporation.

Attendee's Question: How did you handle wheel/rail forces in your non-linear model, especially at flanging?

V. K. Garg: In the non-linear model, we have taken non-linear friction-creep forces which are decomposed into three components, and detail has been given in the paper. In the contact point search scheme, flange contact is automatically detected giving a relatively large lateral rail displacement and contact slope. This results in a large lateral reaction which is a function of rail stiffness.

G. Way: I think we have the same question asked here by two different people both addressed to E. Chang.

Attendee: R. Jeffcoat, Analytic Sciences Corporation.

Attendee's Question: You found that above certain spiral length, your figures of merit L/V and coupler angle actually get worse. That seems surprising. Could you explain physically?

Attendee: F. King, Canadian National Rail.

Attendee's Question: If I understood one of your graphs correctly, you showed an optimum spiral length which generated a minimum lateral force. The graph appeared to me to show that longer spirals actually generated larger lateral forces. I would have thought that there was a minimum spiral length beyond which the lateral forces would not increase. To put the question in another way, is it possible to have a spiral which is too long?

E. Chang: Well, in this study indication has been found that beyond certain spiral lengths the coupler angle attained during curve negotiation did increase slightly.

G. Way: In the simulation model it appears that coupler angle does increase slightly if the spiral is beyond an optimum length, so it is possible to have too long a spiral although the consequences are not disastrous. Is that proper paraphrasing?

E. Chang: Yes.

Attendee: R. E. Johnson, Burlington Northern Inc.
Attendee's Question: Your slide indicates that the NDT band for Grade C is the same as Grade E steel when quench and tempered. Are there any other advantages other than tensil strength for using Grade E steels for couplers, yokes, and knucklers? Any disadvantages? Is quality control to quench and temper E steel more difficult than C steel?

D. Stone: I think that you're correct in assuming that the only advantage that Grade E steel has over the quench and tempered Grade C is the increased yield strength. As a matter of fact, one can use the same steel for both Grade C quench and tempered and Grade E simply by changing the tempering temperature by 100 or so degrees to get the lower stress with a higher tempering temperature for the Grade C. As far as quality control problems go, the general thing I think that you want to avoid is developing a mixed micro-structure. In other words, in the quench and tempered steels you want them to be martensitic and banitic with no pearlite in them. And, that presents a quality control problem. On the other hand, you have the exact situation in reverse with the normalized and tempered steels which are ferrite/pearlite mixtures. There if you get transformation products such as martensite or banite, you're going to decrease the properties of the normalized and tempered steel. So, in both cases, I think the same degree of quality control is required. And, this might be carried a step further in that it could be a risky practice to try to normalize a quench and tempered steel or quench and temper a normalized steel because when you go through that procedure you do have the possibility of getting a mixed microstructure in either of the steels, the quench and tempered or the normalized and tempered.

Attendee: C. E. Hartzell, General Electric Co.
Attendee's Question: Have time duration limits yet been established for critical L/V ratios for rail rollover, rail climb, etc.?

D. Sutliff: It has certainly not been established by us. We've used criteria which have been used for many years and indeed some of the information we've obtained from past tests tend to support the conclusion that those criteria may be unrealistic, perhaps grossly so.

Attendee: E. H. Law, Clemson University.
Attendee's Question: Our analysis of lateral stability using linear models have indicated a strong dependence of stability on the values of creep coefficients used. Could you comment on how you selected the values you used to calculate your results for this linear model?

V. K. Garg: So far, we have not made any parametric study with the linear hunting model. Therefore, we cannot comment on the influence of creep coefficients on vehicle stability. In our analysis, we selected the creep data reported by Professor Law in his previous work.

Attendee: E. H. Law, Clemson University.
Attendee's Question: In the test of configuration the B truck had brand new AAR wheels with a mill scale present while the A truck had the same wheel profiles but with the mill scale worn off. Results indicate large flange to flange motions of the A truck at about 60 mph while large motions of both the A and B truck did not start until 80 to 88 mph. A possible explanation for this behavior is the different wheel surface conditions and hence creep characteristics of the A and B trucks. Was this factor considered in your analysis and if so, how?

V. K. Garg: We are using the same creep characteristics for all the wheels. I don't think we have taken the two different profiles on two trucks.

G. Way: I think the question is that the profile is the same, but one had. . . . .

V. K. Garg: No, we took the same creep.

Attendee: N. Cooperrider, Arizona State University.
Attendee's Question: In what sense do you consider your theoretical analysis validated? For example, we have found the vehicle behaving quite sensitive to factors such as creep coefficient, (this gets back to the previous question) and amplitudes of motion that cross non-linear elements. How did you choose these factors in your analysis? Have you looked at variations of these factors?

V. K. Garg: As I mentioned in the presentation, the validation has not been completed. So far, we have analyzed only one test configuration. We are looking into other test configurations which consist of the AAR and CN wheel profiles. In the non-linear model, the friction-creep curve has been selected by compromising the data published by various investigators. Since at this time we have not conducted any parametric study, we are not in the position to describe the influence of the amplitudes of motion across non-linear elements.

Attendee: N. Cooperrider, Arizona State University.
Attendee's Question: Could you briefly comment on what the QLTS model does or refer me to a document describing it?

E. Chang: Basically the model simulates consists in curve negotiation and it calculates L/V ratio in quasi static sense and also calculates maximum coupler angle; the reference I will refer you to is technical documentation on quasi static lateral train stability model by Larry Thomas and Report No. is E209 and there is also a user's manual that comes with that. That model is developed in Phase I TDT and in this study we have slightly modified a model to calculate the critical speed namely the overturning speed and the acceptable operation speed.

G. Way: Are those modifications documented also?

E. Chang: No, I don't think so.

Attendee: N. Cooperrider, Arizona State University.
Attendee's Question: I am confused about the input and output forces to this model. Are the steady state curving forces wheel/rail forces input to this model? What basis is used for the output L/V ratios per wheel, per truck, per vehicle?

E. Chang: This is based on information supplied by Electro-Motive Division.

Attendee: N. Cooperrider, Arizona State University.
Attendee's Question: We have processed the random response field test data to compute frequencies and damping ratios by both the random decrement technique and by PSD methods. Our values for frequency and damping ratio obtained from the two methods agree quite closely. However, our values are much higher than the values you presented by Configuration 6. Could you comment on how you obtained your values? How many values were averaged, what data was ignored, etc.?

V. K. Garg: We did not use the random decrement technique or the power spectral density method to compute frequency and damping ratio. In our analysis, we used forcer test data and employed two consecutive peaks to compute damping ratios using lograthemic decrement technique. In the analysis, care has been taken to separate the data into segments each representing the response to a single forcer activation. Number of values available for averaging depend upon the length of run-time. On an average about 25 to 30 data points have been used.

Attendee: M. Kenworthy, ENSCO.
Attendee's Question: What is the motivation for fatigue life analysis? Does there exist a real problem with fatigue

failure in rail cars today? If so, what is the order of magnitude in terms of safety, equipment damage, loss of life, or dollars?

V. K. Garg: As I mentioned, the whole objective of this program is to demonstrate how one can use the structural dynamics techniques for better evaluation of the fatigue.

G. Way: I'd like to add something to that. Certainly there is a, not exactly a problem, but cars do suffer from fatigue and the railroads get upset over it, and I call up the car builder and ask what's happening. So, from that standpoint, there certainly could be considered a problem. This all ties into the acceptable life of the vehicle. As far as the cost and the dollars are concerned, I don't have those figures.

Attendee: B. Terlecky, Trailer Train Co.
Attendee's Question: Do you suggest extension of low temperature notch property requirements to other than coupler draft system components, yokes, front-rear stops, etc.?

D. Stone: It would depend on what the experience is with these specific components you mentioned. If they show a brittle fracture when they fail, then yes, I think it would be appropriate to introduce a material with a better notch toughness. And, I think yokes are definitely a case where this would be true.

Attendee: A. Gillespie, Transportation Safety Institute.
Attendee's Question: What are the dominant morphological forces in the Q and T steels and what is the percent closely held carbon?

D. Stone: The quench and tempered steels are primarily a banite/martensite mixture and the carbon range is between approximately .20 and .35 weight percent.

Attendee: A. Gillespie, Transportation Safety Institute.
Attendee's Question: What percent of coupler failures are attributed to temperature related factors in surface couplers?

D. Stone: Virtually all the failed couplers that we looked at showed some fatigue cracking from some sort of a defect whether it was a manufacturing defect or a defect that was introduced by service. This would grow to a critical size and you'd get a brittle fracture. So, in that sense, virtually all of them are temperature sensitive failures.

Attendee: A. Gillespie, Transportation Safety Institute.
Attendee's Question: What processing differences in manufacturing are there between normalized and tempered and quench and tempered steels and, therefore, what are the approximate cost differences?

D. Stone: Well, the difference between normalizing and quenching is essentially that in normalizing you heat the piece to a certain temperature and then allow it to air cooled and in the quenching you heat it to a given temperature and then remove it to normally water or some other quenching medium. So, as far as the manufacturing cost, I don't think there is a large difference. I don't manufacture couplers so don't hold me too tight to that. The real problem seems to be on reclamation, where virtually all of the railroad reclamation shops have normalizing and tempering facilities but not quench and tempering facilities. So, the largest cost if quench and tempered couplers are accepted as a standard will be more on the railroad side in their reclaiming shops where they'll have to add quenching facilities.

G. Way: Lastly we have a comment from H. Law and N. Cooperrider. They wish to express their opinion that the AAR and the Union Pacific made an excellent effort in the field test described. To their knowledge, this effort was one of the most extensive and comprehensive ever conducted for the purposes of validation of lateral dynamics models. The AAR and the UP are to be congratulated for these fine efforts.

SESSION V REPORT ON RAIL SUPPLY INDUSTRY'S R&D ACTIVITIES

IMPROVED RAILROAD ROLLER BEARINGS THROUGH RESEARCH

BY

Thomas C. Keller

Executive Summary

Research at the Timken Company is accomplished in several ways. Timken Research is a separate department of the engineering organization devoted to exploring and developing new ideas. The Physical Laboratories engage in testing full-size bearings and other material to evaluate performance. Service feedback and review is another important input to the accomplishment of research goals.

The Timken "AP" (All-Purpose) Tapered Roller Bearing, which is a standard bearing for use on freight cars as well as other types of rolling stock is the primary object of the research discussed. Material and process improvement can produce up to a seven times improvement in life, as evaluated by full-size fatigue test machines. However, adoption is subject to economic restraints. Higher viscosity oil in the grease used in this type bearing has been adopted as a result of laboratory bearing fatigue evaluations, which showed a doubling of life using SAE 50 versus SAE 20 oil. The water content of the lubricant is another life variable which was explored, where 0.04% water dissolved in oil gave about half the bearing life of that having 0.01% water.

Bearing adjustment is very important to bearing performance and minimum end play or lateral is built into the bearing to give maximum life under running conditions. The operating environment of the bearing is vital to bearing operation thus making the seal a very important element, the efficiency of which is established and maintained by extensive laboratory testing. Other noteworthy components are the vent and locking plate. The vent is important because a 5 psi internal pressure can reduce seal life to about one-half that under zero pressure. The new locking plate has the feature of positive cap screw locking with minimum risk of human error.

REALISM IN RESEARCH FOR RAILROADS

BY

R. H. Beetle

Executive Summary

Railroad freight operations differ from other modes of transport. Research developments must be compatible with trains having coupled cars of differing characteristics operating within the fixed parameters of existing track structures. There is an overabundance of ideas for research and development on track and/or car components. Screening these ideas through railroad constraints is realistic even though viewed as unimaginative by some.

The concerns of early railroading still challenge today's research to find pioneering breakthroughs. Realism in today's research requires handling of many new ideas under changing conditions regarding environmental aspects, availability of materials, energy conservation, and economic factors.

Research and development in trackwork, wheels, and friction braking requires realism in specifications and product performance with proper laboratory and field testing. In addition, the requirement for complete system analysis is vital to railroad progress.

Joining of the rail lines into a transcontinental system back in 1869, was an achievement brought about by cooperation of the pioneer railroads and the Government. Today's comparable achievement is the joining of the resources of not only the United States Government, but the Canadian Government, the AAR, RPI, and the many others, into the TTD and FAST programs. Use of TTD and expansion of the FAST trackage to permit a wider range of tests on longer tangent track, certainly are forward and logical steps to benefit the railroad industry as soon as possible.

INSTRUMENTED WHEEL SET SYSTEMS
FOR PRODUCT PERFORMANCE ANALYSIS

BY

H. G. Tennikait

Executive Summary

There has been, in recent years, an increased and dedicated effort by industry, government and academic groups to more fully understand the dynamics of a railroad freight car. Evaluating the forces and motions imposed on the freight car during "real world" operating conditions has required the application of procedures, techniques, and hardware that were, in the not too distant past, foreign to our industry. Not only has off-the-shelf hardware been brought into wider use, special equipment has been designed, constructed, and employed to develop specific operating parameters.

Of prime importance in any truck study are the forces at the wheel/rail interface. The problem of converting a rotating wheel and axle set to a force measuring transducer is difficult. Even more difficult is the transmission of the relatively low level signals to the recording equipment.

Several types of systems are now being used, however, an approach presented to the industry by Messrs. L. A. Peterson, W. H. Freeman, and J. M. Wondrisco in ASME Paper 71 WA/ RT4, seemed the most practical for ASF's use. Their approach entailed the application of the combined measured bending moments in the axle, and the measured vertical journal forces to the equations developed from free body diagrams of the wheel/axle set. ASF used the same basic force diagram and the resulting equations. However, the methods of sensing the vertical journal forces and the bending moments in the axle are different.

THE DR-1 RADIAL TRUCK
A SIGNIFICANT ADVANCE IN FREIGHT CAR TECHNOLOGY

BY

E. C. Bailey
N. Caldwell
P. P. Marcotte

Executive Summary

During the past two years, significant progress has been made in the development of adapting radial steering arms to a conventional three-piece truck to improve curve tracking characteristics and provide increased stability at maximum operating speeds. The result is a new truck design which has been designated the DR-1 Radial Truck. Extensive tests have been conducted on the DR-1 which provided conclusive proof that the steering arms eliminate hunting within the operating speed range and greatly improve curve negotiation.

On the economic side, actual cost information has been compiled on some of the operational parameters affected by the improved tracking performance of the truck, such as wheel and rail wear and fuel consumption. This data has provided valuable insight into the significant savings that can be realized from the application of the self-steering principle.

The work done to date has demonstrated the DR-1 to be technically and economically feasible. The next step in the program calls for selected revenue service applications of the production version.

The purpose of this paper is to review the progress made on this project in the past two years, explore some of the revealing data on the economics of improved truck performance and outline the plans for the introduction of the DR-1 into field service.

STACKED CONTAINER CAR FOR LAND BRIDGE

BY

L. H. Nations
R. H. Billingsley, Jr.

Executive Summary

The strong need for improved efficiency in rail transshipment of freight containers from container ships prompted the Southern Pacific to undertake the development of a container car specialized for "Land Bridge" operations. The fundamental objective to reduce car lightweight was obtained by stacking containers as compared to usual end-to-end loading. An additional benefit was substantially reduced train lengths. The Southern Pacific is also developing concepts which would retain TOFC capability as well as improve COFC efficiency. ACF, under contract to Southern Pacific, has designed, built, and tested a prototype design version of the stacked container car concept. The prototype design demonstrated the weight reduction objective feasibility while at the same time conforming to all AAR/FRA design and safety requirements. The only restriction involves height, which is the same as multi-level auto rack cars.

The need for a lighter weight car specialized to handle containers has been building for some time. Before the advent of landbridge and minibridge all-container movements and the growth of a network of terminals with lift-off/lift-on handling equipment, the logistic versatility of the all- purpose piggyback car inhibited developing COFC-only cars. Now, however, there is definite justification for developing such cars to achieve maximum economy operating point-to-point within the network.

Reducing car weight for more economical transportation is by no means new or novel; it has been a popular theme for decades. In this case, however, it is not a matter of skinning down existing structure with the attendant complications and hazards of potential fatigue problems, rather it is the selection of an alternate structure. It is an opportunity to take a fresh outlook and to conceptualize taking advantage of many years of piggybacking development and experience behind us.

IMPROVED RAILROAD BEARINGS THROUGH RESEARCH

BY

Thomas C. Keller

Research at the Timken Company is accomplished in several ways. Timken Research is a separate department of the engineering organization devoted to exploring and developing new ideas. The Physical Laboratories engage in testing full-size bearings and other material to evaluate performance. Service feedback and review is another important input to the accomplishment of research goals.

The Timken "AP" (All-Purpose) Tapered Roller Bearing, which is a standard bearing for use on freight cars as well as other types of rolling stock is the primary object of the research discussed. Material and process improvement can produce up to a seven times improvement in life, as evaluated by full-size fatigue test machines. However, adoption is subject to economic restraints. Higher viscosity oil in the grease used in this type bearing has been adopted as a result of laboratory bearing fatigue evaluations, which showed a doubling of life using SAE 50 versus SAE 20 oil. The water content of the lubricant is another life variable which was explored, where 0.04% water dissolved in oil gave about half the bearing life of that having 0.01% water.

Bearing adjustment is very important to bearing performance and minimum end play or lateral is built into the bearing to give maximum life under running conditions. The operating environment of the bearing is vital to bearing operation thus making the seal a very important element, the efficiency of which is established and maintained by extensive laboratory testing. Other noteworthy components are the vent and locking plate. The vent is important because a 5 psi internal pressure can reduce seal life to about one-half that under zero pressure. The new locking plate has the feature of positive cap screw locking with minimum risk of human error.

INTRODUCTION

In 1954 when the Timken "AP" (All-Purpose) Roller Bearing was introduced to the railroad industry, the very best thinking of that time was incorporated into it. Since then, an almost continuous series of improvements and changes have been built into this bearing to better adapt it for the arduous service required for freight cars as well as for passenger cars, transit cars, locomotives, and industrial equipment. The present day tapered roller bearing shown in figure 1, still interchangeable with the original, does not appear to be much different from the original. But it is different in many ways, brought about by several different types of research.

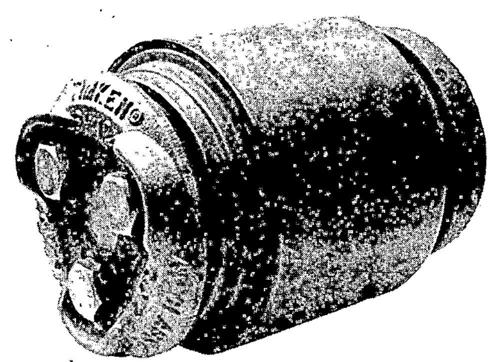


FIGURE 1. TIMKEN "AP" BEARING

Thomas C. Keller is Chief Engineer, Railroad Division of the Timken Company. He received his Bachelor of Science Degree in Mechanical Engineering from the University of Pittsburgh (1939) and also attended the Management Program for Executives at that University in 1967. Keller also is a Registered Professional Engineer in Ohio.

In-depth, advanced thinking and comprehensive work is performed at Timken Research where investigations explore fundamentals of tapered roller bearings relating to metallurgy, geometrical design, and the effect of environmental factors such as lubricants, contaminants, speed, temperature, and misalignment in application.

A second form of research comes from the Timken Physical Laboratories where major activities include bearing fatigue testing, seal testing, and lubricant evaluation. In addition, all sorts of day-to-day investigations are made to evaluate application ideas for improvement or to solve field-generated problems.

Actual service feedback on bearing performance in the field is another important form of research. It is the service endured under the actual equipment which provides the final evaluation of the roller bearing. Here the weaknesses and strengths are reflected. It is also here that the most can be learned about meaningful performance. With an effective feedback system from the field service engineers, provided by the service engineer's report, a very valuable input is added to the other forms of research.

For those who may not be completely familiar with the Association of American Railroads (AAR) standards, it is well to mention that the bearing envelope dimensions are standardized and the AAR requires certain laboratory and field testing for qualification. The Timken "AP" Roller Bearing, having AAR approval number 1A, is shown in an exploded assembly in figure 2 to familiarize the reader with the various parts of the assembly and their arrangement so the following detailed discussion may be understood better. The bearing shown is the latest No-Field-Lubrication (NFL) design which requires lubrication only when the wheelset is in the shop for wheel turning or wheel change. The bearing is completely self-contained, preadjusted and prelubricated with grease.

As further background, it should be mentioned that the "AP" bearing is used in a wide range of sizes, shown in figure 3, having bores ranging in size from 100 mm (4") to 178 mm (7"). The size designations "B" through "G" are applicable to freight cars, although the smaller sizes are not used much in the U.S.A. anymore. The "EE" size is employed mostly on passenger cars, while the large "GG" is used for locomotives.

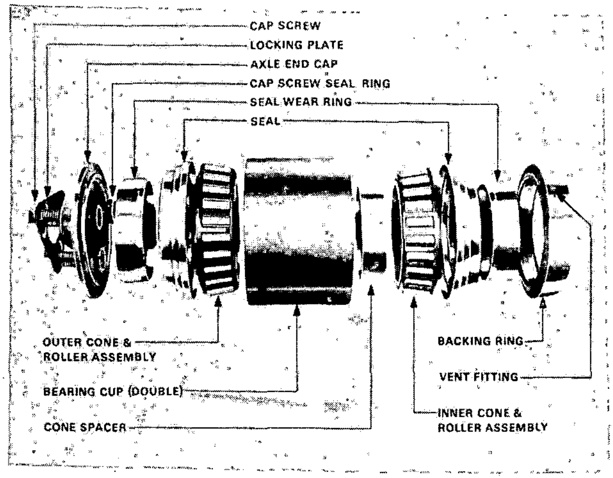


FIGURE 2. TIMKEN "AP" BEARING NOMENCLATURE

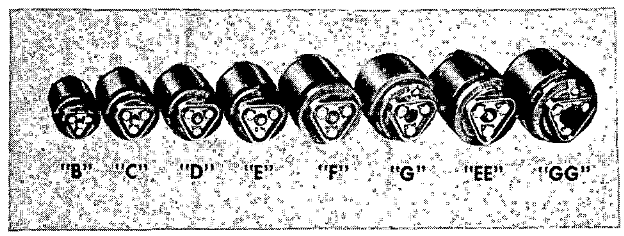


FIGURE 3. RANGE OF TIMKEN "AP" BEARING SIZES

Some of the more significant items of interest concerning research and development are discussed below.

MATERIALS & PROCESSES

Many variations in materials and processes have been evaluated by fatigue testing full-size bearings, generally the Class F (6½ x 12) size normally used on 100-ton freight cars. Table 1 shows the range of life obtained using different metallurgical and process variables. Details are not shown because this information is considered proprietary. The important thing to recognize is that these improvements are possible in the development of bearing technology. Some of them may be applicable today, while others face economic barriers which will have to be overcome for their adoption.

"AP" BEARING FATIGUE RESEARCH

MATERIAL AND PROCESS TESTED	LIFE INDEX
A	1.0
B	1.7
C	3.3
D	7.0
E	1.0

TABLE 1.

Such evaluations are difficult to make in the field so they are performed in the laboratory on fatigue test machines (figure 4). Each machine has two shafts and four bearings per shaft are tested. To accelerate the fatigue process, the bearings are run at over two times normal freight car loading and at 1200 rpm or roughly 190 kilometers per hour (120 mph) equivalent. Even so, 32 bearings must be run two to six weeks to evaluate each independent variable. This is expensive and time-consuming testing.

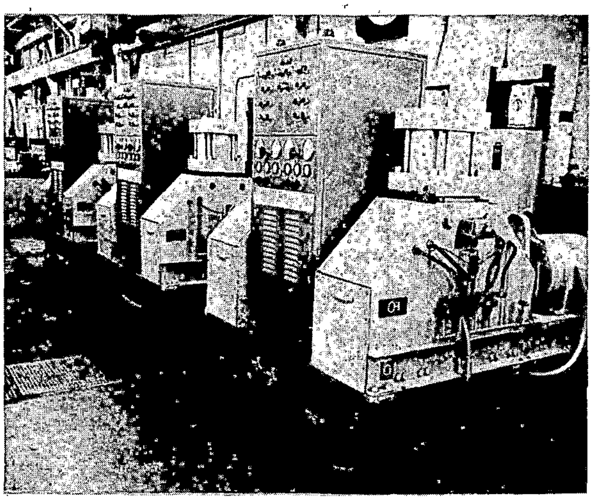


FIGURE 4. TIMKEN BEARING FATIGUE TEST MACHINES

LUBRICANT VISCOSITY

On the same fatigue test machines discussed above, extensive investigations were made to evaluate the effect of oil viscosity on full size "AP" bearings. The decision to undertake this testing followed a comprehensive project at Timken Research, evaluating various lubrication film effect modifying factors, such as base oil viscosity, temperature, speed, and load on tapered roller bearings of relatively small size compared to railroad bearings.[1] The "AP" tests were run to determine the magnitude of influence of oil viscosity on bearing life on these large bearings.

Table 2 clearly shows the bearing life effects of oil viscosity variables. The standard fatigue test lubricant in the laboratory is SAE #20 oil as shown in the first line for comparison. This oil roughly simulated the film thickness under average operating conditions of AAR Grade B grease. Testing with SAE #5 oil, life was reduced to half of standard. On the other hand, SAE #50 oil more than doubled bearing life.

"AP" BEARING FATIGUE RESEARCH

SAE OIL GRADE	LIFE INDEX
20	1.00
5	0.45
50	2.12

TABLE 2.

This experiment illustrated the benefit of higher viscosity oil and resulted in the one grease supplier's development of a new grease embodying it. The new grease was first used in the Timken Xtended Performance ("XP") bearing designed to operate 600,000 miles or 10 years, whichever occurs first without lubrication addition. Subsequently, the AAR adopted a new grease specification, M-942, incorporating the higher viscosity oil.

GREASE STABILITY

In answer to a need to better evaluate grease stability in railroad service, a vibration machine is used (figure 5) to simulate the environment[2] created by steel wheel on steel rail. Traditional bench tests did not predict the performance of a grease in railroad service, but the vibration test utilizing a full-size bearing as shown in the picture has provided entirely reliable results in this respect. The vibrating table produces ±4 G's at 38 hertz and the test duration is 48 hours.

This machine provided a research tool for quick evaluation of grease stability which formerly was very costly, questionable, and time-consuming under field evaluation methods. The previously mentioned AAR Specification M-942 incorporates a requirement for stability evaluation in this same manner.

EFFECT OF WATER IN LUBRICANT

Water in lubricant has a serious effect on roller bearing life.[3] Research testing has determined that

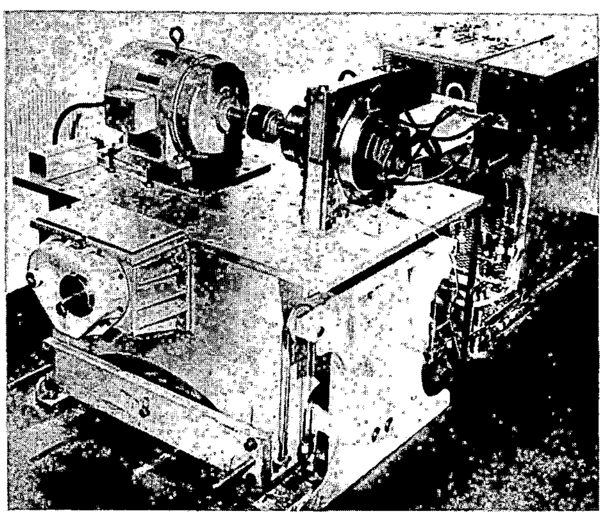


FIGURE 5. VIBRATION TEST MACHINE

bearing fatigue life with SAE #20 oil having a dissolved water content of 400 parts per million (0.04%) is only about half of that having 100 parts per million as shown in Table 3. Data is not available, however, to determine the effect on life of water content in grease. Water content in grease is regularly much higher than used in the research project and generally 0.5% by weight is not considered detrimental to performance. Since most of the water in grease is absorbed by the soap content of the grease, there is little doubt that direct comparisons of water effect in oil cannot be made to that in grease.

TIMKEN RESEARCH STUDY

EFFECT OF WATER IN LUBRICANT ON BEARING LIFE

WATER CONCENTRATION		
PARTS PER MILLION	100	400
% BY WEIGHT	0.01	0.04
BEARING LIFE REDUCTION	—	51%

TABLE 3.

It has been learned from field service experience that excessive water in grease does drastically affect bearing fatigue life, at least under a condition when the soap is saturated and free water is present. Sufficient specific data is not available at this time to quantify the water effect here.

BEARING LATERAL OR END PLAY

The influence of bearing adjustment on fatigue life has been quantified by a computer program based on mathematical analysis of the fundamentals involved. The curve in figure 6 illustrates the relative effect of preload and lateral or end play on bearing life.

A slight preload provides the optimum adjustment and would be the ideal condition to obtain. However, bearing life rapidly decreases with increase in preload as shown by the steepness of the slope on the curve. Another factor is the amount of preload is difficult to measure. For these reasons, on railroad tapered roller bearings, the minimum amount of lateral or end play is considered the best approach to obtain maximum life. To achieve this goal, the spacers which determine the bearing adjustment are provided in incremental lengths to obtain closely controlled preadjustment at the factory or bearing shop.

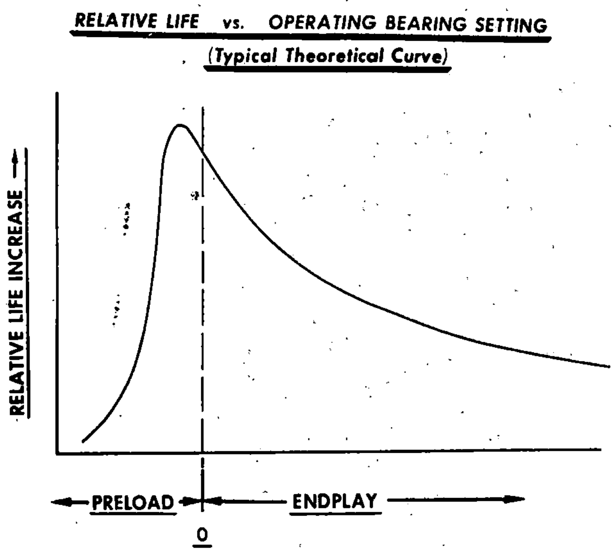


FIGURE 6.

ELASTOMERIC SEALS

Elastohydrodynamic (EHD) seals were first used by the railroad industry in Timken bearings in 1967. The EHD mechanism is located on the field side of the fluid lip of the seal as shown in figure 7. The triangular configuration in this design acts as a pump to impede leakage and enhance the lubricant film in the lip contact area.

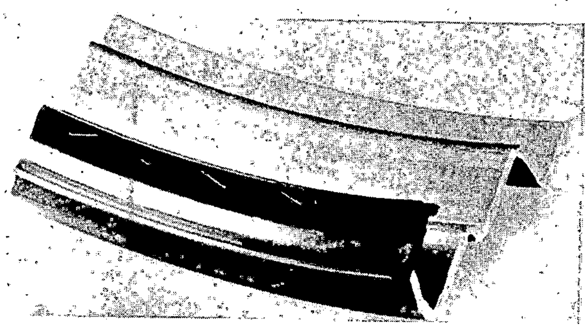


FIGURE 7. THE EHD MECHANISM

Research and experience have disclosed that performance testing is the only reliable way to evaluate seals. Therefore, individual seal designs are qualified by test and each batch of seals is audited in the laboratory. Table 4 shows a list of the extensive test procedures required for a new Timken Company seal design. Once a design is qualified, individual batches are audited under Procedure 2.

Extensive research has gone into seal design for many years because railroad service presents a tough environment for seals. Present-day energy conservation desirability encourages the development of low torque seals, which has resulted in the addition of the torque test under Procedure 9. However, low torque should not be achieved at the expense of adequate performance. Procedures 6 and 7 were added to guard against ingress of dust and water, as well as Procedure 8 to make sure seals won't leak under the vibration conditions experienced in service. These added tests are very important because methods of reducing torque involve reducing fluid lip force on the shaft and elimination or reduction of dust lip fit which can seriously affect sealing efficiency.

THE TIMKEN COMPANY
SUMMARY OF "AP" BEARING SEAL TESTS

TEST PROCEDURE	TEST TYPE	PURPOSE OF TEST TO EVALUATE	APPLIED PRESSURE MAX. PSI
1	AUDIT	COMPOUND AND LEAKAGE	25
2 PART 1	COMPOUND	HEAT AND WEAR RESISTANCE	25
PART 2	COMPOUND	LEAKAGE AFTER EXTENDED SERVICE	25
3	GEOMETRY	SEAL LIP AND MOLD DESIGN	25
4	ECCENTRICITY	SEAL LIP FOLLOWABILITY	0
5	LIFE	AGING AND LIP FORCE RETENTION	0
6	WATER	RESISTANCE TO WATER INGRESS	0
7	DUST	RESISTANCE TO DUST INGRESS	0
8	VIBRATION	VIBRATION AND SHAFT REVERSAL	0
9	TORQUE	DYNAMIC TORQUE	0

TABLE 4.

VENTING EFFICIENCY

Venting of a bearing is important to the sealing efficiency for two reasons. First, to avoid pushing the seals out of the bearing during relubrication and second, to avoid excessive internal pressure which reduces seal life. The curve in figure 8, from work by Brink, shows that seal life under 5 psi pressure is only about half of that under no pressure.[4] To achieve the lowest possible venting pressure, the vent on the extreme right of figure 9 was developed, which has an opening pressure of 0.5 psi, much lower than the previous designs shown.

LOCKING PLATE

A rather mundane piece of hardware is the locking plate whose primary purpose is to prevent loosening of the cap screws which could allow the bearing to loosen and malfunction. However, a lot of thought and experimentation has gone into the new standard lanced tab design on the left in figure 10, which replaces the old design. The new design features heavier material, stronger construction, closer hole fit on the cap screws and most importantly, a revised locking tab orientation. The tabs on the new plate can be bent up against the cap screw head at random, with assurance that at least one tab is in the locking attitude. On the other hand, the old device did not provide this feature since both tabs could be bent against the cap screw head corners and provide no locking whatsoever. The new plate substantially eliminates the probability of human error to do the job right.

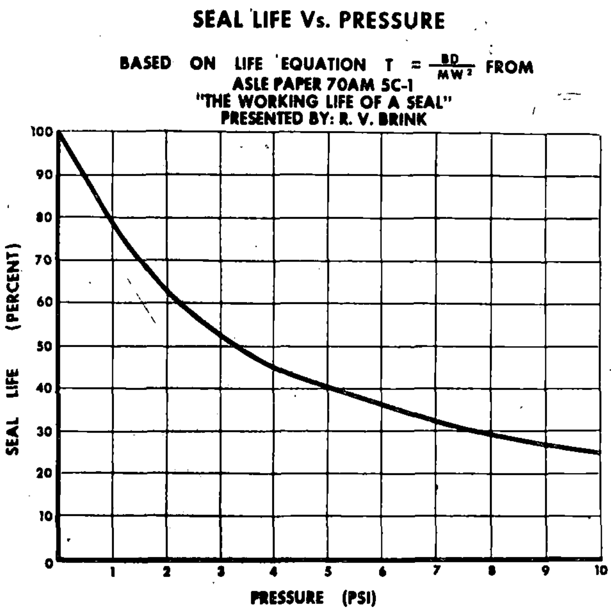


FIGURE 8. SEAL LIFE VERSUS PRESSURE

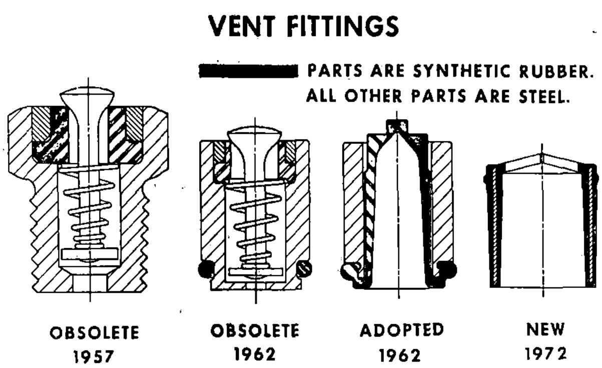


FIGURE 9. VENT FITTINGS

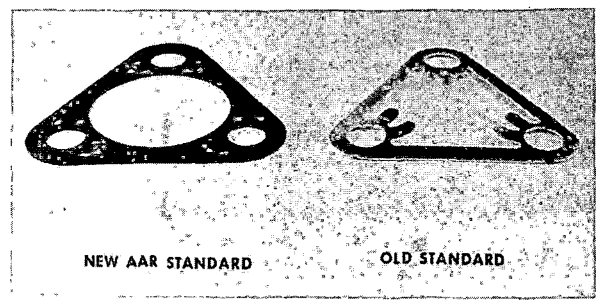


FIGURE 10. CAP SCREW LOCKING PLATE DESIGNS

CONCLUSION

Some of the research achievements have already resulted in steady improvement in standard bearing product without any significant increases in prices attributable to the improvement. Others have resulted in the development of an extended performance bearing at additional cost. Still further improvements in bearing performance are possible, but the demand for such premium performance in a volume attractive for production probably does not exist at this time. Of course the constraints of standardization and interchange of freight cars in railroad service also mitigate against the higher costs involved.

REFERENCES

1. Danner, C. H., "Fatigue Life of Tapered Roller Bearings Under Minimal Lubricant Films", ASLE Transactions, Volume 13, 1970.

2. Leiser, J. E., and West, C. H., "A Vibrating Rig Test for Railway Bearing Greases", Lubrication Engineering, September 1968, pp. 399-408.

3. Cantley, Richard E., "The Effect of Water in Lubricating Oil on Bearing Fatigue Life", ASLE 76-AM-7B-1.

4. Brink, R. V., "The Working Life of a Seal", ASLE 70 AM5C-1.

REALISM IN RESEARCH FOR RAILROADS
BY
R.H. BEETLE

Railroad freight operations differ from other modes of transport. Research developments must be compatible with trains having coupled cars of differing characteristics operating within the fixed parameters of existing track structures. There is an overabundance of ideas for research and development on track and/or car components. Screening these ideas through railroad constraints is realistic even though viewed as unimaginative by some.

The concerns of early railroading still challenge today's research to find pioneering breakthroughs. Realism in today's research requires handling of many new ideas under changing conditions regarding environmental aspects, availability of materials, energy conservation, and economic factors.

Research and development in trackwork, wheels, and friction braking requires realism in specifications and product performance with proper laboratory and field testing. In addition, the requirement for complete system analysis is vital to railroad progress.

Joining of the rail lines into a transcontinental system back in 1869, was an achievement brought about by cooperation of the pioneer railroads and the Government. Today's comparable achievement is the joining of the resources of not only the United States Government, but the Canadian Government, the AAR, RPI, and the many others, into the TTD and FAST programs. Use of TTD and expansion of the FAST trackage to permit a wider range of tests on longer tangent track, certainly are forward and logical steps to benefit the railroad industry as soon as possible.

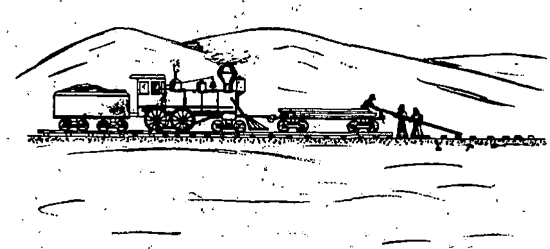


FIGURE 1. EARLY RAILROAD CONSTRUCTION

INTRODUCTION

It was here in the West, that construction and joining of the rail lines into a Transcontinental system was accomplished through the cooperative efforts of the Government and the railroad industry (figure 1). Today, again in the West, near Pueblo, Colo., the Governments of the United States and Canada, along with the railroad industry and the railroad suppliers, have joined together in a common effort.

This new and progressive effort has culminated in a superior facility for researching railroad equipment and track structures. It is appropriately designated the Facility for Accelerated Service Testing, or FAST track.

It is expected that "realism in research" for railroad operations and equipment will result in improvements accelerated by practical testing methods at this excellent facility. Maintaining "realism" in research, while evaluating the increasing number of ideas, will continue to be a challenge.

Some of our company products labeled in figure 2, will be used as examples in this discussion concerning

---

Robert H. Beetle has been the Vice President of Abex Corporations' Railroad Products Group since 1967. He is a NROTC graduate from Renssalaer Polytechnic Institute with a BS in metallurgical Engineering, and gained his MS in Metallurgy while on the staff of Pennsylvania State University.

research.

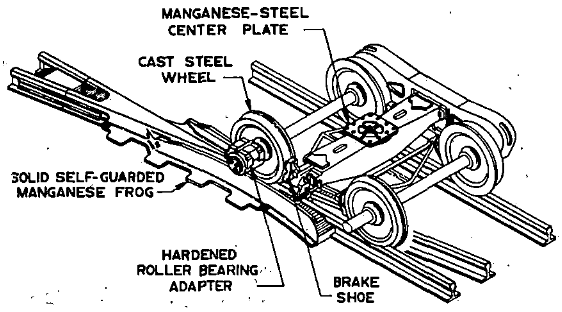


FIGURE 2. MAJOR ABEX RAILROAD PRODUCTS

RESEARCH CONSIDERATIONS

The Flood of Ideas

The flood of ideas, coming to research, can be compared with the flood of water from the mountains which was of great concern during pioneer railroad construction and operations (figure 3). Today, the railroad industry not only must deal with mountain flood waters, but also with the ever-increasing "flood" of new ideas.

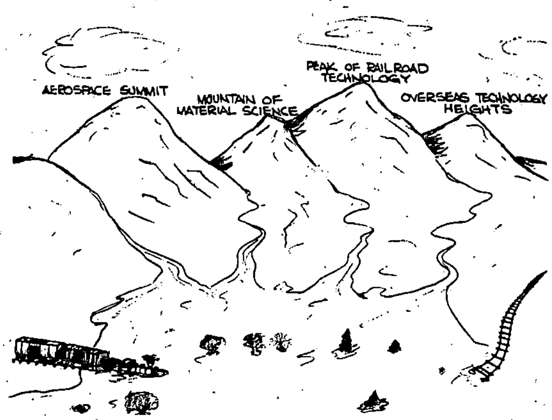


FIGURE 3. FLOOD OF IDEAS

To further demonstrate this point, the illustration shows a typical mountain area, labeled with mountain peaks corresponding to idea sources. Some of the following examples relate to the research involved with our company products:

AEROSPACE SUMMIT

From the aerospace summit, research to develop a new system of Automatic Car Identification (ACI) was launched. This new system used microwave scanners to produce a data-gathering network to maintain location and identification of all interchange railroad cars and equipment. Although it was possible technically, it did not prove to be a commercial success for our company.

Other aerospace technology provided advanced knowledge in electronics and hydraulics, which was successfully applied to improve transit braking, automatic classification yards, and other systems. Unfortunately, the learning curves for these new applications have indeed been too long.

MOUNTAIN OF MATERIAL SCIENCE

From the mountain of material science, we found many metallurgical developments worthwhile for railroad application. Prime examples are the increasing uses of manganese steel for wear parts on freight cars as well as trackwork. The use of manganese steel for center plates and coupler wear plates resulted in at least a five-fold wear life gain over previous materials.

Our introduction of freight car roller bearing adapters in ductile iron, with selective hardening, has resulted in at least a three-fold gain over previous materials.

Our company manufactures castings for vital parts of nuclear submarines. We have applied this same metallurgical technology to cast steel wheels for diesel-electric locomotives. The material was a high yield 0.20% carbon alloy. Wear results throughout the life of these multiple wear wheels indicated that the wear benefits became significant only after the wheels were machined to the last wear surface, as shown in figure 4, along with a corresponding "wear advantage" percent.

PERCENT WEAR ADVANTAGE FOR ALLOY WHEELS VS CLASS B WHEELS

UP TO 1st TURNING	4.5% ADVANTAGE
BETWEEN 1st AND 2nd TURNING	7.0% ADVANTAGE
2nd TURNING TO CONDEMNING	22.8% ADVANTAGE

FIGURE 4.

These multiple wear wheels did increase life, however, these benefits were not sufficient to justify the high cost of the alloy involved.

PEAK OF RAILROAD TECHNOLOGY

The greatest number of ideas originate from railroad technology itself. These ideas are ever-present to a varying degree within all of our products. For example, specialized trackwork designs to improve railroad structures and service life, are particularly dependent upon evaluation through many years of service. In this respect, the FAST track is of special advantage, since realistic evaluation can be accomplished at an accelerated testing rate.

OVERSEAS TECHNOLOGY HEIGHTS

Technology from overseas has been developing a greater interest in the United States. However, much of the overseas-developed railroad equipment requires considerable modification to meet demands of American railroad service. We acquired the patent rights for a high phosphorus alloy developed overseas for making brake shoes. Subsequently, through our technology of brake shoe structure, we successfully adapted the new alloy for American railroad service. This new metal brake shoe, which we call "SAMSON", has revolutionized the opportunities for application of metallic brake shoes in the most modern and demanding railroad braking services.

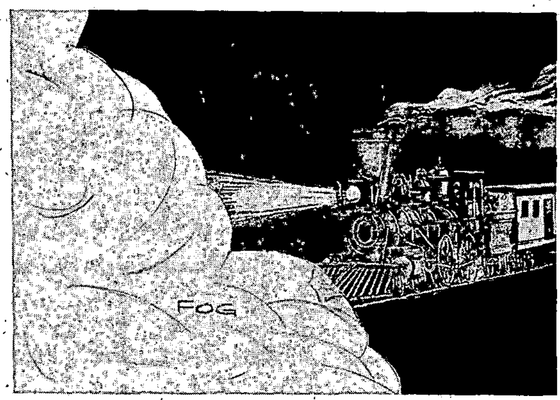


FIGURE 5. FOG OBSCURING THE COURSE OF PROGRESS

The Fog Over Environmental Protection Needs

As we move forward in research, we encounter a "fog", obscuring environmental protection needs (figure 5).

What is historical and proper today may, with future knowledge, raise environmental concerns. Even when there is no actual evidence of potential hazard, environmental concerns may have costly consequences. Research for new products should avoid materials and operations where future indications could raise environmental concerns. For example, this is why our research has been focused on mixes for railroad composition brake shoes, ingredients which are not of concern from the environmental protection aspect. Our new "TIGER" composition brake shoes, having no lead or asbestos, resulted from dedicated research for an improved product without materials, which in other applications, raised questions on the environmental aspects.

Shortages of Material and Energy

As pioneer railroads in desert areas faced a shortage of wood for fuel and crossties, today's railroad research must also be concerned with both material and energy needs (figure 6).

Railroad wheels are either forged and rolled, or cast to shape. Even with an efficient forging and rolling steel wheel plant, it appears at least 60% more thermal energy is required to produce a ton of heat-treated wheels starting from cold scrap, than for a similar tonnage of cast steel wheels.1

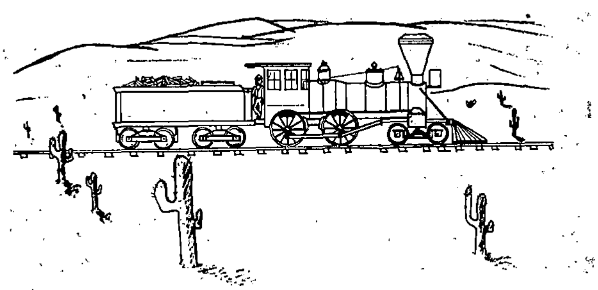


FIGURE 6. SHORTAGE OF FUEL AND TIES IN THE EARLY WEST

Uncertainties in the supply and cost of petrochemical and imported natural resins are examples of the problems involving research development of composition brake shoes. Therefore, research must be realistic and adopt a continuing program to find alternatives. Preferably, domestic materials should be developed which can provide needed performance with economy. It seems difficult enough to select materials based on cost and availability, without additional concerns for choosing natural materials in contrast to synthetic ones.

Biologist Barry Commoner in "Time Incorporated, Energy Conference - 77",2 states in part: "For example, the petrochemical industry produces goods that replace natural products and often have only marginal social benefits. Plastic for heart valves is a socially valuable product; plastic swizzle sticks are questionable." Use of plastics in railroad composition brake shoes, while not as critical as heart valves, and even though consumable, does provide a high degree of performance.

Economic Considerations

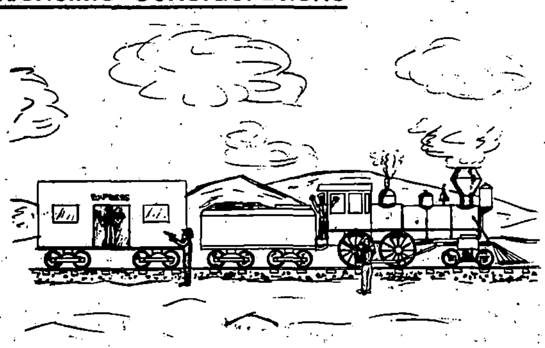


FIGURE 7. RAILROAD HOLDUPS

Modern railroad operations have different types of holdups than those of pioneer railroads. Losses can occur in many ways. One outstanding area is lading damage. Research to build better riding cars and improve the handling of freight trains, is in progress today.

A recent ASME paper by Paul Garin and Klaus Cappel, presented the factors in lateral stability of freight cars.[3] It involved extensive field investigation into the effects of rail-wheel geometry and its relationship to undesired freight car truck hunting.

One of the significant conclusions of this dynamic study, was that the brake shoe type and size can affect the wheel tread profile, to the extent that stability can be altered. Similar cars designed for automobile transport, with comparable mileages, were operated at various track speeds. When one of the two cars exhibited a superior ride at elevated speeds, the cars were carefully examined. The examination indicated only one significant difference in the car equipment--the car with the stable ride was designed for, and fitted with cast iron brake shoes. The other car was fitted with composition brake shoes.

The car with cast iron brake shoes exhibited a more stable ride than the car with composition brake shoes because its wheel treads developed a more favorable contour.

Work is continuing in this important area of research to improve the riding of cars. The application of systems research may be effective in this significant problem area.

It is expected with expanded track facilities, the FAST track operation, and the Track Train Dynamics Program, as well as the continued Truck Design Optimization Project, Phase II (TDOP), could expedite this research and further explore the relationship of brake shoe types and sizes, and the effects on wheel tread geometry in service.

Economics suggest the life of a brake shoe should not be increased at the expense of the wheel, nor the life of the wheel at the expense of special trackwork. Our research for new special trackwork designs is based on the economics of extending service life, while reducing maintenance costs and providing better performance.

Special trackwork, like turnouts and crossings, must be designed to both guide and carry the wheels with minimum impact. Such special trackwork must recognize the changing relationship between the trackwork and wheels from new contours to the worn-out condition. Accordingly, such research requires a complex study of interacting parts, whose changing nature

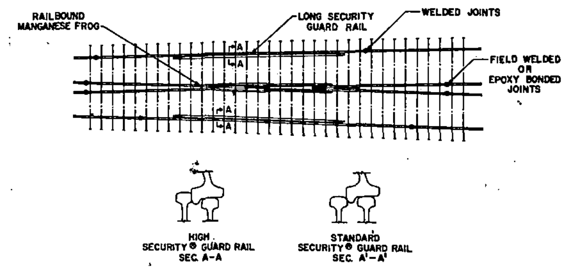


FIGURE 8. IMPROVED TRACK STRUCTURES

favors realistic in-service testing.

Fortunately, new concepts to reduce trackwork maintenance and impact are being evaluated as part of the FAST program. These include longer guard rails to improve tracking of trucks through frogs, long approach flares to reduce impact from cocked trucks, high guard rails which engage the flat back of the wheel flange rather than its flange radius to ensure proper location of each wheelset going through special trackwork (figure 8). The longer arms of the frogs of these turnouts, being rigidly joined to the rail by either field welding or epoxy-bonded joints, require less maintenance than normal bolt joints, and further perpetuates the natural sine wave progressing in advance of the wheelset as the car travels down the track. Perpetuation of the natural sine wave provides a much smoother ride through the turnout and dampens forces encountered by various track components. It is difficult to prove the benefits of mating the wheel and special trackwork under laboratory conditions. As a consequence, the testing on the FAST track realistically evaluates the benefits and economics relevant to both the trackwork and the car.

PERFORMANCE

Specifications

Product specifications and applications must not lead our research to a "mirage" as we must continually focus on the real-world of railroad service (figure 9). Normally, a brake shoe is not viewed in its fundamental role as a converter of mechanical energy to thermal energy. When a brake shoe is required to dissipate excessive thermal energy because it has been misused, we can expect various problems as a consequence. This becomes increasingly important as we research the stress levels induced into wheels at various levels of braking-developed energy.

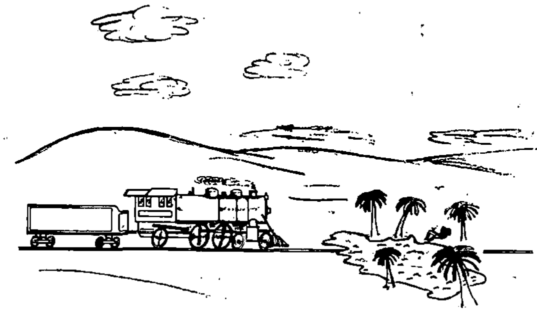


FIGURE 9. LEADING RAILROAD RESEARCH INTO MIRAGES

For example, when wheels are braked with metal shoes the resultant wheel stresses are significantly less than with composition shoes, under the same level of braking energy. The relatively low thermal absorption of the composition brake shoe compared with the metal brake shoe accounts for the biggest difference. When normal braking levels are exceeded, this difference in wheel stress levels becomes appreciable, and favors renewed interest in metallic-type brake shoes.

For the past four years, railroads in Western Europe and Russia have been testing high phosphorus alloy metal brake shoes with Abex SAMSON structures. European test results show three to five times the life and other performance benefits in freight service. Both in Europe and Russia, the performance gains in the heavy-duty high speed passenger and locomotive operations have gained even more notice. In various applications, the braking mechanism must be properly designed to match the characteristics of the brake shoe friction material. We manufacture both composition and metal brake shoes, and believe each type has advantages. Through research and product development, we are continually trying to improve both types of brake shoes. Track Train Dynamics and an expanded FAST track could definitely accelerate evaluation of research efforts resulting in mutual benefits through better products.

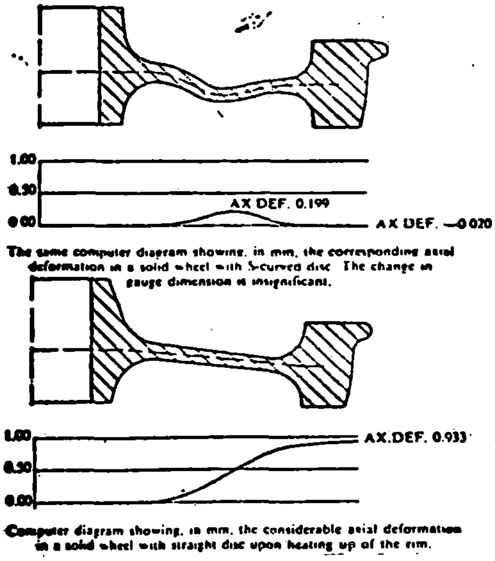


FIGURE 10. SWEDISH WHEEL DESIGNS

A study was conducted in Sweden on the phenomenom of wheel movement resulting from brake shoe heating4 (figure 10). This movement is not a mirage, and has been observed for many years. It was studied by Professor H. R. Wetenkamp in 1973.5 Despite this movement of wheels under certain conditions, there is a lack of information showing that any problems are generated. However, concern over this small movement has brought modifications to the European wheel designs as illustrated by the Swedish study. Thus far, there has not been sufficient concern in the United States to experiment with the greatly changed "S-curved" design because of other potential disadvantages.

Laboratory Testing

Twenty years after the historic joining of pioneer railroads, the first brake shoe testing machine was put in service (figure 11). This more scientific method of testing brake shoes started in 1889 with the first known brake shoe testing machine designed by F. W. Sargent, Engineer of Tests

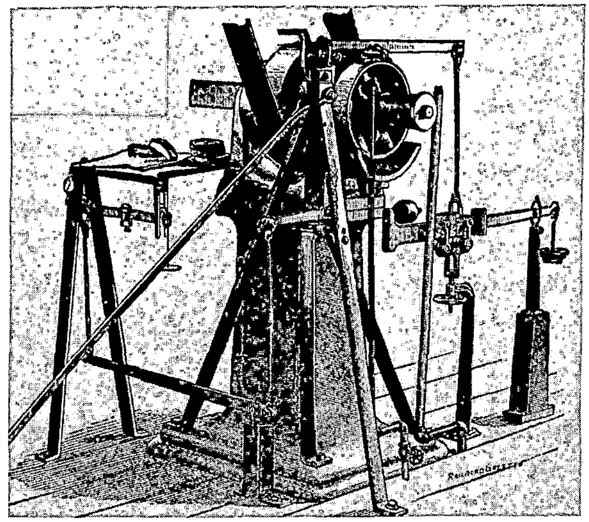


FIGURE 11. THE FIRST BRAKE SHOE TESTING MACHINE

for the Chicago, Burlington and Quincy Railroad, and used by the Master Car Builders Committee in conducting brake shoe tests. It was a small machine that tested miniature brake shoes 4" long and 1" wide on an 11½" diameter chilled iron wheel. From this humble beginning, the scientific testing of brake shoes developed. Mr. Sargent was later Chief Engineer of our company and instrumental in the implementation of dynamometers that led to our newest installation shown in figure 12.

This dynamometer is located at our Engineering Test Center in Mahwah, N.J. It can simulate wheel inertia loads and brake shoe forces for passenger and freight cars, in addition to transit and locomotive equipment. It is anticipated that the real-life results developed from an expanded FAST track or Train Track Dynamics Program will bring about new and improved AAR and other specifications. Additionally, we expect improved laboratory testing guidelines, which may possibly be better correlated with actual service conditions.

Field Evaluation

Lacking modern instrumentation, the pioneer railroads had difficulty in measuring and recording data from field tests. Today, we use equipment such as shown in figure 13.

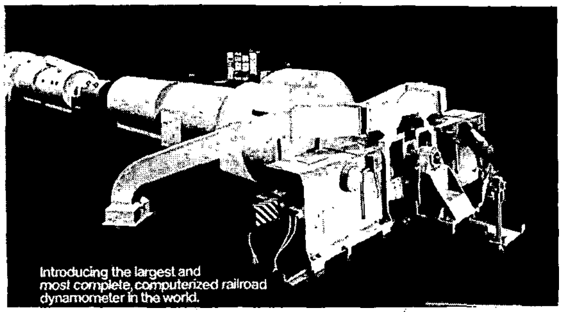


FIGURE 12. THE NEW ABEX DYNAMOMETER

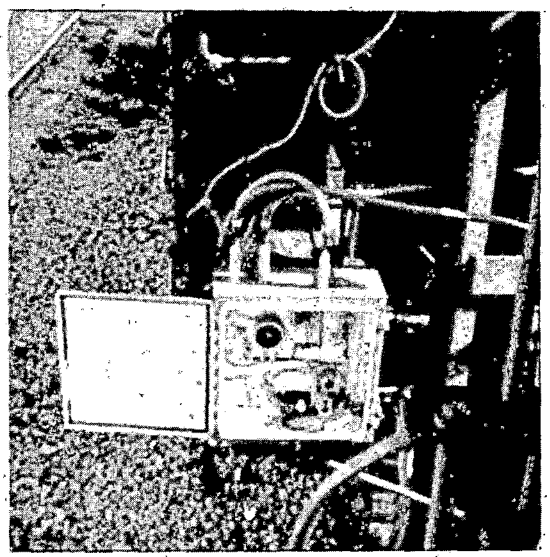


FIGURE 13. FIELD TEST INSTRUMENTATION

This automatic equipment is used to control braking and develop data during single car breakaway tests on loaded 70-ton and 100-ton cars with high friction composition brake shoes, as well as AAR Standard high phosphorus metal brake shoes. Results indicated that a lowering of braking ratios on new cars using AAR Standard high phosphorus metal brake shoes. Results indicated that a lowering of braking ratios on new cars using AAR Standard high phosphorus brake shoes would allow composition brake shoes in mixed consist trains to perform equal work. As is sometimes found, field evaluation results may vary from laboratory dynamometer results which, in the case of friction materials, form the basis for current specifications. Based on L. P. Kennedy's recent paper, field evaluations indicate that a 70-ton car is more subject to overbraking than a 100-ton car when in a mixed consist.6

Systems Analysis

The resources for systems analysis are symbolized here in figure 14.

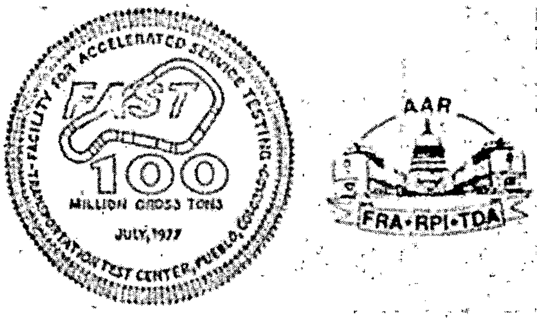


FIGURE 14. FAST AND TRACK TRAIN DYNAMICS

Unlike most modes of surface transportation, railroads in pioneer days and today have used trains rather than single vehicles. Trains can include both loaded and empty cars of various types and characteristics. Economic benefits of train operation are obvious, but the complexities for increased systems analysis, may not be so clear. Despite all the variations possible in such train consists, the specialized trackwork must properly guide and carry all types.

This is another reason why FAST with expanded trackage and Track Train Dynamics are so vital to railroad progress today. They are needed to determine data on the performance and interaction of all components and systems in mixed consist operation of trains under varying conditions.

CONCLUSION

Joining of the rail lines into a transcontinental system in 1869 was an achievement brought about by cooperation of the pioneer railroads and the Government (figure 15). Today's comparable achievement is the joining of the resources of not only the United States Government, but the Canadian Government, the AAR, RPI, and many others, into the TTD and FAST Programs. The combination of these resources sets the stage for even greater railroading achievements through realism in research for railroad products. This realism must consider environment, materials, energy, economics, as well as performance. Products must be properly evaluated and tested both in the laboratory and the field.

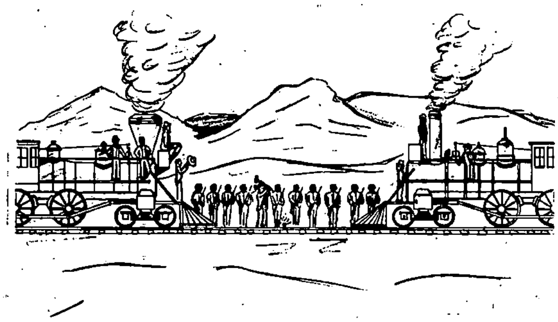


FIGURE 15. THE COMPLETION OF THE TRANSCONTINENTAL RAILROAD AT PROMONTORY POINT

In the year following completion of the transcontinental railroad system, all the railroads together carried only 72.5 million net tons on 53,000 miles of track, compared with today's FAST track accomplishment of 100 million gross tons on 4.8 miles of track in less than its first year of operations.
Expansion of the FAST trackage to permit a wider range of tests on longer tangent track is certainly one forward and logical step to ensure that realism in railroad research results in benefits to the railroad industry as soon as possible.

ACKNOWLEDGEMENTS

The writer wishes to thank members of the Abex Railroad Products Group Engineering Department for their contributions of technical data and editorial comment.

REFERENCES

1. Beetle, R. H., "A Technical Economic Rationale for Railroad Wheel Challenges" International Wheelset Congress, October, 1975.

2. Commoner, Barry, "Opening the Debate", Time Magazine, April 25, 1977.

3. Garin, P., and Cappel, K., "Some Relationships Between Dynamic Performance of Freight Car Trucks and Worn Wheel Tread, Rail Geometry", ASME/IEEE, Chicago, Ill., April 7-8, 1976.

4. Surahammer Wheel Brochure, Swedish manufacturer of railroad wheels, Surahammars Bruks AB, Surahammar, Sweden.

5. Wetenkamp, H. R., "Thermal Stress Developed in S Plates, Straight Plates and Deep Dish Wheels", ASME paper '73-RT-April 1973, St. Louis, Mo.

6. Kennedy, L. P., "Measuring Freight Car Brake Shoe Performance Dynamically", Railway Fuel & Operating Officers Association, September 1977, Chicago, Ill.

7. Statistics from "Historical Statistics of the USA-Colonial Times to 1957".

INSTRUMENTED WHEEL SET SYSTEMS FOR PRODUCT PERFORMANCE ANALYSIS

By

H.G. Tennikait

INTRODUCTION

There has been, in recent years, an increased and dedicated effort by industry, government and academic groups to more fully understand the dynamics of a railroad freight car. Evaluating the forces and motions imposed on the freight car during "real world" operating conditions has required the application of procedures, techniques, and hardware that were, in the not too distant past, foreign to our industry. Not only has off-the-shelf hardware been brought into wider use, special equipment has been designed, constructed, and employed to develop specific operating parameters.

Of prime importance in any truck study are the forces at the wheel/rail interface. The problem of converting a rotating wheel and axle set to a force measuring transducer is difficult. Even more difficult is the transmission of the relatively low level signals to the recording equipment.

Several types of systems are now being used, however, an approach presented to the industry by Messrs. L. A. Peterson, W. H. Freeman, and J. M. Wondrisco in ASME Paper 71 WA/RT4, seemed the most practical for ASF's use. Their approach entailed the application of the combined measured bending moments in the axle, and the measured vertical journal forces to the equations developed from free body diagrams of the wheel/axle set. ASF used the same basic force diagram and the resulting equations. However, the methods of sensing the vertical journal forces and the bending moments in the axle are different.

The following covers the ASF system development and application.

BASIC CONCEPT

The basic concept upon which the ASF system was designed is directly related to a simplified free body force diagram of a wheel and axle set (as applied to a conventional freight car truck) and the resulting equilibrium equations for the system. Figure 1 shows the free body diagrams of the applied forces.

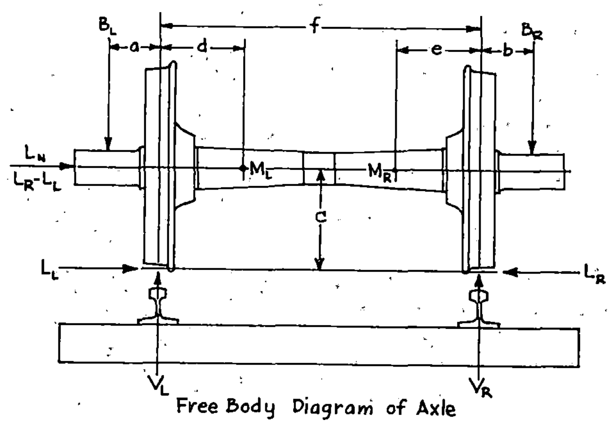


FIGURE 1.

Equations expressing the bending moments in the axle at ML and MR are determined from the diagram as follows:

ML = LLc + BL (a + d) - VLd
MR = LLc+BL(a + (f - e))-VL (f - e)
and
MR = LRc + BR (b + e) - VRe
ML = LRc+BR(b + (f - d))-VR (f - d)

Where:
ML=Axle bending moment - Left Side
MR=Axle bending moment - Right Side
BL=Journal bearing load - Left Side
BR=Journal bearing load - Right Side
LL=Lateral Wheel/rail force - Left Side
LR=Lateral wheel/rail force-Right Side

H. Garth Tennikait was graduated from the University of Missouri - Rolla, in 1950 with a Bachelor of Science - Mechanical Engineering. He is Manager - Test Engineering at the American Steel Foundries Granite City, Ill.

VL=Vertical wheel/rail force - Left Side
VR=Vertical wheel/rail force-Right Side
a b c d e and f are physical dimensions

When the above moment equations are solved simultaneously, expressions for calculating the values of the vertical and lateral wheel/rail forces are obtained as a function of the axle bending moments and the journal loads.

$$VL = \frac{ML - MR}{f - d - e} + BL$$

$$VR = \frac{MR - ML}{f - d - e} + BR$$

$$LL = \frac{ML - MR}{f - d - e}\left(\frac{f - e}{c}\right) + \frac{MR}{c} - BL\left(\frac{a}{c}\right)$$

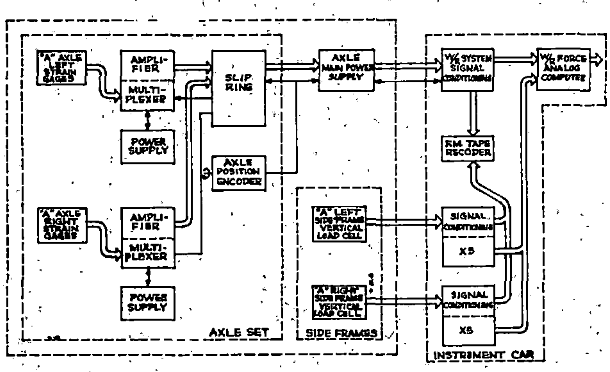
$$LR = \frac{MR - ML}{f-d-e}\left(\frac{f - d}{c}\right) + \frac{ML}{c} - BR\left(\frac{b}{c}\right)$$

$$\text{And } LN = LR - LL$$

Physical dimensions are nominal and although it is recognized that they are not constant in the dynamic application, the variations are small and not considered, because of the added complexity to the system.

THE GENERAL SYSTEM

The complete ASF system for one axle set of wheel/rail force measuring instrumentation is shown in the Block Diagram, figure 2.



BLOCK DIAGRAM OF ONE AXLE SYSTEM

FIGURE 2.

Associated with the test truck are two strain gaged and instrumented wheel and axle sets complete with electronics, position encoders and slip rings. In addition, both side frames of the instrumented truck are functioning load cells designed to sense vertical forces at each journal. The main power supply for the truck mounted equipment is attached to the test car.

The test truck instrumentation is cabled into the Instrument Car which contains all of the signal conditioning and controls for the system. An FM magnetic tape recorder is used for a permanent record of the primary data and a dedicated analog computer is used to furnish "real-time" vertical and lateral wheel/rail force computations.

To better understand how the system functions, each of the above major sub-assemblies will be discussed in detail.

THE AXLE ASSEMBLY

One complete instrumented axle set is shown in figure 3.

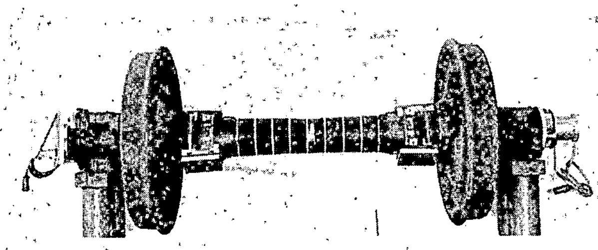


FIGURE 3.

The axle assembly must provide measurements of the axle bending moments ML and MR necessary for the basic vertical and lateral wheel/rail force equations.

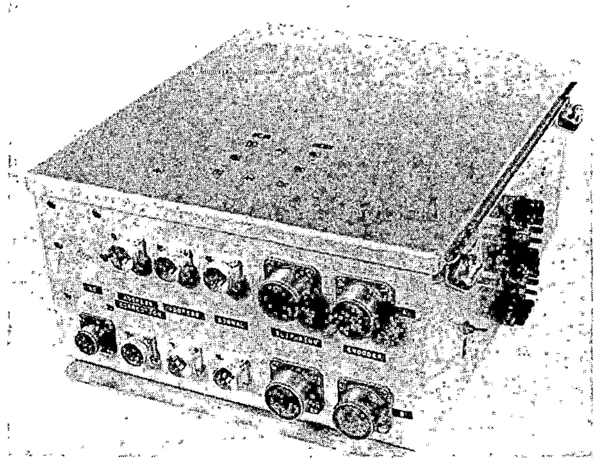
The axle is transformed into a bending moment transducer by the application of strain gages. Sixteen strain gage bridges are applied on the axle, eight located approximately five inches inboard of each wheel and equally spaced about the circumference of the axle. Associated with each group of eight bridges is a regulated bridge power supply package and a multiplexer-amplifier package. These are the metal boxes strapped to the axle near each wheel. At one end of the axle is a slip ring assembly to connect the rotating electronics to the balance of the system. At the other end is an axle position encoder which provides the address for the multiplexer.

As the axle rotates during use, each pair of diametrically opposite strain gage bridges provides the bending moment sensing element when they rotate within the arc $\pm$ 22-1/2° of the vertical plane through the axle. The position encoder senses this angular position and addresses the multiplexer which in turn switches the two pairs of bridges into the electrical measuring circuit and the amplifier. The amplifier output, in turn, is routed through the slip rings to the signal conditioning located in the instrument car.

The strain gages and the electronics on each end of the axle function independently of one another.

THE AXLE MAIN POWER SUPPLY

The axle main power supply is normally attached to the test car in a location readily accessible to the instrumented truck, and to the intercar connections leading to the instrument car. This unit, shown in figure 4, provides the unregulated power for two instrumented axles and serves as a junction point for the interconnection cables from the various components of the system.



THE SIGNAL CONDITIONING AND CONTROL ELECTRONICS

The axle signal conditioning and control electronics are mounted in the equipment racks in the Instrument Car. Figure 5 shows the general arrangement of the axle signal conditioning, the analog computer, and the side frame load cell signal conditioning.

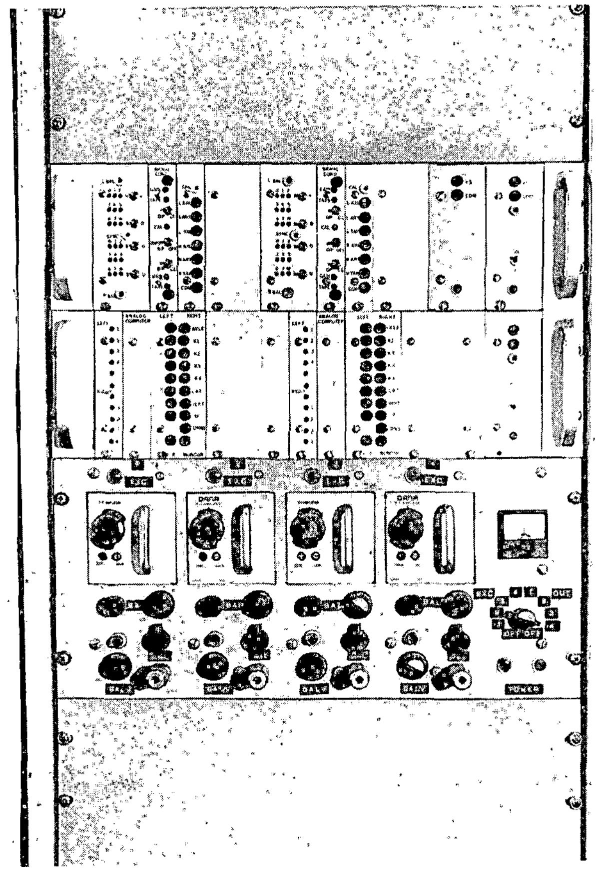


FIGURE 5.

The top unit in the rack is the axle signal conditioning component. The single card rack contains the complete electronics for two instrumented axles. This unit provides the means by which each of the individual strain gage bridges on the axle may be remotely balanced for zero mechanical input. Bridge output caused by the static weight of the car can also be "zeroed" by a control on this module.

Normally, bridge balancing and car weight offset adjustments are made when the car is moving at 10 to 20 mph on straight level track. This procedure reduces the error due to locked in forces in the wheel and axle assembly resulting from static wheel/rail contact.

Other controls permit an electrical compensation for small angular errors in the axle position encoder address supplied to the multiplexers. These errors are normally the result of mechanical misalignment as the encoder is attached to the axle via a tubular flexible coupling. Thus, almost all observed irregularities may be corrected without stopping the train for mechanical adjustments.

The signal conditioning module contains the calibrating means for the system and the necessary amplification and scaling of the output signal. Output from the signal conditioning module is at I.OV F.S. and is available for FM tape recording, analog computations, oscillograph monitoring, etc. Each of the two groups of strain gages on the axle has its own independent set of controls. Included as part of the axle signal conditioning component is a monitoring module which provides test points for the quick access to electrical circuits for system set-up and "real-time" monitoring.

The center piece of equipment shown in figure 5 is the analog computer built by ASF to provide real-time computations of the vertical and lateral wheel/rail forces. The primary purpose of this unit is to provide data for the Test Engineers, to ensure test safety and procedure control. The output from the computer may be recorded on tape or displayed on an oscillograph or similar device.

There are no external controls on the computer. The module to the left of the computer board contains the necessary controls to set up the circuit constants for a particular axle - the physical measured dimensions of the axle determine the values. Modules are set up in the laboratory for each axle assembly at the time the axle is calibrated. In the event that axles are changed, four plug-in resistor boards within the modules are also changed to reflect the new constants.

The lower piece of equipment in the photograph is the four channel signal conditioning unit for the side frame which has been strain gaged to provide the vertical journal force measurements required by the vertical and lateral wheel/force equations. The unit was built by ASF for this and similar applications. It is basically a general purpose signal conditioning system for strain gage type transducers. It has wide range bridge balancing capabilities, individual bridge excitation, calibrating means, and galvanometer drivers. The four channel unit is built around high quality Dana model 3420-V2 differential D.C. amplifiers.

Side frame load cells (strain gage bridges on the tension and compression members) are cabled directly to this signal conditioning unit via intercar connections. Output of the amplifier is normally recorded on magnetic tape to be used at a later time when the lateral and vertical wheel/rail forces are computed. Output signal may also be routed to the analog force computer for "real-time" computation.

It should be noted that ASF records, on magnetic tape, only the primary data required by the basic force equations of the system (i.e., the vertical forces at the four journals and the axle bending moments at the left and right side of each instrumented axle assembly).

CALIBRATION

The instrumented axle assembly is calibrated in the laboratory with all of its normal signal conditioning units in operation.

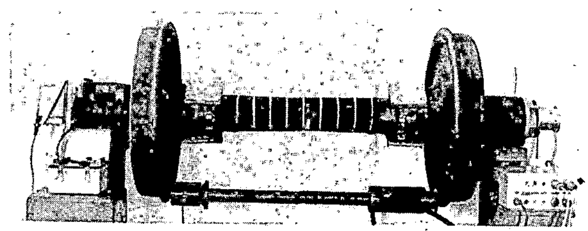


FIGURE 6.

The axle is set-up on jacks to permit the wheels to rotate. A load cell and hydraulic jack assembly as shown in figure 6 is positioned between the wheels to exert a force on the inside rim. Distance from the center of the axle to the load application point is measured. Output of each strain gage bridge on the axle is determined for the measured force and calculated bending moment applied. One calibration value is selected for each group of four paired strain gage bridges (i.e., four for the left side and four for the right side). The nominal full scale calibration for a 36" diameter wheel is 40,000 foot-pounds or approximately 27,000# of force applied at the rim.

Physical dimensions, required for the general force equations of the system, are determined and the "constant" values for the analog computer are calculated.

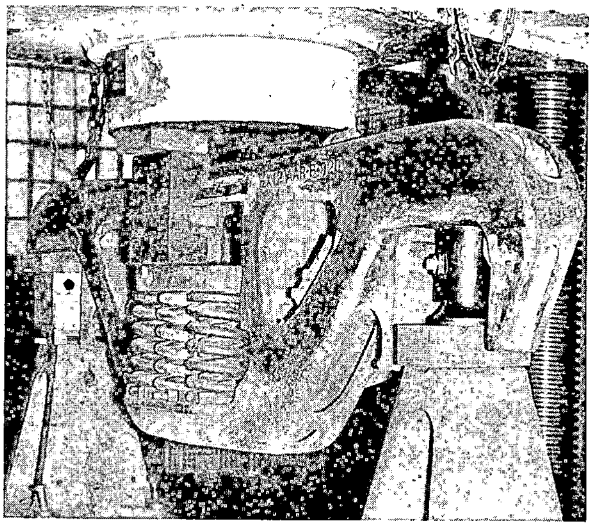


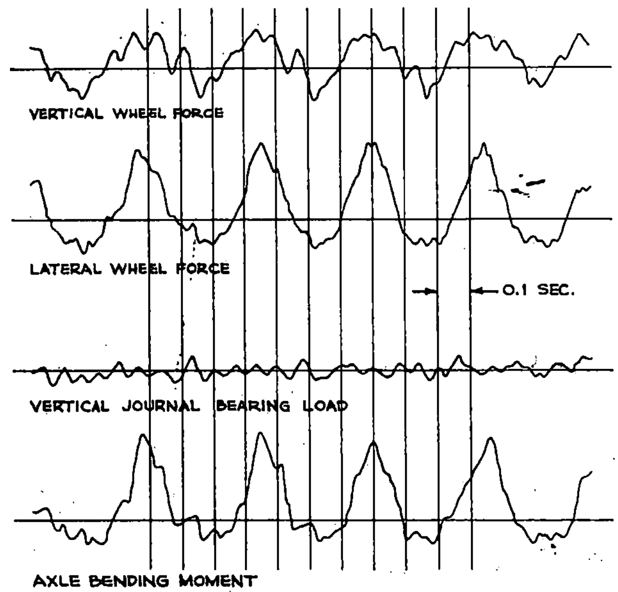
FIGURE 7.

Side frame load cells used to measure the vertical journal forces are calibrated in a laboratory universal static testing machine. The set-up used by ASF is shown in figure 7. Machine load is applied to the side frame through a normal spring group. A compression type load cell is used to support one pedestal and to measure the reaction force at that location. From this data the actual load at each pedestal can be determined.

A series of loads are applied and the output of the strain gage bridges at each end of the side frame is measured. From this data, full scale calibration values are selected. Normally these values would be on the order of 40,000 pounds.

DATA OUTPUT

During test operation, the primary data channels are recorded on FM tape (i.e., bending moments from the axle, and vertical journal forces from the side frame.) This procedure requires eight FM recorder channels, however, maximum flexibility is achieved when processing tapes on the mini-computer. For instance, corrections may be applied to individual parameters should it be necessary before final computation. The option is also available to use the dedicated analog computer to reduce the primary data to wheel/rail forces outside the mini-computer, thereby saving core for other purposes.



TYPICAL OUTPUT SIGNALS

FIGURE 8.

Figure 8 shows typical signals taken from oscillograph charts during an ASF Test Series at TTC at Pueblo, Colo., in 1974. This chart duplicate shows the primary parameters (axle bending and vertical force) for one wheel along with the same parameters after computation through the analog computer. On-board observation of these wheel/rail forces is one form of test control. The recording was made during a high speed hunting test on a 100-ton truck. Speed was in excess of 70 mph. Figure 8 is typical of the output at each of the four wheels of the test truck. L/V ratios are also within the capability of the on-board computer, however, in most studies, there is more interest in actual forces than the ratio of forces.

TEST SITE

ASF has used the wheel/rail measuring system in tests conducted at the Transportation Test Center at Pueblo. Three separate test sections are used in developing the results discussed herein:

1. Curve "A" - Balloon Track - 300' spiral leading into a superelevated curve of 7°30'. Balance Speed - 32 mph.

2. Curve "C" - Train Dynamics Track - 300' spiral leading into a superelevated curve of 4°. Balance speed - 47 mph.

3. Curve "E" - Train Dynamics Track - 0°50' curve. Balance Speed 100+ mph.

All curves were to the right causing clockwise rotation of the truck relative to the car. Balance speed was used on curves "A" and "C" to negate, as much as possible, the effect of out-of-balance of the car body. These two curves were used to study the effect of different curvatures on wheel forces. Curve "E" was used for truck hunting at speeds up to 80 mph. Balance speed in "E" was unattainable, but test speeds were adequate to develop truck hunting characteristics.

EXPLANATION OF GRAPHS

The ASF Test Train System consists of several cars: (1) ASFX-1946 (Power and Equipment Support); (2) ASFX-1965 (Instrumentation and Crew Support); and (3) Test Car of suitable capacity. Recorded speeds are taken from the ASFX-1946 Car equipped with nominal 33" diameter wheels. Primary source for the speed pick-up is a magnetic sensor with a 60 plus per wheel revolution output. Output is put directly on the magnetic tape, making it possible to use any number of revolutions or portion of a revolution as data selection points.

Using the ASFX 1946 Car (power car) as a reference, one data sample was stored for every parameter, for every wheel revolution into the curve. A 1946 Car wheel was used for reference since these wheels were never changed, and the distance into the curve for a given sample point would be the same for every test regardless of the truck arrangement under the test car. As a result, a data sample was stored once for every 8.689 feet (circumference of 1946 Car wheel) along the curve. Due to tolerances on the track marker pick-up system and in the computer itself, a given point would be the same distance into the curve plus or minus approximately 1-1/2' for any given test. Since the circumference of the standard instrumented wheels under the test car was 9.31', a data sample was stored for every .933 revolutions of the test car wheel through the test curve.

All data contained herein are plotted on graphs with a common abscissa, while individual wheel parameters were plotted on the ordinate. The common axis is labeled "distance Traveled Into Test Section (Feet)". This label is self-explanatory and is expanded further on the graphs to show that the first 300 feet of test section represents the spiral going into the curve, and the second 300 feet represents the steady state curve. Data processing began at the point of tangent/spiral.

Even though a typical plot looks like a continuous smooth line, it actually represents a series of very short straight lines between data sample points. Each curve contains 67 to 68 data points representing the number of base car wheel revolutions through the curve.

In summary, each individual data curve plotted on the graphs represents 67 or 68 discreet data points comprising some parameter magnitude on the ordinate versus some particular wheel revolution into the curve (along the abscissa).

In curve "E" (truck hunting section) the abscissa is again plotted as "Distance Traveled into the Test Section". In this case there are no tangent or spiral conditions to be considered. Data points were stored and then plotted in a manner similar to that described above. The only difference was the number of points stored per 1946 Car (reference wheel) wheel revolution. In curves "A" and "C", one point per wheel revolution was stored. In curve "E", two points per wheel revolution were stored to ensure adequate resolution of the truck hunting frequency data curves.

As a result, a data sample (for all channels) was stored once every 4.345 feet (1/2 the circumference of the 1946 Car reference wheel). This provided about 10 data points per cycle of truck swivel at 70 mph which was deemed adequate to define the shape and magnitude of the curve, effect of curvature, and the underbalance speed on truck hunting.

All graphs show the leading truck of the car with the wheels in the following position relative to the curve.

1. L-4 wheel - Leading Outside
2. R-4 wheel - Leading Inside
3. L-3 wheel - Trailing Outside
4. R-3 wheel - Trailing Inside

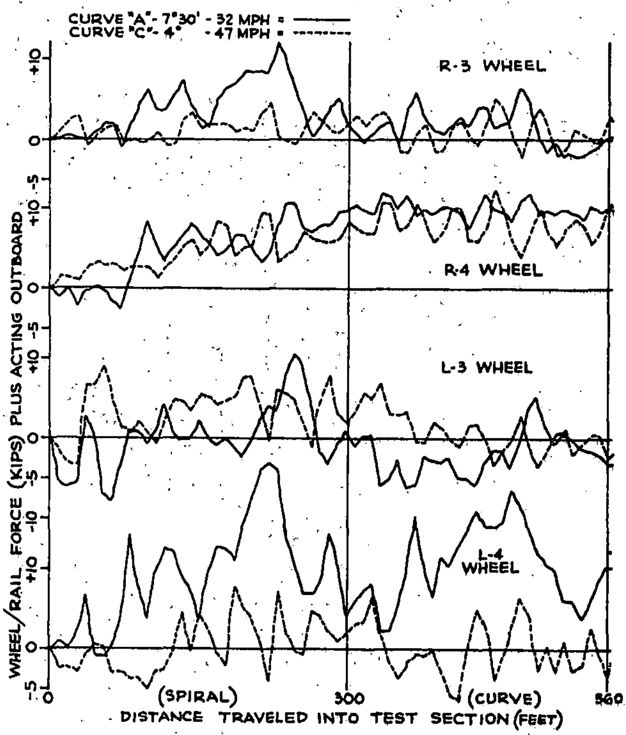
The truck used in the experiments was a standard 100-ton capacity, with 15" centerplace and with double roller side bearings.

DATA

Discussion up to this point has described the system and generally how it works. Attention is now focused on the output: What does it look like? And, how can it be used? Undoubtedly, the number of "best" ways to use the data can be expected to be a linear function of the number of individuals involved in the analysis. Nevertheless, the following represents ASF's current format for examining wheel/rail forces in service.

Data presented here are samples of actual data taken from a study conducted in 1975 at Pueblo. Displayed values are real, but not particularly important to this presentation. The main objective of this presentation is to show that a quantitative study can be based on the ASF wheel/rail force transducer system.

To provide some basis for comparison, a variety of test conditions was selected. Samples include light and loaded car wheel/rail forces through two curves of different radii. All four wheels of the leading truck under the test car are examined simultaneously. In addition to looking at lateral and vertical wheel/ rail forces, another graph is presented which shows the ratio between the two (L/V ratio). Finally, a sample of lateral wheel/rail force output under truck hunting conditions is presented. These samples are of sufficient diversity to illustrate that the ASF wheel/rail force system is capable of detecting changes in force characteristics, related to basic changes in the test configuration.



LATERAL WHEEL FORCES
260,000 LB. RAIL LOAD

FIGURE 9.

Figure 9 illustrates the simultaneous output at four wheel positions through the test section under loaded car conditions. Curves "A" and "C" are superimposed on this graph to show the relative affects of curvature at the different wheel positions. Both curves were traversed at their respective balance speeds.

Previous discussion defined the L4 wheel as the leading outside wheel in the test mode. Historically, this wheel position has been shown to be the most likely to wear under heavy curving circumstances. Figure 9 seems to point out clearly why this wheel would wear out first. However, it is also interesting to note that in the shallower curve (curve "C") the leading inside wheel (R4) experienced the higher lateral wheel/rail forces.

The trailing axle (R3 and L3 wheels) doesn't appear to provide any particular trend except that, perhaps spiraling into a 7-1/2° curve may result in higher forces at the R3 position than spiraling into a shallower 4° curve. The trailing axle output helps illustrate a point which perhaps should be expanded upon slightly. The point is, data of this type should be examined in terms of trends, as opposed to an exact quantitative analysis. Many factors can have a variable effect on the data, not the least of which is rolling friction, friction at the truck bolster center plate, etc.

A final note regarding figure 9 concerns the lateral forces in the spiral compared to those in the steady-state curve. The leading inside wheel apparently experiences a steady force buildup while passing through the spiral and then tends to maintain that force through the steady-state curve. The trailing-axle wheels generally tended to experience some force buildup through the spiral and then drop off as the car passed through the steady-state curve. The leading outside wheel experienced a larger buildup in the sharper curve than did any other wheel position. This might be expected since this is the main steering wheel when entering a curve.

Obviously, figure 9 contains more information than brought out above. However, the intent here is to illustrate one way the data can be interpreted. Summarizing this very brief analysis, it seems reasonable to state that, under these particular conditions, the leading outside wheel of the truck is considerably more sensitive to the degree of curvature than is any other wheel. It also seems fair to state that the leading inside wheel, although not particularly affected by the degree of curvature (at least for the samples presented) might also be expected to experience some wear under these conditions. This is, of course, assuming that lateral wheel/rail force and wear are relatable.

Thus, figure 9 presents information relating to wheel wear as well as potential rail wear. Obviously, other factors could be investigated in the same manner. For example, what are the effects of passing through curves considerably above or below balance speed? Or, what is the difference in lateral wheel/rail forces under long and short cars?, etc.

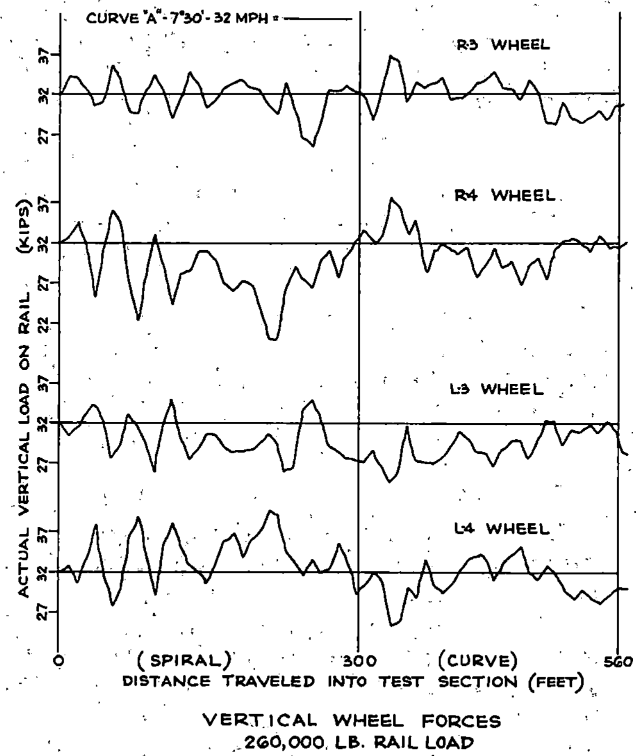
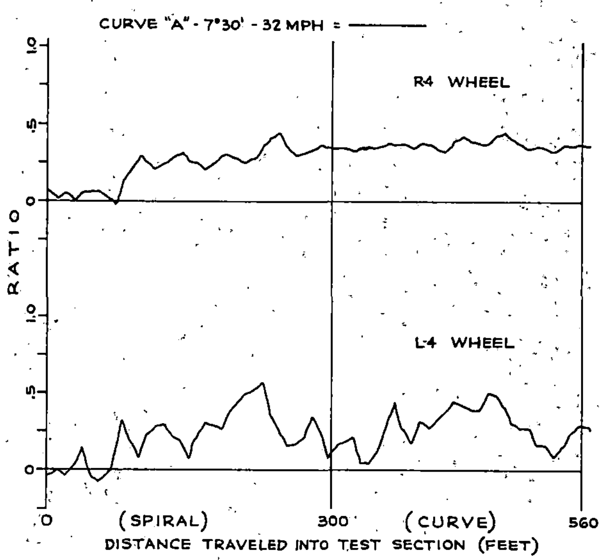


FIGURE 10.

Figure 10 illustrates measured vertical wheel/rail forces recorded simultaneously with lateral forces presented in figure 9. Only curve "A" data are presented in this sample.

Of particular interest is the L4 vertical wheel/rail force and its behavior relative to the lateral forces at the same position. The vertical force tends to fluctuate in the same general manner as the lateral force. This is not "cross talk" as might be expected (since considerable effort was expended to minimize this possibility). Instead, it is felt that the coincidental vertical peaks are caused when the leading outside wheel tries to climb the rail while passing into and through the curve. As the rail head travels into the throat of the flange, the leading outside wheel will try to move up. As the wheel tries to move up, it increases the force on the end of the side frame (which is being held down by the carbody weight) and thus results in increased vertical force at this point. Note that the R4 wheel (leading inside) has less tendency to exhibit this reflective behavior. This is reasonable since the leading inside wheel tends to roll away from the rail head, thereby minimizing the possibility of the wheel trying to climb the rail.



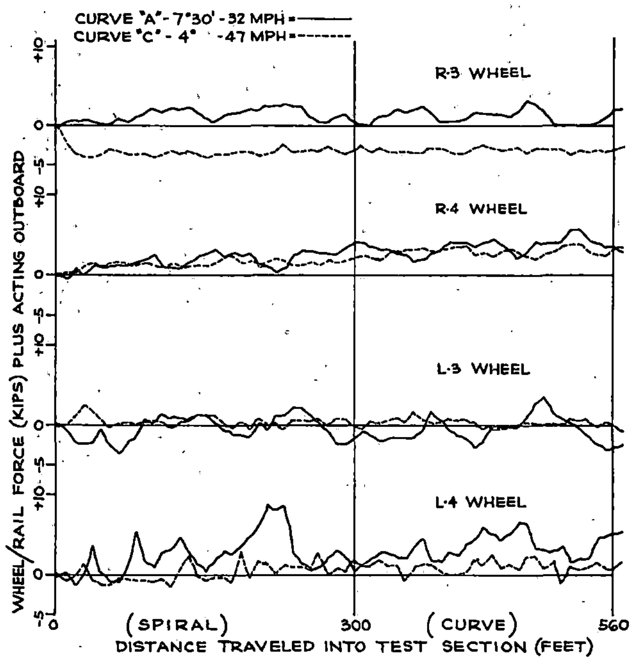
LATERAL/VERTICAL WHEEL/RAIL FORCE RATIO
260,000 LB. RAIL LOAD

FIGURE 11.

Combining the L4 wheel data from figures 9 and 10 leads to figure 11 which illustrates the instantaneous ratio between the lateral and vertical wheel/rail forces. Only the single wheel position is shown for illustrative purposes.

Figure 11 shows that, although the L/V ratio generally follows the lateral wheel/rail force trend, the change in the ratio may not be as drastic as expected when examining the lateral force alone. Again, it is felt that this reflects the tendency of the vertical force to increase with the lateral force, at least at the leading outside position. Obviously, if both the lateral and the vertical forces increase, the ratio between the two is going to tend to remain relatively constant. This would indicate that a predicted L/V ratio might tend to be higher than an actual L/V ratio measured at a position where a wheel flange was trying to climb the rail head.

The possibility of a misleading L/V prediction is probably best illustrated by comparing the R4 and L4 wheel positions, figure 11. For all practical purposes, the maximum L/V ratio for both sides of the leading axles are about the same. However, examination of figure 9 shows that the peak lateral wheel/rail force at the L4 position was at least twice as high as anything noted at the R4 position.

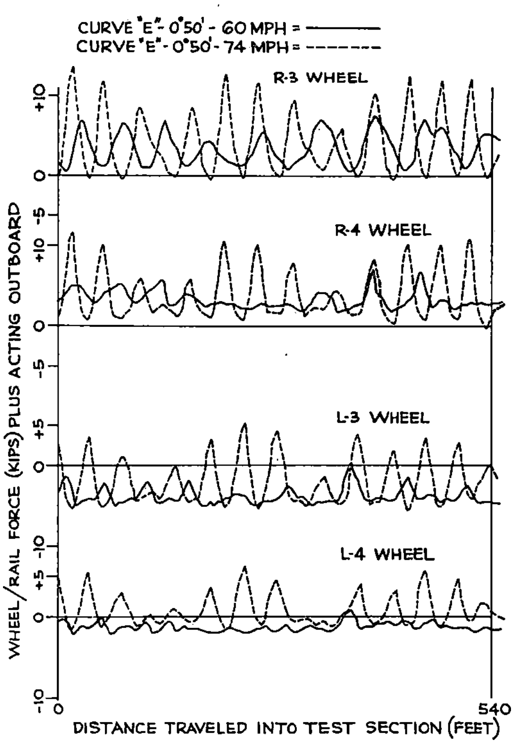


LATERAL WHEEL FORCES
86,000 LB. RAIL LOAD

FIGURE 12.

Figure 12 illustrates typical lateral wheel/rail forces under light car conditions. The test curves and wheel positions are the same as displayed in figure 9. Only the rail load has been changed.

Comparing the curves of figure 12 to figure 9, certain similarities are revealed under the two different rail loads. For instance, at the L4 position, the lateral force tends to build through the spiral and then maintain some positive value through the steady state portion of curve "A". The shape of the L4 data trace is remarkably similar under light and loaded car conditions. Of course, the magnitudes are considerably reduced under light car conditions. The same pattern of behavior can be seen at all wheel positions in curve "A". For instance, the leading inside wheel showed a similar force buildup through the spiral and into the curve under both rail load conditions. Only the magnitude of the response seemed to differ appreciably.



LATERAL WHEEL/RAIL FORCE DURING TRUCK HUNTING
86,000 LB. RAIL LOAD

FIGURE 13.

The similar data curve shapes are undoubtedly due to irregularities in the rail which caused the wheels to respond in the same general manner with each test pass. The magnitude of the response is, of course, dependent upon the weight and/or forces that might respond to these irregularities.

Figure 13 is the final data sample used to illustrate the capabilities of the wheel/rail force measuring system. Illustrated are lateral wheel/rail forces measured during truck hunting under light car conditions in a 0°50' curve at two different speeds. The approximate threshold for this particular arrangement was 60 mph. Lateral wheel/rail forces were considerably reduced from those encountered at 74 mph.

The slight bias (either positive or negative) noted at the various wheel positions is due primarily to the fact that both test speeds were considerably below the balance speed in the curve. Since the curve was superelevated for speeds over 100 mph, the test car had a natural tendency to run closer to the inside rail of the curve. Under these conditions, the inside wheels would tend to experience a positive or downhill bias while the outside wheels would tend to exhibit a negative bias. Figure 13 indicates that this was generally the case.

Data curves of figure 13 indicate the trailing inside wheel position (R3) experienced the greater force fluctuations through the test curve. Also, note the general phase relationship between the two wheels on the right side of the truck and the two wheels on the left. Relative force and phase relationships may differ significantly for a similar test run on tangent track.

WHEEL POSITION	MAX/MIN (KIPS) 260,000#	86,000#	RMS (KIPS) 260,000#	86,000#
	CURVE A FORWARD			
R-3	10.0/-3.6	5.4/2.2	3.2	3.5
R-4	13.8/-0.8	7.8/1.4	10.0	4.7
L-3	16.6/-1.8	7.8/0.4	6.3	3.9
L-4	22.0/-2.4	8.6/-1.2	10.2	3.7
	CURVE C FORWARD			
R-3	3.0/-4.4	5.4/1.4	1.9	2.3
R-4	11.4/-0.8	5.8/2.0	6.1	4.1
L-3	9.6/-2.8	4.2/0.8	4.2	2.2
L-4	10.6/-3.4	3.8/-0.4	4.5	1.9

LATERAL WHEEL FORCE SUMMARY

FIGURE 14.

Figure 14 summarizes some of the aforementioned graphed data and presents it in tabular form. Even in this very simple form, noticeable, and perhaps significant trends are evident.

Admittedly, the foregoing discussion of results has been very brief and general. However, the purpose of the discussion was not to present specific results and findings, but to illustrate the capabilities and the information potential provided by the ASF wheel/rail force measurement system.

ACKNOWLEDGEMENTS

ASF wishes to express their appreciation to Mr. L. A. Peterson, then with the B&LE Organization for his initial contribution which led to ASF's wheel/rail force system.

THE DR-1 RADIAL TRUCK,

A SIGNIFICANT ADVANCE IN FREIGHT CAR TRUCK TECHNOLOGY

BY

E.C. BAILEY
N. CALDWELL
P.P. MARCOTTE

During the past two years, significant progress has been made in the development of adapting radial steering arms to a conventional three-piece truck to improve curve tracking characteristics and provide increased stability at maximum operating speeds. The result is a new truck design which has been designated the DR-1 Radial Truck. Extensive tests have been conducted on the DR-1 which provided conclusive proof that the steering arms eliminate hunting within the operating speed range and greatly improve curve negotiation.

On the economic side, actual cost information has been compiled on some of the operational parameters affected by the improved tracking performance of the truck, such as wheel and rail wear and fuel consumption. This data has provided valuable insight into the significant savings that can be realized from the application of the self-steering principle.

The work done to date has demonstrated the DR-1 to be technically and economically feasible. The next step in the program calls for selected revenue service applications of the production version.

The purpose of this paper is to review the progress made on this project in the past two years, explore some of the revealing data on the economics of improved truck performance and outline the plans for the introduction of the DR-1 into field service.

INTRODUCTION

Despite the many years of successful performance behind the conventional three-piece freight car truck, it has some performance limitations which are increasingly evident in today's railroad operations. The major deficiencies fall into two classes of vehicle guidance, curving and high speed tracking.

The first problem involves the inability of the truck to provide satisfactory curving performance without excessive lateral forces on the rails. Heavy rail wear on curved track and high rates of wheel flange wear are now being experienced in many types of service because of these forces, particularly on unit trains. More explicitly, rail wear is in the form of gauge face and head wear on the high rail and head flow and corrugations on the low rail. Increased track replacement and gauge and alignment maintenance are direct consequences of inadequate curving performance.

The second deficiency lies in the inability of freight car trucks equipped with roller bearings to provide stable guidance on tangent track, particularly on light cars traveling in excess of 40 to 50 mph. This condition is more pronounced when the wheel profiles

---

Ed Bailey is Director of Engineering and Quality Assurance for the Transportation Equipment Division of Dresser Industries Inc., Depew, NY. He received his B.S., M.S. and MBA in Civil Engineering from the University of Missouri.

William Nelson Caldwell is Senior Research Engineer for the CN Railway. In 1964 he was awarded an Athlone Fellowship and took an educational leave of absence from the CN to pursue post graduate studies at the Imperial College of Science and Technology, London, England.

Pierre P. Marcotte is a Research Engineer for the CN, having graduated in Mechanical Engineering from Ecloe Polytechnique University of Montreal in 1966. On an Althone Fellowship, Marcotte obtained a Master of Engineering Degree from Sheffield University in England.

have acquired the degree of wear normally found after about 25,000 miles of service for 1 in 20 taper wheels. The instability manifests itself in the form of lateral cyclic vibration commonly referred to as "hunting". This behavior ultimately leads to extensive wear and damage to truck and carbody components and identifiable damage to lading.

In the search for a cost effective solution to the curving and stability problems, several years ago Dresser began investigating various approaches proposed to alleviate them. There were some modifications available which eliminated the hunting problem associated with the conventional three-piece truck for speeds up to approximately 60 mph. Also, some new types of trucks were in existence or being developed that had been shown to be stable during high speed tracking trials. However, these trucks did not specifically address the curving problem. Therefore, they only partially meet the performance requirements for present and future freight operations.

At the same time, others in the industry were directing their attention to the curving problem. Consequently, a cooperative development program was undertaken in early 1975 by the Canadian National Railways, Dresser Transportation Equipment Division, Dominion Foundries and Steel, Limited and Railway Engineering Associates. The purpose of the project was to develop a commercially viable truck based on considerable conceptual and investigative work performed by Mr. Harold List of REA on the principle of radial steering of truck wheelsets.

Since North American railroads have accumulated much successful background experience with the conventional three-piece truck, and the supply industry is well equipped for its manufacture and maintenance, it was decided that the trucks basic components should be preserved. After a thorough evaluation of numerous concepts, the one selected as most practical was the application of a radial steering mechanism to a conventional truck which has resulted in the development of the new DR-I Radial Truck, as shown in figure 1.

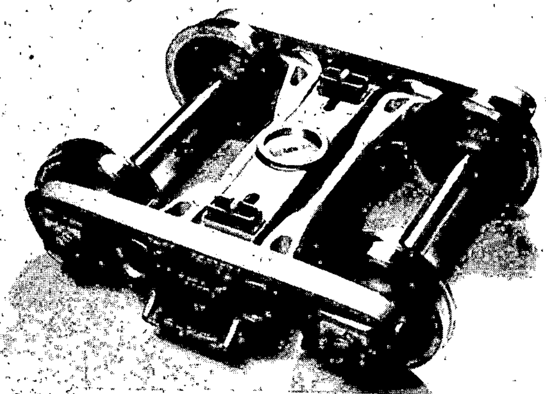


FIGURE 1.
MODEL OF DR-1 TRUCK

DESIGN CONCEPT

The DR-1 consists of the standard side frame and bolster assembly, with conventional suspension components, plus a steering arm arrangement made-up of two "C" shaped arms. These are attached at their extremities to the bearing adapters and universally connected to each other by center posts through one of the existing holes in the bolster. Resilient shear pads are applied over the bearing adapters to permit radial positioning of the wheelsets. The steering arms, thus, have been designed such that they can be retrofitted to existing trucks with standard components and no modifications are required to the carbody. Compatibility is thereby maintained with existing truck maintenance support facilities and practices. This offers distinct advantages over other steering trucks that utilize non-standard side frames and bolsters and other nonconventional, rigid frame trucks.

To appreciate the benefits to be derived from the steering feature, the curving behavior of various types of trucks must be considered. To achieve flange free curving performance, the axles must align themselves radially to the curves. Existing freight trucks naturally assume a parallelogrammed or lozenged shape, illustrated in figure 2, such that the angle of attack between the vertical plane of the wheel and the direction of travel is large, causing high lateral forces and wear. In rigid

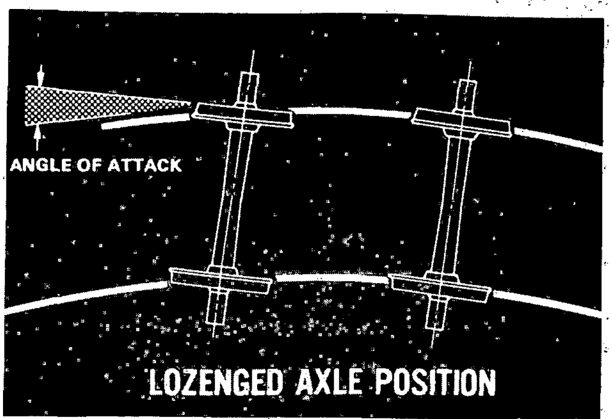


FIGURE 2.
LOZENGED AXLE POSITION

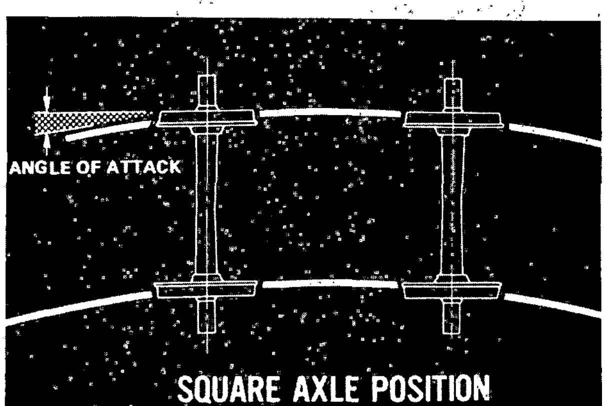


FIGURE 3.
SQUARE AXLE POSITION

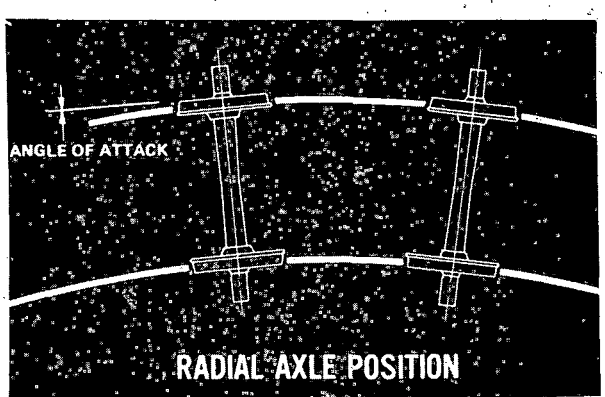


FIGURE 4.
RADIAL AXLE POSITION

or square trucks the condition is improved, but the angle of attack is still significant, as shown in figure 3. The DR-1 Radial Truck inhibits any tendency of the wheelsets to assume a lozenged position in curves, since the axles must maintain the proper radial alignment as shown in figure 4, because of the constraint provided by the interconnected steering arms.

Before truck design concepts could be generated which would produce the desired performance, it was first necessary to conduct an extensive study to gain a thorough understanding of the factors that affect a truck's curving and tangent track stability behavior. Early theoretical analyses were carried out by CN Rail Research and REA using computer modeling techniques to evaluate the tracking characteristics of a self-steering type truck. The two most important parameters that control tracking performance, which can be modified by the designer, were ultimately identified. These are interaxle lateral and yaw stiffness. Assuming the predominant condition of service worn wheels, computer optimization of these design parameters showed that both the desired high speed tracking stability of empty cars and radial alignment of loaded cars in curves could be attained if the appropriate combination of lateral and yaw stiffness was used to interconnect the wheelsets.

To clarify these terms, interaxle lateral stiffness is defined as the resistance of the truck assembly to relative lateral motion between the wheelsets when equal and opposite forces are applied, as shown in figure 5. In the DR-1 truck, this stiffness is achieved by the cast steel steering arms. Increased interaxle lateral stiffness has a major effect on hunting stability of conventional trucks.

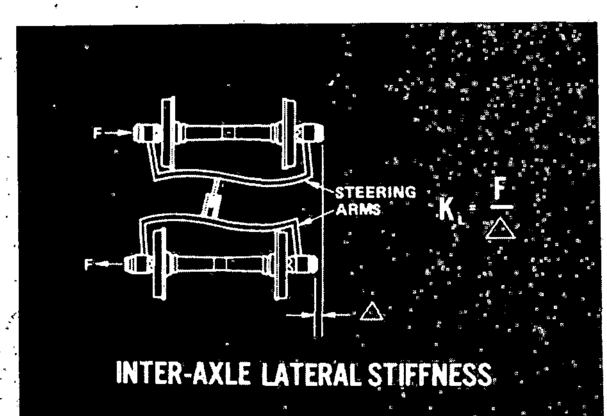


FIGURE 5.
INTER-AXLE LATERAL STIFFNESS

Similarly, interaxle yaw stiffness is the resistance of the truck assembly to relative yaw motion of the wheelsets when equal and opposite moments are applied, as shown in figure 6. In the DR-1 it is achieved by elastically coupling the wheelsets to each other through the truck side frames. This permits fore and aft longitudinal movement of the journal bearings under resilient control of the elastomeric pads, which allows radial positioning of the wheelsets in the frame. The reduced interaxle yaw stiffness of the DR-1 compared to a conventional truck is the most influential factor on curving performance since it permits radial steering action to occur. The forces developed to steer the wheelsets are generated at the wheel/rail interface which depend to a large degree on the wheel tread profile.

The results from the computer model analyses showed that interaxle yaw stiffness not only influences curving characteristics, but also has a considerable effect on hunting stability. Conversely, the interaxle lateral stiffness has a beneficial effect on curving. Therefore, curving performance and critical hunting speed are interdependent which requires that a compromise be made between the combination of lateral and yaw stiffnesses. On the DR-1, the first priority was to achieve high speed stability up to a minimum of 80 mph since dynamic instability on tangent track is highly undesirable in the operating speed range. Fortunately, the yaw stiffness required for adequate hunting control at this speed yields a very substantial improvement in curving performance that is sufficient for flange free curving on most mainline curves.

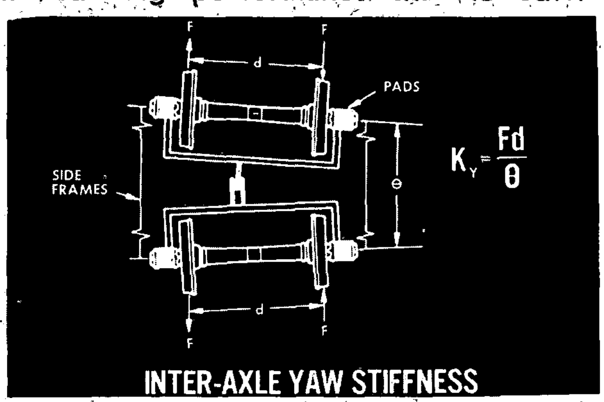


FIGURE 6.
INTER-AXLE YAW STIFFNESS

DEVELOPMENT/TEST PROGRAM

In addition to the theoretical analyses, an extensive series of tests have been conducted over the past four years to verify the effectiveness of the steering arm concept. The first two years consisted of experimenting with an all new steering truck which has been reported on previously. In the latest series of tests, which will now be discussed, a carset of retrofit steering arms was applied for the first time to standard 100-ton freight car trucks by Dresser in 1975. The steering arms were fabricated from structural steel sections to expedite construction, as shown in figure 7. This truck was designated the DR-1 Radial Truck.

Curving and stability tests were conducted on the DR-1 truck in early 1976 at the DOT Test Center in Pueblo, Colorado. Although the DR-1 demonstrated that the curving characteristics were significantly improved, this prototype version did not achieve the stability we had anticipated as the trucks hunted at approximately 60 mph. Static tests of the truck were conducted to determine the interaxle lateral stiffness. These tests showed that the stiffness was considerably below the minimum value established by the computer analysis because of large torsional flexibility in the steering arms.

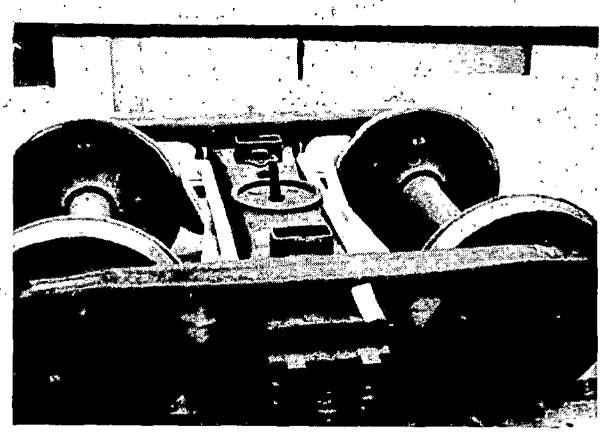


FIGURE 7.
DR-1 TRUCK WITH FABRICATED STEERING ARMS

The prototype was modified to increase the lateral stiffness to the maximum practical level for this particular fabricated design. When the truck was reassembled, dynamic lateral and yaw stiffness tests were performed by CN Rail Research, as illustrated in figure 8. The tests showed that the interaxle lateral stiffness was considerably improved over the level available in the Pueblo tests but still was not as high as it should be. The interaxle yaw stiffness was slightly higher than the design value because standard elastomeric pads were used. The dynamic test procedure was necessary to insure that the stiffnesses of the complete truck assembly were representative of those exhibited during dynamic over-the-road operation.

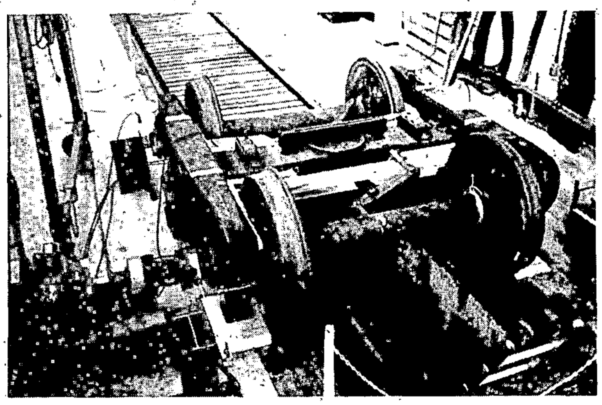


FIGURE 8.
DYNAMIC STIFFNESS TEST FIXTURE

High speed, light car hunting stability tests were then made by CN Rail Research on both a High-Cube box car and a torsionally flexible open top hopper car. Each car was alternately equipped with conventional trucks with new wheels (6,000 miles of service) and then with DR-1 trucks with worn wheels to determine the carbody input influence.

Runs were made on a section of continuously welded tangent track starting at 40 mph and increasing in 5 mph increments up to a maximum test speed of 75 mph. Laterally oriented accelerometers mounted on each end of the cars were used to identify hunting instability. Two television cameras monitored the action of a set of wheels on the rails with a continuous display on a split screen in the instrument car.

Unlike the conventional trucks which hunted on both types of cars beyond 55 mph, the DR-1 remained stable up to approximately 75 mph where intermittent hunting occurred. The results are illustrated in figures 9 and 10. The fact that the hunting onset speed was significantly improved by the DR-1 on both cars, confirmed the theoretical prediction that the higher interaxle lateral stiffness increases the critical speed. Since the stiffness was not optimal, it was not surprising that the hunting threshold was still below the desired 80 mph level. Also, the increase in the critical speed of the DR-1 in these tests compared to the original prototype test in Pueblo where the lateral stiffness was too low, further illustrates the importance of having the appropriate stiffnesses predicted by the computer model. In addition, these tests indicated that a cast production design with optimal stiff-

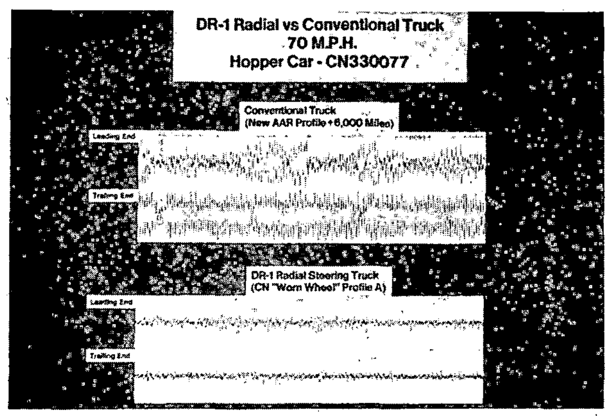


FIGURE 9.
ROAD TEST RESULTS FOR
OPEN TOP HOPPER

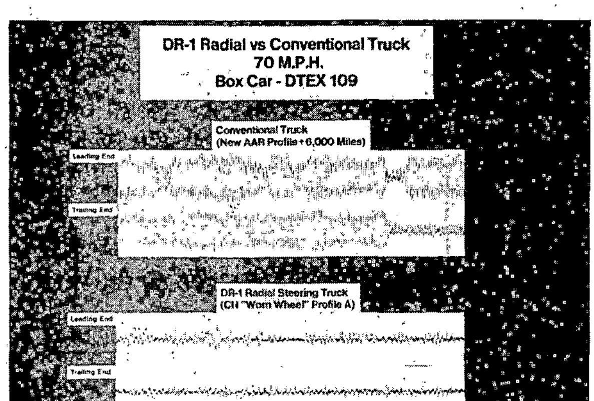


FIGURE 10.
ROAD TEST RESULTS FOR
BOX CAR

nesses would provide the desired stability to at least 80 mph.

A few additional comments should be made at this point regarding truck performance. While it has been established that a suitable combination of interaxle lateral and yaw stiffnesses is the key to truck stability and curving, proper operation can be obtained over a relatively wide range of values which can be easily controlled in the manufacturing process. Still, the minimum value of interaxle lateral stiffness required is well beyond that of conventional freight car truck designs. Secondly, since both the DR-1 and the conventional trucks behaved consistently under two very different types of cars, it was concluded that the carbody does not contribute significantly to the critical speeds observed on conventional freight trucks as long as the trucks themselves remain stable. This agrees with data from previous CN Rail Research tests.

After demonstrating that stable operation could be achieved with the DR-1, its curving characteristics were investigated using fully loaded cars. Tests were performed with the DR-1 and conventional trucks on a 5° and a 12° curve to measure the lateral track force. Numerous passes were made in each direction over an instrumented tie plate. The DR-1 reduced the lateral track force by an average of 60% on the 5° curve and by 20% on the 12° curve.

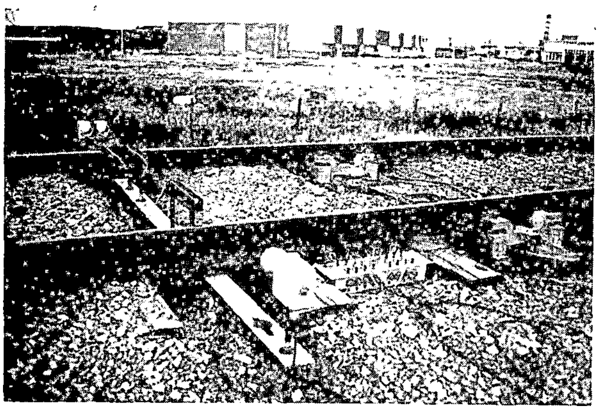


FIGURE 11.
ANGLE OF ATTACK MEASUREMENT INSTRUMENTATION

A laser beam, similar to the device shown in figure 11, was then set up to measure the exact angle of attack of the leading wheels on the 5° curve. This measurement showed that the DR-1 reduced the angle of attack by 75% compared to the conventional truck.

Since well known theoretical works as well as independent studies conducted by CN Rail Research indicate that flange and rail wear are proportional to the product of the lateral track force and the angle of attack, wheel flange wear and track gauge face wear on 5° curves should be reduced to approximately 1/10 of that caused by conventional trucks with AAR contour wheels, i.e., .40 x .25 = .10.

Over a thousand miles of tests were then made with the DR-1 truck to check its total performance. During these runs, the trucks were instrumented to measure curving, strain gaged to measure forces in critical areas of the steering mechanism and continuously monitored for hunting using the television cameras and accelerometers. At several intervals during the runs, the trucks were checked for any signs of unusual wear or potential problems. Performance during these tests confirmed satisfactory operation of the DR-1 truck in all respects.

One noteworthy test was made during these runs to record truck steering during brake applications in curves and determine whether braking interferred with steering action. It was discovered that the truck continued proper steering during light and medium braking. When full service or emergency brakes were applied, normal brake shoe forces pressed the bearing adapter against the outer pedestal stop lugs. This forced the truck to adopt a square configuration which is more desirable than the lozenged attitude of conventional trucks in curves.

The elastomeric pads have also been tested at both normal temperatures and extremes of hot and cold, without structural deficiencies. It should be noted that the pads used between the bearing adapter and side frames in the DR-1 design have been successfully applied for several years in similar railway applications. By eliminating hunting, the DR-1 will considerably extend the service life of these pads beyond current experience in conventional trucks.

After the successful completion of these tests, we were confident that we had a viable solution to the stability and curving problems. In the next phase of development, the final design of cast steering arms was completed by Dresser utilizing the forces measured during the Canadian National road tests. This involved generating and repeatedly adjusting a finite element model to simultaneously achieve the required stiffnesses and strength. The initial production units of the steering arms were then cast by Dominion Foundries and Steel, assembled into trucks and dimensionally checked for adequate clearances under the most adverse operating conditions as shown in figure 12. Comprehensive testing of the truck is following a plan similar to the one used on the fabricated prototype.

First Dresser performed a preliminary static test to ensure the arms had the optimum lateral stiffness. CN Rail Research later confirmed during dynamic laboratory testing of the truck that the lateral and yaw stiffnesses were within the range of values established in the computer simulation. Strain gage data was recorded during the static test as a preliminary check on the strength of the arms. This data was also used for correlation with a detailed finite element model of the assembly.

Stability tests have been conducted by CN Rail Research on the DR-1, this time with empty 100-ton coal gondolas which will be used later for service tests of the truck. While the conventional trucks began hunting at approximately 48 mph, the DR-1 was completely stable at 80 mph. Additional curving tests are planned and will follow the same sequence as the earlier fabricated version.

During these tests, strain gage data on the steering arms will be recorded so that it is possible to determine the operating environment forces and develop a load spectrum to be used in a fatigue test of the arms. Sufficient cycles will then be applied to the steering components to accurately simulate 40 years of service to prove no structural deficiencies exist.

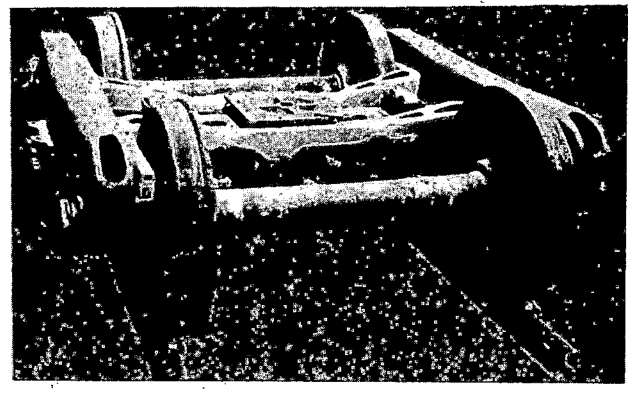


FIGURE 12.
PRODUCTION VERSION OF THE DR-1 TRUCK

As data was accumulated from the numerous laboratory and road test series and correlated with the results from the computer model, a great deal of confidence in the understanding of the DR-1 Radial Truck concept developed. Achieving both curving performance and stability, while preserving the desirable characteristics of the three-piece freight car truck, represents substantial progress in rail truck efficiency. Also of great importance is the fact that a validated computer model has been developed which accurately predicted the performance of the truck and will serve as a unique and valuable tool for future research.

ECONOMIC JUSTIFICATION

Although the performance of the DR-1 has been proven, a discussion of its benefits would not be complete without consideration of the economics involved. With this in mind, CN Rail Research compiled an extensive and impressive set of data which permits the following evaluation of the potential savings that can be realized from the application of the DR-1. These calculations have been compared with other financial information available to the railway industry and, in contrast, they appear to be conservative.

Rail Wear

The major thrust of the DR-1 truck design is to improve curved track wear. From a 1975 wear study conducted by Mr. F. E. King of the Canadian National Railroad on their British Columbia Southline, it was shown that a savings of $180 per million gross ton miles (MGTM) of transportation was attainable if curved track wear could be reduced to the level presently experienced on tangent track. This assumes rail replacement costs at the 1975 figure of $100,000 per mile.

Wheel Wear

In this same study, wheel wear was evaluated based on the practice of using a two-wear wheel which normally gives 340,000 miles of service on this line. Assuming the life of the wheels would be increased by 30% to 442,000 miles, wheel wear savings were estimated at $18 per MGTM based on a 1975 cost of $2,000 per carset of wheels. Other estimates of the cost to replace a carset of wheels have been as high as $3,600.

Fuel Savings

Another benefit of an improved curving truck is reduced rolling resistance in curves. A train performance calculation (TPC) was made to evaluate the maximum fuel economy attainable if the rolling resistance in curves was reduced to that experienced on tangent track. A loaded coal train of 98 gondola cars was moved in a TPC computer simulation from Winniandy coal mines in Alberta to the Neptune Terminals in Vancouver, British Columbia (680 miles) and returned empty. The rolling resistance of cars on standard freight car trucks is generally assumed to be 0.8 lb. per ton per degree of track curvature, i.e., the curving drag is equivalent to climbing a 0.04% grade for each degree of track curvature. This value of curving resistance used for one run and zero curving resistance was used for the other.

The results show a fuel consumption reduction of 1,308 gallons for the round trip, with 10,620 gallons consumed on the standard train with conventional trucks. At 42 cents a gallon for diesel fuel, this would result in a $550 saving for each round trip. The transportation service supplied in moving the consist for the round trip from Winniandy to Neptune is 11.56 MGTM which includes the weight of the locomotive and caboose. Hence, the maximum fuel savings is: $550/11.56 = $47.58 per MGTM.

Economic Summary

At present, coal gondola cars in this service average 60,000 miles/yr. based on car mileage records. One-half the trip is loaded at 130 gross tons and the return trip at 30 gross tons. Hence, each car supplies 30,000 (130 + 30) = 4.8 MGTM of transportation each year. The potential annual savings for rail wear, wheel wear and fuel are therefore:

Rail Wear	$180 per MGTM
Wheel Wear	$ 18 per MGTM
Fuel	$ 47 per MGTM
	$245 x 4.8 = $ 1,176 savings per car per year

Based on a present value calculation using an economic life of 10 years and a 15% cost of capital, the $1,176 per year saving would justify investing $5,900 more per carset for DR-1 trucks versus conventional trucks to gain just the savings in rail wear, wheel wear and fuel consumption.

Other Savings

In this study, no economic allowance has been claimed for certain "soft dollar" benefits due to decreased derailments in curves brought about by lower curving forces or those caused by high speed hunting of empty cars. Nor has any allowance been claimed for decreased track alignment and gauge maintenance or improved track occupancy during such maintenance.

Additional tangible savings would be derived by eliminating hunting from a reduction in truck component wear, carbody wear and lading damage which is know to be worth at least $250 per carset in capital cost, judging from the cost of existing anti-hunting devices. Also, there is a distinct possibility of increased car utilization from operating the empty car trains at higher speeds.

FIELD APPLICATION

This year, a limited number of carsets will be placed on several railroads in the U.S. which have representative service. A similar number will be installed by the CN. In 1978 Dresser will be prepared to place at least another 100 carsets in service. The CN plans to retrofit some existing unit trains as the DR-1 proves it will produce the anticipated savings.

SUMMARY

It has been proven by computer modeling and verified by extensive tests that the DR-1 Radial Truck represents a significant advance in freight car truck technology. The DR-1 is an economically justifiable means of effectively controlling the curving behavior and stability of conventional freight car trucks.

We are now at the point where introduction of the DR-1 is imminent. We, as an industry, are in a position where a practical solution is available to solve problems that have been plauging us for years.

STACKED CONTAINER CAR FOR LAND BRIDGE

BY

L. H. NATIONS
R. H. BILLINGSLEY, JR.

The strong need for improved efficiency in rail transshipment of freight containers from container ships prompted the Southern Pacific to undertake the development of a container car specialized for "Land Bridge" operations. The fundamental objective to reduce car lightweight was obtained by stacking containers as compared to usual end-to-end loading. An additional benefit was substantially reduced train lengths. The Southern Pacific is also developing concepts which would retain TOFC capability as well as improve COFC efficiency. ACF, under contract to Southern Pacific, has designed, built, and tested a prototype design version of the stacked container car concept. The prototype design demonstrated the weight reduction objective feasibility while at the same time conforming to all AAR/FRA design and safety requirements. The only restriction involves height, which is the same as multi-level auto rack cars.

The need for a lighter weight car specialized to handle containers has been building for some time. Before the advent of landbridge and minibridge all-container movements and the growth of a network of terminals with lift-off/lift-on handling equipment, the logistic versatility of the all- purpose piggyback car inhibited developing COFC-only cars. Now, however, there is definite justification for developing such cars to achieve maximum economy operating point-to-point within the network.

Reducing car weight for more economical transportation is by no means new or novel; it has been a popular theme for decades. In this case, however, it is not a matter of skinning down existing structure with the attendant complications and hazards of potential fatigue problems, rather it is the selection of an alternate structure. It is an opportunity to take a fresh outlook and to conceptualize taking advantage of many years of piggybacking development and experience behind us.

The essential lift-on/lift-off nature of container handling and the point-to-point character of volume container movements naturally channels weight reduction thinking along two paths, skeletonizing and articulation. The car structure can be skeletonized to the extent that its function is limited only to providing point supports and restraints at standardized container corner locations and to protecting the containers from the longitudinal loads of the yard and train environment. Articulation reduces car weight both by eliminating portions of end structure and reducing the number of trucks.

Investigating container car configuration alternatives, the SP found that a concept of stacked containers nested in a drop-center or "well" type car unit possessed the most substantial potential for decreasing car weight. As shown in figure 1, the stacking concept in a three-section articulated configuration has a feasible 120,000-pound

Lloyd H. Nations is Assistant General Manager, Intermodal Traffic (Services) for the Southern Pacific Transportation Company, San Francisco, Calif. Nations received his B.S. degree from Texas Western, performed graduate work at the University of Texas and later attended Harvard University.

Robert H. Billingsley, Jr., is Senior Director - Technical for AMCAR Division, ACF Industries, Inc., St. Charles, Mo. Billingsley received his B.S. degree in Mechanical Engineering at the University of Florida in 1952.

car lightweight and a 171-foot length over pulling faces.

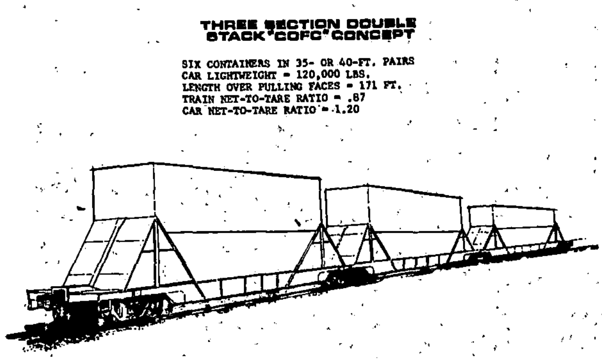


FIGURE 1.

By comparison, the three standard all-purpose COFC-TOFC cars needed to transport the six containers would have a 210,000-pound lightweight and 276 feet over pulling faces. Thus, the stacking concept, in addition to its substantial reduction in lightweight, also permits a much shorter train length for the same payload. Stated another way, for a given length of train, or terminal dock, the stacking concept provides a 60% increase in lading capacity compared to conventional COFC-TOFC.

The index used by the SP for evaluating the degree of effective weight saving is the net-to-tare ratio, that is, the ratio of payload weight to the tare weight (car plus containers or trailers lightweight). For true comparison, the "weight of power" must be included in determining the tare weight on a car unit basis. For "weight of power", the SP uses 3 HP/trailing ton, which equates to about 342 locomotive pounds per trailing ton, based upon expedited trains powered by SD 45's (3600 HP; 410,000 lbs. R.W.). Other ratios of HP/ton and size and weight of locomotives will materially affect this allotment of locomotive weight in the calculation of the train net/tare ratios.

In Table I, the three-section stack-container car concept is compared to conventional COFC-TOFC on a net-to-tare ratio basis. For the comparison, 7,500 pounds each is used for container light-weight and 12,000 pounds for trailer lightweight were applicable. Typical lading weight is based upon 33,000 pounds of product per container. The three-section stack-container unit, with a 0.87 train net/tare, represents a 45% improvement over the 0.60 train net/tare of the conventional 89-foot all-purpose car in container movement. On a design basis, excluding locomotive weight, the car net/tare ratio for the three-section stack container unit is 1.20 as compared to a conventional 89-foot container car with a car net/tare ratio of .78. On this basis, a 54% improvement can be realized with the three-section double stack design.

40-Foot Dry Container and Trailer Movements - Net Tare Comparison

	TYPE OF MOVEMENT			
	Standard TOFC-COFC		Articulated Three Section Double Stack	Net/Tare % Improvement Articulated Three Section Versus
	TOFC	COFC	COFC	COFC
Car Tare Weight	70,000	70,000	120,000	
Container Tare Weight	--	15,000	45,000	
Trailer Tare Weight	24,000	--	--	
Total Tare Weight	94,000[1]	85,000	165,000	
Locomotive Weight	27,300[2]	25,800	62,000	
Total Tare Weight	121,300	110,800	227,000	
Product Weight	66,000	66,000	198,000	
Train - Net/Tare Ratio	.54	.60	.87	45%
Car - Net/Tare Ratio[1]	.70	.78	1.20	54%

[1]Excludes locomotive weight.

[2]Based on 3 HP/ton, SD45 type locomotives (3600 HP, 410,000 pounds) = 342 pounds per trailing ton. Output, weight, and HP/ton ratio will materially influence this locomotive weight allotment in the determination of train net/tare ratio.

TABLE 1.

From these numbers you can see the dramatic change in net cargo carrying capacity that can be achieved using the double stack configuration. The main restriction on this car will be because of height. With two 8½-foot high containers, the top clearance will be 18½ feet above top of rail (empty containers), and this will permit going any place that multilevel rack auto cars will go.

The SP is also pursuing other concepts to utilize the advantages of articulation. In these other concepts, one goal is to retain TOFC capability in dedicated train, lift-on/lift-off, landbridge type movements. These concepts include two-section and four-section articulated COFC-TOFC arrangements.

The four-section COFC-TOFC concept, figure 2, features 47-foot sections with fixed trailer hitches and lightweight of 100,000 pounds. End trucks are 50-ton, interior trucks, 70-ton. This arrangement will handle either a 40-foot container or a 40- or 45-foot trailer on each section, including nose refrigeration units. For COFC, the train net/tare ratio for an 89-foot car of .54 for TOFC and .60 for COFC, an improvement of 24% for TOFC and 27% for COFC by use of the four-section car. In length of train there is no significant difference.

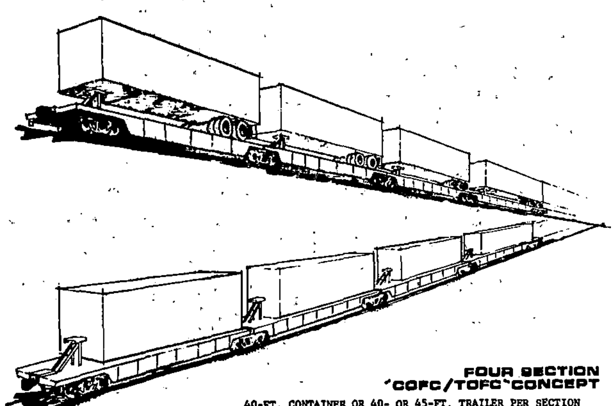


FIGURE 2.

The two-section concept arrangement, figure 3, will accommodate three 40-foot containers, with the center container straddling the articulation joint, or two 40- or 45-foot trailers secured on retractable trailer hitches. The middle container would mount on swiveling bolsters, one of which could also move longitudinally, to permit angling and length variations accompanying curve negotiation by the car. The double unit lightweight is about 72,700 pounds, yielding a train net/tare for containers of .77 and for trailers, .53 train net/tare. For COFC, the two-section concept with a length over pulling faces of about 130 feet does reduce train length slightly, about 3%, but in operating TOFC, it suffers a 30% train length handicap versus the conventional all-purpose car.

In summary, both these COFC-TOVC concepts retain some significant benefit in increased net-to-tare ratios for container movements, although less than the stack-container concept. For conventional TOFC loadings, the net/tare ratio is improved for the four-section concept. Regarding train length, there is no substantial change from the conventional, when handling containers, but some penalty is incurred in the two-section case when running trailers. These appear to be the limitations imposed to obtain TOFC versatility.

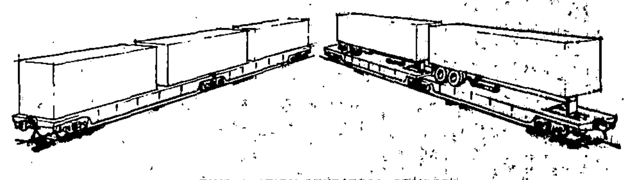
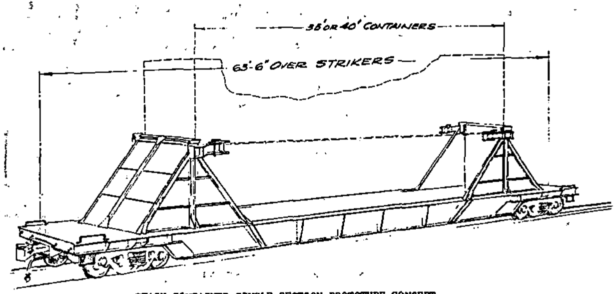


FIGURE 3.

The three-section stacked container concept was selected by the SP as the first for design development. For development purposes, the SP contracted with ACF to design, build and test a prototype single section version of this concept. The prototype was to be designed to stack pairs of 40-foot (ISO) and 35-foot (nonstandard) containers, and the design had to be compatible with SP lift-on/lift-off equipment at port terminals.

The prototype version as proposed for development by ACF is shown sketched in figure 4. A 50,000-pound lightweight target was set for the prototype, which would give it a .76 train net-to-tare ratio, calculated on the same basis as in Table I. The design was to conform to all applicable AAR Specifications and FRA Safety Standards.

From ACF's point of view, the "well" type car concept provided an attractive opportunity for innovative weight-savings design. The car is a center sill-less car (figure 4), a type with which ACF has substantial experience. The bulkheads which are primarily for longitudinal and lateral restraint of the upper container also effectively function to structurally augment the transition section between the "well" and the stub sills. The side sills, forming the sides of the container "well" also provide torsional rigidity for the car. Supporting the container as closely as possible to the car bottom makes the stacked concept possible. The result is minimizing of both car weight and length.



STACK-CONTAINER SINGLE SECTION PROTOTYPE CONCEPT

TWO 35- OR 40-FT. CONTAINERS
CAR LIGHTWEIGHT = 50,000 LBS.
LENGTH OVER PULLING FACES = 68 FT.
TRAIN NET-TO-TARE RATIO = .76
CAR NET-TO-TARE RATIO = 1.02

FIGURE 4.

As a new car design, this was an opportunity to apply design technology based upon the body of experience and environmental data background at ACF. Fatigue failure is a major pitfall of weight reduction. In recent years, ACF has concentrated on developing data and technology to design more closely to material limits. During the basic design phase, and again during post-test analysis, the design is carefully reviewed and monitored to assure fatigue failures are prevented. Our design approach is identical to that outlined in the Interim AAR Guidelines for Fatigue Analysis of Freight Cars, which has recently been developed in Phase II of the cooperative Track-Train Dynamics Program.

The design phase was initiated in September 1976 after the basic criteria and load conditions for design were agreed upon. A maximum gross container weight of 75,000 pounds was selected for design, a not uncommon maritime condition, as opposed to the ISO standard maximum of 67,200 pounds. The car was to be designed for seven basic load conditions, which are:

a) Light car - no containers
b) One lightweight (empty) container
c) Two stacked lightweight (empty) containers
d) One gross weight container
e) Two stacked gross weight containers
f) One gross weight container stacked on a lightweight container
g) One lightweight container stacked on a gross weight container

The light car condition involves empty-load brake requirements, truck hunting, upper coupler height limitation, and other lightweight design criteria. The two empty container's condition sets the maximum vertical clearance height for the car. The two fully loaded containers govern the structural strength design and the other limit of the braking requirements and lower coupler height limit. A loaded container on top of an empty, although considerd a rare case because crews are instructed not to load in this manner, is critical due to the very high resulting center of gravity.

ACF gave special consideration to problems anticipated due to loaded car high center of gravity and light car-weight. Advice from American Steel Foundries was solicited. From this, suspension and brake design evolved which includes:

a) New design ASF 70-ton trucks incorporating D-7 spring nests, improved damping Ride Control, constant contact side bearings and 16-inch center bowls.
b) Empty-load truck-mounted brake system.
c) Ten-inch end-of-car hydraulic cushioning.
d) Cylindrical tread wheels.

Features developed during the design phase include:

1) Self-storing adapters for longitudinal/lateral securement of the alternate 35-foot stack-container lading.
2) Bulkheads of semi-monocoque design.

Semi-monocoque design refers to a hybrid truss design where the loads are carried in part through panels (steel plates) welded between truss members; rather than carrying the loads wholly through truss members with heavily reinforced joints, as in simple truss design. Thus, with semi-monocoque it was possible to redistribute essentially the same amount of material to more efficiently utilize it as structural reinforcement.

Data covering the prototype car are listed in Table 2. The prototype car lightweight turned out to be 52,700 pounds, slightly over the estimated 51,600 pounds shown in the table. One thousand pounds of the excess was found to be attributable to overweight trucks and the remainder due to warehouse purchased steel thicknesses being on the high side of tolerances. Refinement of the design, after post-test fatigue analysis is completed, will result in weight reduction.

PROTOTYPE STACKED CONTAINER CAR STATISTICS

Length Over Strikers	63'-10 1/4"
Truck Centers	50'-0"
Wheel Diameter	33"
Deck Height ATR at Bolster (Light Car)	3'-5 7/16"
Car Width Over Side Sills	10'-4 1/4"
Bulkhead Height ATR (Light Car)	12'-4"
Weight of Carbody	27,860 pounds
Weight of Bulkheads	7,040 pounds
Truck Weight (1WW)	16,690 pounds
Estimated Lightweight of Car	51,600 pounds
Gross Weight on Rails	220,000 pounds
Truck Capacity (Nominal)	70 Ton

TABLE 2.

Figure 5 is a photo of the completed and container-loaded prototype car, preparatory to the testing phase of the program. This phase includes a battery of fully instrumented tests to evaluate the design both structurally and operationally. Specifically, these tests include:

1) Loaded car static tests
2) Impact tests
3) Loaded car squeeze to one million pounds
4) Jack loaded car at coupler
5) Torsional rigidity tests
6) Over-the-road environmental test - St. Louis to Oakland via Houston and return.

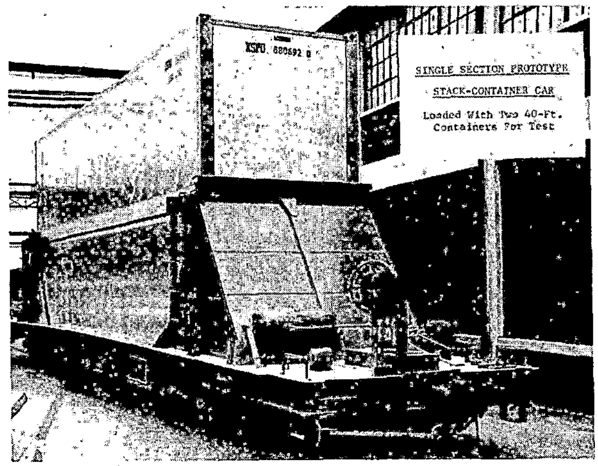


FIGURE 5.

Extensive strain, deflection and acceleration data are collected for car performance evaluation. As mentioned, test data evaluation will include additional fatigue analysis, to verify that fatigue failures will not occur. Again, this analysis complies with the Interim AAR Guidelines for Fatigue Analysis of Freight Cars.

QUESTIONS SESSION V

Session Chairman - George Reed

Attendee: W. E. Trantham, Marine Electric Railway Products
Attendee's Question: You mentioned you are now using NFL type lubricants. Do you have any data available on effect of this lubricant on bearing operating temperatures as opposed to petroleum based greases?

T. C. Keller: In regard to this question between the two greases, the two greases are M917 Grade B grease that was the AAR standard prior to the M942 which is the improved grease used in NFL bearings. And, I think the primary differences between the two greases are the old grease or the Grade B used a 500 second oil or a lower viscosity oil, the M942 new grease uses a higher viscosity oil with nominally about 850 seconds. That's 850 compared to 500 seconds. That's the difference in the viscosity. Also there's some additional minor changes in the additive package in the new grease as well as there being a requirement for stability on the new grease that wasn't existent on the old M917 grease. But the question still has substance in the matter of temperature of operation in that the higher viscosity oil in the new grease does result in a somewhat higher operating temperature and the quantification of this that we've done by laboratory means is that it's approximately ten degrees higher for the higher viscosity M942 oil and the M942 grease. Now this on the face of it is a higher temperature if you have equal quantities of grease, but I will add something that's not asked by the question and that is that the NFL concept visualizes no addition of grease in the field and so in doing this the temperature of operation is considerably reduced because according to our tests one addition of grease to the M917 will bring the operating temperature up higher than with the standard amount of new grease. I hope that answers that question. We have not gotten into synthetic lubricants.

Attendee: Pierre Marcotte, Canadian Rail Research
Attendee's Question: Is roller bearing life significantly affected by the truck hunting problem of freight cars at high speeds?

T. C. Keller: My answer to this is that anything that increases load on the bearing is going to reduce the fatigue life. Now to quantify this is something else again. Truck hunting is almost standard for most all freight cars and it is difficult to get a comparison between an operation not having truck hunting and one having truck hunting. Also, we know that truck hunting occurs primarily with lightly loaded cars. This tends to minimize the effect on life because the lightly loaded car is not depleting bearing life very fast. But there are side effects with truck hunting that are not proportionate as far as load effects are concerned. That is truck hunting on light cars causes the very active condition to exist between, for example, bearing adaptor and the side frame and when this condition exists the adaptor is worn rapidly and the load relationship on the bearing is adversely affected. In addition, this knocking about that the bearing receives tends to loosen seals, cause seals to leak, and it also produces excessive thrust loading which indeed does have an adverse affect on bearing life. But, to quantify it I don't have a specific answer.

Attendee: C. R. Kaelin, Santa Fe Railroad

George Reed has served as Director of Railroad Sales with ACF since 1972. He is a graduate Engineer with a BS from Wayne University and an MSE from the University of Michigan.

Attendee's Question: You inferred composition shoes contribute to truck hunting, you also inferred composition shoes are susceptible to thermal damage of wheels. How do you explain the higher derailment rate and wheel failure rate on cars equipped with metal shoes?

R. Beetle: The first part of the question dealt with truck hunting, and I was simply referring to the paper that Paul Garin of the Southern Pacific and Klaus Cappel of Wyle Laboratories presented in their study of lading damage. They had two classes of cars, similar except for the types of brake shoes and in this case by substituting the wheels from one car to the other, the phenomenon reversed between the test cars and was attributed to the difference in wheel tread profile. The wheel tread profile differences were attributed to differences in the wear pattern of the wheel resulting from the type of brake shoe. The phenomenon, I think was recognized but not reported in detail as being a brake shoe factor. The fact was that the metal brake shoe apparently allowed normal rail action to shape the tread of the wheel whereas the wear from the composition shoe was sufficient to condition the wheel in a manner that the original taper was maintained and in reality perhaps accentuated some. This was not dealt with in the paper except for the fact of reporting the circumstances.

With reference to thermal damage and the type of brake shoes, we manufacture both AAR standard high phosphorus iron brake shoes as well as AAR high friction composition brake shoes. Both can perform well without wheel damage, however, both can cause problems with over braking. The nature of the two products is that the metal shoe may be more inclined with over braking to five visual heat checking on the tread, whereas the composition shoe tends to induce higher internal stresses. For the same braking effort with composition shoes, stresses may be 50 percent higher and these are internal in the wheel, without any evidence that can be seen.

About 60 percent of the freight fleet today is equipped with metal shoes, the remaining 40 percent with high friction composition. The high friction composition shoes, of course, are more common on new cars, particularly the 100-ton cars. Studies show that 70-ton cars on 33-inch wheels are doing a greater share of the braking effort than the 100-ton cars on 36-inch wheels. No wheel problem arises when you place a metal shoe, which has lower friction, on a car equipped for high friction composition brake shoes. However, when the reverse situation occurs, where a high friction composition shoe is erroneously misapplied to those 60 percent of the cars that are equipped for metal shoes, you have twice the designed brake shoe load, all the time, on that shoe. Misapplied composition shoes can generate very high internal stresses which can remain in the wheel - without being observed.

A problem may not occur while a misapplied composition shoe is on the wheel. It may be that the next time a shoe (metal or composition) is put on the wheel, or the following shoe after that, that the problem occurs. We made a study early this year and reported to the AAR our concern for the percentage of metal shoe cars that have misapplied high friction composition brake shoes. In a study of over 9400 cars, we showed that on an average, 8.5 percent of the cars had misapplied high friction composition shoes. In some particular locations, this rose to almost 12 percent. Each time shoes are applied to a metal shoe car, the probabilities add up that its wheel would be more than likely to have seen sometime in their course of performance, the brake effects of misapplied high friction composition shoes which would have twice the force applied than was intended. With statistics like these, it should be obvious that there will be thermal damage reported for such wheels. We think corrective measures should be taken and we have so recommended. The problem is there could be a general feeling that this is a metal shoe problem if there was a metal shoe on the wheel at the time of removal, where in reality the problem could have occurred as a result of misapplication of composition sometime previously.

Thank you. I should say that this question is not one that is limited to the United States because the high phosphorus brake shoe has been evaluated for some years throughout the world, in Europe and in Russia. One prime interest favoring with high phosphorus iron brake shoes is the protection of wheels and so they are using them on demanding passenger, commuter, and locomotive applications. But, as I say, we are manufacturing both products and we think that both have a future, however, it is important to insure that they are properly applied and properly used. Thank you.

Attendee: T. Taylor, Trans-West Associates
Attendee's Question: On true unit train operations, as a practice, we occasionally reverse the direction of the entire train to equalize wheel wear. Could we not improve equal wear patterns by preventive maintenance procedures in rotating each truck, at six-month intervals for example, to make the lead axle set become the trailing axle set? That is change the L4 to R3.

G. Tennikait: By turning the trucks under each end of the car I suspect that Tem, you're wondering if we could reduce the amount of wheel wear caused by curving. Well, this is one type of wheel wear. In some of the data displayed it was noted that the trailing axle set in the lead truck had the most active truck hunting. And, we'd have to balance that to say whether the truck hunting effect on the trailing axle was worse than the amount of curving and I really couldn't put a handle on it at this time. I think there has to be a lot of over the road study on that Tem. But, to really turn that truck around. . .we do it on our plant cars. We have uni-directional curves in our plant. So we do rotate the truck. But, if you run into that condition, yes it would help. But on the over the road running I don't think so because your trailing axle is very active and also your trailing truck also has a leading axle. And at that point your end axle on the trailing truck is taking the greater amount of swivel or truck hunting.

Attendee: H. A. List, REA, Inc.
Attendee's Question: What tests were made to establish freedom from cross talk, such as a torsion in the axle?

G. Tennikait: We did this basically when we set up the axle strain gage bridge system by calibration and we, I don't recall now, but we must have had eight or ten trial gage applications where we loaded in torsion in bending at the center of the axle and on the ends of the axles to get rid of this cross talk in the bending. Does that answer your question? We think we took care of it when we set up the original bridge conditions.

Attendee: E. Dailey, Koppers Company
Attendee's Question: How is the stiffness of the side frames in the DR-1 offset by the steering arms?

E. Bailey: This is not too clear to me, but we'll try to see what we can do with it. First, the interaxle lateral stiffness is very low on the conventional truck and it really isn't effected by the side frames that much. The addition of the steering arms with their interaxle lateral stiffness provides the stiffness that is required. On the interaxle yaw stiffness, the elastomeric pads being placed between the bearing adaptors and the side frames and the coupling, therefore, is transferred through the side frames; but they essentially act as a rigid body you might say in that respect. So, their influence again is not significant.

Attendee: R. Radford, Canadian National Railways
Attendee's Question: Was consideration given to a stacked car to carry four 20-foot ISO containers at 45,000 pounds each thereby requiring 100-ton truck?

L. Nations: Consideration was given to building the double-stacked container car to handle four 20's, two 35's, or two 40's. The 20-foot container in our business, the Southern Pacific Company's business, only takes about 10 percent of our effort in the handling of piggyback. We have plenty of cars to handle a 20-footer. We do not foresee the need of double stacking 20-foot containers on a car in order to reduce the train length. It's not that significant a volume with us. For that reason we did not ask that the car be designed to handle them. If we did design the car to handle four 20-foot containers, then you get the bending force in the middle of the car which would add significantly to the total weight of the car. We felt that by designing for four 20's there would not be a significant reduction or improvement in the tare weight of the car for the handling of 40-foot equipment which is the item that we're truly after, and it would appear to us in years to come that the 40-foot container in domestic containerization of traffic in the United States is the way it's going to go. It's going to be 40-foot or longer and not 20.

Attendee: D. Reynolds, Southern Railway
Attendee's Question: You've covered the positive applications of the DR-1 truck very thoroughly and I hope you will say something on possible negatives: (1) What happens if steering arms become bent due to some accident leaving the axles in a permanently yawed aspect? How do you protect against this? (2) What happens when the truck hits a single heavy rail defect? Will it not go into a sustained yaw vibration? Have you tested against this?

Pierre Marcotte: Concerning the bad aspects of the truck, we are also very anxious to see what might develop with this truck. This is why we have plans to put some cars in service. Concerning the truck following an accident; the steering arms being bent. I don't think that they could be bent in a yawed position more than they can be with a standard truck. In fact, there's a rubber elastomeric pad on the side frame which is centered and which tries to maintain the bearing adaptor in the middle of the pedestal opening. If after a derailment they would get bent, they might get bent in a parallelogramming position. I don't think it would be worse than a standard car is. What would happen is the wheel wear would certainly be evident on these cars that would be bent.

SESSION VI REPORT ON STATUS OF FAST & RAIL DYNAMICS LABORATORY

FAST TRACK STRUCTURE RESULTS TO DATE AND FUTURE PLANS

BY

R. Michael McCafferty

Executive Summary

Under a jointly funded, cooperative effort the Facility for Accelerated Service Testing (FAST) was created to determine comparative life cycles of railroad systems and components. Track structure systems, rail, fasteners, ties, ballast, special track work and rolling stock are subjected to service conditions in a short period of time. This paper describes preliminary track structure observations after one year of operation consisting of 129 million gross tons and 70,214 miles of simulated traffic. Under a unique unit train type operation five rail types can be grouped as to wear rates on a five degree curve. Overall, most track test components are performing up to expectations. There have been some items and maintenance functions which have been removed or modified and in some cases low failure rates have been experienced. Included are future plans for FAST Loop I for the present test and other near-term activities.

FAST MECHANICAL EQUIPMENT TEST RESULTS TO DATE - FUTURE PLANS

BY

D. E. Gray

Executive Summary

The Facility for Accelerated Service Testing (FAST) is the newest test track located at the DOT's Transportation Test Center in Pueblo, Colorado. Developed under the cooperative government/industry Track Train Dynamics Program, FAST allows for the controlled service testing of both track components and mechanical equipment at accelerated rates up to ten times that achieved in normal revenue service.

Test operations began in September 1976. The mechanical equipment tests involve the wear and performance measurements of twelve component experiments including wheels, trucks, brake shoes, roller bearings and adapters on a total of 89 test vehicles. Two of the test cars are also instrumented to measure their dynamic characteristics. During the first 11 months of operation, the test train has accumulated approximately 66,000 miles. Although the mileage accumulated to date has not been sufficient to access the comparative wear rates and performance of several of the mechanical components under evaluation, some have developed definite wear and performance characteristics.

The component experiencing the major wear and replacement has been wheels. Due primarily to the high percentage of curves in the FAST Loop, wheel flange wear has been excessive whereas tread wear has been minimal. Results for the first 20,000 miles indicate that the rate of flange wear for untreated Class U wheels is approximately twice that for treated Class C wheels. Several wheels were removed after 40,000 miles for cracked flanges which have been related to flange fatigue resulting from the unique wheel/rail wear pattern developed in the curved track sections. Some tread shelling and cracking has also occurred.

Results on other components have been less dramatic to date. Some of the premium trucks have experienced component failures and have undergone minor design changes as a result. One constant contract side bearing design has been removed from the test as a result of cage and elastomeric block failures.

The current test configuration of 100-ton equipment is planned to continue for a total of 450 MGT or 230,000 total miles. During this remaining period, some of the mechanical measurements will be deleted while others are planned to be continued or expanded.

RAIL DYNAMICS LABORATORY PERFORMANCE REQUIREMENTS AND HARDWARE CONFIGURATIONS

BY

A. Gross

Executive Summary

The railroad and transit industries have frequently encountered dynamic operating problems with their vehicles leading to: injuries and fatalities, accidents and derailments, lading damage, excessive maintenance costs, and rough train rides for passengers. The Federal Railroad Administration since the inception of the Transportation Test Center (TTC) has been dedicated to building a Rail Dynamics Laboratory (RDL) to conduct fundamental research in a controlled environment on the many dynamic factors affecting vehicle performance and safety. The RDL at TTC is near completion and will house: the Vibration Test Unit (VTU) and the Roll Dynamics Unit (RDU).

The VTU shall provide the capability for a 320,000 pound (145,150 kg) loaded rail vehicle equipped with two, two-axle trucks or to one truck of a vehicle having three or four axles per truck, to the vertical and lateral vibrations environments which the vehicle and its components would "see" in traveling over track with representative profile and alignment variations. Major subsystems of the VTU are as follows: a) vertical excitation modules (one for each test vehicle wheel), b) lateral excitation modules (one for each test vehicle axle), c) vehicle restraint mechanism (one for each coupler), d) support elements such as reaction masses and service structures, e) hydraulic pumping and distribution system and f) hybrid control and monitor system.

The RDU will provide the capability for driving, or absorbing power from the wheel sets of a four-axle 400,000 pound (181,437 kg) loaded vehicle or a three or four axle locomotive truck. One roller module shall be provided for each wheel set. Through rotation of the rollers, the RDU will simulate tangent track at various vehicle velocities, and will permit investigation of vehicle performance on "perfect" tangent track such as truck hunting. "Perfect" track is defined as track with no lateral or vertical irregularities. Major subsystems of the RDU are as follows: a) drive trains, b) roller module units, c) RDU support structures, reaction masses and structures, d) vehicle restraint system, e) service structures and f) control and monitor system.

Acceptance tests for both the VTU and RDU are currently scheduled for late spring 1978. These tests are to demonstrate that contract performance requirements have been met.

Once operational, the VTU and RDU will permit researchers to perform much needed analytical and experimental tests of full-scale locomotives, passenger and freight cars, and transit vehicles under controlled conditions. Lessons learned in the RDL should lead to safer and lower cost equipment before it is built, not after mistakes are demonstrated in the field.

RAIL DYNAMICS LABORATORY TEST PLANNING, SCHEDULING, AND BUDGETING

BY

Wade D. Dorland

Executive Summary

The Transportation Test Center (TTC) facilities have been constructed to expedite improvements and solve problems in railroad and mass transit transportation. Within this highly specialized center of the Federal Railroad Administration, an agency of the U.S. Department of Transportation (DOT), the unique dynamic test capabilities embodied in the Rail Dynamics Laboratory (RDL) are available not only to DOT agencies such as the Urban Mass Transit Agency and the Transportation Systems Center, but also to other agencies of the Federal Government and private industry.

TTC is an extensive complex of rail test facilities comprised of tracks, guideways, and specialized facilities including the Tank Car Torch Test Facility and the Rail Dynamics Laboratory. The Center is employed in practical research and development (R&D) testing of railroad systems, transit systems, and other ground transportation concepts with the objective of promoting a safe, adequate, economical, and efficient national transportation system. This testing is performed in an objective and impartial manner by trained and experienced personnel without the test vehicle ever traversing revenue trackage. These R&D operations play a vital role in obtaining maximum return in equipment and track development or upgrading investments.

The Rail Dynamics Laboratory is being activated to investigate one of the most complex facets of rail engineering --wheel/rail interactions and vehicle dynamics. Currently, the laboratory is in the final stage of a seven-year activation period which will result in the commencement of test operations in mid-1978. RDL testing will be performed for a wide variety of users: the Federal Railroad Administration (FRA) Office of Research and Development (OR&D), the Urban Mass Transit Administration (UMTA) Office of Technology Development and Deployment, other Federal agencies, the American Association of Railroads (AAR), railroads, and rail equipment suppliers. Since RDL users will maximize test results by thorough planning, this paper has been prepared to provide information which will facilitate the planning. General policies and definitions will be found in the subsequent section; then sections on pretest planning, test program scheduling, test program planning, test conduct, and budgeting follow.

FAST TRACK STRUCTURE RESULTS TO DATE AND FUTURE PLANS

BY

R. Michael McCafferty
Federal Railroad Administration

Under a jointly funded, cooperative effort the Facility for Accelerated Service Testing (FAST) was created to determine comparative life cycles of railroad systems and components. Rail, fasteners, ties, ballast, special track work, track structure systems, and rolling stock are subjected to long term service conditions in a short period of time. This paper describes preliminary track structure observations after one year of operation consisting of 129 million gross tons and 70,214 miles of simulated traffic. Under a unique unit train type operation, five rail types can be divided into three groups as a function of wear rates on a five degree curve. In a few cases, test components have experienced some failures; however, overall most track test systems and components are performing to expected levels. There have been some initial maintenance operations for some test zones that have been modified. In some cases low failure rates have been experienced. Also, included in this paper are future plans for the present test variables on FAST Loop I, other near-term activities, and a list of contributors to the FAST Program.

INTRODUCTION/BACKGROUND

The Facility for Accelerated Service Testing was created to answer near term railroad problems in a shorter time period than revenue service tests. Full scale tests in a centralized environment are to determine comparative life cycle performance and to provide data to verify analytical models. FAST is a total systems approach to track structure and rail vehicle problems. A one of a kind test facility in the United States, FAST is designed for testing simultaneously track structures, rail, fasteners, ties, ballast, special trackwork, maintenance methods, and rolling stock under heavy demand conditions.

The FAST program is under the jointly funded Track Train Dynamics (TTD) program. Membership to this international government-industry research program consists of the Association of American Railroads (AAR), Railway Progress Institute (RPI), Transportation Development Agency (Canada), and the Federal Railroad Administration (FRA). Early discussions and planning for FAST began in the fall of 1975 and through a gigantic cooperative effort the first train operations began on September 22, 1976.

Located at the DOT/FRA Transportation Test Center (TTC) in Pueblo, Colorado, the FAST track loop was created from existing track, which in some cases was slightly modified, and with new construction. While many track components were in the track or from other FRA research projects, the railroad industry and suppliers provided many track test items and the use of locomotives, cars, and other rolling stock components. FRA funded the track construction, purchased some track test components and instrumentation (including a track geometry car capability) and is funding operation, maintenance, and data collection and processing. Extensive assistance, monitoring of progress, and guidance is provided from various sources through the TTD program.

Task XI of TTD, Phase II, is designated the FAST project and presently Mr. W. W. Simpson of the Southern Railway is the task chairman. A large technical review group meets

---

R. Michael McCafferty is Program Manager of the Improved Track Performance Research Program of the Improved Track Structures Research Division, Office of Research and Development for the FRA. He attended the University of Missouri at Rolla and the University of Missouri at Columbia with B.S. and M.S. degrees in Civil Engineering. McCafferty completed his Ph.D. in Engineering at the University of Colorado in 1972.

quarterly to assess progress and consider future requirements.

Many, many people have helped make FAST a success. Today the main people in the FAST organization at TTC (figure 1) consist of: FRA/ TTC - Greg McIntosh and Pete Cramer; AAR-Sergei Guins and Ron Begier; and Operations and Maintenance Contractor - Doug Tharp, Dan Frankowski and Curley Walker. Overall program direction is from the heads of the various organizations involved; but program management (figure 2) is primarily the responsibility of Don Spanton (FRA-Washington) with the concurrence of Jim Lundgren (AAR-Chicago).

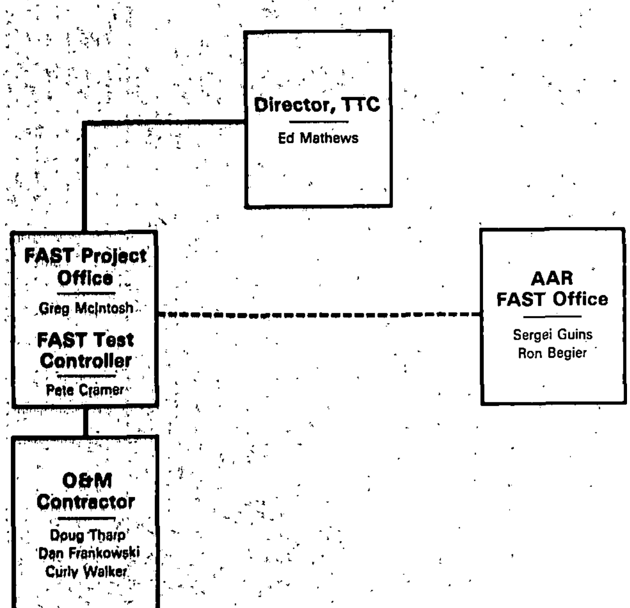


FIGURE 1.

In July of this year a FAST Experiment Coordinatng Group (FECG) was officially created to coordinate the efforts of a newly formed group of researchers termed Experiment Managers. This group consists of Sergei Guins, Greg McIntosh, Phil Olekszyk (FRA-Washington) and myself.

The track Experiment Managers are divided into five areas, each with a "coordinating head". These five areas and primary people are: Track Systems/Maintenance of Way - Dick Murphy (DOT, Transportation Systems Center (TSC), Cambridge, Mass.), Special Trackwork - Bill Cruse (AAR-Consultant), Rail - Roger Steele (TSC), Ties/Fasteners - Howard Moody (FRA-Washington) and Ballast/Subgrade - Andy Sluz (TSC). The five "coordination head" rolling stock Exerpiment Managers are also shown in figure 2. In many of the 10 experiment areas, more than one Experiment Manager is assisting on the program. Experiment Manager duties include conception and planning for future tests, evaluation of ongoing tests, and overall experiment monitoring through the final report stage.

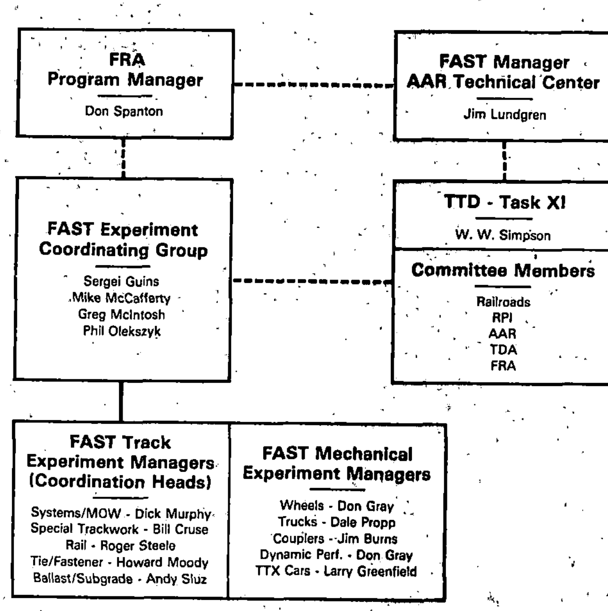


FIGURE 2.

FAST began operation on September 22, 1976, and one year later 129 million gross tons (MGT) and 70,214 miles of simulated traffic have been generated. During this period volumes of data have been collected and, following appropriate processing, are being placed in a data base at the AAR Technical Center in Chicago. For future reference, the original data are stored at the Transportation Test Center.

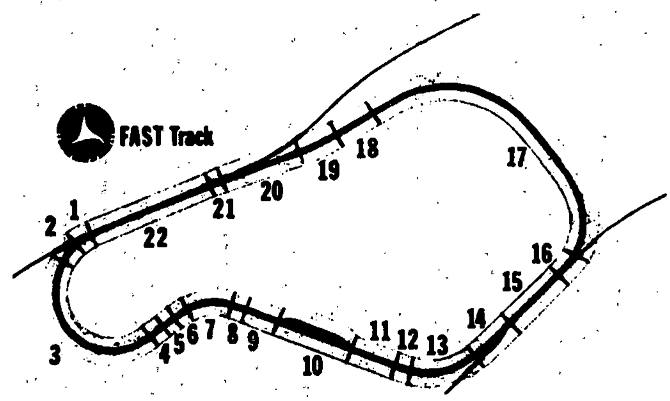
AAR, under contract to FRA, is to maintain a data management system and when fully operational will provide data access for those who are interested. Presently there are several organizations evaluating, to a limited degree, certain pieces of the raw data. TTC is performing data quality control functions, analysis to plan maintenance needs, and evaluation of present experiments and instrumentation. The AAR is working on methods to reduce the raw data for storage and is looking closely at wheel and rail results. Contractors under the FRA Improved Track Structures Research Program are evaluating data from specific experiments, for example, concrete ties and the ballast/subgrade area.

Reports on the preliminary results to date consist of an AAR report (No. R-264), "Facility for Accelerated Service Testing, Progress Report No. 1" dated April 1977, and a second progress report which is scheduled for the Fall of 1977. An FRA report (No. FRA/ORD-77/29), "Track Structures Performance" prepared in April and dated September 1977 will be released shortly.

The FRA report covers more than just FAST results and was prepared as a result of needs of the FRA Office of Northeast Corridor Project. Of particular interest were the comparative evaluations of the performance of wood and concrete tie track systems, the performance of concrete tie track components, the effectiveness of varying ballast type, depth, and shoulder width in improving the long term track behavior and the comparison of rail metallurgy wear rates. The above report includes FAST data through as much as 50 MGT for some of the experiments.

RESULTS TO DATE

FAST Loop I is divided into 22 track test sections (figure 3). They are described in greater detail in an FRA publication, "The FAST Track" dated September 1976 and an AAR brochure, "FAST - The First Experiment." Some test sections are simply transition zones between other sections while others contain more than one experiment. Therefore, to make comparisons or draw conclusions many times more than one test section must be studied. While the same basic loaded consist operates over the track, in a controlled cycle, the point to point differences of rail lubrication, grade, speed, and old or new construction must be considered in any evaluation. The following paragraphs describe results, trends, or observations of various track experiments.



TEST SECTION DESIGNATION

1. Existing Turnout
2. Rubber Pads; Wood Ties
3. Plate Cant; Rail Metallurgy
4. Existing Rail, Field Welded
5. Bonded Joints
6. Steel Ties
7. Rail/Tie Fasteners
8. Continuous Welded Rail
9. Reconstituted and Laminated Ties
10. Turnouts; Frogs; Spikes; Wheel Fax
11. Joints; Frogs
12. Existing Track
13. Rail Metallurgy; Spike Hole Filler
14. Existing Turnout
15. Ballast Shoulder Width
16. Glued Turnout
17. Concrete Ties; Tie Pads
18. Ballast Depth
19. Wood Ties, Hard vs. Soft
20. Ballast Type Depth; Rail Anchors
21. Welded Turnout
22. Rail Anchors; Spike Patterns

FIGURE 3.

Track sections 1, 10, 14, 16, and 21 contain complete No. 20 turnouts. Those in section 10 (two turnouts) and section 14 are straight through moves. The other three are facing point moves during the complete operational cycle of the train. All turnouts have required considerable maintenance which was to be expected, to hold good surface.

Switch point life can be stated in gross terms as to how often they have been replaced. Variables of lubrication, new wheel profiles, and grinding procedures must be considered in any qualitative conclusion. For example, the standard point in section 1 has been replaced three times. The standard points in sections 16 and 21 were replaced at 30 MGT with a field installed manganese tip. The one in section 16 was again replaced at about 90 MGT. This was due to a flaw in the material rather than the point wearing out. The one at section 21 installed at 30 MGT is still in service. Section 16 is an all glued turnout while the one at section 22 is all welded. We anticipate that maintenance data will provide an insight as to the relative merits of these types of turnout construction.

Frogs, which are located in sections 10 and 11, are performing similar to those in revenue service conditions. Again, considerable maintenance is necessary to retain surface. Six of the eight frogs in section 11 have cracked in the heel of the frog. Two have been removed for closer observation. To date, there appears to be little difference in the performance of the standard cast and hammer hardened manganese frogs.

Bonded and insulated joints in sections 5 and 11 were all installed under field conditions. Often spot tamping is necessary to maintain surface and uniformity of service conditions for all joints. Some joints are experiencing greater movement between parts than others; however, all continue to perform satisfactorily. Continued observation is necessary prior to any comparison between the various types under test.

Dowel, laminated, and reconstituted wood ties on tangent, jointed track all exhibit little tie plate cutting and are performing satisfactorily. These are in section 9. Section 6 had steel ties which were removed after 27 MGT of traffic. The fastener system developed cracks and could not retain gage. Revenue service tests exhibited the beginning of the same problem at approximately the same tonnage level. The hardwood and softwood ties in section 19 show little difference in tie plate cutting. The rail has been replaced once in this section and it is hoped that the ties will not become spike killed before the comparison can be completed.

Spike pull out tests of cut spikes with two types of hole filler show little difference to date. Tests of an elastic spike indicate large vertical forces are needed to remove the anchor. These are located in section 10 and provide both gage and longitudinal rail restraint capability. The smaller tie plate which is a flat piece of steel (cant is in the tie) has shown some tendency to curl. Future results may show increased tie plate cutting as compared to the larger tie plates. However, tests are necessary to determine the type of wood since these ties were supplied with the fastener system.

Section 2 has a rubber pad between the tie plate and wood tie. Many of these pads show a recent tendency to flow. It does not appear that these pads will have a sufficient life cycle to reduce tie plate cutting.

Of the eight types of pads in the concrete tie section (17), one is not performing to expectations. It is located on the five degree curve on the tangent and three degree curve and exhibits various degrees of deterioration continued observation will determine the relative performance of the various pads.

The various wood tie fastener combinations in section 7, a five degree curve, were evaluated for gage retention and tie plate cutting. Little difference could be determined except that the elastic clip had no tie plate cutting after 33 MGT and the others showed very small amounts. A few of the heads of one type fastener broke allowing the tie plate to ride up onto the portion left on the tie. In one instance this caused the tie plate to break. It is believed that maintenance of way equipment initiated the situation. Due to non-uniform installation of the fasteners and extensive tie splitting from spike killing, this section is being rebuilt for additional tests of fasteners.

Tie plate cant is varied in section 3 from the standard 1:40 to 1:30 and 1:14. While the data is scattered for the five rail metallurgy types, in general the gage point wear is greater for the rail on the 1:14 cant tie plate. On the average, little difference in rail wear is observed between the other two tie plate zones.

Concrete ties and fasteners in Section 17 were analyzed previously for the Northeast Corridor Project. In general the conclusions were (1) measured tie bending moments were lower than the American Railway Engineering Association Specification requirements, (2) insulator and clip movement and breakage has not been a critical maintenance or performance issue although component replacement sometimes has been necessary, and (3) large rail corrugations are detrimental to overall performance. Measured wheel/rail loads average 32,000 pounds which have produced peak moments of 130 and 80-inch kips at the tie centerline and rail seat, respectively. Visual inspections at 50 MGT of 80 new ties found only two ties with rail seat cracks. Old ties from the Kansas Test Track, which were subjected to about 20 MGT of traffic and already had small cracks, continue to exhibit acceptable performance.

The five degree curve, two percent grade area of section 17, has required considerable maintenance. It is thought that the rail corrugations eventually lead to "soft" spots in the track. This then creates ballast degradation, tie skewing, and fastener problems. Several elastic clips have failed recently mainly on the gage side of the low rail in a region of "soft" track and inadequate pad performance. Also some concrete ties have center cracks in this region. The cracks extend only an inch downward and are not considered structural cracks. These were probably the result of center bound ties which occurred in "soft" spots.

All of the above leads to the conclusions that for concrete tie track the regions of poor support, either due to a temporary joint or rail corrugations, can produce areas of ballast degradation and high maintenance requirements.

Vertical and horizontal track stiffness tests have been conducted and show greater resistance to load (less deflection at the same load) for concrete tie track as compared to wood tie track. Increased vertical stiffness ranged from two to four times the wood tie track. Horizontal pull tests in general required twice the load for concrete tie track as compared to wood tie track to deflect the track 0.25 inches.

Ballast results in terms of track settlement have shown little difference with ballast type as a variable. With changes in ballast depth it is observed that overall settlement is somewhat greater with increased depth. More important is the condition of the roadbed which at FAST was either new or had gone through three winter cycles and about 0.5 MGT of traffic (old track). Mean rail settlement values at 30 MGT were 1.02, 0.63, and 0.20 inches for tracks of jointed rail-new, jointed rail-old, and continuously welded rail-old, respectively.

Two wood ties were removed at 40 MGT in regions of 8 and 16 inches of ballast under the tie. In both cases a somewhat center bound tie was selected to attempt to observe a "worst" case condition. The thin ballast layer had about one inch of rutting and subballast fines within 3 inches of the tie bottom. In the 16-inch section no rutting was observed; however, fines were with 6 inches of the tie bottom. This would suggest that the better track design for the FAST support conditions is closer to a ballast depth of 16 inches than 8 inches.

Both ballast and subballast strains are being measured. High ballast strains were observed for the first 2 to 5 MGT following a tamping operation. Subgrade settlement through 50 MGT had not stabilized in the new track construction. Pressures under both concrete and wood tie track at the subballast/ subgrade interface were 5 ± 1 pounds per square inch.

Rail metallurgy experiments are in sections 3 and 13. Five different types are being tested in these curved test zones to evaluate both gage point wear and head area loss as comparisns between the various types. While section 13 has only four types and 115 pound rail, based on data to date, it would appear to have better wear characteristics. This may be due to the difference in lubrication in this area. The higher observed wear rates on section 3 (132-136 pound rail) has apparently not allowed the development of shelling which is becoming common in section 13.

Figure 4 shows some gage point wear data from section 3 on rail with increased silicon content on 1:40 tie plates. A linear regression analysis was performed by TTC staff on all data from section 3 to predict replacement requirements. Figure 5 shows the gage point wear for the five rail types on 1:40 tie plates vs. tonnage. The wear rates can be grouped into three zones: (1) highest-standard rail; (2) medium - fully heat treated and hi-silicon and (3) lowest - chrome moly and head hardened. While the exact order sometimes changes with location on the track or tie plate cant, the three groupings are statistically significant.

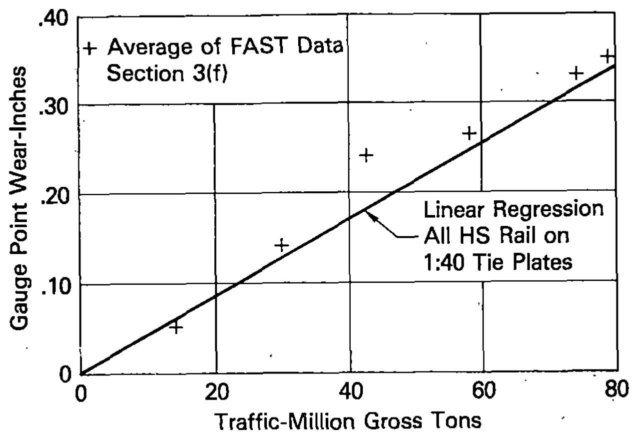


FIGURE 4.

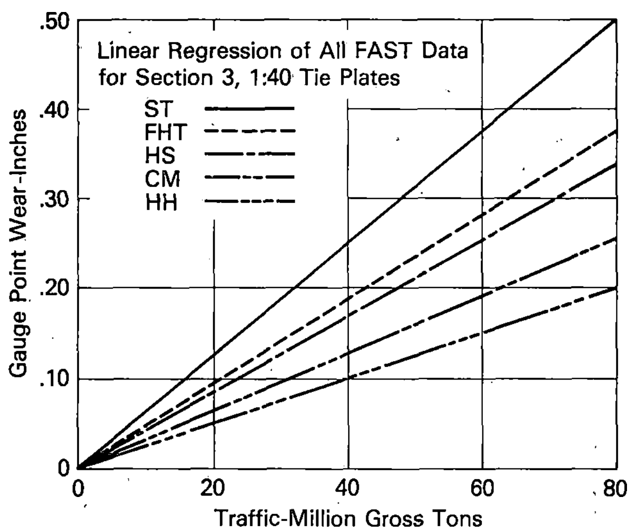


FIGURE 5.

FUTURE FAST LOOP I ACTIVITIES

Future activities and experiments for loop I are now being planned by the various Experiment Managers. It is anticipated that the present consist with four-axle locomotives and mainly 100-ton cars will operate for 400 to 450 million gross tons. This is necessary to adequately determine the life cycle of various track components and to subject the rolling stock to 200,000 miles of service. For example, ballast performance and tie plate cutting will require considerable tonnage before significant differences can be observed. Several surfacing operations will be needed to determine relative performance of wood and concrete tie tracks. Due to the observed variations in rail wear around the track, it will be necessary to wear out rail of different types at the same location to completely address the comparative rail wear rates.

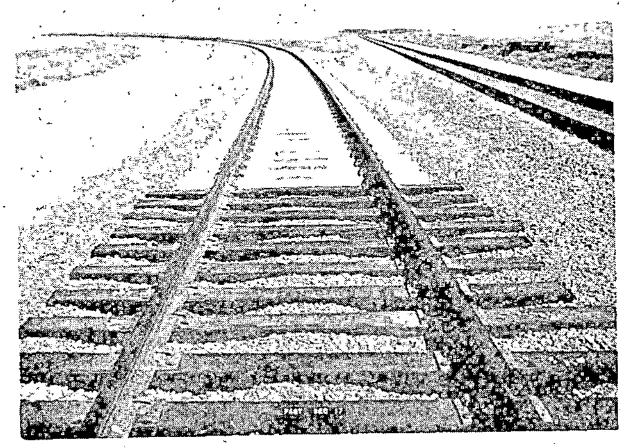


FIGURE 6.

Following the above 400-450 MGT test, it is planned to change the consist to 70-ton cars (locomotive type is to remain constant) and then repeat many of the experiments. This is to determine the comparative performance of various systems and components as a function of the lower axles load environment. Hopefully, this will begin to answer the question as to which system is more economical to operate (100-ton vs. 70-ton).

As a first phase under the new Experiment Manager concept, all track experiments are being evaluated as to their goal/objective and the approach being used to obtain the goal. Data collection and analysis techniques and requirements are being studied to determine if they are adequate and cost effective. As experiments are completed, modified or expanded, additional test items will be necessary. A review and selection process has been initiated to standardize the selection and definition of experiments.

Near term activities include a rebuild of the rail metallurgy and wood tie fastener test zones. Longer segments (200 rather than 78 feet) of different types of rail will be joined together with frozen joints rather than welding. It is hoped that this will reduce the initial problem of weld failures and wear differences when plugs are inserted with a new rail head profile.

Many ties in the fastener test became spike killed which destroyed the capability to determine gage holding and tie plate cutting. Therefore, all new ties are planned with a revised fastener scheme. The revised test will allow removal of worn out rail without the problem of spike killing ties.

FAST CONTRIBUTORS

Since the FAST program was conceived there have been many railroads, companies, organizations, and individuals involved. As new experiments are defined the list of participants will continue to grow. Without their contribution the FAST program would not be a success in order to continue the search for answers to today's railroad problems.

The following lists constitute, to the best of my ability, those railroads and suppliers which are contributing to the program. If anyone is missing, I apologize and request that it be brought to my attention.

CONTRIBUTORS-RAILROADS

Atchison, Topeka and Santa Fe Railway Co.

Bessemer and Lake Erie Railroad Co.

Burlington Northern, Inc.

Canadian National Railways

Canadian Pacific Ltd.

Chessie System

Chicago, Milwaukee, St. Paul & Pacific Railroad Co.

Consolidated Rail Corp.

The Denver & Rio Grande Western Railroad Co.

Elgin, Joliet and Eastern Railway Co.

Grand Trunk Western Railroad Co.

Illinois Central Gulf Railroad Co.

Louisville & Nashville Railroad Co.

Missouri Pacific Railroad Co.

Norfolk & Western Railway Co.

St. Louis - San Francisco Railway Co.

St. Louis Southwestern Railway

Seaboard Coast Line Railroad Co.

Southern Pacific Transportation Co.

Southern Railway System

Toledo, Peoria and Western Railroad Co.

Union Pacific Railroad

CONTRIBUTORS-SUPPLIERS

A. Stucki Co.

Abex Corp.

ACF Industries (AMCAR Div.) Shippers Carline

Allegheny Drop Forge Co.

Bethlehem Steel Corp.

C. W. Blakeslee & Sons

Brenco, Inc.

Cardwell Westinghouse Co.

Cedrite Corp.

C.F.&I. Steel Corp.

Chemetron Corp.

Colt Industries (Crucible Spring Div.)

Conforce Costain Concrete Tie Co., Ltd.

Dayton Malleable Inc.

Dow Chemical U.S.A.

Dresser Industries, Inc.
Dresser Transportation Equipment Division

E.I. DuPont DeNemours & Co., Inc.

Edgewater Steel Co.

Fabreeka Products Co.

Fairmont Railway Motors, Inc.

General American Transportation Corp.

Griffin Wheel Co.

Grinaker Precast Ltd.

Harmon Industries Inc.

Hawker Siddeley Canada Ltd.
Canadian Steel Wheel Div.

Henry Miller Spring and Manufacturing Co.

International Track Systems, Inc.
Railroad Products Div.

Intma, Inc.

L. B. Foster Co. (Weir Kilby Div.)

Lewis Rail Service Co.

Manganese Steel Forge Co.

McConway and Torley Corp.

Midland-Ross Corp.

Minnesota Mining & Manufacturing Co.

Moore & Steele Corp.

NDH Bearing Service

New Departure Hyatt Bearings

Newton County Stone Co.

North American Car Corp.

Pandrol Limited

Pennsylvania Power & Light Co.

Pettibone Corp.

The Polymer Corp.

Portec, Inc.

Pullman-Standard

Racine Railroad Products,Inc.

Railroad Friction Products Corp.

Santa Fe-Pomery, Inc.

A. Schulman, Inc.

Servo Corp. of America

Shell Oil Co.

Standard Car Truck Co.

Timken Co.

Titanium Metals Corp. of America
Standard Steel Div.

Trailer Train Co.

True Temper Corp.

Union Spring & Manufacturing Co.

Union Tank Car Co.

Unit Rail Anchor Co.

United States Railway Equipment Co.

United States Steel Corp.

Vanguard Corp.

Virginia Plastics Co.

Vulcan Materials Co.

WABCO-Westinghouse Air Brake Div

Warner Co.

Woodings - Verona Tool Works

FAST MECHANICAL EQUIPMENT TEST RESULTS TO DATE-FUTURE PLANS

BY

D.E. GRAY

This paper summarily describes the results to date, after approximately 66,000 miles of train operation on the mechanical experiments being conducted at the Facility for Accelerated Service Testing (FAST). Specific results are presented on wheel flange wear, wheel failure modes and premium truck performance. Future plans for FAST mechanical testing are also described.

INTRODUCTION

As reported in a number of recent AAR and FRA technical reports and presentations, the FAST program is a cooperative FRA, AAR, RPI research program concerned with both track systems and mechanical equipment components. This paper will be concerned with only the mechanical equipment aspects of the FAST Program.

The FAST test consist is composed of a total of 89 test vehicles. Specifically they are made up of the following types and sizes. Sixty-five of the test vehicles are 100-ton open-hopper cars. There are three 100-ton capacity bathtub coal cars and three 70-ton Trailer-On-Flat-Car (TOFC) units also are included. The remaining 18 test vehicles are 100-ton tank cars.

The typical test consist in any one day is normally made up of 76 cars. The motive power has been typically four, four-axle diesel-electric locomotives providing a test train of approximately 9,500 total gross tons.

With regard to FAST operations the following information is provided. The FAST train is operated up to 16 hours per day, five days per week at an average speed of approximately 42 miles per hour. The remaining eight hours of each test day are used to take measurements and perform track and vehicle maintenance.

Each day a block of four test cars are removed from the FAST train and routed to the shop for measurements. This shopping cycle is repeated every 22 test days.

Car position in the consist is rotated by removing eight cars from the front of the train each day and placing them at the rear. In addition, to equalize wear on both track and rolling stock components under test, the direction of the train movement as well as its orientation are reversed in a four-day cycle.

As mentioned above, four cars are removed from the FAST consist each day. Depending on which experiments are included on these cars, literally hundreds of measurements are made. For example 77 cars require the following wheel measurements. Three types of measurements are made at two locations (180° apart) on each of the eight car wheels. Flange thickness, rim thickness and flange height are measured using the Standard AAR Finger Gage. Wheel profiles along with tread and rim hardness are also measured at these two locations.

Twenty-four cars are specifically involved in the truck experiment of which 12 car sets are equally divided among four premium trucks. The remaining 12 car sets are comprised of two types of commonly used trucks under six 100-ton hopper cars and the trucks used on the three "Bathtub" and three TTX cars. As with wheels, a great number of measurements are made on the trucks under test and include the following: Both wear and surface hardness measurements are made on the friction castings and mating surfaces, bolster gibs, side

---

Donald E. Gray has served as the Evaluation Program Manager, Office of Freight Systems (R&D) for the FRA since 1976. He received his B.S. degree in Aeronautical Engineering from the University of Maryland (1964), and his M.S. degree in Mechanical Engineering from George Washington University (1971).

frame column wear plates and column guides. In addition, measurements are also taken on bolster and side frame rotation stops as well as bolster and transom lateral stops.

There are 10 other component/ system areas under investigation in FAST and they too undergo a similar measurement and inspection cycle. However, to date, the amount of wear on these components has not been significant or the results statistically significant. In addition to the static measurements obtained on the various freight car components, selected cars in the FAST consist have been instrumented to measure their dynamic response characteristics. Both a low mileage car and a car normally accumulating mileage in the consist were each instrumented with a total of 20 channels of accelerometers to assess the effects of the various track section configurations, their wear and the car component wear on freight car dynamic performance. An instrumented wheelset was also installed on the low mileage car to measure the dynamic lateral and vertical rail/wheel loads continuously as the car traverses the FAST loop. In addition to running on the FAST track, each car is also operated on a tangent section of the Railroad Test Track (RTT) to provide a relatively invariant reference track input for obtaining car transfer functions.

RESULTS

During the first 11 months of FAST operations, the test train has accumulated approximately 66,000 total miles. The average individual car mileage, however, is somewhat lower than this maximum due to lost time for scheduled car measurement and maintenance shoppings and unscheduled bad orders. Although the mileage accumulated to date has not been sufficient to assess the comparative wear rates and performance of several of the components under evaluation, others have developed definite wear and performance characteristics and are discussed in the following paragraphs.

Wheel Flange Wear

The component experiencing the major wear and replacement to date has been wheels. Wheel flange wear has been excessive since the beginning of the test due primarily to the high percentage of curves in the FAST Loop. In contrast, tread wear has been minimal. An example of the flange wear experienced on FAST is shown in figure 1. Flange thickness measurements have been analyzed for a limited population of wheels in the wheel wear experiment.

FIGURE 1. TYPICAL WHEEL FLANGE WEAR ON FAST

For each of the variable six parameters (wheel manufacturing process, hardness, one wear, two wear, profile, center plate size, and truck type) contained in the experiment matrix, a nominal sample of 16 wheels was used to compare wheel flange wear. For each parameter considered, an equal mix of wheels representing the other parametric variations was included in the sample of 16 wheels. For each 22-day measurement cycle, the mean value of flange thickness decrease was calculated. In addition to the six parameters identified in the original experiment, a comparison was also made to determine if there is any significant difference in flange wear for the two types of brake rigging.

The only statistically significant results to date in this experiment are shown in figure 2. As shown for the first 20,000 miles the rate of flange wear on Class U wheels has been approximately twice that experienced on Class C wheels.

One additional phenomenon that was observed is depicted in figure 3. The difference between rates of flange wear on each axle set of the truck shown was observed on all of the cars in the wheel experiment and, the data on each car was typical of that shown, but not identical. The exact cause of this effect has not been determined however, it is generally accepted as occurring in revenue service.

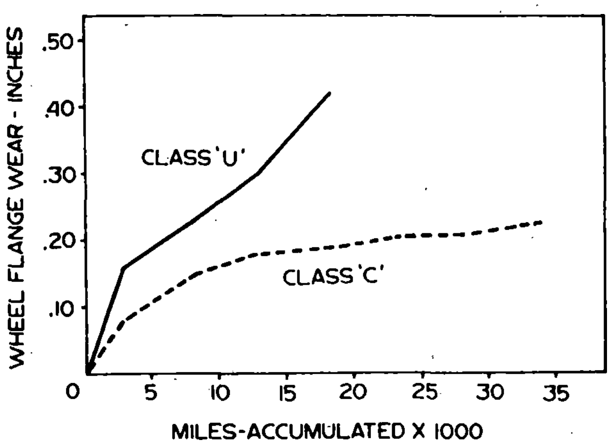


FIGURE 2. COMPARISON OF FLANGE WEAR ON CLASS U AND CLASS C WHEELS

Wheel Failure Models

After approximately 40-45,000 miles a number of wheels were found to contain small cracks across the flange of the wheel. This phenomenon is depicted in figure 4. A probable cause for this phenomenon is the unique wheel/rail wear pattern as shown in figure 5. This pattern is due to the use of cars with the majority of wheels having uniform flange heights. Figure 6 shows an actual cracked wheel flange section mated with a worn section of high rail. An enlargement of the flange apex (Area A) is provided in figure 7.

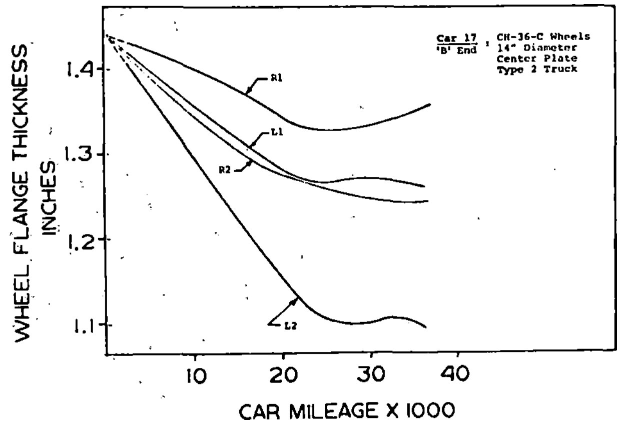


FIGURE 3. EXAMPLE OF FLANGE WEAR VARIATION WITH WHEEL POSITION

This enlargement shows that on the gage side the flange material has plastically deformed and small subsurface cracks have been identified. This preliminary AAR study has indicated that the nucleation of the flange cracks is a subsurface phenomenon associated with the plastic deformation of the wheel steel which occurs near the apex of the flange due to wheel/rail contact in this area. As mentioned previously, such a condition has developed because of the characteristic wheel and rail wear at FAST. It is suggested that the flange cracks nucleate from the longitudinal cracks which in turn develop due to subsurface rolling contact fatigue. The phenomenon appears to be very similar to the formation of shells and detail fractures in rails.

The second failure mode that has been noted is comprised of tread cracks and wheel rim shelling. These phenomenon are depicted in figures 8 and 9. This condition has just recently been observed, and investigations to date have not resulted in a conclusive determination for the cause of this failure.

Table 1 gives a summary of the total number of wheel sets removed in the 32 car wheel wear experiment. The cause of removal is depicted as well as the distribution according to wheel class.

FIGURE 4. TYPICAL WHEEL FLANGE CRACKS

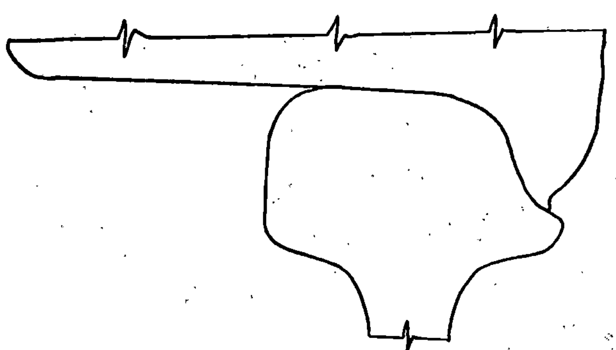


FIGURE 5. WHEEL/RAIL WEAR PATTERN

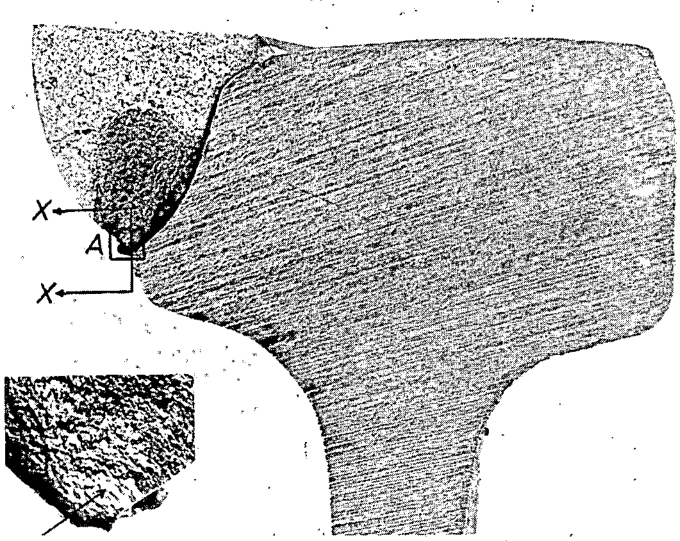


FIGURE 6. SECTIONS OF CRACKED WHEEL FLANGE AND WORN HIGH RAIL

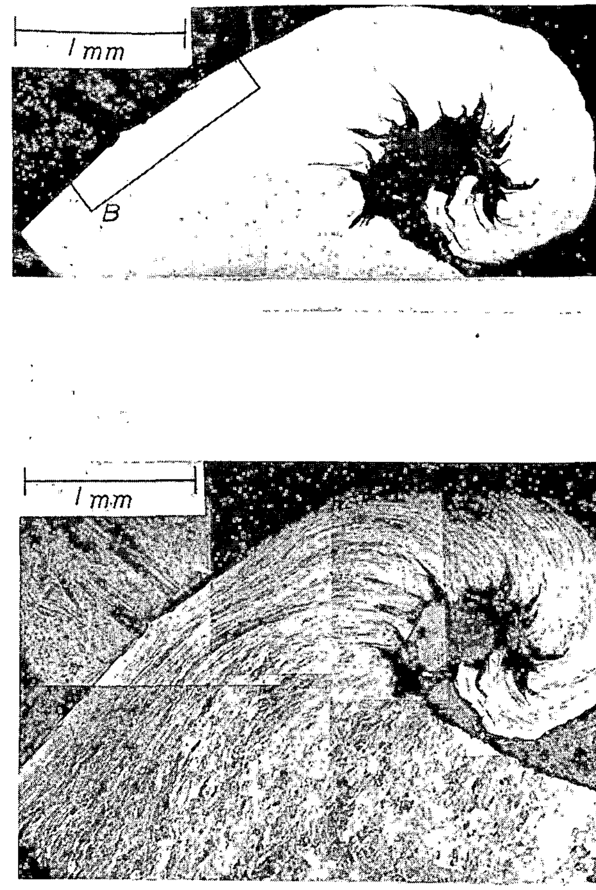


FIGURE 7. UNETCHED AND ETCHED SECTIONS OF FLANGE APEX (AREA A) SHOWN IN FIGURE 6

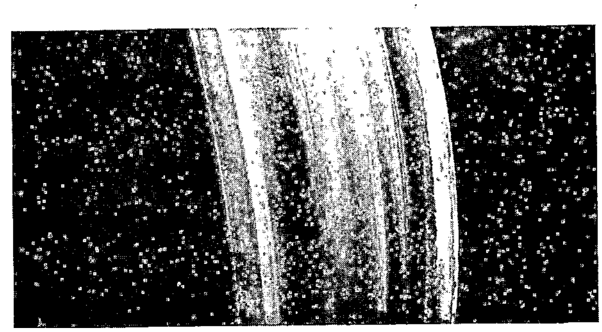


FIGURE 8. WHEEL TREAD CRACKS

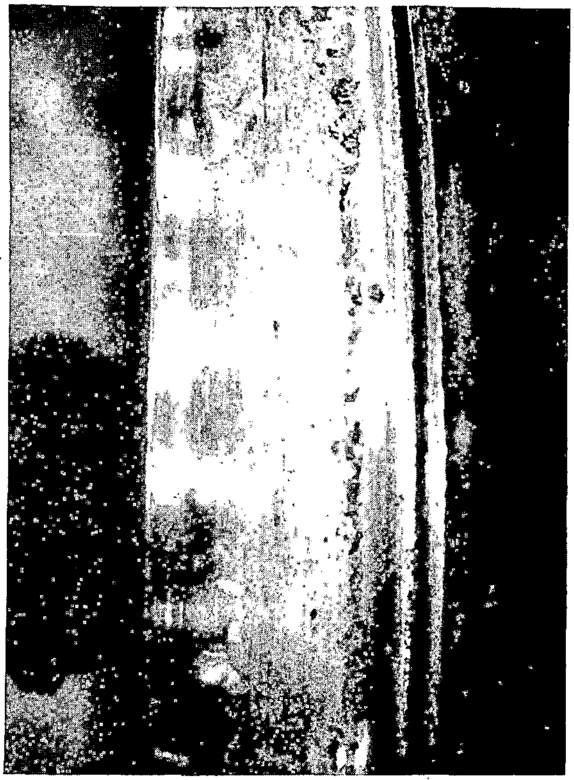


FIGURE 9. WHEEL RIM SHELLING

TABLE 1. TOTAL NUMBER OF WHEEL SETS REMOVED IN WHEEL WEAR EXPERIMENT

Cause For Removal	Class U	Class C
Thin Flange	70	11
Cracked Flange	10	3
Tread Cracks/ Shelling	20	49
Thin Rim	5	1
Over Heated Bearing	2	4

Premium Truck Performance

The results to date indicate that one of the three-piece premium truck designs is experiencing greater gib wear than the other two. One of the other three-piece truck designs has experienced cracks in some of the side frame column wear plates which have since been replaced by the manufacturer. This truck design has also experienced a small number of spring failures. Lastly, the fabricated truck design has undergone changes in the body bearing mount design and the hydraulic snubber design as well as experiencing a number of broken springs.

It should be emphasized that the amount of data collected to date has not been significant enough to draw firm conclusions on the performance of these premium trucks. This data is reported, however, to indicate the type of results that are being achieved in the FAST program with regard to truck performance and that the manufacturers of these trucks are able to take advantage of this FAST experience.

Other Components Performance

As mentioned above, a great deal of mileage has not been accumulated on the test cars in FAST to date, however, other component performance can be reported at this time. For example, some grease loss was reported on one supplier's roller bearing early in the FAST program. As a result of detecting this failure in the bearing seals, the manufacturing process has been modified to include a vibration test as part of the quality control procedure for these bearings. Since this process has been initiated there has not been a reoccurrence of this failure.

One of the constant contact side bearing designs has been removed from the FAST test program due to the development of cage fractures and tears in the elastomeric blocks. These failures are being reviewed by the manufacturer. Lastly, there has been some minor deterioration of composition brake shoes due to metal accumulated from the rail and resultant sparking. This is not a typical operational situation, but is reported for consideration should this phenomenon occur in a unique operational railroad application.

Dynamic Performance

Conclusions related to the objective of quantifying the dynamic response of freight vehicles to different track structures are as follows: Variations in track structures such as ballast shoulder width and depth, spiking patterns, tie material, and rail anchors had little if any effect on truck and carbody accelerations or wheel forces. In contrast, curves greater than 4°, and discrete events such as turnouts had a marked effect on vehicle dynamics. The highest carbody accelerations were experienced on Section #5 of the FAST track which contains unsupported bonded joints. Since truck mode accelerations were moderate to low over this same section of track, it can be hypothesized that this particular track structure excites a resonance in the vehicle suspension system.

FUTURE PLANS

Using the existing consist, which is nominally a 100-ton unit train, the current test configuration is planned to continue for a total of 400 to 450 MGT which is equivalent to approximately 230,000 vehicle miles. With regard to the mechanical experiments it is planned that the 32 car set wheel wear experiment will be repeated with the addition of some high flange wheels in the balance of the consist to prevent the unique wheel/rail wear pattern that was encountered in the data shown above.

Also, a new concept of management has been added to the FAST program in the form of FAST Experiment Managers. Ten Experiment Managers, five related to track and five related to rolling stock, now have the responsibility to review the existing FAST experiments and plan future experiments. These individuals have been earnestly working for several months now and the results of their constructive efforts are already being felt.

For example with regard to mechanical experiments, the side bearing experiment and the brake shoe experiment have been terminated because of the limited speed capability on FAST and the inability to have long periods for programmed air brake testing, respectively. Also, in the interest of efficiency and costs, the truck spring experiment has been deleted since the AAR mechanical committee has approved the use of alloy springs. Failure rates on springs of both alloy and carbon steel will be maintained to provide assurance on the capabilities of alloy springs.

And lastly, in the interest of efficiency and accuracy a large number of measurement fixtures have been designed and fabricated and are being put into service to assure repeatable accurate measurements on components such as center plates, side bearings and gibs.

RAIL DYNAMICS LABORATORY PERFORMANCE REQUIREMENTS AND HARDWARE CONFIGURATIONS

BY

A. GROSS

This paper describes the Rail Dynamics Laboratory (RDL) facility at the Transportation Test Center (TTC), Pueblo, Colorado. Two unique test machines, the Vibration Test Unit (VTU) and the Roll Dynamics Unit (RDU) are to be housed in this facility to perform dynamic tests of full-scale railroad and transit industry vehicles. Both the VTU and RDU performance requirements and hardware configurations are described.

INTRODUCTION

The railroad and transit industries have frequently encountered dynamic operating problems with their vehicles leading to: injuries and fatalities, accidents and derailments, lading damage, excessive maintenance costs, and rough train rides for passengers. Since the inception of TTC, the Federal Railroad Administraton (FRA), has long recognized the need for a rail dynamics laboratory as a research tool to conduct fundamental research in a controlled environment on the many dynamic factors affecting vehicle performance and safety. While the RDL facility is not fully operational as of yet, the goals and objectives through the years of development have remained relatively the same.

The RDL goal is to provide a facility to perform dynamic tests of full-scale locomotive, passenger and freight cars, transit vehicles and advanced track systems under controlled conditions. Such a facility will permit the evaluation of various hardware designs in a safe, controlled and reproducible scientific laboratory environment, allowing the performance of a variety of tests with minimal risk to personnel and equipment.

The objectives of the FRA RDL program for the past several years have been to provide an operational facility within reasonable costs which can be utilized by railroad and transit industry researchers in dynamic studies such as: passive and active suspension characteristics; vehicle rock and roll tendencies; component stress analysis; component and vehicle natural frequencies; adhesions; ride comfort; acceleration; braking; lading responses; hunting and analytical model validation as well as supporting causes of derailment.

This facility will help to isolate the causes of and aid in the solutions to various dynamic operating problems encountered in the railroad and transit industry. Through study of vehicle dynamics in the RDL, the number of dynamic-related accidents and derailments and their attendant costs should be reduced significantly.

RDL HISTORY

Todays RDL facility is considerably different than FRA originally planned at the inception of the program. Prior to the development of DOT's TTC, no test facility was available in the United States to extensively evaluate and determine the solutions to dynamic operation problems. Just before 1970, FRA contractor studies recommended a full-scale roller rig (a rail dynamics simulator) with capability to handle cars and locomotives at full speed and power, with vibrations applied through the wheels to simulate track conditions. Representatives of railroads and suppliers assisted FRA in preparing performance specifications for the simulator.

FRA engineers opened communica-

Arnie Gross has served as the Rail Dynamics Laboratory Program Manager, Office of Freight Systems (R&D), FRA since 1975. He received his BME from City College of New York (1958) and his MSME from Drexel Institute of Technology (1967). Gross is also a member of ASME.

tions with experts in other countries who had operated similar facilities, using their experience in preparation of the specifications. In order to leave options open for testing advanced high speed systems, such as the tracked air cushion vehicles, the simulator speed capability was designed for approximately 300 mph (483 km/h). The Urban Mass Transportation Administration joined in funding part of the RDL project so that transit vehicles could also be tested in the laboratory and agreed to locate the rail dynamics simulator (RDS) in a laboratory at TTC.

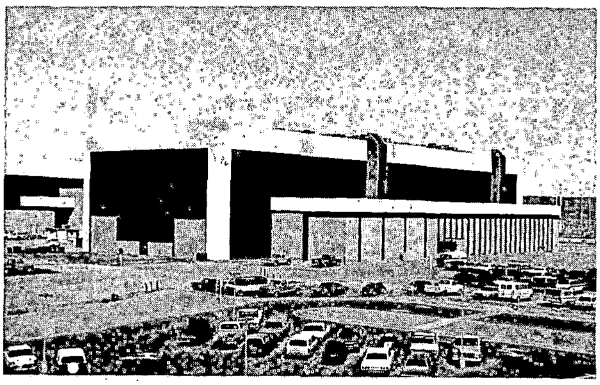


FIGURE 1. RAIL DYNAMICS LABORATORY BUILDING

RDL Building

FRA placed the RDL building construction contract in 1972 for a high bay (the testing area) and a connecting low bay office wing, a two-story structure which contains offices, control room and other facility support areas. The principal dimensions of the high bay are 352' x 108' x 65' (107.3m x 32.9m x 19.8m) while the low bay is 264' x 50' x 30' (80.5m x 15.2m x 9.1m).

This modern steel and reinforced-concrete structure RDL building (figure 1) was accepted in April 1974. Some notable features of the RDL building include: a) two high bay 100-ton overhead cranes for loading and off-loading the test machines, b) calibration laboratory for instrumentation, c) electronic shop for equipment repairs and maintenance, d) clean rooms for disassembly, inspection and cleaning equipment.

Rail Dynamics Simulator (RDS) and Subsystems

The high bay portion of the RDL building was to house the RDS as well as service areas and a vertical shaker. Starting in 1972, FRA let contracts for the following subsystems comprising the RDS:

a. Drive train, which was to provide rotation to the track module rollers;

b. Track module, which was to simulate the tracks on which the test vehicle rests and had the capability to simulate vertical and lateral irregularities;

c. Carriage assembly, which acted as the support and reaction structure for the track module;

d. Instrumentation and control subsystem.

e. Computer subsystem.

f. Communication system.

A separate contractor was involved with each subsystem.

Vertical Shaker System (VSS).

While the RDS subsystem was in the design phase, FRA awarded a contract to Wyle Laboratories to design and construct a Component/Vehicle Preliminary Evaluation System, later named the Vertical Shaker System (VSS), envisioned as a pre-test tool prior to complex testing on the RDS. The VSS was to be used for the determination of rough estimates of response modes and frequencies and for studying the responses of truck assemblies and total vehicles to vertically applied periodic excitation. In 1975, the VSS (figure 2) was activated, it essentially consists of four independently operated vertical actuators which can be placed under four wheels of a two axle truck on one end of a rail vehicle. Each actuator can accommodate wheel loads of up to

40,000 pounds (18,144 kg). The acceleration, frequency, and displacement of these actuators can be varied over a wide operating range to simulate operating environments of most test specimens.

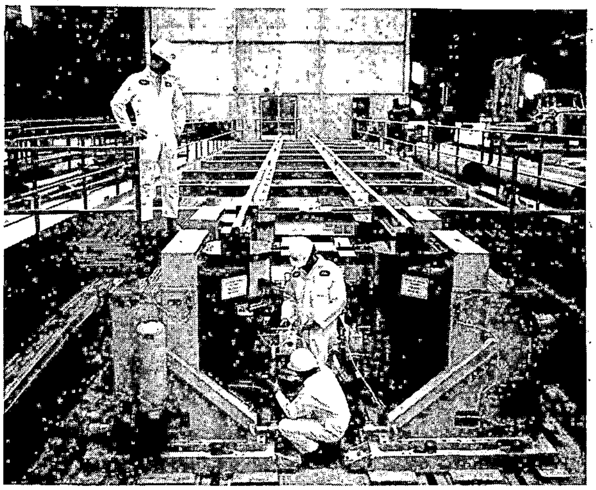


FIGURE 2. VERTICAL SHAKER SYSTEM

Input capabilities for this system include: a) vertical translation of the truck and railcar, b) roll motion of the truck and railcar and c) pitch motion between the forward and aft axles sets. Prior to changes in the RDL program that affected the VSS configuration, two test programs were conducted in the RDL on the VSS:

a) The Trailer-on-Flatcar (TOFC) Optimization Program (see figure 3) which was designed primarily to determine the sensitivity of lading response to suspension system component variations and load distribution; and
b) The AAR Structural Dynamics evaluation of the TOFC configuration which was structured to collect data for verification of a mathematical model of the flat car body.

RDL Program Redirection

During the development of some of the RDS subsystems, unforeseen technical problems arose which resulted in severe schedule delays and associated risks of great concern to DOT.

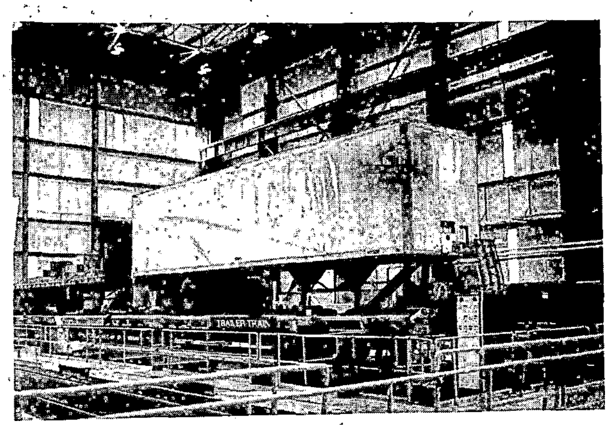


FIGURE 3. TOFC ON VSS

In mid-1975, after the RDL program had continued to encounter R&D development and management problems, a DOT task force review resulted in the redirection of the RDL program so that it could be completed in a timely manner, relatively free of technical risk, and with minimum cost. The RDS was replaced by the Vibration Test Unit (VTU), an upgraded VSS, which will provide vertical and lateral excitation at both ends of a test vehicle, and the Roll Dynamics Unit (RDU), a basic roller rig. The RDS formerly combined both vibration and roll in one simulator. The redirected RDL program now has one prime contractor, Wyle Laboratories, instead of several major contractors.

The subsystems and systems which formerly supported the RDS, now as Government Furnished Property (GFP), will be modified for VTU and/or RDU operation, whenever possible. These subsystems/systems included the VSS, drive trains, hydraulic subsystem, integrated computer subsystem network, analog acquisition and control subsystem, and communication system, and structures subsystems. The RDL building floor plan including the VTU and RDU test pits is shown in figure 4.

VTU DEFINITION/BASIC REQUIREMENT

The VTU will provide the capability for subjecting a 320,000 pound (145,150 kg) rail vehicle equipped with two, two-axle trucks or to one truck of a vehicle having three or four axles per truck, to the vertical and lateral vibrations environments which the vehicle and its components would "see" in traveling over track with representative profile and alignment variations.

RDU DEFINITION/BASIC REQUIREMENT

The RDU will provide the capability for driving, or absorbing power from the wheel sets of a four-axle vehicle or a three or four axle locomotive truck. One roller module shall be provided for each wheel set. Through rotation of the rollers, the RDU will simulate tangent track at various vehicle velocities, and will permit investigation of dynamic phenomena characteristics of "perfect" tangent track such as truck hunting. "Perfect" track is defined as track with no lateral or vertical irregularities.

VTU PERFORMANCE REQUIREMENTS

A brief summary of the major VTU performance requirements are noted here. Table 1 identifies the test vehicle weight and size limitations for

TABLE 1. VTU/RDU VEHICLE WEIGHT AND SIZE LIMITATIONS

Vehicle Length (max)	90.0 ft (27.43m)	108.0 ft (32.92m)
Vehicle Width (max)	12.0 ft (3.66m)	12.0 ft (3.66m)
Vehicle Weight (max)	320,000 lb	400,000 lb
	(145,150 kg)	(181,437 kg)
Axle Load (max)	80,000 lb	100,000 lb
	(36,287 kg)	(45,360 kg)
Truck Center Distance (min)	20.0 ft (6.10m)	20.0 ft (6.10m)
(max)	70.0 ft (21.34m)	80.0 ft (24.38 kg)
Truck Axle Spacing (min)	54.0 in. (1.37m)	54.0 in. (1.37m)
(max)	110.0 in. (2.79m)	110.0 in. (2.79m)
Gauge (min)	56.5 in. (1.44m)	56.5 in. (1.44m)
(max)	66.0 in. (1.68m)	66.0 in. (1.68m)
Coupler Centerline to Railhead		
(min)	17.5 in. (0.44m)	17.5 in. (0.44m)
(max)	34.5 in. (0.88m)	34.5 in. (0.88m)
Center of Gravity to Railhead		
(min)	18.0 in. (0.46m)	18.0 in. (0.46m)
(max)	98.0 in. (2.49m)	98.0 in. (2.49m)

VTU. The VSS vertical actuators (as GFP) was the primary factor for the maximum vehicle weight requirements.

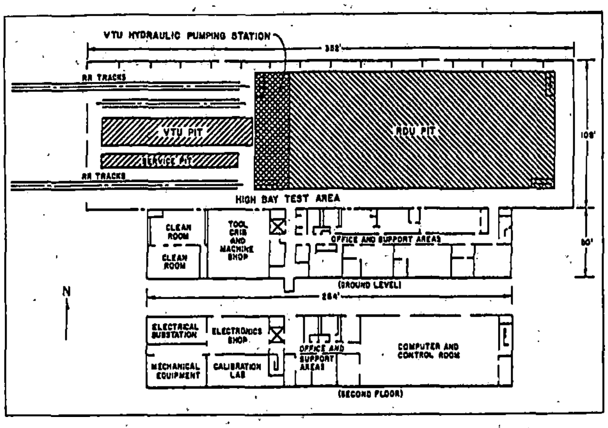


FIGURE 4. RDL BUILDING FLOOR PLAN

Table 2 summarizes the vertical and lateral excitation motion requirements.

reasonable daily test period and (c) VTU configuration changes per test vehicle requirements (i.e., different truck axle spacings, etc.) in a reasonable time period. In addition, system safety requirements have been specified to prevent the VTU from damaging itself, the test vehicle or causing any hazard to operating/maintenance personnel.

VTU HARDWARE CONFIGURATION

As previously identified, the VTU hardware had to be designed to provide the capability to subject a variety of rail vehicles to vertical and lateral vibratory environments similar to that experienced during over-the-road operations.

Owing to the variety of vehicle configurations to be accommodated, each with a unique set of dimensions associated with such elements as axle spacing, truck center distance, overhang, coupler height (in the case of transit vehicle) and inertial properties, a modular approach to hardware implementation was required. In addition to modularizing for the purpose of handling the broad spectrum of vehicles, serious consideration had to be given to ease of reconfiguration in order to minimize test program turn around time.

An artist's rendering of one end of the VTU as designed and currently under construction and assembly at the RDL is presented in figure 6. The

TABLE 2. VTU VERTICAL AND LATERAL EXCITATION

	Excitation	
	Vertical	Lateral
Frequency Range	0.2 to 30 Hz	0.2 to 30 Hz
Displacement	±2"(5.08cm)	±1.5"(3.81cm)
Velocity	25 inch/sec. (63.5 cm/sec)	15 inch/sec. (38.1 cm/sec)
Acceleration	3.5 g's	3.1 g's

The VTU is to provide the following types of vibratory motions to the test vehicle: vertical translation, pitch motions, roll motions, lateral translation, yaw motions, time delayed motions, combined rigid body motions, combined time delay motions and arbitrary wheel vibrations. These modes of vibration are shown in figure 5.

Requirements have been specified from (a) continuous VTU operation (periods up to 10 hours), (b) time required to start up/shutdown (four hours or less) and thereby permit a

VTU hardware as partially shown consists of the following major subsystems:

- o Vertical excitation modules (one for each test vehicle wheel)
- o Lateral excitation modules (one for each test vehicle axle)
- o Vehicle restraint mechanism (one for each coupler)
- o Support elements such as reaction masses and service structures
- o Hydraulic pumping and distribution system
- o Hybrid control and monitor system

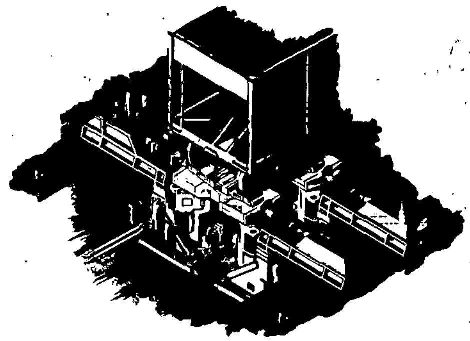


FIGURE 6. VIBRATION TEST UNIT

RIGID BODY MOTIONS

- VERTICAL TRANSLATION
- PITCH
- LATERAL TRANSLATION
- YAW
- COMBINED

TIME DELAYED MOTIONS

- VERTICAL TRANSLATION
- ROLL
- LATERAL TRANSLATION
- COMBINED

ARBITRARY WHEEL VIBRATIONS

- VERTICAL EACH WHEEL
- LATERAL EACH WHEEL SET
- DISPLACEMENT TIME HISTORIES

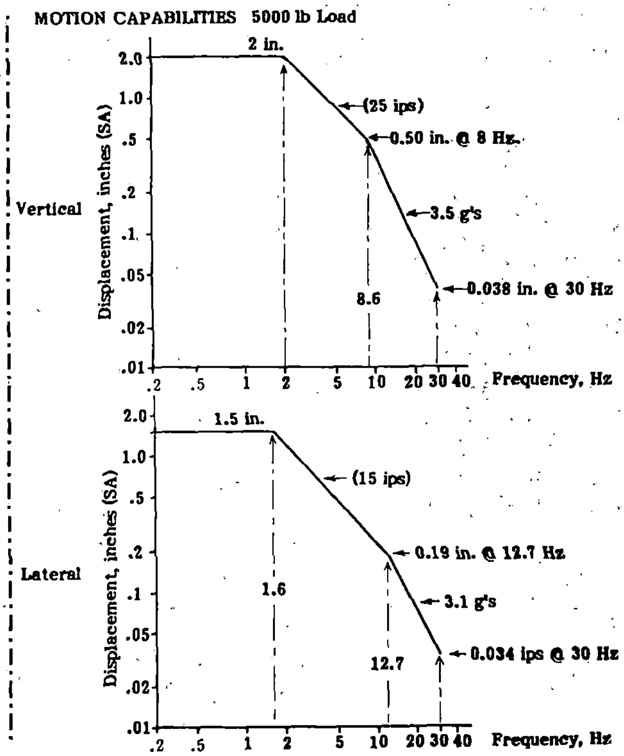


FIGURE 5. VTU--MODES OF VIBRATION

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 (.0216m^3/s) high performance servo-valve. Part of the actuator assembly is an air/oil biasing system designed to support the particular wheel load being tested, such that the degradation of actuator dynamic force capability is minimal. The vertical moving elements are constrained to move in a vertical plane by three hydrostatically lubricated journal bearings designed to carry the attendant transverse forces during operation.

The lateral excitation modules (again each under independent servo control) are the most complicated of the VTU hardware elements in that the following parameters had to be accounted for:

- o Lateral translation (per axle basis)
- o Lateral translation with a phase shift (per truck basis)
- o Allowance for out of phase vertical motion (i.e., roll)
- o Provision for longitudinal vehicle expansion and contraction as experienced during excitation of the lower body bending modes
- o Minimum impact on truck polar moment of inertia
- o Allowance for wheel lift-off

The combination of constraints, high instantaneous loads and overall performance demands resulted in a hardware configuration illustrated in more detail in figure 7. The lateral excitation modules are centered around a 45,000 lb. (20,412 kg) actuator assembly equipped with a 70 gpm (.0044m^3/s) high performance servo valve. In this case, oil/air biasing is provided on both sides of the primary moving elements in order to "sandwich" or preload the entire assembly. This approach was taken in order to provide minimum moving weight and roll moment of inertial of the moving elements. The motion capabilities of this subsystem as governed by the overall motion requirements previously identified are summarized below:

ΔA (In): + 1.78/ - 2.04 (+4.52/-5.18)*

ΔB (In): $\pm$2.0 ($\pm$5.08)*

ΔC (In): $\pm$3.84 ($\pm$9.75)*

Δp (In): $\pm$0.81 ($\pm$2.06)*

θp (Deg): $\pm$6.78 ($\pm$.118)**

θR (Deg): $\pm$3.85 ($\pm$ .067)**

* ( ) Cm
** ( ) Rad

The vehicle restraint mechanism is designed to limit the longitudinal rigid body motion of the test car and minimize spurious forces on the excitation modules. The device consists of a universal coupler adaptor, cable and preloading mechanism, with force measuring capabilities.

The support elements such as reaction masses and service structures are designed as permanent or moveable elements as appropriate to react vibratory loads and provide access as required for the variety of test configurations.

The hydraulic flow demands of the various excitation modules and hydrostatic bearing elements at peak excitation levels can be as high as 1,000 gpm (.0631m^3/s) at 3,000 psi (20,684,271 N/m^2). This was provided for via three 360 gpm (.0227 m^3/s) variable volume pumping systems each capable of delivering the rated flow at 3,000 psi (20,684,271 N/m^2). The distribution manifolds provided allow for connection of the excitation modules in the required combinations of axles spacing and truck center distance.

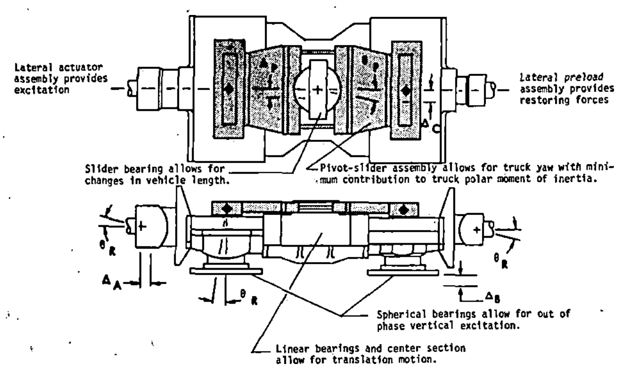


FIGURE 7. VTU-MOTION CAPABILITIES

The hybrid control and monitor system will permit the operation of the VTU from the remotely located control room of the RDL. The control consoles provide the approximate devices and displays for operation of the VTU in either a manual or automatic mode. This system consists of two major subsystems. A digital computer subsystem which will provide synthesized signals representing the "track environment" to the analog control and monitor subsystem in the automatic mode.

Portions of this control and monitor system are shown in figure 8. The complete analog control and monitor subsystem is illustrated as well as the master computer operations station.

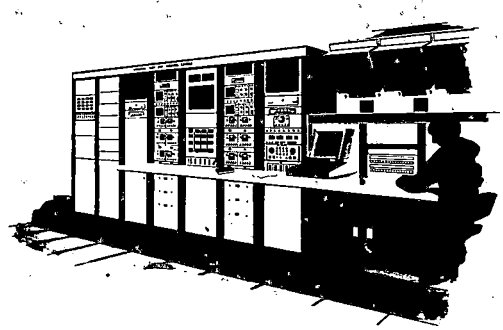


FIGURE 8. VTU-CONTROL CONSOLE

RDU PERFORMANCE REQUIREMENTS

A brief summary of the major RDU performance requirements are noted here. Table 1 identified the test vehicle weight and size limitations for the RDU, which is essentially the same as the former RDS requirements. Vehicle speed simulations are specified at 3-288 mph (4.8-463km/h) for up to 50,000 lb (22680 kg) axle load and 3-144 mph (4.8-232 km/h) for 50,000 to 100,000 lbs (22680 kg to 45,360 kg) axle loads with special tolerance determined by GFP (drive train) capabilities. Likewise, simulation of wheel/rail traction forces have been specified but will be largely determined by modified GFP (drive train) capabilities.

In addition to driving both ends of the two-axle per truck vehicle, or one end of a three or four axle per truck vehicle, the RDU is required to have the capability to simulate steady state curve track operation (minimum curve radius) as follows:

a). 100 ft. (30.5m) for truck center of 50 ft. (15.2m) or less.

b). 150 ft. (45.7m) for truck center between 50 and 80 ft (15.2 and 24.4m).

Requirements have been specified for reasonable times to reconfigure the RDU such as changing gage or axle spacing between rollers, or test vehicles of different length or different curve-radius track. Identical start up/shutdown requirements (four hours or less) have been specified for the RDU as the VTU. During test operation, the RDU operator is only capable of increasing or decreasing the operator speed by manual adjustments. System safety requirements have been specified to prevent the RDU from damaging itself, the test vehicle or causing any hazard to operating/maintenance personnel. The RDU design capabilities have also been specified for the following future installations:

a). Body (lateral and roll) exciters to assess the effect of vehicle dynamic motion and forward speeds,

b). installation of equipment for static loading to simulate the effect of super elevation unbalance during steady curve negotiations and

c). automatic control of the RDU.

RDU HARDWARE CONFIGURATION

Like the VTU, the RDU hardware had to be designed to provide the capability to subject a variety of rail vehicles to dynamic tests, necessitating a modular hardware approach. The RDU, as shown in figure 9, will support and drive (or absorb power from) the wheelsets of a four-axle rail vehicle or a three or four axle locomotive truck. The rotation of the rollers will simulate vehicle speed on tangent track, and make possible the investigation of those phenomena which are independent of track irregularities, such as hunting modes. The RDU can also be configured to simulate steady state curve negotiations on tangent track.

The RDU consists of the following

major subsystems:

- o Drive trains
- o Roller module units
- o RDU support structures, reaction masses and structures
- o Vehicle restraint system
- o Service structures
- o Control and monitor system

Each of the four drive trains is powered by a 600 hp (447.6 kw) variable speed motor. There is a master control station for synchronous operation of selected drive trains.

The roller module units (RMU), each driven by a drive train, will be equipped with two interchangeable sets of rollers, one set with a 42 in. (1.07m) diameter and a second set with a 60 in. (1.52m) diameter. The smaller set will be used for simulation of vehicle speeds up to 144 mph (231.7 kmph) for axle loads under 100,000 lb (45,360 kg).

Each of the two RDU support structures (RDUSS) supports two drive trains and two RMU's. The RDUSS is equipped with air bearings to permit relocation of the RDUSS for various truck center distances and rotation to provide for simulated curves of up to 100 ft (30.48 m) in radius.

The vehicle restraint system controls the longitudinal position of the test vehicle, with respect to the RDU. This system consists of a cable, a flexibility element, a preloading device and a load measuring device. A reaction mass and structure are provided at each end of the test vehicle to react the loads generated by the vehicle, and transmitted through the vehicle restraint system.

Service structures consist of platforms, stairways and ladders required to provide access to and around the drive trains, roller module units, vehicle restraint systems, and test vehicle.

The control and monitor system will permit operations of the RDU from consoles located in the RDL control room. Speed of drive train rotation is commanded via a thumbwheel switch. Operational parameters are monitored and interlocked to prevent damage to the RDU or the test vehicle. The entire RDU control and monitor console is shown in figure 10.

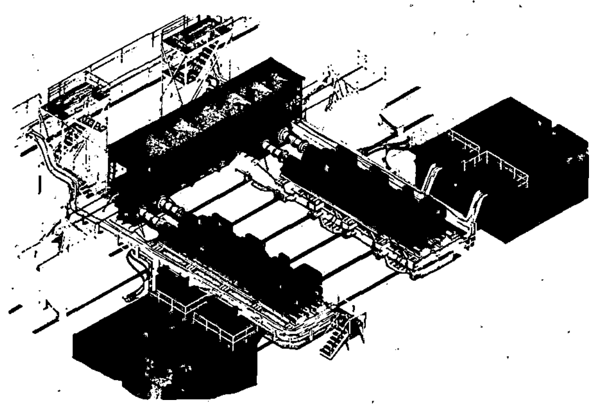


FIGURE 9. ROLL DYNAMICS UNIT

SPECIMEN DATA ACQUISITION SYSTEM (SDAS)

The VTU and RDU each have requirements for necessary data collection as implemented by SDAS. Figure 11 is a general description schematic of the SDAS. Additional data acquisition equipment as follows are available at TTC: (1) calibration scanner, (2) photo motion analyzer, (3) closed circuit television, (4) video recording capability and (5) acoustic recording and analyzing capability.

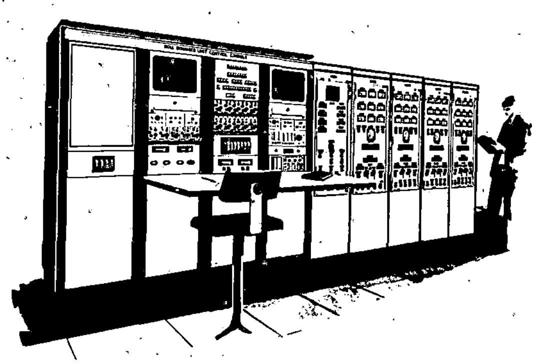


FIGURE 10. RDU-CONTROL CONSOLE

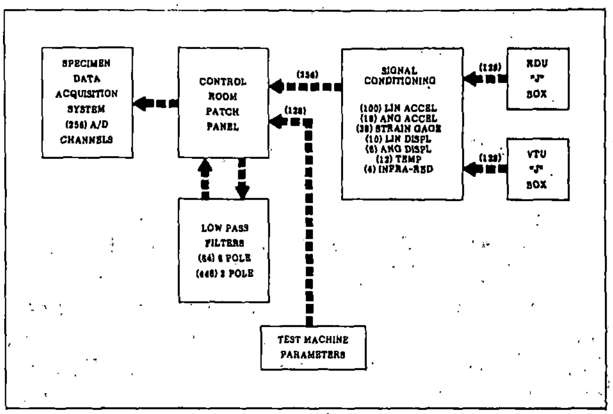


FIGURE 11. SDAS GENERAL DESCRIPTION

VTU/RDU SYSTEM ACCEPTANCE TESTS

Acceptance tests for both the VTU and RDU are currently scheduled for late spring 1978. During these tests, Wyle Laboratories, the RDL contractor, will demonstrate that the performance requirements per contract statement-of-work have been met. During this same timespan training of TTC personnel to operate and maintain the VTU and RDU will be conducted.

SUMMARY

This paper has presented an overview of the RDL's VTU and RDU performance requirements and hardware configurations. At the time of preparing this paper, a large percentage of the VTU and RDU designs were complete and fabrication underway. It is also noted, that when the RDL program was redirected via the DOT Task Force, a finite budget was also imposed. Depending on the final program costs, the final VTU and RDU systems may be different than described in this paper. FRA is doing all that is fiscally possible to have the RDL facility operational as soon as possible, currently late spring or early summer 1978.

Once operational, the VTU and RDU will permit researchers to perform much needed analytical and experimental tests of full-scale locomotives, passenger and freight cars, transit vehicles and advanced track systems under computer controlled conditions. Lessons learned in the RDL should lead to safe and lower cost equipment before it is built, not after mistakes are demonstrated in the field.

RAIL DYNAMICS LABORATORY TEST PLANNING, SCHEDULING, AND BUDGETING

BY

WADE D. DORLAND

INTRODUCTION

The Transportation Test Center (TTC) facilities have been constructed to expedite improvements and solve problems in railroad and mass transit transportation. Within this highly specialized center of the Federal Railroad Administration, an agency of the U.S. Department of Transportation (DOT), the unique dynamic test capabilities embodied in the Rail Dynamics Laboratory (RDL) are available not only to DOT agencies such as the Urban Mass Transit Agency and the Transportation Systems Center, but also to other agencies of the Federal Government and private industry.

TTC is an extensive complex of rail test facilities comprised of tracks, guideways, and specialized facilities including the Tank Car Torch Test Facility and the Rail Dynamics Laboratory. The Center is employed in practical research and development (R&D) testing of railroad systems, transit systems, and other ground transportation concepts with the objective of promoting a safe, adequate, economical, and efficient national transportation system. This testing is performed in an objective and impartial manner by trained and experienced personnel without the test vehicle ever traversing revenue trackage. These R&D operations play a vital role in obtaining maximum return in equipment and track development or upgrading investments.

The Rail Dynamics Laboratory is being activated to investigate one of the most complex facets of rail engineering --wheel/rail interactions and vehicle dynamics. Currently, the laboratory is in the final stage of a seven-year activation period which will result in the commencement of test operations in mid-1978. RDL testing will be performed for a wide variety of users: the Federal Railroad Administration (FRA) Office of Research and Development (OR&D), the Urban Mass Transit Administration (UMTA) Office of Technology Development and Deployment, other Federal agencies, the American Association of Railroads (AAR), railroads, and rail equipment suppliers. Since RDL users will maximize test results by thorough planning, this paper has been prepared to provide information which will facilitate the planning. General policies and definitions will be found in the subsequent section; then sections on pretest planning, test program scheduling, test program planning, test conduct, and budgeting follow.

GENERAL POLICY AND DEFINITIONS

The purpose of this section is to *familiarize potential test sponsors and* users with the operating procedures involved in test scheduling, test planning, and test conduct at the Rail Dynamics Laboratory. All procedures delineated in this document are in accordance with DOT, FRA, and TTC policies and regulations.

The TTC mission is to operate and administer an intermodal center for the conduct of comprehensive testing, evaluation, and associated development of ground transportation systems and their components by DOT organizations, other Government agencies, and related elements of the private sector.

RDL is one of the test capabilities of the TTC. The principal mission is to provide dynamic testing of rail

---

Wade Dorland is the Rail Dynamics Laboratory Manager at the TTC. He received his Engineering Degree from the University of Nebraska. He has spent most of his professional career working for NASA in various capacities in rocket and spacecraft testing. Dorland was transfered from NASA to FRA in March 1977.

vehicles using either linear vibratory or rotary excitation of the wheels. Furthermore, the RDL will update test capabilities by advancing test techniques and instrumentation.

At the head of the TTC organization is the Director who is responsible for accomplishment of the mission. The line and staff civil service personnel are responsible for overall planning, direction, scheduling, priorities, and funding required to perform the mission tasks. The RDL Manager is responsible to the TTC Director for performance of the associated functions in the RDL. Management, operation, and maintenance of test facilities (including the RDL), support facilities, and related utilities are accomplished by an operations and maintenance contractor who has expertise in track testing and all related disciplines.

The Federal staff at TTC assists in the development and approval of the objective, scope, schedule, and relative priority of all testing performed at the Center. Accomplishment of the tests is the joint responsibility of the test sponsor, the operating contractor, and the Federal staff.

The Operations and Maintenance (O&M) Contractor is responsible to the TTC Federal staff for the efficient and professional conduct of all test activities, including those in the RDL, and for the attainment of test objectives which result from test planning and coordination as outlined in the next section.

A number of terms are commonly used at RDL. Some of these are compiled here for a handy reference:

a. TEST SPONSOR -- The Government agency or private group (i.e., commercial firm) that initiates and coordinates test requirements and provides funding for a test to be conducted in the RDL.

   1. <u>Sponsor Representative</u>-- The individual designated by the test sponsor to ensure that test planning and coordination requirements are met before and during the course of an RDL test. He also serves as the sponsor point of contact and has access to decision authority relative to test requirements.

b. USER -- The organization that directly utilizes the RDL data. The user is normally under contract to the sponsor but can also be the sponsor.

   1. <u>User Representative</u> -- A member of the user organization who coordinates test requirements and agreements with TTC, with the O&M, and with the test sponsor. He also serves as the user point of contact and has access to decision authority relative to supporting coordinated test requirements.

c. TEST MANAGER -- The designated member of the TTC Federal Staff who (a) provides the technical interface with the O&M staff at the RDL; (b) coordinates the test requirements and agreements with the O&M, sponsor, and user; and (c) is responsible for the overall scheduling of RDL test machines.

d. TEST ENGINEER -- O&M engineer in charge of accomplishing a test project.

e. PRETEST PLANNING CONFERENCE -- A meeting between the sponsor, user, O&M, and TTC civil service personnel to establish feasibility of testing, define equipment and hardware requirements, delineate responsibilities for further actions, and establish scheduling considerations. Preliminary estimates of test machine occupancy time may be made at the conference, but a more detailed study of unique test requirements and costs may be made before a formal estimate and test schedule are supplied. A user test request usually results from this conference followed by a TTC Test Plan.

f. TEST PLANNING CONFERENCE -- A conference held prior to the start of a test where the final schedule, scope, and objectives of the test can be firmly established between the sponsor, user, O&M, and TTC and documented in a Test Specification document.

g. PROPRIETARY TEST -- A test sponsored by a non-Government establishment (e.g., a railroad, a supplier, etc.) and resulting in test data considered to be the exclusive property of the sponsor. Since the data resulting from a proprietary test is owned by the sponsor who reimburses TTC for the costs of the tests, the sponsor can restrict the distribution of test information; the effective proprietary instructions must be specifically defined by the sponsor.

h. TEST TIME -- The time, measured in hours to the nearest tenth, during which an RDL test unit is being utilized for actual testing of test hardware or data processing as shown in the official test logs.

i. TEST PECULIAR COST -- Cost derived from unique requirements for laboratory modification, special instrumentation, unique software, overtime, or other abnormal service, material, or equipment.

j. O&M -- The Operations and Maintenance Contractor which operates and maintains the test and support facilities of the TTC under contract with the Federal Railroad Administration.

k. TEST AGREEMENT -- A formal document defining the sponsor, user, and TTC responsibilities and commitments. A test request and test plan are usually included along with funding provisions. For a Proprietary Test the Test Agreement will be in the form of a contract executed by the Sponsor and TTC; the contract will also delineate reimbursement provisions.

l. TEST READINESS REVIEW -- A formal meeting convened three to five days prior to the first test run in a program to assess readiness to proceed with test operations.

PRETEST PLANNING

Maximum schedule flexibility and assurance of meeting the requirements of a test program at TTC can best be achieved through early planning and coordination of test requirements. Experience gained from many rail test programs has shown that proper planning is essential in maximizing the timely benefits of testing and effective application of test results. Therefore, early contact is encouraged for TTC planning and programming assistance in order that years of test experience can be used to help determine what to test, when, and under what conditions.

From descriptive documentation on RDL capabilities (i.e., Part II), sponsors and users can generally establish whether a prospective test need can be accommodated. However, factors such as unique test requirements, test unit adaptations, special test equipment, test alternatives, lead time requirements, scheduled commitments, etc., usually necessitate an early meeting between sponsor, user, and TTC to establish feasibility of testing, scheduling considerations, resources allocation, and further planning activities.

When the need arises, sponsors and users should contact as early as possible by mail or phone:

Rail Dynamics Laboratory Manager, RTC-1
Transportation Test Center
Pueblo, CO 81001
Phone (303)545-5660.

As a result of this contact, the RDL Manager will arrange a pretest planning conference and will establish information requirements to make the conference most productive. The sponsor and user, who should plan on spending a minimum of one day at the conference, should prepare the following information for discussion and review at the initial conference:

a. Objectives and scope of the proposed test.

b. Scheduling considerations.

c. Preliminary test plans to show desired test configurations, test conditions, instrumentation, data processing, reporting, and general test support requirements.

d. Other considerations that might assist the RDL staff in studying the overall test requirements.

Review of this information will be used in an RDL estimate of funding and schedule projections which will be provided to the sponsor and user.

SCHEDULING TEST PROGRAMS

Test projects are allocated RDL test time on a first-come, first-served basis. Reallocations resulting from uncontrollable delays, experienced by either user or TTC, will be arranged on the basis of TTC judgment of precedence. Such rearrangements will be worked out in consultation with all users whose commitment is impacted by a schedule problem.

After determining that a specific test can and should be conducted at the RDL, many decisions and actions ensue. The following items are representative of these activities.

The RDL Manager will establish:

a. Preliminary allocation of test unit occupancy.

b. Extent of TTC support requirements.

c. Whether the O&M should proceed with preliminary planning, engineering, and simulation studies for the test. (A simulation study simulates the test setup with a mathematical model which is used to determine the performance margin available to safely meet the test requirements.)

d. The ground rules for further planning on coordination activity.

e. Applicable test peculiar costs.

The RDL technical staff will determine and coordinate with the sponsor and user:

a. Test resources required.

b. Lead times required.

c. Occupancy time estimates.

d. Recommended extensions of test requirement technical analysis.

NOTE: Detailed funding considerations will be coordinated between the test sponsor and the RDL Manager during the planning conference described in a subsequent section.

After the foregoing information is coordinated, the sponsor and user can determine whether to proceed with a formal test request.

PLANNING TEST PROGRAMS

Proceeding with a test will require the sponsor to initiate a formal request to TTC in a letter which includes the following information:

a. Test title.

b. Description of work.

c. Requested test unit.

d. Safety considerations relating to test vehicle.

e. Desired start date.

f. Test requirements availability (date).

g. Test vehicle availability (date).

h. Name, address, phone number of user and sponsor representatives.

i. Data distribution limitations.

j. Cost reporting requirements.

The description of work in the test request should be as comprehensive as possible and can be thought of as a pretest report or technical appendix of the test request. The description of work should include to the fullest possible extent the following:

a. Objectives.

b. Scope of excitation.

1. Linear or rotary vibration.
2. Frequency or speed range.
3. Number of vibration modes to be applied.
4. Vertical and lateral excitation input combinations.
5. Arbitrary inputs.

c. Number of runs.

d. Desired schedule.

e. Test vehicle description.

f. Fixtures required.

g. Instrumentation.

h. Data reduction requirements.

i. On-site test success criteria.

Prompt TTC response to a formal test request will be accomplished by the RDL Manager who will establish schedule and resource requirements commensurate with current directives. This response will be in the form of a proposed test plan which consists of the test request combined with designated statements of TTC facility, equipment, and staff capabilities to be assigned to meet the requirements of the test request. The test plan will recognize contingencies for typical schedule slippages dependent upon project complexity. The letter transmitting this plan will contain a cost estimate.

Once the proposed test plan has been incorporated into a Test Agreement approved by the sponsor, user, and TTC, the TTC schedule and resource commitment is firm, and the Center will assure the schedule, test, and data commitments are met. For a proprietary test, the test plan becomes the statement of work for the test contract, and execution of the contract is required to establish schedule and resource commitments and to authorize O&M planning and test preparations.

Detailed test planning will also proceed upon receipt of the signed test plan. These activities will progress with due consideration of test complexity, lead time requirements, and clear definition of test objectives.

Upon TTC approval of the formal test plan, a work order number is assigned, and the sponsor and user are authorized direct contact with the O&M Test Engineer. All meetings and coordination involving test scope, schedule, and resources will include the Test Manager. The coordination of engineering and procedural details may be accomplished directly with the cognizant O&M staff under the guidance of the Federal Test Manager. The Test

Manager will be responsible for keeping the sponsor apprised of status and trend of test activities as required for overall program planning and scheduling.

Once a test plan is approved, a test specification is prepared which details the following information. In general, pretest conferences provide the best means of finalizing test planning details. The Test Specification should include:

a. Objective and scope of test in detail.

b. Vehicle configurations to be tested.

1. Vehicle description, configuration, specifications.
2. Lading type, amount, distribution, and restraints.
3. Fixture performance requirements.
4. Detailed drawings.
5. Structural analysis, if any, required.
6. Hazard analysis.

c. Test run sequences and procedures.

1. Run schedules and rationale.
2. Excitation input combinations and rationale.
3. Priority of various test points.
4. Vehicle safety monitoring requirements.
5. Abort and alarm limits.

d. Instrumentation and data reduction requirements.

1. Amplitude and frequency ranges, accuracies, recording format, and priority of instrumentation.
2. Locations, orientation, station notations, and identification of sensors.
3. Definition of engineering unit displays required from test.
4. Definition of axis system.
5. Digital recording format requirements.
6. Data reduction displays: realtime, quick look, and post test.
7. Post test disposition plan.

e. Logistics.

1. Hardware delivery and test schedule.
2. User equipment to be furnished for the test.
3. Post test disposition of test hardware.
4. User services to be made available.
5. Spare parts availability and acquisition procedures.

f. Procedures and reports.

1. Chronology report.
2. Test unit operating procedures.
3. Data handling procedures.
4. Safety rules.
5. Equipment verification procedures.
6. Quick-look review reports.
7. Anomaly disposition.
8. Failure disposition.
9. Test report.
10. Dry run reports.

Although test planning frequently merges into preparations with no discernable transition, the Test Readiness Review serves to identify the completion of pretest planning activity (figure 1).

CONDUCT OF TESTING

The O&M will perform the work as set forth in the test plan and in accordance with the test specification. The O&M will cooperate with the sponsor and the user to ensure that the user will be provided maximum flexibility to meet his objectives and to obtain the most vitally needed information at the earliest possible time consistent with safe and efficient utilization of the RDL.

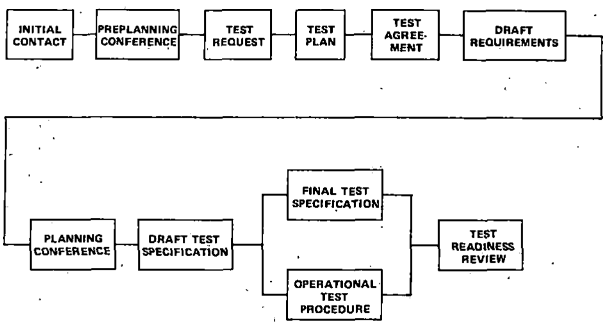


FIGURE 1. TEST PLANNING WORKFLOW

Variations in the approved scope and objectives necessary to the successful accomplishment of the test may be agreed to between the user, sponsor, and O&M during the test, provided time and costs are not affected. Deviations from the approved scope and objectives which impact time and costs must be submitted to the Test Manager for approval action. The approval of the sponsor will have to be obtained in advance. If changes are outside the scope of the test plan or contract, these documents must be modified accordingly.

Test sponsors are responsible for assuring that user commitments and agreements are coordinated and accomplished during all phases of testing in accordance with TTC requirements.

The amount, kind, and degree of reduction of data will be that previously agreed upon at test conferences and documented in the Test Specification. This does not preclude additional reduction of the data desired by the sponsor if such a reduction will result in a better evaluation of test results. If requested, the O&M will furnish to the user representative (and sponsor) test data (preliminary and unverified) as soon as they are available. Preliminary and unverified data are given to the user to expedite the use of test results, and as such they do not constitute an official transmission of the test data. Preliminary data discrepancies should be brought to the attention of the Test Manager immediately. Desired changes in the method or amount of data reduction must be reflected by a modification to the test plan if necessary.

After the test is completed, a compilation of the test data will be prepared in the specified form, and designated reports will be prepared and distributed by TTC.

In order for a sponsor to maintain current knowledge, progress, and status of testing and test results, frequent information exchanges and engineering meetings are encouraged at TTC during the course of testing. Such meetings have proven to be indispensable because of the number of decision-making situations arising from the testing of complex hardware where large quantities of test data are involved.

Whenever possible, engineering coordination can be accommodated when sponsor representatives and user representatives are present at RDL at least periodically during the course of actual testing. The RDL control room includes a crew station for a user (or sponsor) representative for maximum visibility into the momentum of a test operation. When such presence is not possible, the Test Manager will attempt to maintain close liaison and coordination by telephone and telecopier.

The planning and conduct of extended or follow-on test programs is carried out in the same manner as described in the preceding sections. In closely spaced test programs, however, maintaining a knowledge of test lead time requirements and test program scope is essential in providing effective test support at TTC and preserving schedules in time-critical programs. A planned approach and clearly defined test objectives and procedures are the keys to success of any test program.

Although the O&M is responsible for safety in test preparations and operations, TTC will discharge its overall responsibility for test safety by maintaining continuous surveillance of the RDL workflow. In test procedures involving user operation of specialized user-furnished equipment (e.g., operating remote control consoles for traction motors), TTC will require the user to cooperate with the O&M in assuring safe test operations. The user will be responsible for providing adequate hardware, applicable hazards analyses, and qualified operating personnel. TTC will authorize the O&M to inspect the test vehicle and specialized user equipment and to review procedures to assure that it has been prepared in accordance with test documents. The Test Engineer will notify the user representative of discrepancies, and any of these which remain unresolved may be the basis of a Test Manager decision to delay start of testing.

In the event of a test vehicle malfunction which the Test Engineer judges hazardous, the test operation will be delayed pending rectification of the hazardous condition. Should a major failure occur, the O&M will remove the vehicle from the test unit. A determination to extend the duration of an approved test or to cancel the test program will be made only by TTC and the sponsor after consultation with both user and O&M personnel.

The test operations will be culminated by a post test meeting convened by the Test Manager. This meeting will review the test objectives and scope with the user and the test sponsor and assure that all expected test objectives were met. Secondly, the reduced data will be inspected and discussed to determine its accuracy and usefulness and to ensure that both sponsor and user fully understand the proper interpretation of the data obtained. Thirdly, the content and distribution of the final test report is reaffirmed. Finally, the views of both the sponsor and user are solicited regarding improvements which may be worth making in future tests of a similar type and how well the objectives of the test were met.

It is hoped that by a free exchange of information among TTC, sponsor, user, and O&M, the test procedures and capabilities of TTC can be further improved. Another result of the meeting is the possibility of anticipating future needs for additional testing.

BUDGETING

TTC operates as a test and evaluation cost funded organization within the budget and accounting operations of the Department of Transportation. From a user viewpoint, this means that the user must program, budget, and pay the costs for any services obtained from TTC. Furthermore, proprietary users must pay depreciation and interest on the Government investment in the RDL.

Based upon a composite of internal requirements and a consolidated projection of user requirements, TTC contracts in advance for the levels and types of effort required to satisfy projected workload demands. TTC programing and contracting is very sensitive to changing customer requirements. Accordingly TTC should be immediately advised of any program changes which will affect the nature and scope of TTC support expected. Planning and programming between TTC and its users is a continuum with each having an obligation to plan together and to inform each other of any program changes.

Once pretest planning, as discussed in the foregoing section, has progressed to the point of test authorization, certain actions to commit resources are necessary. When pretest planning between TTC, sponsor, users, and the O&M has progressed to a point of mutual agreement regarding what is to be done, when, and for how much, the Test Manager will, following the receipt of a sponsor funding order, prepare a test specification. This document is a compilation of the negotiated agreements as to how the test will be conducted in sufficient detail to satisfy the user requirements. The user or sponsor will approve the specification, and upon receipt of the approved document, the test project formally starts. The O&M initiates internal work orders for commitment of resources to make good the project objectives in keeping with schedule parameters. Sponsors and users are also provided copies of the Test Specification.

As changes occur or project amendments are made, a Test Specification Change Notice will be issued. These changes are coordinated by the Test Manager with the sponsor, user, and O&M and are provided to all involved parties. Generally, the changes are of such a nature as not to require amending the funding order or contract; however, any change in time, scope, or funding necessarily requires amendment to the original order.

Upon receipt of the funding order or contract, a project fund code is established with a limit of the cited funds. As work is performed, the cost is reported internally against this fund code. Cost reports will be furnished to the sponsor as provided in the Test Agreement.

For those tests which are industry sponsored where the user intends to reimburse the Government directly, a test contract will be negotiated between the user and TTC. The following costs will be included in the test contract.

a. Direct labor costs (burdened with payroll loadings) of the O&M.

b. Indirect labor and miscellaneous costs which will be a factor (e.g., 40%) of the direct labor costs.

c. Materials and subcontracted services which directly support the test.

d. A depreciation fee for the Government investment in the RDL. (This fee is computed on the basis of logged hours of use of each major test unit in the laboratory.)

e. An interest fee for the Government investment in the RDL. (This fee is also computed on the basis of equipment usage in hours.)

If the proprietary user is willing to share his test data with the Government, then the depreciation and interest fees may possibly be waived based upon a determination by the Government. Details of the type of costs included in each of these categories are delineated in appendix A.

In order to guide potential users in envisioning test budgets for RDL operations, an example has been prepared. This example describes a test scenario which involves the Vibration Test Unit, Specimen Data Acquisition System, and the Integrated Computer System Network to perform a vibration test of a boxcar, and the example includes schedule and cost details. This example is described in appendix B in briefing chart format.

APPENDIX A

RDL TEST BUDGET DETAILS

Test costs at TTC are basically categorized as "direct" which includes labor, materials, and subcontracted services, or "indirect". However, some limited services are furnished to the test projects where the costs are supported by the Center institutional budget. Identification of specific costs as "direct", "indirect", or "institutional" varies with the specifics of each test program. Hence, the individual items in the following lists should be considered as examples of specific costs which can fall into a category; accordingly the categories are not necessarily limited to items delineated here.

I. <u>Labor Costs</u>
- -- Test planning
- -- Unique or special procedure preparation
- -- Work coordination and control
- -- Design and fabrication of special hardware; i.e., mounting pads for instrumentation sensors or cribbing for lading

-- Installation (and removal) of dummy lading
-- Configuration of test unit (RDU or VTU) to fit test vehicle dimensions
-- Installation, connection, and checkout of instrumentation (including SDAS)
-- Configuration of SDAS patch panels
-- Unique software development; i.e., produce special data plotting
-- Definition and inputting of variables for control and data processing software
-- Wheel truing
-- Conduct the test runs including daily test unit start-up, excite test vehicle, and test unit shutdown
-- Access control if a dedicated guard is required to accommodate proprietary security requirements
-- Photography including stills, television
-- Data processing and handling
-- Logbook keeping and report writing
-- Instrumentation removal
-- Logistic movement of test cars to the RDL over the TTC tracks
-- Calibration of sensors and electronic measurement instruments
-- Safety and Quality Assurance surveillance

II. Indirect Cost
-- O&M management and supervision
-- Material procurement, receipt, and distribution.

III. Materials and Subcontracted Services
-- Steel, aluminum, wood, and plastic stock and fasteners, and adhesives to mount instrumentation sensors
-- Strain gauges
-- Dummy lading (e.g., baled wastepaper, or gravel, or canned dog food)
-- Wood, paper, plastic stock used for cribbing
-- Materials for special positioning or loading fixtures
-- Heavy-lift rigging services
-- Laboratory processing for color photography (still or movie)
-- Electrical energy

IV. Depreciation Fee
Depreciation for the Government investment in the following test units:
-- RDU (Roll Dynamics Unit)
-- VTU (Vibration Test Unit)
-- ICSN (Integrated Computer System Network)
-- SDAS (Specimen Data Acquisition System)

This fee will be charged on an hourly basis for hours involving actual test time (including start-up, test excitation, shutdown); for each test unit the billed hours will be based on usage records in the test log compiled during the operation. The time required for test article installation and storage, and configuration of the test units will not be included in actual test time for calculating depreciation. However, the machine time for post-test data processing on ICSN will be billed. RDL building and crane investment has been partitioned and allocated as a portion of the RDU and VTU investments.

V. Interest
Interest on the aforementioned investments will be charged on an hourly rate for the same number of hours included in the depreciation costs.

VI. Institutional
Other cost elements (e.g., janitorial services, fire protection) related to a test project in a peripheral context will probably (but not necessarily in every project) be considered an institutional cost.

APPENDIX B

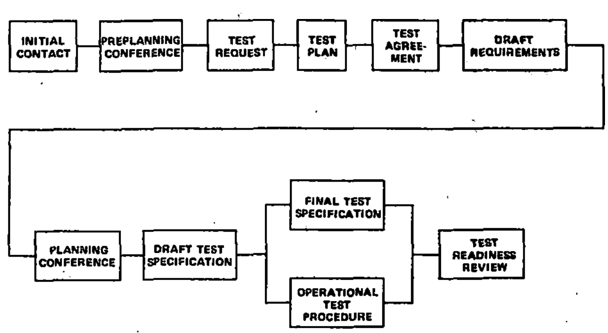
EXAMPLE RDL TEST SCENARIO

"BOXCAR SNUBBER EVALUATION"

INTRODUCTION

- o Scenario illustrates planning and budgeting with a hypothetical proprietary test example.
- o Scenario exemplifies significant planning and budgeting details included in RDL usage.
- o Topics:
  - -- Planning Workflow
  - -- Preplanning Activity
  - -- Initial Planning Scope
  - -- Reduced Planning Scope
  - -- ROM Cost Breakout
  - -- Test Request Introduction
  - -- Test Request Description of Work
  - -- Test Plan
  - -- Test Plan Schedule
  - -- Chronology
  - -- Final Detailed Cost Breakout

TEST PLANNING WORKFLOW



PREPLANNING ACTIVITY

o Initial Contact:

-- From a boxcar fleet operator; February 2, 1977.
-- Perform vibration test to determine if experimental snubber improves ride quality of a boxcar.
-- Conference set for February 10, 1977.

o Conference Input By Sponsor:

-- Objective is to vibrate typical car (and fragile lading) with current snubber and experimental snubber and measure improvement in ride quality.
-- Perform test during the next summer.
-- No definition of instrumentation, data processing, or reporting

o Technical Approach:

-- Perform a series of frequency sweeps.
-- Apply linear sinusoidal, excitation to various boxcar configurations.
-- Assess snubber performance by evaluating processed data in transfer function and mode shape formats.
-- Perform track geometry simulations to complete excitation.

INITIAL PLANNING SCOPE

o Configurations:

1. Bare car with no snubber
2. Bare car with operational snubber
3. Bare car with experimental snubber
4. Loaded car with no snubber
5. Loaded car with operational snubber
6. Loaded car with experimental snubber

o Vibration Excitations

-- Sinusoidal frequency sweeps 0.2 to 30 Hertz

1. Vertical Translation
2. Lateral Translation
3. Pitch,
4. Roll,
5. Yaw
6. Programmed Max responses

-- Following are track simulations

7. Geometry Profile 1 (Mild)
8. Geometry Profile 2 (Severe)
9. Geometry Profile 3 (Extreme)

o Amplitudes

1. 100% - High
2. 80%
3. 60% - Medium
4. 40%
5. 20%
6. 10% - Low

o Measurements:

-- 16 Car Displacements
-- 92 Car Accelerations
-- 6 Angular Car Rates
-- 12 Shaker Displacements
-- 12 Shaker Accelerations
-- 12 Shaker Forces

INITIAL PLANNING SCOPE (COMPLETED)

- Data Processing:
  - -- Transfer functions (amplitude of response/force ratio vs. frequency and response/force phase shift vs. frequency); all response displacements and accelerations.
  - -- Amplitude vs. frequency; angular rates, shaker displacements, and shaker accelerations.

- Run Total:

--	6 Configurations, 5 Excitations, 6 Amplitudes:	180	Runs
--	6 Configurations, 4 Excitations, 3 Amplitudes:	72	Runs
	TOTAL	252	Runs

- On-Site Data Evaluations: 15, 2 days each

- Reports:
  - -- Quick-look (no more than 10 pages): one per configuration
  - -- Final (4 volumes)
    - Volume 1: Narrative, Chronology
    - Volume 2: Processed Data Compilation
    - Volume 3: Photographs
    - Volume 4: "As-run" Operational Test Procedure (OTP)

- Estimates:
  - -- Schedule: Approximately 90 days for test operations and vehicle preparation.
  - -- Final Report: Issued 6 weeks after final run.
  - -- Rough-Order-Magnitude (ROM) Cost: $190K
  - -- Cost most sensitive to volume of runs and data processing.

SCOPE REDUCTIONS

- o Cost estimate over sponsor budget.
- o Cut number of runs substantially.
- o Data reduction will be proportional to reduction in number of runs.
- o Reduce reporting.
- o Don't reduce configurations, excitations, or instrumentation.
- o Consider reduction in amplitudes.
- o Number of data evaluation periods should be reduced appropriately.

REDUCED PLANNING SCOPE

o Run Total:

- -- Configuration 1; Excitations 1-9; Amplitudes 1, 3, and 6: 27 Runs
- -- Configuration 3; Excitations 1-9; Amplitudes 1-6: 34 Runs
- -- Configurations 2, 4, 5, and 6; Excitations 6-9, Amplitudes 2, 4 and 6: 48 Runs
- -- Contingency to resolve data evaluation uncertainties: 20 Runs
- -- TOTAL: 149 Runs

o Reports:

- -- 2 Quick-look reports
- -- Final test report; data compilation to include results from only 50 selected runs.

o Data Evaluations Reduced from 15 to 11

o Estimates:

- -- Schedule: Approximately 75 days
- -- ROM Cost: $130K

ROM COST BREAKOUT

Cost Item	Amount ($000)
Labor (5000 manhours at $13/mh)	65
Materials	15
Equipment Depreciation and Interest	27
Subtotal	107
Contingency (approximately 20%)	23
TOTAL	130

- o Sponsor discussed several options to reduce cost further but finally agreed on March 7 to $130,000 as minimum test budget to achieve objective.

- o Sponsor initiated test request.

- o RDL Manager allocated test period: June 15 to September 1; established authority-to-proceed (ATP) milestone as May 1.

TEST REQUEST

- o Letter of transmittal

  - -- Addressed to RDL Manager.
  - -- Affirms intent to negotiate contract for test.
  - -- Transmits test request as attachment.
  - -- Designates John Smith as sponsor technical representative and James Jones as Contracting Officer; lists phone numbers and addresses of each.
  - -- Requests test plan and cost estimate.
  - -- States draft requirements available in one month.
  - -- Signed by Company Officer (E.J. Brown, Vice-President of Engineering).

- o Attached technical scope consists of two parts:

  1. Introduction
  2. Description of work

TECHNICAL SCOPE INTRODUCTION

- o Title: Boxcar Snubber Evaluation (BSE)

- o Objective: Obtain vibration response data to support evaluation of experimental snubber performance.

- o Requested Test Unit:

  - -- VTU (Vibration Test Unit)
  - -- ICSN (Integrated Computer System Network)
  - -- SDAS (Specimen Data Acquisition System)

- o Test Vehicle Availability: June 1, 1977

- o Data Distribution Limitations:

  - -- Company Private
  - -- 6 copies of reports: 5 to sponsor representative; 1 kept at TTC in controlled-access storage
  - -- 3 copies of data plots and tabulations: 2 to sponsor representative; 1 to test engineer

TEST REQUEST DESCRIPTION OF WORK

- o Boxcar NO. 123456 with modified trucks and two test sets of snubbers (1 operational, 1 experimental);

- o Lading: 50 tons of baled waste paper evenly distributed within volume; cribbed to be non-resonant below 10 Hertz.

- o Instrumentation:

  - -- Displacements: Axle end to truck side frame; 8 places truck side frame to carbody; 8 places shaker; 12 places
  - -- Accelerations : Along side displacements, 28 places
    - : truck bolsters; 14
    - : body bolsters; 14
    - : truck and body kingpins; 12
    - : body mode shape; 48 (8 planes, 6 meas/plane).
  - -- Angular Roll Rates: Car ends around three axes; 6
  - -- Shaker Forces: 12

BSE RUN MATRIX

	Configuration	Excitation	Amplitudes	Runs
1.	Bare Car/No Snubber	1. Ver, Tl., 2 Lat. Tl., 3. Pitch, 4 Roll, 5 Yaw 6. Prog. Max. Resp's 7. Geo. Prof. 1, 8. Geo. Prof. 2, 9. Geo. Prof. 3.	100% 60% 10%	27
2.	Bare Car/Operational Snubber	6, 7, 8, 9	80%, 40%, 10%	12
3.	Bar Car/Experimental Snubber	1 through 9	100%, 80%, 60%, 40%, 20%, 10%	54
4.	Loaded Car/No Snubber	6, 7, 8, 9	80%, 40%, 10%	12
5.	Loaded Car/Operational Snubber	6, 7, 8, 9	80%, 40%, 10%	12
6.	Loaded Car/Experimental Snubber	6, 7, 8, 9	80%, 40%, 10%	12
7.	Contingency	TBD*	TBD	20
			TOTAL	149

*To Be Determined

DESCRIPTION OF WORK (CONTINUED)

- Data Processing for Each Run:
  - -- Transfer Functions: Each car acceleration/master force; 104 plots
    Each car displacement/master force; 16 plots
  - -- Amplitude vs. Frequency: All other measurements; 30 plots
  - -- Mode shape for 24 resonances; 24 plots
  - -- Damping values for 24 resonances; 1 tabulation, 24 values
  - -- Total: 174 plots and 1 tabulation stored in data management files on disc in ICSN

- Data Evaluation:
  - -- Inspect, evaluate, and edit, data using data management files in interactive mode through ICSN CRT Terminal.
  - -- Transfer selected data to evaluation files for correlation with data from other runs.
  - -- Evaluate and manipulate data in evaluation files using interactive mode.

- No Fixtures Required

- On-Site Test Criteria: Acceptable data from each run as assessed within two days by sponsor data evaluation team

- Reports:
  - -- Two quick-look reports
  - -- Final test report including hard-copy reproductions of data evaluation

- Miscellaneous:
  - -- Still Photographs: Measurement locations, setups, 15 other views
  - -- Cinematography (24 Frames Per Second): Six runs with significant motions (4 cameras)
  - -- Closed Circuit Television (CCTV): 4 cameras
  - -- Rigorous controlled access required for control room and data evaluation room
  - -- Normal access control required for test area

TEST PLAN

- Defines Boxcar Installation on VTU
  - -- Removal of coupler, brake system, doors
  - -- Fixture and procedure outline for snubber installation and removal
  - -- Fixture, tools and components for lading installation, non-resonant cribbing, and removal

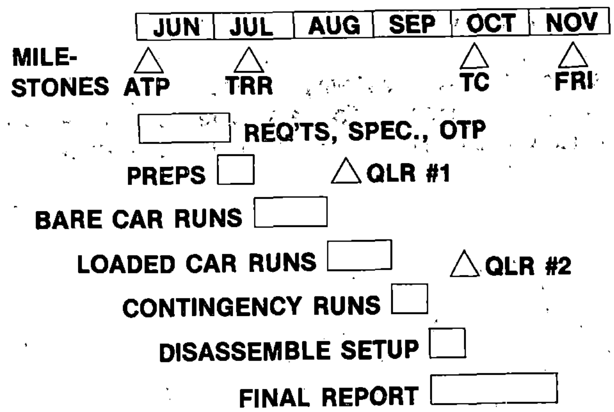
- Defines Time Spans for Test Operating Tasks
  - -- Instrumentation sensor installation
  - -- Car installation
  - -- SDAS, ICSN, VTU System operations verification
  - -- VTU start-up, run execution, shutdown
  - -- Data processing (per run)
  - -- Reconfiguration (snubber and lading installation/removal)
  - -- Specification, procedure, report preparation
  - -- Car and sensor removal

- Defines Sensor Types (including size and sensitivity)

- Schedule (next sheet)

- Letter of Transmittal
  - -- Affirms occupancy time allocation
  - -- Detailed cost estimate enclosed
  - -- Contract form enclosed signed by TTC Contracting Officer
  - -- Proposes date for planning conference
  - -- Addressed to company officer
  - -- Signed by RDL Manager

SCHEDULE



CHRONOLOGY

- Negotiating Contract took longer than expected; result was delay in ATP (Authority to Proceed) to July 7, 1977.

- Subsequent planning and test operations ran close to test plan time lines.

--	TRR	Sept. 6, 1977
--	Bare Car Runs Complete	Oct. 17, 1977
--	Loaded Car Runs Complete	Nov. 10, 1977
--	Contingency Runs Complete (TC)	Nov. 29, 1977
--	Test Disassembly Complete	Dec. 8, 1977
--	Final Test Report Issued (FRI)	Jan. 17, 1978
--	Final Cost Report Issued	Mar. 16, 1978
--	Government Reimbursement Paid	May 1, 1978

- Technical objective achieved with few data anomalies

SUMMARY COST

Labor*	$ 57,898
Indirect (40% of Labor)	$ 23,159
Material/Other	$ 11,350
Depreciation and Interest	$ 29,632
TOTAL	$122,039

NOTE: *Labor calculated with average rate; for actual test, payroll data would be used.

DETAILED LABOR BREAKOUT

	ITEM	NUMBER OF HOURS	RATE ($/HOUR)	LINE TOTAL ($)
1.	Planning and Pretest Analysis	600	8.85	5310
2.	Work Planning and Control	390	8.85	3452
3.	Reports	363	8.85	3213
4.	Fixture Design	120	8.85	1062
5.	Test Preparations*	1128	8.85	9983
6.	Test Operations (including overtime)	3301	8.85	29214
7.	Cross-Charges (calibration, photo, CCTV, fixture fabrication, rail logistics)	280	8.85	2478
8.	Safety and Quality Assurance	360	8.85	3186
	TOTAL	6542		57898

*Including instrumentation, installation, system preparation, reconfiguration, disassembly.

DETAILED MATERIALS/OTHER DIRECT COST BREAKOUT

	ITEM	QUANTITY AND RATE	LINE TOTAL ($)
1.	Sensor Mounting Blocks	150 @ $1 each	150
2.	Dummy Lading and Cribbing	Lump sum	2150
3.	Fixture Materials	Lump sum	875
4.	Data Processing Supplies	Lump sum	4150
5.	Photo Processing	Lump sum	825
6.	Electrical Energy	164 Megawatt-Hours @ $19.50/MWHR*	3200
	TOTAL		11350

*MegaWatt-Hour

DEPRECIATION AND INTEREST FEE BREAKOUT

	ITEM	NUMBER OF HOURS	RATE ($/HOUR)	LINE TOTAL ($)
1.	VTU	168	117.03	19661.04
2.	SDAS	168	18.63	3129.84
3.	ICSN	199.5	34.29	6840.85
4.	RDU	Not Used	112.37	-0-
	TOTAL			29631.73
	ROUNDED TOTAL		29632	

NOTE: Rate will increase as investment in upgrading is added.

SUMMARY

- o Scenario uses hypothetical boxcar vibration test to illustrate key points to be considered in RDL test planning and budgeting.
- o Vital components of test request delineated.
- o RDL cost elements, rates and magnitudes delineated.
- o Typical lead time denoted.
- o No actual test will simulate scenario since effects of several minor technical and planning complexities have been eliminated for brevity.
- o Sponsors with specific problems are invited to contact the RDL Manager to arrange a preplanning conference and ROM cost estimate.

QUESTIONS SESSION VI

Session Chairman - Sergei G. Guins

Attendee: R. A. Boerke, North American Car Corporation.
Attendee's Question: References to specific components in your paper avoided manufacturers' names. This is probably good policy to avoid premature judgment based on preliminary data which is not qualified by other related conditions. Will the final reports fully identify such failed components?

D. Gray: If I understand the question properly, in the donor's agreement for FAST, it was agreed that the suppliers of equipment would be given copies of their results, however, the results of other suppliers would be coded so that only the generic name would be available. So, it would not identify manufacturers in the published results.

Attendee: J. B. Smythe, Office of the Secretary, DOT.
Attendee's Question: What differences in the quantity of track maintenance between the different track types has been noted, and what conclusions can be drawn from these data. In particular, concrete versus wood ties.

M. McCafferty: That's an interesting question. The fact that we're at only 130 MGT means, of course that it's a fairly short time period. For most good tracks, you would not be doing an out-of-face maintenance operation. That's basically the case in FAST in that there's been spot maintenance done in some sections. But you can find other sections, for example the two tangent sections of continuous welded rail and jointed rail, where there's been no maintenance. FAST has the other complex situation where some of it was new track and some of it was old track. For the new track, after a run-in period of about five or six MGT, there was a skin lift and a complete retamping. So, it's difficult to go in and look at the data itself and say there's so many man-hours spent on this section and so many spent on that section. I think the major thing that is obvious to most people is especially that once you've got a broken rail, you've got to get in and perform a field weld. If you leave a joint in concrete ties for a while, it will end up a bad piece of track. The longer you leave it in, the worse it gets. The worse it is, the longer it takes to get it rehabilitated back to a good section. The same is true of the rail corrugations. If you let them get too big, you start getting a problem, again the worse the problem gets, the harder it is to bring it back. Some of the differences between concrete and wood are in maintenance operations. The concrete ties themselves, of course, are deeper, they weigh more; therefore, you've got to make some adjustments to the tamper in order to get the tie tamped up in order for it to give you good track. We are collecting all the data, all the man-hours, all the resources expended; and as we get on through a complete life cycle, I think we'll know a little more about what the overall economics might be.

Attendee: R. A. Boerke, North American Car Corporation.
Attendee's Question: Descriptions of both the VTU and RDU show an upper limit of 98 inches for center of gravity. Is this actually a mechanical restraint on the test equipment or does it simply reflect AAR regulations?

D. deBenedet: Ninety-eight inches is a tentative design parameter. Higher CG's can be handled, but you will have to reduce the magnitude of lateral ac-

---

Sergei G. Guins is currently acting as AAR Director of the FAST Project at the Department of Transportation Test Center in Pueblo, Colo. He is a graduate of the University of Michigan with a BS in Aeronautical Engineering and an MS in Applied Mechanics. He retired from the Chesapeake and Ohio Railroad in 1970, as Assistant Director of Research.

celeration in order to not overtax a restraint system which has been built into the VTU configuration to prevent overturning in case of prime power failures. Whether that's an absolute limit, I couldn't answer in terms of AAR regulations. But, the machine would only have to be limited in terms of how much excitation you introduce laterally. You could go to 100-inches or something higher with a commensurate reduction in lateral input. The RDU has no problem in higher CG's. Note: Mr. D. deBenedet is the RDL Program Manager at Wyle Laboratories, Colorado Springs.

Attendee: G. Platt, Phillips Petroleum Co.
Attendee's Question: What plans have been made for combined tests from different suppliers or requesters?

W. Dorland: The answer to that one is easy, none. We are soliciting test ideas. The Center itself will not sponsor tests. We are a performing organization. Tests of that nature I would like to see come to us from RPI or AAR. We'd be most receptive to developing a management plan for combined tests much as is done on FAST.

Attendee: G. Liebig, Surface Transportation International.
Attendee's Question: In your evaluation of truck components have you observed accelerated center plate wear? Is 16-inch diameter better than smaller diameters? Are wear liners effective? Also, is car body center plate wearing comparably?

D. Gray: I don't think to date we have enough data that really indicates if there's any significant difference between the 14-inch or 16-inch center plate.

Attendee: P. Marcotte, Canadian National Rail Research.
Attendee's Question: I believe that actual measurements of a truck position and tracking attitude of the truck in the curves would be of great help to analyze the wheel/rail wear problem. Are there any plans to acquire any such instrumentation for FAST test in the future?

S. Guins: There has been a question of how various car configurations and car location in the train affect lateral forces. A special test is being planned to attempt to resolve the above question. During a normal train operation, data will be collected for ten laps. The instrumentation will consist of Battelle tie plates and strain gages on the rail. While this type of instrumentation may not give us absolute values, we should have a good relative values.

Attendee: H. A. List, REA Inc.
Attendee's Question: What is your present wheel profile strategy?

D. Gray: Well, as I mentioned, we're planning to repeat the wheel wear experiment. Initially when we were comparing the standard AAR profile with the CN profile, the CN profile wasn't available for some of the wheel hardnesses as tested. As a result, we had to turn these from the AAR standard profile.

In these next series of tests, we're planning to get some new CN wheels, treated wheels, with a new CN profile. We'll also run a few of the ones turned from the AAR profile to see if in fact there is any difference in the wear. As I also mentioned, we're planning to add some worn wheels (deep flange wheels) to hopefully control the wear pattern we had with the lip building up on the gage face of some of the curve portions of high rail.

Attendee: F. Houser, Railroad Research Information Service.
Attendee's Question: Are RDU tangent and curve test modes separate or can movement of vehicles into and out of curves be reproduced?

A. Gross: The design requirement for the Roll Dynamics Unit (RDU) is essentially for two distinct conditions, the tangent track and for a steady state curve condition. The RDL user could test his vehicle in either or both of these test conditions. The RDU cannot instantaneously switch from one to another test condition.

Attendee: D. Smit, General Motors Corp.
Attendee's Question: In FAST there are some 22 sections of track being tested. Has this caused any appreciable delay in running the train due to the construction of track? If so, has consideration been given to a parallel temporary track construction to facilitate testing continuance while reconstruction of the affected track section is completed?

M. McCafferty: I think it's an obvious yes! Of course, it causes problems. If you shut down, you can't run the train. We've got that situation right now, and we're not able to show you the train running over the track. We do have the conflict that exists between the high speed 14-mile loop and the FAST loop. We are in the process of building a bypass at that point. We had not, I don't think, considered a bypass primarily because of funding. Building basically a second 4.8 mile loop would cost in the area of a million or so dollars. Our budget just doesn't allow that type of an operation.

S. Guins: In this type of accelerated testing, it is vital that in comparing effect of various equipment or track operating conditions must be kept as constant as possible. This means that train speed, track curvatures, etc. must be maintained as nearly constant as possible. With as many different sections as we have, all wearing out at different times, double tracking would be impractical.

Attendee: D. Smit, General Motors Co.
Attendee's Question: Is there interest and/or a plan to utilize the FAST testing as a means of focus on research and development to affect new failsafe designs for railroad components?

S. Guins: At present the rules are that we don't test anything that hasn't been approved by one of the AAR committees, or at least given tentative approval. But, obviously, as we proceed there's going to be newer components being introduced. We have requested at this stage to introduce a variety of additional types of wheels. We've been only testing Grade C and Grade U wheels, there are requests to include some Grade B wheels and Grade C wheels of different types of heat treatment. So, we are going to do some developmental work as we go along prior to introducing these components, but it's going to go through the normal bureaucracy that we have to face. You know these components will be submitted to the AAR committees, for example, in wheels it will go to the WABLE Committee. The WABLE Committee will review the proposal, recommend whether we do or don't, then we're going to have to see, for our Experiment Managers, how it fits in the program, how it fits in our finances and timing, and gradually they may be included as some of the wheels wear out. I think this will apply to a variety of other components. So, the answer is yes and no.

Attendee: A. Krauter, Shaker Research Corp.
Attendee's Question: During steady state curving runs on the RDU, how does the run cause the simulated outside rail to move faster than the inside rail?

A. Gross: I think you pointed out a feature where the RDU has some limitations. What we plan to do for curve simulation is to have the outside rollers of a larger diameter than the inside rollers. This would cause the simulated outside rail to move faster than the inside rail.
Attendee: No Name Given
Attendee's Question: Data identifying the cause for wheel replacement indicates a significant improvement in the treated wheels over the untreated wheels in all categories except for cracks and shelling. Is the comparison based on usage of an equal number of wheels and is there any effort underway to correct the cracks and shelling on the treated wheels?

D. Gray: I had mentioned when I showed the slide that the number of C Class wheels or I should say that the mileage for the C Class wheels was greater than some of the U Class wheels because of the obvious replacement due to thin flange wear. Probably a better way to have shown this comparison would be on equivalent mileage. As far as correcting the shelling, I think that in revenue service it has been shown for about the same equivalent mileage (I think was about 50,000 miles) for that unit coal trains where they were half empty for half the time this is roughly comparable to shelling that they've seen as well.

S. Guins: One of our problems is lack of experience in interpretation of AAR definitions as written in the wheel and axle manual. Railroads have people who specialize in this area. Lack of this experience and the desire to extend the life of wheels in the consist caused early removal of wheels with indication of thermal cracks and shelling. In the future, we will have a more precise definition for the Test Center to use.

Attendee: M. F. Hengel, Missouri Pacific Railroad.
Attendee's Question: Does the test train operate in the speed range of 10 to 22 miles per hour for any substantial period of time? If so, have any of the usual operating characteristics associated with this speed range been observed?

S. Guins: The answer is really no. We have not operated at low speeds. We try to operate at the continuous speed of an average of 42 mph so the only time we operate at low speed is to start up and slow down. The rest of the time we run at speeds designed to maintain a constant condition for track/train relationship on the curves. We're working with two inches unbalanced superelevation which creates a fairly heavy wear on the high rail but this is one of the decisions of our operating conditions. No, we haven't had any low speed operation and we haven't had any rock and roll type of things that I think Mark is directing his attention to. We did have some on occasions where we had a softness on the track on occasions, but this hasn't been continuous enough to really see the rock and roll problems.
Attendee: M. Jacobs, Federal Railroad Administration.
Attendee's Question: What plans are being made to obtain a constant set of locomotives for FAST?

S. Guins: The problem is that in order to distribute the load between the different contributors, it's very unfair to ask anyone to donate a set of locomotives for long periods of time and donate something that's 2½ million dollars worth of investment. So far we've been working on the four units per donor for every three months and this creates all kinds of pain and problems. There is a consideration being made in creating a means of providing a full-time locomotive here in the future.

SESSION VII TRANSPORTATION TEST CENTER

UPDATE OF TTC ACTIVITIES AND FUTURE PLANS

BY

E. R. Mathews

Executive Summary

From a singular and rather specialized role as a high speed vehicle test facility, this installation has taken on larger responsibilities as the Transportation Test Center. The range of testing has broadened considerably. Testing of the exotic and the futuristic has been placed in proper perspective and we are now conducting testing related to the very real and immediate problems that conventional railroad and rapid rail transit systems are facing.

The FAST Program, which is covered in other Conference papers, has provided the added capability for testing track structures as well as rolling stock.

The Rail Dynamics Laboratory is nearing completion. Another paper presented today has provided more detailed information on this subject. This laboratory will be included in the afternoon tour of the Test Center.

Since testing is the only reason that the center exists, it's appropriate that I begin by describing our major program accomplishments in the past year and discuss briefly the tests that will be conducted in the future.

I believe that the broad scope of the test programs planned for the Test Center together with the planned facilities and equipment acquisitions clearly indicate the intent of the Federal Railroad Administration to make significant contributions to the state-of-the-art in rail systems. A wide variety of rail system test facilities is -- and will continue to be -- made available or use by the various elements of the rail industry. Inquiries regarding the use of the Test Center's facilities are cordially invited.

UPDATE OF TTC ACTIVITIES AND FUTURE PLANS

BY

E. R. MATHEWS

Since most of you have visited here before, I have elected to bring you up to date on the activities at the Transportation Test Center and provide you a preview of test and construction planning.

From a singular and rather specialized role as a high speed vehicle test facility, this installation has taken on larger responsibilities as the Transportation Test Center. The range of testing has broadened considerably. Testing of the exotic and the futuristic has been placed in proper perspective and we are now conducting testing related to the very real and immediate problems that conventional railroad and rapid rail transit systems are facing.

The FAST Program, which is covered in other Conference papers, has provided the added capability for testing track structures as well as rolling stock.

The Rail Dynamics Laboratory is nearing completion. Another paper presented today has provided more detailed information on this subject. This laboratory will be included in the afternoon tour of the Test Center.

Since testing is the only reason that the center exists, it's appropriate that I begin by describing our major program accomplishments in the past year and discuss briefly the tests that will be conducted in the future.

RECENT TESTING ACTIVITIES

Perhaps the most critical of all the tests are those relating to railroad safety. For instance, three test programs last year involved various aspects of Hazardous Material Tank Cars.

The first of these programs was the Tank Car Torching Study. A large, high velocity propane torch was built at an isolated site on Test Center property. It is used to project a high temperature flame against steel plates and actual tank cars that were coated with special thermal insulation materials designed to retard heat buildup. These tests have indicated that such insulation materials would be effective in preventing a catastrophic explosion if the tank car is exposed for a prolonged period to a fire resulting from an accident.

FIGURE 1. TANK CAR TORCHING STUDY TEST

The second program in this category was a series of impact tests simulating switchyard accidents in which a tank car filled with a hazardous material such as propane fuel is punctured by the coupler of another car. A puncture of this type almost invariably results in ignition of the contents of the tank and can lead to "chain reaction" explosions of adjacent tank cars.

The impact tests were set up by positioning an empty hopper car several feet in front of a standing hundred-ton tank car, then impacting the hopper car with one or more loaded cars moving at switchyard speeds. The impact would frequently cause the

Edward R. Mathews has served as Director of the Transportation Test Center for the Federal Railroad Administration since June 1976. He received his Bachelor of Civil Engineering degree from George Washington University (1952) and earned his Master's Degree from the Massachusetts Institute of Technology (1971) through award of the Alfred P. Sloan Fellowship by the Kennedy Space Center.

opposite end of the hopper to rise and, in turn, impact the standing tank car.

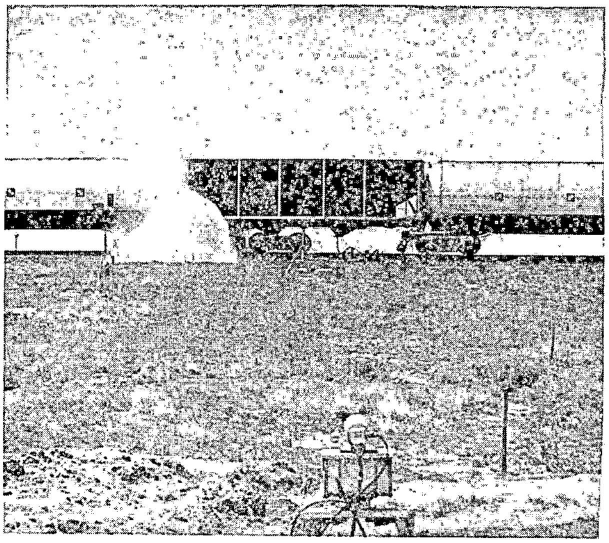


FIGURE 2. SWITCHYARD IMPACT TEST

These tests involved tank cars both with and without head shields, and some were also equipped with type-E shelf couplers that are designed to keep the mating coupler knuckle from slipping up and out of the tank car coupler. Each tank car was loaded with a measured amount of water under pressure to provide the same characteristics as a load of propane. Much information was obtained about the mechanics of this type of accident by the time the test program was completed last December.

Another program in this category, which had begun at the time of last year's Engineering Conference, is the Accelerated Life Test Program. These tests are for the purpose of discovering how well the tank car thermal insulation coatings should hold up under prolonged revenue service conditions. Hundred-ton tank cars coated with various types of the insulating material are used in the tests. After they were subjected to coupling impacts, these cars are now being periodically exposed to a strong salt-water spray and run for thousands of miles in simulated revenue service. A little over a year ago this test program was merged with FAST. By now, the Accelerated Life Test tank cars are approaching an average of 70,000 miles each. That's almost half way to the goal of a 160,000 miles, which should be achieved within a year.

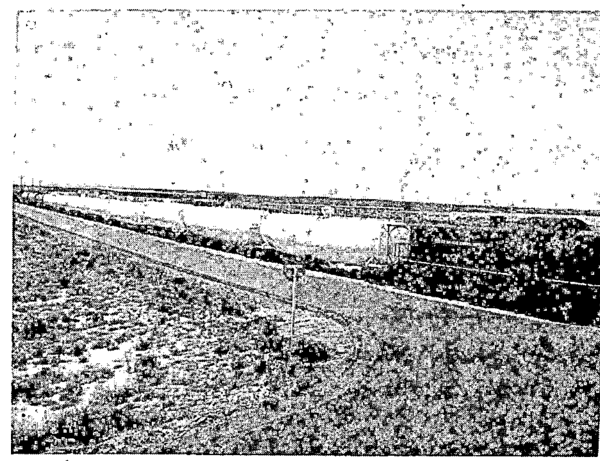


FIGURE 3. TANK CARS IN ACCELERATED LIFE TEST PROGRAM

Other safety related tests are also under way at the Test Center. A number of wayside safety inspection devices are being tested. Last year at this time, for example, the installation of a commercial wheel flaw detecting device was just completed. Later it was extensively tested and then installed on the FAST bypass track for use in checking the wheels of the FAST consist for cracks. The latest effort in this area involves evaluating a Wayside Braking Inspection System that is now being installed on a tangent track section on the west side of the Railroad Test Track. This equipment is designed to detect abnormal train braking performance.

In the category of freight equipment, the first series of tests of the Aerodynamic Trailer-on-Flat-Car program was completed less than a month after the previous Engineering Conference last year. This program began with wind tunnel tests on a model of a TOFC train at the California Institute of Technology back in 1975. The ultimate objective is to develop practical and simple streamlining of the trailer bodies and containers used in this kind of service to reduce the rather severe air drag of the present designs. The Test Center's part in this effort was to validate the results of wind tunnel testing of the model by measuring the air drag forces on a trailer-on-flat-car consist. Later this month, a second series of tests should begin, using a new measurement system designed to give more consistent results at high speed.

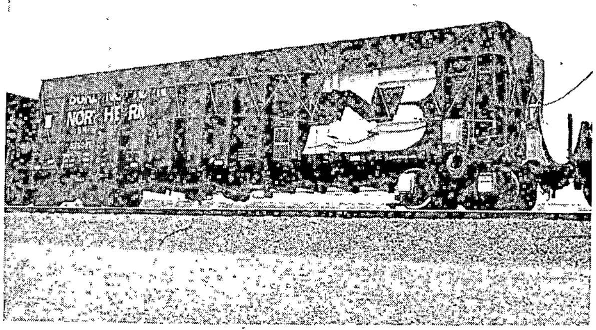


FIGURE 4. AUXILIARY SNUBBING DEVICES TEST CAR

Last June and July, tests were conducted for the AAR to determine the performance of a number of auxiliary snubbing devices installed on the standard trucks of a 4,000 cubic foot rotary dump hopper car. This car was run in a consist, both empty and loaded with 100 tons of ballast, over various perturbed track conditions.

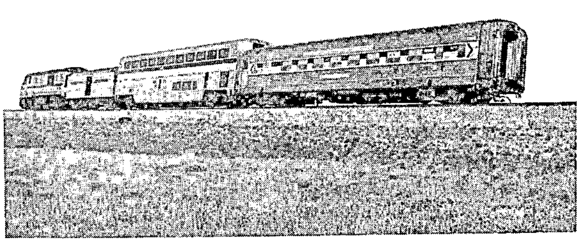


FIGURE 5. AMTRAK PULLMAN BI-LEVEL COACH DURING TESTS

Testing of equipment for passenger service in the past year included a series of tests on Amtrak's new bi-level coaches built by Pullman. These tests began last May with one car, which was supplemented in the late summer by three additional cars. Ride quality and acceptance tests were conducted along with road qualification tests on simulated revenue runs. This test series was completed on October 6, 1977.

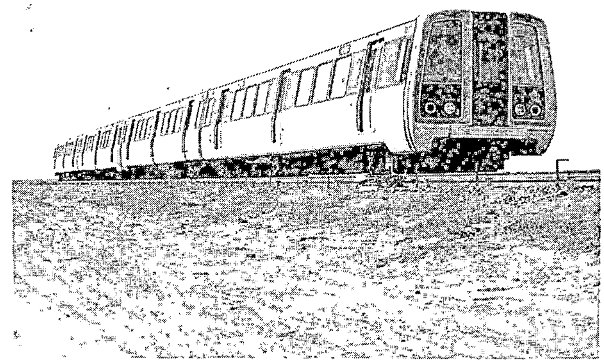


FIGURE 6. WASHINGTON METRO (WMATA) CARS

Another area of passenger service of importance at the TTC is that of transit system testing for the Urban Mass Transportation Administration. At the time of last year's conference, four Washington Metro cars had just been received and were being prepared for testing on the Test Center's electrified transit loop. By August 5, all the tests to evaluate their operational characteristics had been completed and each of the cars had run over 20,000 miles of simulated revenue service to prove their reliability. The four cars are still at the Test Center, and will be seen during the tour this afternoon.

Another recent arrival at the Test Center is the Advanced Concept Train ... or ACT-1. Preparation of the cars for testing should be finished by December. Then a lengthy series of tests will be conducted during the following six-month period. The two cars of this train contain a large number of advanced concepts and subsystems. The most productive advancements over today's state-of-the-art in transit car design are in the areas of regenerative braking and onboard energy storage, reduced car weight, air conditioning and equipment cooling.

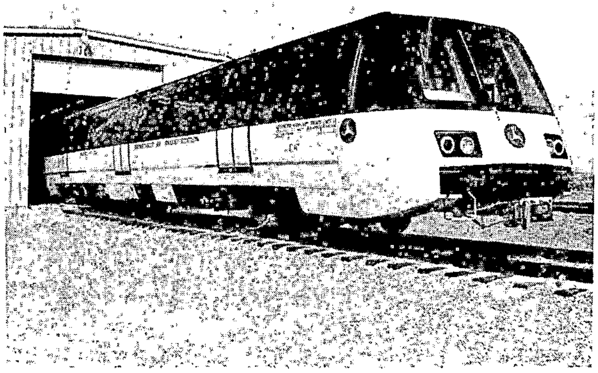


FIGURE 7. ACT-1 ADVANCED CONCEPT TRAIN

Another advanced concept currently being tested is the linear induction motor propulsion system. Last July, the final phase of testing of the double-sided motor originally installed in the Linear Induction Motor Research Vehicle was completed. As you probably remember, this is the vehicle that set a new world's speed record for steel-wheel-on-rail vehicles of 255.4 miles an hour, back in August of 1974. More tests are planned with the vehicle and a different motor configuration.

FUTURE TESTING ACTIVITIES

Future tests related to railroad safety include a follow-on to the Hazardous Material Tank Car Pool Fire tests that were conducted at White Sands Missile Range in New Mexico four years ago. In two of these tests, a full-scale tank car containing propane was placed in a large pit fed wth jet fuel. That way, the car could be kept surrounded with a hot fire until the tank ruptured and exploded. An uninsulated tank car and one with a one-eighth-inch coating of thermal insulation were tested. The insulation proved very effective in prolonging the time to rupture and reducing the severity of the explosion.

A large pit and other facilities have been built in a remote location in the northern part of the Test Center where, in December 1977, a similar test will be conducted with a double-wall tank car with insulation in the annular space. This test should prove the effectiveness of the tank car insulation on tank cars with high flow relief valves.

Other safety-related test programs to be conducted at the Test Center include a freight car brake shoe performance investigation for the Association of American Railroads. Twelve brake shoes of each type will be tested by subjecting them to stopping distance tests, drag tests and static holding tests. It's anticipated that these tests will be completed early in 1978.

In the category of future plans for testing passenger equipment performance, it is expected that there will be testing and extensive evaluation of a prototype radial truck design for high speed passenger cars beginning in 1979. The total program should take about a year, with nine months of tests and demonstrations following three months of preparations.

In the area of transit car testing, the Advanced Concept Train will be the center of attention in the immediate future. The testing of these two cars should be completed by midsummer of next year and the test program for new Massachusetts Bay Transit Authority cars from Hawker Siddeley will begin. These tests are intended to demonstrate that the new Boston transit cars meet their design specifications. These cars will also be run through the general vehicle test series that was developed especially for transit car test programs. Four cars will be tested, with the first two scheduled to arrive at the Test Center in April.

In November of 1978 the two State-of-the-Art Cars ... or SOAC's, as they are more commonly called ... will return to the Test Center after an absence of more than four years. These cars will be modified with new propulsion, braking and truck designs which will be tested and compared with the performance of the components originally installed in the cars and similar components on other transit cars.

Looking farther down the line, transit cars designed for the Metropolitan

Atlanta Regional Transportation Authority should be received late in 1978. These cars will undergo the same tests as the Boston cars plus thousands of miles of reliability test runs.

A year from this coming January, construction of a guideway loop for advanced "people-mover" vehicles should begin in the Test Center core area. This will be the first step in the Advanced Group Rapid Transit test program that will evaluate the components and procedures unique to this type of system. It should take about a year to build the guideway and its associated passenger and service facilities, so actual testing should begin early in 1980.

In the area of advanced systems testing, the conversion of the motor in the Linear Induction Motor Research Vehicle to a single-sided configuration is now being completed. The old vertical aluminum fin, or reaction rail, was removed from the track and will soon be replaced with a horizontal reaction rail laid flat at the height of the rails. Changing the tall vertical fin to a flat horizontal member placed low between the rails, will eliminate a number of obstacles to operation, such as making it possible to have crossings at grade and facilitating switching. Check-out and initial performance testing should begin in December.

RECENT FACILITY CONSTRUCTION

I don't intend to go into details on all of the new items that have been built for the FAST Program in the last year, since that has been covered in a previous paper this morning, but I would like to mention in passing the new FAST Service Facility with its yard tracks and locomotive servicing equipment, all located next to the south tangent of the FAST Track. We also now have a paved road, some three miles long, connecting the FAST Service Facility area with the Test Center core area.

The Test Center's nine-mile Transit Track is being equipped with two new substations that will provide precisely regulated electric power. The electrical equipment is presently being installed in the substation buildings and around the track, and the system should be operational by March 15, 1978.

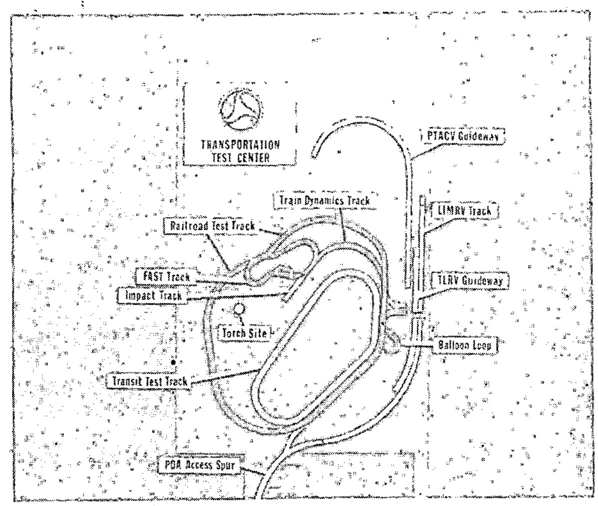


FIGURE 8. TEST CENTER TRACKS AND GUIDEWAYS

FUTURE FACILITY CONSTRUCTION

The Test Center will have the facilities to test high speed electric locomotives and high-speed self-powered electric passenger railcars by early 1979. Construction is due to begin next spring on a Northeast Corridor type of high voltage catenary system on the 14-mile Railroad Test Track loop. In all, there will be about 18½ miles of catenary.

The high speed catenary on the Railroad Test Track will be a constant tension type designed for 150 miles an hour. It will provide a choice of 12,500 volts, 25,000 volts or 50,000 volts at a continuous power rating of 13 megawatts from a single substation. The catenary for the FAST Track and the Train Dynamics Track will be a 70 mile-an-hour design, while the reversing loop for the Railroad Test Track will have a low-speed 15 mile-an-hour catenary.

Another major construction project, which is still in the preliminary planning stage, is the construction of a second FAST track. Experience with the original FAST track has shown that higher speeds, longer tangents and longer test sections are needed. More detailed requirements are now being gathered and the funds for developing the design of the track are anticipated in 1978. It is intended that this track will be used in addition to the present FAST track.

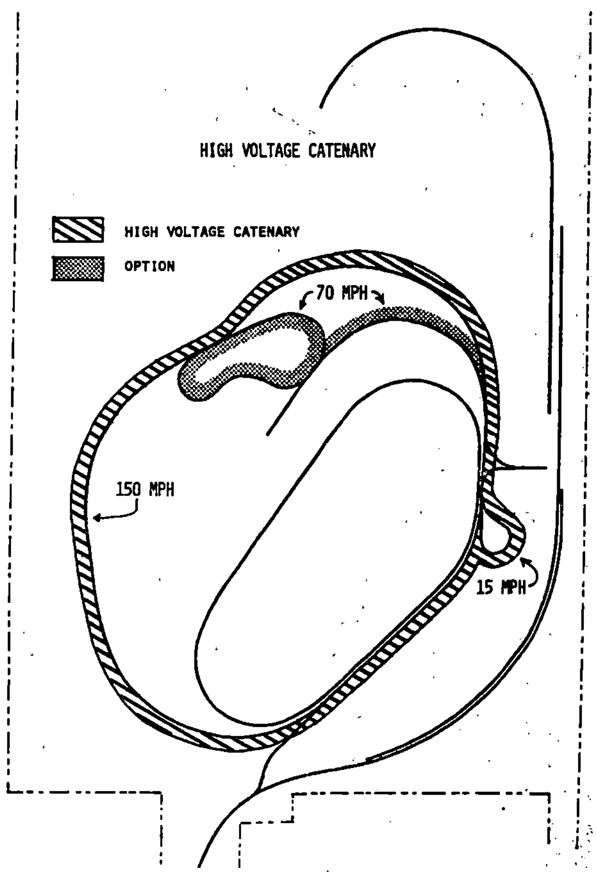


FIGURE 9. HIGH VOLTAGE, HIGH SPEED CATENARY LOCATIONS

A contract was recently let for a new switchlock control system that will allow all of the Test Center's main line switches to be remotely locked from the Operations Control Center after they have been manually thrown. This way, there wll be a positive indication of the position of every critical switch in the Control Center -- a safety feature that will become more and more important as rail traffic volume increases. The installation of this system will begin soon.

Construction is scheduled to begin next spring on a 40,000 square-foot warehouse that will accommodate metrology, calibration and instrumentation staffs as well as supply personnel. It should be completed before the end of 1978, and may have a solar heating installation to handle 80% of the heating load.

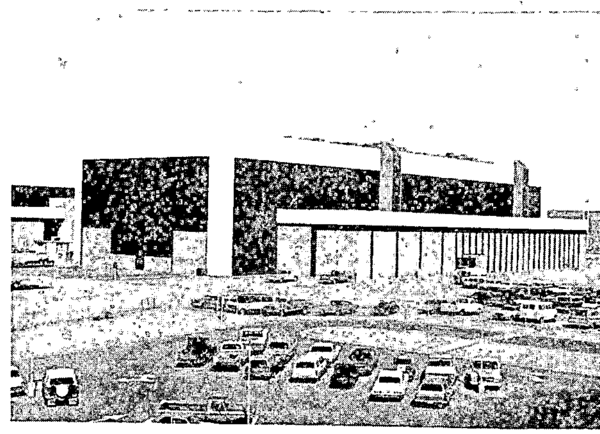
As the Test Center has grown in recent years, so have the demands for reduction and evaluation of engineering data. As a result, the two minicomputers, intelligent terminal and peripherals now at the Test Center are overloaded. A series of actions has been initiated to obtain a mid-sized computer which should greatly enhance the Test Center's capability to reduce and evaluate data in accordance with users' needs. It is expected that this computer will be operational by early 1979.

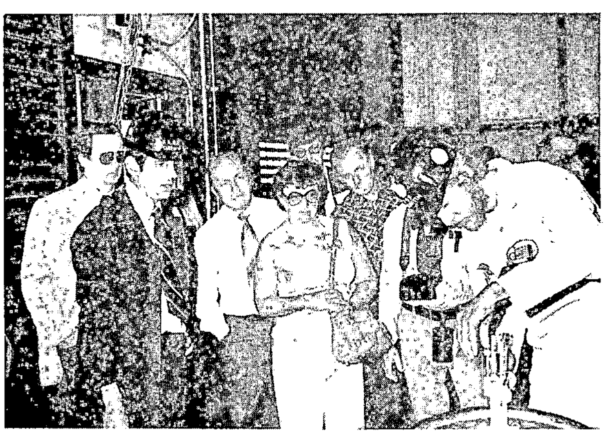
The resources of the Test Center Instrumentation Group have been modestly increased in the past year and will continue during 1978. The TTC Instrumentation Car is nearing completion and will be outfitted with a digital data acquisition system in the very near future.

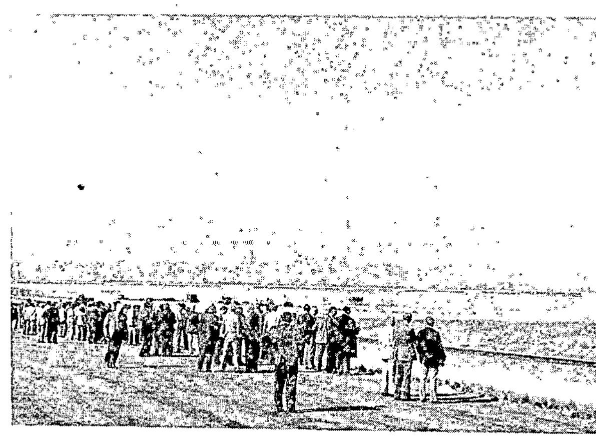
SERVICE TO THE RAIL INDUSTRY

I believe that the broad scope of the test programs planned for the Test Center together with the planned facilities and equipment acquisitions clearly indicate the intent of the Federal Railroad Administration to make significant contributions to the state-of-the-art in rail systems. A wide variety of rail system test facilities is -- and will continue to be -- made available for use by the various elements of the rail industry. Inquiries regarding the use of the Test Center's facilities are cordially invited.

TRANSPORTATION TEST CENTER TOUR







EVENING SESSIONS

REMARKS TO THE FRA ENGINEERING CONFERENCE

By

Robert E. Gallamore
Deputy Administrator

It is a pleasure for me to be here with you tonight. This conference gives me an opportunity to meet you and find out how you are approaching the problems that face the railroad industry. I hope I can be of help to you in your deliberations and that you will continue to share your ideas with me at convenient times in the future.

I would like to talk with you tonight about what I see are the major problems of the railroad industry, some general ideas about the nature of DOT's response to the problems, and a few more specific comments on R&D in that broader context. I think it is important for you, who daily work on engineering contributions, to know where FRA's general policies are headed.

To be sure, Secretary Adams, Administrator Sullivan and I have only been in office a brief time -- not long enough to formulate final policies. Right now, for example, we are in the very middle of a series of statutorily-mandated policy reports that will have great impact on Carter Administration Rail Policies. We need to complete our studies, formulate some tentative proposals, and consult more with carriers, suppliers, employees, other Government representatives, academicians and citizens. This time next year I would hope we could be back together working hard on implementation of our newly established or recently reaffirmed policies.

My own background is in economics and governmental policy formulation. So I would like to start with a characterization of the overall railroad problem and then see if there isn't an analogy to R&D specifically.

I believe that the best summary of the problem of America's railroads today is that they must make large scale, complex adjustments within a very rigid environment. Just to scan through those adjustments is sobering:

0 Railroads must adjust physical facilities to shifting markets; some product lines are improving, others declining; some regions are growing, others retrenching. Adjustments in location and type of physical facilities are needed.

0 Railroads must shift managerial and employee skills to meet changes in market demand and technological opportunities. Optimization of the operation of existing plant resources holds great profit potential.

0 Adjustments in pricing of railroad services is a natural means of achieving optimum output from existing facilities.

0 Railroads from time-to-time and, in some cases more than others, need to make changes in their financial structure or sources of outside funding.

These adjustments are made in an environment that is, at once, changing itself and rigid with respect to permitting needed change within. Railroads are where they are because their markets changed, their competition changed, and public policy changed. But their ability to respond is constrained. Regulation of entry and exit from markets and ICC rate regulation are only the most obvious rigidities in the railroad environment. The basic rail technology imposes a tremendous rigidity; indeed, my experience in

---

Robert E. Gallamore has served as the Deputy Administrator for the FRA since July 1977. Gallamore has been Associate Administrator for Transportation Planning for the Urban Mass Transportation Administration since 1976. Gallamore received an A.B. in Government (1963), his M.P.A. from Harvard University's Graduate School of Public Administration (1965) and his Ph.D in Policy Economy and Government (1968).

UMTA was that urban planners and developers liked rail transit's limiting technology (fixed guideways) -- at least as much because of this inflexibility as because of rail's economic and environmental performance. Other rigidities stem from labor contracts, traditional traffic patterns and partnerships, management styles, institutional financing relationships (particularly in the event of bankruptcies), and merger regulation. Of course, some of these constraints are necessary to serve other social objectives.

The poor financial condition of the industry is not itself "The Railroad Problem" but is the result of these other difficulties. If adjustments are impossible, the mighty market-place eats away at traffic--then revenues, then profits, and finally, capacity. We are all the losers -- the railroads, their customers, their employees, their neighbors.

I'm not yet prepared to offer solutions, but I feel very strongly that there is no single panacea. We cannot hope to cure the problem with infusions of Federal tax dollars alone or just with regulatory changes, adjustments in labor-management relationships, technological breakthroughs, or new public policies. We shall have to press forward on most or all of these fronts. We must probe the full range of possible solution-types and innovatively combine our best ideas into an aggressively implemented policy. That's not easy, and I can't promise success. Others have tried hard before. But we have a talented and energetic team in Washington today and we will give it our best shot. I ask for your counsel and support.

Let me turn now to some comments on R&D. I'm not an expert on technology, but I am keenly aware of the relationship between an industry's technology and its production function; that is, its output and costs. I've also participated in studies of rail productivity which addressed needed technological improvements. And I've been a student of the progress of technology diffusion.

It seems necessary to come to grips with two fundamentals. First, what are the proper roles and relationships of industry and Government with respect to R&D sponsorship. Second, what are today's priorities for R&D projects. As with general policy, my ideas are still in the formulative stage, but I'd like to make a few comments.

With respect to roles, I think the key word is cooperation. We want to aid the industry in its operations. Some of us have wondered how we can get more Federal assistance to the industry without interfering with its management and internal incentives; R&D is one way. While our perspective can and should be different from the industry's, we need to give first thought to what it is the industry thinks needs to be done, because it must complete the technological diffusion process; it must make the innovation work. In simple fact, Federal research and development funds will go farther if our efforts are cooperative with the industry. The FAST project here at the Pueblo Test Center is a good example of the kind of cooperation we need. I congratulate all of you here who have helped and are helping to make this effort a model success story.

Then there is the question of basic and applied research. Here I am somewhat schizoid. I believe Government is often best at basic research, leaving development and implementation design to practitioners. And there are basic, important, unanswered questions of train roll and stop dynamics, track and wheel dynamics, rail metallurgy, freight car structure, signaling technology and data retrieval. At the same time, however, I want our research to be directed at relatively near-term, operational goals. We perhaps have given too much attention to advanced technology, especially for passenger systems, and I think it is right that we have turned our focus to projects like FAST and the Rail Dynamics Lab.

I said earlier that we won't find a single solution to the rail industry's problem of adjustment. Perhaps the R&D analogy is not too far-fetched. It won't work for Government alone to fund a research project or push a particular innovation. It won't work for the industry's operating people just to demand new technologies--they need engineering support. And rail management needs to know both the operating problem and the range of possible technical solutions before a concerted campaign can be launched. So again we need an across-the-board approach; we need to move on all fronts. "Supply-Push" or "Demand-Pull" innovation alone isn't enough-just like "Cost-Push" and "Demand-Pull" theories of inflation by themselves have proven wanting. Something more like a "Demand-Shift" theory is needed--a total shift of all R&D resources: Government, industry-technical and industry-managerial--plus the cooperation of rail labor.

On the subject of R&D priorities, you have to inform me. I know Bob Parsons has already established consultative processes--notably in cooperation with the AAR through the Transportation Research Board--to assure the industry's participation in our R&D policy formulation. But this is a never-ending task, for needs and opportunities change.

More for your entertainment than your edification, let me report my recent paltry research on rail industry innovations. I was at the Transportation Research Forum's annual meeting in Altanta yesterday, having lunch at a table with four long-time colleagues, each of whom is distinguished as a rail analyst. The subject of discussion was my talk here with you tonight, and I decided to take a poll. I made up one list of post-war innovations and a second of technological developments still needed. My friends then rank-ordered these innovations by past or potential effectiveness. My conclusions are the following:

1. The only clear consensus was the dieselization was the most important past innovation.

2. Microwave communications, mechanized maintenance-of-way, computerized car location and per diem systems, and automatic yards were all grouped far below dieselization, but in no significant order.

3. There was even less consensus on ranking or needed innovations. Improved brake systems might have had a slight lead, and automatic couplers scored well. Better ACI systems received two first-place votes, but one respondent put it at the bottom of the list. Better steel rail and new signal technologies also were in the sample.

4. Remember your training in small sample errors.

5. Never try to conduct serious business at TRF annual meeting lunches. I won't tell you some of the other suggestions I received.

Well, of course you can't conclude much from this exercise, but it was interesting and it is the type of discussion we need to have with experts like you here tonight.

Finally, I'd like to turn to the FRA Research Program. Under the able leadership of Bob Parsons, and Ed Mathews here at the Pueblo Test Center, a strong research and test program has been developed. You have heard today that the program is divided into freight systems, passenger systems and safety. Since this last area probably is least familiar to you and since we are particularly proud of it, I would like to review that area briefly.

Railroad safety is one of FRA's major responsibilities. Our technological improvements in rail safety are aimed at preventing accidents from happening, and reducing their severity. One good example of our efforts is the testing of various glazing materials for use in the windows of locomotives and cabooses. Everyone is in agreement on the need for action in this area, the unions, management and FRA. We know we can reduce the toll of injuries and deaths caused by mindless vandals who use our trains as moving targets in their ugly sport. Soon, we will have the full documentation, prepared in conjunction with the unions and the railroads, upon which the industry can base window replacements.

Another example was the joint Government-industry tank car research effort. The safety problem was the incidence and severity of accidents involving tank cars transporting hazardous materials. Statistics showed that from 1969 through 1975, 519 tank cars were involved in derailments. These accidents caused 18 deaths, 832 injuries, and 45 major evacuations involving over 40,000 persons. Four of these accidents resulted in a combined total loss estimated at more than $100 million.

The research on solutions covered a range of head shields, thermal protection and shelf coupler devices. From this research, a new safety regulation was born, with industry support. New standards were created for puncture resistance from impacts and thermal protection from fire exposure for new, as well as existing cars. This program should result in considerable safety benefits at acceptable levels of additional costs.

We are also developing criteria to support other parts of DOT's regulatory and policy-making role and to aid FRA's safety field work. It is under this program that we developed track geometry measuring cards to perform inspections and help carriers plan more effective maintenance programs.

Let me conclude by again soliciting your views on where we should be headed. The Railroad Industry needs your ideas, your creativity, your skill, your dedication. One of you out there may have an idea for the advanced coupling system, or the ultimate automatic car identification system. Or your contribution may be more immediate--such as finding the available off-the-shelf window glazing material that only had to be tested to be proven effective. Obviously, we need to keep up a stream of technological successes while we work together to resolve financial and operating issues besetting this once grand, still productive and ever promising industry. Thank you for giving me the opportunity to share my thoughts with you.

Dr. John J. Fearnsides
Acting Chief Scientist
Department of Transportation

As Bob Parsons mentioned in his introduction, I have a lot of jobs at present. Only two jobs are important for tonight's discussion -- the Chief Scientist and the Executive Assistant to the Deputy Secretary. The Chief Scientist job enables me to talk with and advise the Secretary on many technical matters such as the Alaskan pipelines, both oil and natural gas/and on SST's. I don't know how many of you know but Bob Parsons used to be the Deputy Director of the Department's SST Program before Congress decided to close its development. I have also been involved in the recent secretarial decisions regarding the air bag. So you can see the Chief Scientist is involved in current DOT issues that relate to technology and its impact on the public and, therefore, is very interested in the content and relevance of the Department's R&D programs.

Let me take a minute to explain Secretary Adams' approach to the DOT organization to demonstrate that there is no de-emphasis of technical matters in the Department. He felt, and I agree to a large extent, that having an R&D staff organization that also managed R&D programs, tended to lead to something that was more of a line organization than a staff organization. I think there have been many of us, including Bob Parsons, who have agreed with that observation. The Secretary has established a new research and special programs directorate which will take the advanced R&D programs, the intermodal and multi-modal programs out of the Office of the Secretary -- a staff organization -- and put them in a line organization. There is absolutely no R&D de-emphasis from this at all. In fact, by putting them into a line organization where everybody knows what it is they are doing should strengthen these R&D activities. At present I am acting as director of this new research program management activity.

The Secretary also plans to transfer the R&D staff people to those other elements of the Office of the Secretary that have more influence on Secretarial decisions. For example, the program evaluation R&D personnel will work directly with the budget and program review offices and will be available to advise the budget director.

I fully expect that this will mean more clout for the R&D staff people in the long run.

From this perspective as the Department's Chief Scientist, I naturally believe strongly that the <u>right</u> R&D can make a significant difference in the operational efficiencies of all transportation modes. On the other hand, however, it is DOT policy that the private sector -- both suppliers and operators -- take initiative and provide those investments, including R&D, that will promote efficiencies in their respective modes. History clearly shows that the private sectors profit motives and ingenuity will result in improved products and transportation efficiencies, provided, of course, that there is a relatively free market place. Advances in commercial aviation and the automobile industry were so dramatic, we almost take them for granted.

We all know, however, that there is no "free market" in transportation. That is, for varying reasons the Government is involved in every transportation mode -- some modes more than others and not all are equitably treated. DOT has and will continue to propose legislation aimed at overcoming these modal imbalances. My point is that from a DOT R&D perspective, it is very difficult to treat Federal R&D support to the various modes on an

---

John Fearnsides received his B.S.E.E. and M.S.E.E. degrees from Drexel University Philadelphia, Penn. in 1962 and 1964 respectively and his Ph.D degree from the University of Maryland (1971). Since November 1972, he has been with the Office of the Secretary, U.S. Department of Transportation. He is currently a Science Advisor to the Secretary and Executive Assistant to the Deputy Secretary.

apples-to-apples basis, since our involvement in each mode differs. Being in the Department for five years, I have come across many programs and have been able to develop an overall broad departmental perspective. One must look at the modes separately and together -- the modes being each of the operating administrations of DOT, the Coast Guard, UMTA, Federal Highway Administration, etc. For someone who has come out of the university sector and the aerospace business, my basic observations are that the overwhelming amount of government R&D is sponsored by the Defense Department and NASA. This is still true. I am used to governmental organizations that use the products of their R&D. In these cases, management understands what is needed, they tell their operations people what to do, and instruct the R&D people in what has to be produced and all work together - that is some of the time. Our Coast Guard and Federal Aviation Administration use this system.

Another thing that exists in DOD and in NASA that does not exist in the Department of Transportation is one of a confidence in technology - a confidence that leads to a part of the investment portfolio (R&D programs) going toward long range potentials. My past observation in the Department of Transportation is that some elements of DOT seem to be out to save the world with technology and have a tendency to rush through what used to be considered in normal government R&D as the traditional phases of R&D. These are the two issues that I'd like to talk about tonight. First, what is the notion of the government as a provider of technological leadership? These are my words and I will be happy to explain a little bit further if they need explaining. The second issue is the question of the R&D delivery system.

Let me take the second of these issues first because the delivery system question is the most interesting one to those of us who have had to deal with DOD and NASA -- organizations where they use the products of their own R&D. This is in contrast to DOT where the really challenging problem particularly with UMTA and FRA is the fact that the end R&D users are not part of FRA and not part of UMTA but they are part of the private sector. That presents, as you folks know, some very interesting problems. Especially if you tie it together with the notion that DOT should provide technological leadership. For this process to work, however, it is clear there needs to be some kind of cooperative program. It is somewhat difficult to get funding for cooperative programs within the government budget process. We in the Office of the Secretary are involved as well as is OMB and the Congress. At each level someone will say "well, if you are going to go out and design a brand new rail transit car, how do you know the people are going to use it, how do you know the properties are going to use it, how do you know suppliers are willing to supply it?" And the answer is simple we don't really know. The same thing is true in the rail industry where we are not users -- the suppliers and private railroads are R&D users for the most part. The important thing is that there has to be a cooperative approach, and cooperative approach means to me that the industry does not tell the government what we should do and the government doesn't tell the industry what you should do. While there is much cooperative rail R&D, my observation is that there is still an imperfect relationship. There are a lot of common rhetorical goals and we must strive to make them <u>truly</u> common goals.

I believe R&D offers many opportunities to improve service, reliability, and safety. You must admit in all these areas you have problems and should be using any tool to improve. We in DOT are willing to help but to reach our joint goals, that is, a better transportation system for the American people and shippers, we must work together. Here at Pueblo -- at our Transportation Test Center -- we are providing large federal investments to do our share. But you must carry an increasing share of the experimental costs -- particularly as your financial lot improves. My concern is that DOT as a provider of federal funds and looking out for the average citizen must assure a balanced investment portfolio including long range R&D.

Here in Pueblo we could have a real laboratory situation. Laboratories are a common part of most R&D programs. For example, if we didn't have laboratories in NASA, we would have been in deep trouble if the first Apollo booster didn't work the way we had hoped it would work.

There is a need for advanced planning and maybe substituting more prototype testing or prototype evaluation before product deployment will save you money in the long run.

In my second role as Executive Assistant to the Deputy Secretary, I provide the Deputy Secretary advice on such matters as budgets and I think it is important for you to know the basis upon which some of this advice is given. I am looking for how the industry and FRA are cooperating in providing both a good delivery system and an R&D program that is balanced between long range and short range. I want to be sure that FRA is not just putting things together that nobody will ever use. In DOT we must and do worry about short term programs. We have a big capital grants program that exceeds the size of the R&D Program many fold. But, it is my feeling that DOT should have a balanced R&D program. If we ask a stockbroker to manage our money, we expect it to be a balanced portfolio -- not all AT&T. If DOT has about four hundred million dollars for R&D, we surely should be willing to put a little of that money into long range R&D.

In any event we are excited about the current FRA test and evaluation projects at Pueblo. As we move forward together to additional FAST-type projects and large RDL experiments, we at the OST level in DOT are awaiting large participation by the private sector. For only in that way will Bob Parsons be able to muster the Administration support for this government's share of these larger R&D activities.

I'll close with this challenge and I'm confident you'll respond favorably as you did in instituting the first FAST experiment.

Thank you.

Abernathy, M.B.
Manager, Surface Trans. Sales
Continental Oil Company
5 Greenway Plaza
Houston, TX 77046

Acord, F. D.
C.M.O.
Union Pacific
1416 Dodge
Omaha, NB 68179

Albanese, D. J.
Vice President, Tech. & Internat'l
Midland-Ross Corp.
2570 Woodhill Rd.
Cleveland, OH 44104

Alber, Rudy H.
Sales Manager Brake Shoes
Griffin Wheel Co.
200 W. Monroe, Floor 2300
Chicago, IL 60606

Alexander, T. R.
Asst. Director-Rail Services
Railway Transport Committee
275 Slater Street
Ottawa, Ontario, Canada

Anderson, Charles Jr.
Research Physicist
U. S. Army Ballistic Research Lab
Fragmentation Branch/TBD-Bldg. 393
Aberdeen Proving Ground, MD 21005

Arnott, T. H.
Manager, Equipment Engineering
E. I. DuPont de Numours & Co.
Transportation & Distribution
Wilmington, DE 19707

Ault, Kenneth
Sales Manager
Sperry Rail Service
Shelter Rock Road
Danbury, CT 06810

Autrey, W. S.
Chief Engineer
Santa Fe Railroad
80 East Jackson
Chicago, IL 60604

Babich, Thomas L.
Manager Eng. & Q.C.
Hadady Corp.
17506 Chicago Avenue
Lansing, IL 60438

Baicy, Edward
Chief Fragmentation Branch
U.S. Army Ballistic Research Lab
Aberdeen Proving Ground, MD 21005

Bailey, E. C.
Director of Eng. and Quality Assurance
Dresser Industries
Transportation Equipment Division
Depew, NY 14043

Bakken, Gordon
Manager of Program Development
Wyle Laboratories
4620 Edison Avenue
Colorado Springs, CO 80908

Bang, Arne
Chief, Freight Service Division
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Barriger, Stanley
Consultant
Box 1102
Manchester, NH 03105

Bartley, Robert D.
Mgr. Market Division
Railway Age
29 East Madison
Chicago, IL 60602

Bartolini, Dean
District Sales Manager
Alco Spring Industries, Inc.
23rd & Euclid Avenue
Chicago Heights, IL 60411

Beckwith, Walter C., III
Washington Representative
Systems Control, Inc.
1911 N. Ft. Myer Dr., #402
Arlington, VA 22209

Beetle, Robert H.
Vice President
Abex, R. P. G.
Mahwah, NJ 07430

Begier, Ronald J.
Sr. Research Engineer
AAR
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Bender, Erich K.
Manager, Applied Technologies
Bolt Beranek & Newman Inc.
50 Moulton Street
Cambridge, MA 02138

Berg, Norman A.
Central Regional Sales Manager
Griffin Wheel Company
200 West Monroe
Chicago, IL 60606

Billingsley, R. H., Jr.
Senior Director, Technical
ACF, Inc., Amcar Division
Main & Clark Streets
St. Charles, MO 63301

Bird, H. R.
Director of Railroad Safety
FRA
Rm. 1807, 911 Walnut Street
Kansas City, MO 64106

Blanchfield, James R., Jr.
Research Mgr./Intermodal Systems
FRA/Office of Freight Systems
2100 Second Street, S.W.
Washington, D.C. 20590

Boerke, R. A.
Vice President-Engineering
North American Car Corp.
222 S. Riverside Plaza
Chicago, IL 60606

Bray, Don E.
Assistant Professor
Department of Mechanical Engineering
Memphis State University
Memphis, TN 38152

Brink, Henry
Acting Chief, Test Control
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Brohaugh, R. G.
Chief Engineering Materials
Burlington Northern Railroad
176 E. 5th Street
St. Paul, MN 55101

Brown, James
Assistant Chief Engineer
Bethlehem Steel Corporation
Johnstown, PA 15907

Bunker, Russ E.
Regional Track Engineer
FRA
Rm. 1807, 911 Walnut
Kansas City, MO 64106

Cappel, Klaus
Chief Design Engineer
Wyle Laboratories
7800 Governors Drive West
Huntsville, AL 35758

Cartwright, John B.
Regional Track Engineer
FRA
900 S.W. Fifth Ave., #450
Portland, OR 97204

Caruth, Richard
Vice President
The Maxson Corporation
P. O. Box 43585
St. Paul, MN 55164

Ceccon, Harry
Engineer
TSC
Kendall Square
Cambridge, MA 02142

Chang, Edward H.
Research Engineer
AAR
3140 South Federal Street
Chicago, IL 60616

Chellis, Willard B.
General Plant Manager
Chessie System-Raceland Shop
Russell, KY 41169

Chidley, W. H.
Chm.-Tech. Committee of R. R. Materials
American Iron & Steel Institute
230 North Michigan Ave., #2118
Chicago, IL 60601

Child, Michael
Program Analyst
FHWA
Nassif Bldg., 7th & D., S.W.
Washington, D.C. 20590

Christoph, A. E.
Sales Engineer
FORTEC, Inc.-RMC Division
P. O. Box 1888
Pittsburgh, PA 15230

Cooperrider, Neil K.
Professor-Mechanical Eng. Dept.
Arizona State University
Tempe, AZ 85282

Cope, Geoffrey W.
Manager of Research & Development
Dresser Ind., T. E. Division
Two Main Street
Depew, NY 14043

Courlos, John G.
Engineering Program Manager
DOT
400 Seventh Street, S.W.
Washington, D.C. 20590

Cruse, W. J.
Consultant-Track Engineering
149 Sunnyside Avenue
Elmhurst, IL 60126

Cunningham, F. E.
Assistant Vice President - Car
Chicago & North Western Trans. Co.
500 West Madison
Chicago, IL 60606

Curati, Marino, Jr.
Chief Engineer-Rail Appl. Division
True Temper-Central Engineering
185 Water Street
Geneva, OH 44041

Dailey, Eugene E.
Specialty Sales Administrator
Koppers Company
900 Koppers Building
Pittsburgh, PA 15219

Danahy, F. A.
Executive Director
AAR
1920 L Street, N.W.
Washington, D.C. 20036

Davidson, Dale R.
Group Manager
Civilian Agencies Mkt.
P. O. Box 3525
St. Paul, MN 55165

Davis, Louis
President
American Steel Foundries
1005 Prudential Plaza
Chicago, IL 60601

Day, John C.
Vice President Marketing
American Steel Foundries
1005 Prudential Plaza
Chicago, IL 60601

De Claire, Alton G.
Development Engineer
General Motors
Transportation Systems Division
Technical Center
Warren, MI 48090

Dean, Francis E.
Principal Research Scientist
Battelle Columbus Laboratories
505 King Avenue
Columbus, OH 43201

Demmert, W. R.
Vice President Marketing
Griffin Wheel Co.
200 W. Monroe
Chicago, IL 60606

Dick, Harvey F.
V.P. & General Manager
Pullman Standard
Suite 206-4045 Bonita Road
Bonita, CA 92002

Dickhart, William III
Assistant to Technical Director
The Budd Company Tech. Division
300 Commerce Drive
Ft. Washington, PA 19034

Dickinson, Lemaine V., Jr.
Professional Staff-Trans. Group
Office of Technology Assessment
U. S. Congress
Washington, D.C. 20510

Dorcy, William
General Manager
The Alaska Railroad
Pouch 7-2111
Anchorage, AK 99510

Dorland, Wade D.
Manager, Rail Dynamics Lab
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Ekberg, Eldon
Acting Chief, Facility Mgt.
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

English, E. R.
Track Engineer
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Ephraim Max, Jr.
Chief Engineer
General Motors Corporation
c/o Electro-Motive Division
La Grange, IL 60525

Erdmann, Robert H.
Manager Transportation Sales
American Koyo Corporation
P. O. Box 45028
Westlake, OH 44145

Evans, Robert
Project Director
AAR
3140 South Federal Street
Chicago, IL 60616

Evans, Thomas E.
Regional Track Engineer
FRA
819 Taylor Street, Rm. 11A23
Fort Worth, TX 76102

Fay, Grace
Research Manager
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Fearnsides, John
Atg. Chief Scientist
DOT
400 Seventh Street, S.W.
Washington, D.C. 20590

Feigenbaum, Edward L.
Program Manager
The Aerospace Corporation
955 L'Enfant Plaza, SW
Washington, D.C. 20024

Ferguson, John D.
Program Manager, Office of Rail Safety Res.
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Ferguson, Max A.
Regional Track Engineer
FRA
1568 Willingham Drive
College Park, GA 30337

Ferrara, James M.
Program Analyst, RRD-30
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Fletcher, William H.
General Engineer
National Transportation Safety Board
800 Independence Ave., S.W.
Washington, D.C. 20594

Flohr, Bruce M.
President
Rail Tex, Inc.
300 Elizabeth Road
San Antonio, TX 78209

Fogarty, Thomas
Project Manager
ENVO, Inc.
800 Follin Lane
Vienna, VA 22180

Frank, E. E.
Chief Engineer
Abex Corp., R.R. Product Group
65 Valley Road
Mahwah, NJ 07430

Frasier, Cline
Deputy Director
Office of Ground Systems
TSC
Kendall Square
Cambridge, MA 02142

Gaeth, Dale L.
Manager Mounted Wheel Sales
Griffin Wheel Co.
200 West Monroe, #2300
Chicago, IL 60606

Gallamore, Robert E.
Deputy Administrator
FRA
400 Seventh Street, S.W., Rm. 5424
Washington, D.C. 20590

Gannett, M. Clifford
Chief, Passenger Equipment Division
FRA
2200 2nd Street, S.W.
Washington, D.C. 20590

Garg, Vijay K.
Manager Dynamics Research
AAR
3140 South Federal Street
Chicago, IL 60616

Gierlach, Robert W.
Sales Manager
Standard Car Truck Co.
332 South Michigan Avenue
Chicago, IL 60515

Gieskieng, Fred
Product Manager
Wheel Checkers-Marine Electric RPD
2020 South Bannock Street
Denver, CO 80223

Gieskieng, Mike
President
Wheel Checkers
2020 South Bannock Street
Denver, CO 80223

Gile, Earle
State Leg. Director
United Transportation Union
4800 Wadsworth Plaza, Rm. 207
Wheatridge, CO 80033

Gill, Ross
Assistant Manager of Research
Southern Pacific Transportation Co.
One Market Plaza
San Francisco, CA 94105

Gillespie, H. Aldridge
Lab Mgr. & Materials
Transportation Safety Institute
6500 South MacArthur
Oklahoma City, OK 73125

Godfrey, C. E.
Mgr. Trackwork Product Sales
Abex Corp., RPG
Mahwah, NJ

Gray, Donald
Evaluation Program Manager
Office of Freight Systems
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Gray, Thomas
Track Superintendent
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Greenwood, W. F.
Director of Marketing
Dresser, Ind.
Two Main Street
Depew, NY

Grejda, F. J.
Mgr. Railroad Products & Market Res.
Westinghouse Air Brake Co.
Wilmerding, PA 15148

Gross, Arnold
RDL Program Manager
Office of Freight Systems
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Grunwald, Kalman J.
FRA Program Representative
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Guitaut, Francois P.
General Enginer (hon) French Railways
16-18 BLVD De La Republique
S.T.E.D.E.F.
92100 Boulogne, France

Guiterrez, Miguel Ruiz
Ferrocarriles Nacionales De Mexico
Av. Central & 140
Mexico 3 D. F.

Gutmann, Paul, Jr.
Vice President Engineering
Scullin Steel Co.
6691 Manchester
St. Louis, MO 63139

Hales, H. F.
Vice President
Florida East Coast Railway Co.
P. O. Box 1048
St. Augustine, FL 32084

Hansen, Knut
Field Engineer
Buffalo Brake Beam Co.
11779 Unity Drive
Bridgeton, MO 63044

Hanson, Michael
Engineer-Track
Soo Line Railroad
Box 530, Soo Line Bldg.
Minneapolis, MN 55440

Harbour, G. W.
Rail Car Representative
Gulf Oil Co. - W.S.
P. O. Box 1627
Longview, TX 75601

Harley, E. T.
General Mechanical Sup't.
CONRAIL
Room 233, 30th Street Station
Philadelphia, PA 19104

Harralson, Leonard
Mgr. Lease Fleet Operations
Evans Products, U.S. Railway
2200 East Devon Ave.
Des Plaines, IL 60018

Hartzell, Edward C.
Design Engineer
General Electric Co.
2901 E. Lake Road
Erie, PA 16501

Hawthorne, V. T.
Vice President - Engineering
Railroad Dynamics, Inc.
10 East Athens Ave.
Ardmore, PA 19087

Hengel, Mark F.
Mechanical Engineer
Missouri Pacific Railroad
210 North 13th Street
St. Louis, MO 63103

Holabeck, Gus
Chief Engineer
Bethlehem Steel Corporation
Johnstown, PA 15907

Houser, Fred N.
Manager
Railroad Research Information Service
2101 Constitution Ave., N.W.
Washington, D.C. 20418

Immelt, Fred
General Mechanical Sup't.
CONRAIL
6 Penn Center Plaza
Philadelphia, PA 19104

Inabinett, Daniel W.
State Rail Safety Program Coordinator
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Ingrao, Hector C.
Transportation Systems Center
Kendall Square
Cambridge, MA 02142

Jackson, Keith
V.P. & Gen. Mgr./GSI Eng. Div.
GSI Engineering
P. O. Box 2396
St. Louis, MO 63114

Jacobs, Marilynne
Research Engineer-Mech. Sys.
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Jamison, Warren E.
Associate Professor
Colorado School of Mines
Basic Engineering Department
Golden, CO 80401

Jeffcoat, Robert
Technical Staff
The Analytical Sciences Corp.
6 Jacob Way
Reading, MA 01867

Jeffers, William A.
Director of Railroad Safety
FRA
900 5th Street, S.W., Rm. 450
Portland, OR 97204

Jeter, Thomas
Chief, Physical Test
Ballistic Research Laboratory
Bldg. 393
Aberdeen Proving Ground, MD 21014

Johnson, E. Q.
Senior Asst. Chief Engineer
Chessie System
P. O. Box 1800
Huntington, WV 25718

Johnson, R. E.
Director, Mechanical Engineer
Burlington Northern Inc.
176 East 5th Street
St. Paul, MN 55101

Jones, R. D.
Director of Sales & Development
Hawker Siddeley Canada, Ltd.
1900 Dickson Street
Montreal, Quebec, Canada H1N 2H9

Jones, Tom
Program Manager
ENSCO
2560 Huntington Ave., Suite 204
Alexandria, VA 22303

Kalina, Harry
Manager, Design Engineering
FreightMaster, Div. Halliburton
P. O. Box 40555
Fort Worth, TX 76140

Karlovic, Martin J.
District Engineer
Southern Pacific Transportation Co.
Rm. 1007, SP Building
One Market Plaza
San Francisco, CA

Kassab, G. M.
Transportation Officer
Pittsburgh, Naval Reactors
P. O. Box 109
West Mifflin, PA 15122

Kaufman, William M.
Vice President Engineering
ENSCO, Inc.
5408 A. Port Royal Road
Springfield, VA 22151

Kavet, R. A.
Project Engineer
Sandia Labs, Div. 1713
Kirtland Airforce Base East
Albequerque, NM 87115

Keeler, Harold L.
Regional Track Engineer
FRA
2 Embarcadero Center #630
San Francisco, CA 94111

Keller, Paul F.
Manager, Business Development
TASC
1730 N. Lynn Street
Arlington, VA 22304

Keller, Thomas
Chief Engineer/R.R. Division
The Timken Company
1835 Dueber Avenue, S.W.
Canton, OH 44706

Kelly, James E.
Supervisor Unit Trains
Detroit Edison Co.
2000 Second Avenue
Detroit, MI 48226

Kenworthy, M. A.
Engineer
ENSCO
2560 Huntington Avenue
Alexandria, VA 22303

Kenyon, Michael D.
Assistant Chief Engineer
Denver & Rio Grande Western R.R.
P. O. Box 5482
Denver, CO 80217

Kern, D. W.
Eng. Mgr.-Couplers & Draft Gears
Midland Ross Corp.
2570 Woodhill Road
Cleveland, OH 44104

King, F. E.
Senior Technical Advisor
Canadian National Rail Research
935 De La Gauchetiere Street. W
Montreal, Quebec, Canada H3C3N4

Klinger, David L.
Senior Engineer
Systems Control, Inc.
1801 Page Mill Road
Palo Alto, CA 94304

Kostolansky, David J.
Mgr.-Product Development
Schaefer Equipment Co.
Phoenix Road
Warren, OH 44483

Krauter, Allan
Senior Mechanical Engineer
Shaker Research Corporation
Northway 10 Executive Park
Ballston Lake, NY 12019

LaCorte, Fred
Sales Manager
Alco Spring Industries
23rd & Euclid
Chicago Heights, IL 60411

Lanman, R. W.
President
17530 Chicago Avenue
Hadady Corp.
Lansing, IL 60438

Law, E. Harry
Associate Professor of Mech. &
Mechanical Engineering
Clemson University
Department of Mechanical Engineering
Clemson, SC 29631

Layton, James
President
JL Industries, Inc.
2200 Carlson Drive
Northbrook, IL 60062

Lemon, Lucien
Manager R&D
Pacific Car & Foundry Co.
1400 Fourth Street
Renton, WA

Levergood, John V.
Director Peaking Unit & Unit Train Div.
Detroit Edison Co.
2000 2nd Ave.
Detroit, MI 48226

Lewing, Lawrence
Asst. Professor of Mechanical Eng.
Department of Mechanical Engineering
Colorado State University
Fort Collins, CO 80523

Lieb, Gerry
Manufacturing Manager
True Temper Corp./RA Division
102 West Railroad Street
Lake City, PA 16423

Liebig, Gil W.
Vice President-Marketing
Surface Trans. Internal't. Inc.
151 S. Milton Avenue
Glen Ellyn, IL 60137

Lin, Cheng-jer
Rail Systems Analyst
Dynalectron Corp.
Transportation Test Center
Pueblo, CO 81001

List, H. A.
Railway Engineering Assoc., Inc.
38 W. University Ave.
Bethlehem, PA 18015

Lockridge, Pres
Public Affairs Office
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Lohfeld, Robert E.
Director of Mkt. & Adv. Programs
OAO Corporation
5050 Powder Mill Road
Beltsville, MD 20705

Lombardi, Edward J.
Engineer of Tests
AMTRAK
400 N. Capital Street, N.W.
Washington, D.C. 20001

Long, L. Dale
Asst. Manager of Applied Research
Santa Fe Railway Co.
1001 N.E. Atchison
Topeka, KA 66616

Love, Robert
Asst. VP-Engineering
American Steel Foundries
1005 Prudential Plaza
Chicago, IL 60601

Lundgren, James R.
Manager-Track Research Division
AAR
3140 South Federal Street
Chicago, IL 60616

MacDonnell, R. W.
President
R. W. Mac Co.
525 Craige Avenue
Crete, IL 60417

MacNiece, G. Rodriguez
Chief Engineer
Nationals Ry of Mexico
Pv. Central #140
Mexico 3 DF, Mexico

Manos, William
Vice President of R&D
Pullman Standard
1414 Fields Street
Hammond, IN 46320

Marcotte, Pierre P.
Research Engineer
Canadian National Rail - Research
3950 Hickmore Avenue
St. Laurent, Quebec, Canada H4T1K2

Mathews, Edward R.
Director
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

May, Joe T.
Senior Engineer
EIT
1860 M. Faraday Drive
Reston, VA 22090

McCafferty, R. M.
Program Manager
FRA
2100 Second Street, S.W.
Washington, D.C. 20590

McDonough, J. F.
ACMO
Union Pacific
1416 Dodge, Rm. 903
Omaha, NB 68179

McGovern, William
Manager, Research Projects
AAR
3140 South Federal Street
Chicago, IL 60616

McGranahan, Eugene E.
Vice President-Operations
Midland Ross Corp.
National Castings Division
700 South Dock Street
Sharon, PA 16146

McHenry, F. H.
Chief, Mechanical Office
P&LE Terminal Building
Pittsbugh & Lake Erie Railroad
Pittsburgh, PA 15219

McIntosh, Gregory P.
FAST Project Manager
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

McLean, W. E.
Sales Mgr. Rail Passenger Equipment
MLW Industries
1505 Dickson Street
Montreal, Quebec, Canada

Meeker, William
Railroad Analyst
National Transportation Safety Board
800 Independence Avenue
Washington, D.C. 20594

Miller, Charles G.
Project Manager-NE Corridor Project
Bechtel, Inc.
P. O. Box 24038
Washington, D.C. 20024

Miller, J. Ted
Project Director
ARINC Research Corporation
2551 Riva Road
Annapolis, MY 21401

Mitchell, Myles B.
Director, Office of Passenger Systems
FRA
2100 2nd Street, S.W. Rm. 4110
Washington, D.C. 20590

Moehling, Charles
Manager-Truck Engineering
American Steel Foundries
1005 Prudential Plaza
Chicago, IL 60601

Molitoris, Michael
District Sales Manager
Dresser, Inc. T.E.D.
1522 Cotton Tree Drive
St. Louis, MO 63141

Money, Lloyd J.
Acting Assistant Secretary
DOT
400 Seventh Street, S.W.
Washington, D.C. 20590

Morella, Norman A.
Manager Eng. & Dev.
Midland Ross Corporation
2570 Woodhill Road
Cleveland, OH 44126

Morris, Howard J.
Vice President-Sales
Evans Products, Creco Division
P. O. Box 429
Woodstock, IL 60098

Mortensen, Donald G.
Senior Project Engineer
F.M.C.
1185 Coleman Avenue
Santa Clara, CA 95052

Mowatt-Larssen, Erling
Chief Design Engineer
General American Transportation Corp.
P. O. Box 53E
Sharon, PA 16146

Mowatt-Larssen, R.
Director
Office of Safety
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Moyar, Gerald J.
Mgr. Task IX TTD II
AAR Technical Center
3140 South Federal Street
Chicago, IL 60616

Murphy, Richard A.
General Engineer
Kendall Square
TSC
Cambridge, MA 02142

Murray, J. L.
Manager-Locomotive Truck Sales
Rockwell International
2135 W. Maple Road
Troy, Michigan 48084

Muzychemko, Walter
Mgr. Railway Div. Engrs.
American Koyo Corp.
29520 Clemens Road
Westlake, OH 44145

Nations, L. H.
Assistant General Manager Interm.
Southern Pacific Railroad
Number 1 Market Plaza, Rm. 735
San Francisco, CA

Netherton, T. C.
General Manager
Pittsburgh & Lake Erie Railroad
Smithfield & Carson Streets
Pittsburgh, PA 15219

Novotny, Richard A.
Chief, Passenger Systems
FRA
2100 Second Street, S.W.
Washington, D.C. 20590

Olsen, Kent A.
Senior Rehabilitation Planner
CONRAIL
Rm. 601, 6 Penn Center
Philadelphia, PA 19104

Olson, P. E.
Manager Transportation Systems Dept.
ASEA Inc.
4 New King Street
White Plains, NY 10604

Owings, Raymond P.
Chief Engineer
ENSCO, Inc.
5408 Port Royal Road
Springfield, VA

Palmer, D. E.
Chief Engineer, R&D Equipment
New York Air Brake Co.
Starbuck Avenue
Watertown, NY 13601

Parke, Harry G.
President
Marine Electric Corp.
600 Fourth Avenue
Brooklyn, NY

Parsons, Robert E.
Assoc, Administrator for R&D
FRA
2100 Second Street, S.W.
Washington, D.C. 20590

Patel, Ratee T.
Sr. Project Engineer
U.S. Army Meradcom
ATTN: DRXFB-HM
Fort Belvoir, VA 22060

Paxton, William R.
Chief, M/W Division
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Pazera, Eugene E.
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Peterson, Leavitt A.
Director, Office of Rail Safety Research
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Phillips, Earl A-
Vice President, Eng. & Dev.
Union Tank Car Co.
151st Street & Railroad Avenue
E. Chicago, Indiana 46312

Phipps, Phil
Manager Signal Processing
Sperry Univac MS UIT24
Univac Park, Box 3525
St. Paul, MN 55165

Pinnes, Robert W.
Division Chief (TST-42)
Office of the Secretary
DOT
Washington, D.C. 20590

Platt, L. George
Senior Development Engineer
Phillips Petroleum Co.
Room 8B4
Bartlesville, OK 74004

Powell, E. J.
General Manager - Sales
Portec, Inc., RMC Division
P. O. Box 1888
Pittsburgh, PA 15230

Price, B. H.
Senior Development Engineer
Bessemer & Lake Erie Railroad
P. O. Box 471
Greenville, PA 16125

Propp. Dale H.
Engineer of Tests
Burlington Northern
176 East 5th Street
St. Paul, MN 55101

Punwani, S. K. (John)
Senior Research Engineer
AAR
3140 South Federal Street
Chicago, IL 60616

Quindry, Thomas L.
Environmental Engineer (Noise)
Environmental Protection Agency
1921 Jefferson Davis Hwy.
Arlington, VA 20460

Radford, R. W.
Chief Mechanical & Elec. Engr.
Canadian National Railways
P. O. Box 8100
Montreal, Quebec, Canada

Radwill, Robert
Vice President-Engineering
American Steel Foundries
1005 Prudential Plaza
Chicago, IL 60601

Read, John W.
General Manager
B&LE Railroad
Box 471
Greenville, PA 16125

Reed, George
Director-Railroad Sales
ACF Industries, Inc. Amcar Division
Clark & Main Streets
St. Charles, MO 63301

Reiff, Glenn A.
Chief Technical Services
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Reynolds, D. J.
Asst. Manager Tests
Southern Railway Systems
409 S. Henry Street
Alexandria, VA 22316

Richter, Frank
Publisher
Progressive Railroading
20 N. W. Acker Drive
Chicago, IL 60606

Robichard, Richard H.
Design Engineer
TSC
Kendall Square
Cambridge, MA 02142

Robinson, John
Senior Engineer
Boeing Vertol Co.
P. O. Box 16858
Philadelphia, PA 19142

Rodriguez, Guillermo
Ferrocarriles Nacionales De Mexico
Av. Central and 140
Mexico 3 D.F.

Ronald, Christopher C.
Senior Engineering Staff
OAO Corporation
50/50 Powder Mill Road
Beltsville, MD 20705

Ruprecht, W. J.
Director Engineering
ACF Industries
620 North Second Street
St. Charles, MO 63301

Sage, Marshall
Legislative Research Director
United Transportation Union
400 First Street, N.W., Rm. 704
Washington, D.C. 20001

Sammoui, K. Y.
Manager-Railway Applications
MTS Systems Corporation
P. O. Box 24012
Minneapolis, MN 55424

Sanko, Thomas M.
Project Engineer
Rockwell International
2135 W. Maple Road
Troy, MI 48098

Sarkisian, Edward A.
Electronic Engineer
TSC
Kendall Square
Cambridge, MA 02142

Saroop, Rajendra
Project Manager
FRA
Office of Rail Safety Research
2100 Second Street, S.W.
Washington, D.C. 20590

Schiffers, Albert, Jr.
Executive Secretary
Railway Supply Ass.
332 South Michigan Avenue
Chicago, IL 60604

Schofield, Thomas
Vice President/Sales
A. Stucki Company
2600 Neville Road
Pittsburgh, PA 15225

Schwarm, Edward G.
Sr. Staff Member
Arthur D. Little, Inc.
25 Acorn Park
Cambridge, MA 02140

Scofield, Robert
Program Engineer
ENSCO, Inc.
20 South Quaker Lane
Alexandria, VA 22303

Sellers, William W.
Vice President Sales
Standard Car Truck Co.
332 South Michigan Avenue
Chicago, IL 60604

Semmler, Paul F.
Track Geometry O&M Manager
ENSCO, Inc.
20 South Quaker Lane
Alexandria, VA 22314

Settle, P. S.
Vice President
Portec, Inc.
1000 Ride Plaza, Room 204
Pittsburgh, PA 15238

Shannon, W. R.
Vice President-Equipment
Trailer Train Co.
300 South Wacker Drive
Chicago, IL

Sharpe, J. J.
Director of Safety-Chicago
FRA
536 South Clark Street
Chicago, IL 60606

Shedd, T. C.
Editor
Modern Railroads
5 South Wabash Avenue
Chicago, IL 60603

Sherfy, Michael A.
Program Manager
Office of Transportation Research
Iowa Department of Transportation
Ames, Iowa 50010

Shull, J. Robert
Program Manager
AiResearch Mfg. Co. of California
2525 West 190th Street
Torrance, CA 90509

Shute, Bruce W.
Supervisor
Air Brake Test Lab
New York Air Brake Co.
Starbuck Avenue
Watertown, NY 13601

Sierleja, Edward D.
President
Transportation & Distribution Ass.
600 North Jackson Street
Media, PA 19063

Simpson, W. W.
Vice President Engineering
Southern Railway
P. O. Box 1808
Washington, D.C. 20013

Sloat, Joseph F.
Manager National Accounts
Schaefer Equipment
Phoenix Road
Warren, OH 44483

Smit, Richard L.
Transportation Systems Engineer
General Motors Corp.
GM Technical Center
Warren, MI 48090

Smith, Harry F.
Account Manager-Transportation Sales
Continental Oil Co.
2500 Devon Avenue
Des Plaines, IL 60018

Smith, Carl R.
Project Engineer
General Motors Corp.
Electro-Motive Division
LaGrange, IL 60525

Smith, Richard E.
Program Director
The Aerospace Corporation
955 L'Enfant Plaza S.W.
Washington, D.C. 20024

Smythe, John B.
OST-FRA Monitor
DOT
7 N & D Streets, S.W.
Washington, D.C. 20590

Solomon, James E.
Engineer Group Manager-Trucks
Buckeye Steel Castings
2211 Parsons Avenue
Columbus, OH 43207

Souter, T. T.
AGM - Mechanical
Kansas City Southern Railway Co.
114 West 11th Street
Kansas City, MO 64105

Spanton, Donald L.
Director, Office of Freight Systems
FRA
2100 2nd Street, S.W.
Washington, D.C. 20590

Spohrer, Walter
DIPL-JNG
Deutsche Bundesbahn
Bundeshbahn-Zentralamt
8 Munchen, Arnvlfstra 19 FRG

Springer, J. S.
General Transportation Specialist
Transportation Test Center
P .O. Box 11008
Pueblo, CO 81001

Staten, Lorin C.
Test Controller
Transportation Test Center
Box 11008
Pueblo, CO 81001

Steber, W. C.
W. C. Steber Ass.
717 N. Overlook Drive
Alexandria, VA 22305

Steele, Roger K.
Metallurgist
TSC
Kendall Square
Cambridge, MA 02142

Stephenson, J. G.
Vice President-Engineering
FreightMaster Div. Halliburton
P. O. Box 40555
Fort Worth, TX 76140

Stone, Daniel H.
Director-Metallurgy
AAR
3140 South Federal Street
Chicago, IL 60616

Stravasnik, Luke F.
Project Engineer
Sandia Laboratories, Div. 1713
P. O. Box 5800
Albuquerque, NM 87115

Surkamp, Arthur T.
Senior Program Manager
Unified Industries, Inc.
205 South Whiting Street
Alexandria, VA 22304

Sussman, Nathan E.
Technical Staff
The Mitre Corporation
1820 Dolley Madison
McLean, VA 22101

Sutliff, David R.
Director, Track/Train Dynamics Program
AAR
3140 South Federal Street
Chicago, IL 60616

Taylor, Thomas P.
Manager of Sales
General Steel Ind., Inc.
P. O. Box 2396
St. Louis, MO 63114

Tennikait, Garth
Manager, Test Engineering
American Steel Foundries
1700 Walnut Street
Granite City, IL 62040

Terlacky, Boris
Manager Engineering
Trailer Train Co.
300 South Wacker Drive
Chicago, IL 60606

Tharp, Douglas B.
Operations Mgr. (FAST Project Mgr.)
Dynalectron Corp.
Transportation Test Center
Pueblo, CO 81001

Thomet, Michael
Transportation Planner
Bechtel Inc.
50 Beale Street
San Francisco, CA 94105

Thompson, J. Edmund, Jr.
Mechanical Engineer
Ballistic Research Lab
Fragmentation Br. Bldg. 393
Aberdeen Proving Ground, MD 21005

Trantham, W. E.
Vice President-Railway Products
Marine Electric Railway Products
GPO Box 11
Falls Church, VA 22046

Tsai, N.
Program Manager
FRA
Office of Freight Systems
2100 2nd Street, S.W.
Washington, D.C. 20590

Vadeboncoeur, James R.
President
Commonwealth Research Corp.
11800 Sunrise Valley Drive
Reston, VA 22091

Vogel, Hal Hunt
Member Technical Staff
Mitre Corporation
Westgate Research Park
McLean, VA 22101

Walker, G.
Sr. Project Engineer-FAST Loop
Dynalectron Corp.
Transportation Test Center
Pueblo, CO 81001

Walters, Bill
Instrumentation Engineer
Transportation Test Center
Pueblo, CO 81001

Wandell, W. H.
Associate Engineer
Dynalectron Corp.
Transportation Test Center
P. O. Box 11008
Pueblo, CO 81001

Wandrisco, J. M.
Chief Research Engineer
U.S. Steel Research Lab
Jamison Lane
Monroeville, PA 15146

Waring, E. H.
Chief Engineer
Denver, & Rio Grande Western Railroad
P. O. Box 5482
Denver, CO 80217

Way, G. H.
Assistant Vice President
AAR
1920 L Street, N.W.
Washington, D.C. 20036

Webb, C.E.
Assistant V.P. Engineer & Research
Southern Railway
Box 1808
Washington, D.C.

Weber, Hans B.
Engineering Manager-Trucks
National Castings Division
Midland Ross Corp.
2570 Woodhill Road
Cleveland, OH 44104

Weseloh, Roger J.
Senior Design Engineer
Cardwell Westinghouse Co.
433 West 93rd Street
Chicago, IL 60620

Wettach, William C.
Director Facilities & Equipment
United States Railway Ass.
2100 2nd Street, S.W.
Washington, D.C. 20590

Wiebe, D.
V.P. Research & Engineering
A. Stucki Co.
2600 Neville Road
Pittsburgh, PA 15225

Wilhelm, J. Roy
Manager-Marketing
Portec Inc., RPD
300 Windsor Drive
Oakbrook, IL 60521

Wilhite, Don
District Manager
Dresser Ind.
14447 Griffith Avenue
San Leandre, CA 94577

Willaman, P. O.
President
New York Air Brake Co.
Watertown, NY 13601

Wilson, John
President
Dynamic Sciences Ltd.
359 St. Croix, Suite 200
St. Lawrent, Quebec, Canada

Winn, James B.
Chief Improved Inspection
Office of Rail Safety Research
FRA
2100 Second Street, S.W.
Washington, D.C. 20590

Wiseman, Robert
OACI Task Force Man.
TSC
55 Broadway
Cambridge, MA 02142

Wulff, Cal W.
Vice President-Engineering
Holland Company
747 East Roosevelt Road
Lombard, IL 60148

Yang, Ta-lun
Chief Engineer, Trans. Group
ENSCO, Inc.
5408 A Port Royal Road
Springfield, VA 22151

Yearwood, Kevin
Mechanical Engineer
TSC
Kendall Square
Cambridge, MA 02142

Zarembski, Allan M.
Senior Research Engineer
AAR
3140 South Federal Street
Chicago, IL 60616

Zebrowski, Joseph R.
Regional Track Engineer
FRA
434 Walnut Street, Rm. 1020
Philadelphia, PA 19106

☆U.S. GOVERNMENT PRINTING OFFICE: 1978-261-264/191