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EFFECT OF HEAVY AXLE LOADS ON TRACK

12TH ANNUAL RAILROAD ENGINEERING CONFERENCE

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Sessions included track developments, suspension developments and track/train interaction. Eighteen papers were presented by authors from the United States, Canada, Mexico and Europe.

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PROCEEDINGS OF THE 12TH ANNUAL RAILROAD ENGINEERING CONFERENCE HELD AT UNIVERSITY OF SOUTHERN COLORADO PUEBLO, COLORADO, OCTOBER 23-24, 1975

THEME: EFFECT OF HEAVY AXLE LOADS ON TRACK

SPONSORED BY U.S. DEPARTMENT OF TRANSPORTATION
FEDERAL RAILROAD ADMINISTRATION

Bruce M. Flohr
Deputy Administrator
Federal Railroad Administration

Bruce M. Flohr was appointed Deputy Administrator of the Federal Railroad Administration in September 1975. Prior to assuming his new post, Flohr headed the Southern Pacific Transportation Company's San Antonio Division for four years. He had been with Southern Pacific since 1965.

Flohr, a native of Wallace, Idaho, received his B.S. degree in Industrial Engineering at Stanford University and his M.S. in Industrial Administration at Purdue University. As Deputy Administrator, he serves as chairman of the FRA Safety Committee, which reviews all railroad safety regulations and standards and recommends action on proposed regulations, and of the Railroad Operating Rules Advisory Committee. He is affiliated with the American Association of Railroad Superintendents and is a member of Tau Beta Phi.

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 has served for the past four years 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 until 1971 he was Acting Deputy Director, Office of Supersonic Transport Development. Prior to his federal experience Parsons was with the Martin Company, an aircraft and electronic systems manufacturing firm in Baltimore, 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.

Edward J. Ward
Conference Coordinator

Mr. Ward holds a Bachelor's degree in Civil Engineering from Union College, Schenectady, N.Y. and also Masters' degree in Civil Engineering from the University of Illinois and in Industrial Management from the Sloan Fellowship program at Massachusetts Institute of Technology. He has had a long association with the Air Force, serving as an R&D officer assigned to the Materials Laboratory during the Korean War and as head of the fatigue research group. Later he served as a civilian in planning and programming for the Aeronautical Systems Division. In 1963 he joined the Office of the Assistant Secretary of Defense--Comptroller as Director of Program Systems Development.

In 1965 he joined the Northeast Corridor Project Office in the Department of Commerce. He directed the beginning of the first railroad R&D program in the Federal government, including the establishment of the Transportation Test Center.

Mr. Ward recently served as Acting Associate Administrator for Research, Development and Demonstration of the Federal Railroad Administration. He was responsible for planning and carrying out the FRA R&D program for the Department of Transportation. He is a member of Sigma Xi

OPENING SESSION

The opening session of the technical program for the 1975 12th Annual Railroad Engineering Conference, conducted and sponsored by the Department of Transportation of the Federal Railroad Administration, was called to order by Edward J. Ward, Conference Coordinator.

He greeted the delegates and noted that the pressure of Congressional business had prevented the announced appearance of Congressman Fred B. Rooney (D-Pa.), Chairman of the Subcommittee on Transportation and Commerce of the House Committee on Interstate and Foreign Commerce, and Asaph H. Hall, Administrator, Federal Railroad Association. In Mr. Hall's absence, Robert E. Parsons, Associate Administrator of FRA, Office of Research and Development, was called on to give the address of welcome.

WELCOME ADDRESS

by

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

It is a pleasure for me to welcome you here this morning to the 1975 Railroad Engineering Conference. My only regret is that Administrator Hall could not deliver his welcome to you in person. I know he was looking forward to being here and meeting with you, but urgent legislative business forced him and Congressman Rooney to remain in Washington.

This is the 12th of the series of Railroad Engineering Conferences started by Dresser Industries and the second to be sponsored by the Federal Railroad Administration. We are here to focus on a problem of great importance to the rail industry: the effect of heavy axle loads on track life. This has a direct relationship to both the financial and the safety aspects of rail operations.

This problem is so complex that extensive research will no doubt be necessary in order to develop solutions. By exploring the present status of R&D efforts, along with some of the remedies or preventive measures already applied, we should be able at this meeting to identify new thrusts that will advance our understanding of the problems and possible solutions.

Before we get on with our task, however, I would like to highlight some of the excellent results of the first phase of a joint industry/government program and briefly discuss a new facility at our test center. Axle loads and their effect on track represent just one aspect of the broader study of track/train interactions. The study of track/train dynamics is, indeed, a formidable task. And to accomplish it, a formidable team was formed, combining the technical expertise, manpower, and financial resources of the Association of American Railroads, the Railway Progress Institute, the Federal Railroad Administration and the Canadian Transportation Development Agency.

The people engaged in this study are to be congratulated for their accomplishments, for they have been substantial. With the completion of Phase I of the National Research Program on Track/Train Dynamics, there now exists a complete bibliography on the dynamics of track and train systems. The three-volume set has been incorporated into the FRA's Railroad Research Information System and is available to all.

Train-handling methods have been developed, and a detailed list of these methods has been distributed to the railroads. With these tools, carriers can improve their operating efficiency and safety with their present equipment. The "invisible savings" can be substantial in terms of repairs not required, cargo not damaged, trains not derailed, and lives not lost.

Models of train actions are now available to predict force levels in moving trains, to determine safe or risky train makeup, to test equipment in areas considered unsafe for real equipment and personnel, to test proposed equipment before subjecting components to costly and time-consuming real-world tests, and to study and improve train-handling techniques.

The actions of engineers have also been studied in conjunction with several train-handling aids and new training programs. All in all, Phase I involved an intense data gathering effort which has laid a strong foundation for future research.

Work on Phase II of the project has already begun. It will devĕlop performance specifications for track and equipment, building on the modeling and testing efforts I've just described. Hopefully, new standards will be developed with an eye to low-cost implementation and operation.

Now there is another exciting joint effort of the government and industry. The acronym of this program is FAST, for Facility for Accelerated Service Testing. Basically, the proposed testing procedure will involve almost continuous operation of a test train over a closed loop, to accumulate years of over-the-road experience in a compressed time period. Such a facility will supply safety and performance data on rolling stock, freight systems, track, and other railroad components.

Our original plan was to construct FAST at the Transportation Test Center for operation in 1978. However, in order to begin the program earlier, we are making modifications to existing trackage at the center to provide for an interim facility. You will see the site on tomorrow's tour. While a permanent FAST facility has not yet been approved as part of FRA's budget, we expect to gain valuable experience with the interim FAST.

When it is fully operational, FAST is expected to apply up to 360 million gross tons per year to track test sections and up to 360,000 miles per year to rolling stock. Accepting 40 million gross tons as average heavy-density U.S. main line track utilization, we'll have achieved a time compression factor of 9. The rolling stock time compression factor may be as high as 18 if we use 20,000 miles per year as average car travel and 100,000 miles as representative of typical unit-train car mileage.

Obviously, with this kind of use or abuse of our track, we'll be available for testing new track maintenance equipment. Our test center can supply firsthand information on how well the maintenance equipment works, how difficult it is to operate, and what it costs to run. Maintenance-of-way planning--that is, determining how to allocate maintenance resources best--is presently in the pilot study stage, and the FAST program will help us validate the predictive maintenance-of-way models.

Yes, we have many plans for our test center here in Pueblo. You may have noticed that the name of the center has changed from the High Speed Ground Test Center to the Transportation Test Center. Today our research is geared more toward solving conventional railroad and transit problems and implementing near-and intermediate-term technological improvements.

On tomorrow's Transportation Test Center tour you will be taken to the Rail Dynamics Laboratory, where you will see the first piece of test equipment, a vertical shaker, now in use. We expect the Rail Dynamics Laboratory and its specially designed equipment to provide data that will contribute to solutions of vehicle dynamics problems and, perhaps, to the heavy axle-load problems we are here to discuss at this conference.

I mention our facilities and our programs to give you an idea of our capabilities and to remind you that the test center is intended for use by industry as well as government. Several companies have already used the facilities of our center; as a matter of fact, American Steel Foundries and Dresser Industries are here now. We hope many more will follow. We will consider the Transportation Test Center a success only when government and industry, both foreign and domestic, use it steadily.

All of us in FRA are pleased at the high level of technical capability of the delegates to this Conference. I would like to extend a particular welcome to our foreign visitors; we are honored by your presence.

I look forward to the papers and the discussions to follow. Your contributions may help improve operations for an economically troubled industry with the beneficial by-products of better, safer, and more economical service to shippers and consumers. In your hands is the raw material that can shape the future of rail technology. With your ideas, give it substance; with your engineering skills, give it form. We offer our test facilities for determining its value. I wish you all success.

MESSAGE FROM THE U.S. SECRETARY OF TRANSPORTATION, WILLIAM T. COLEMAN, JR.

Presented by Michael O'Rourke, Staff Assistant to the Secretarial Representative

This is a happy occasion for me personally, because it is always a genuine pleasure to meet again with old friends and colleagues of the Federal Railroad Administration. I remember when I first came out to Pueblo to look at the site of the Transportation Test Center, which some of you have visited in the past and will visit again tomorrow. In those days, that site consisted of a water hole, some sagebrush, a little controversy, and a lot of vision.

When I visit the Transportation Test Center today, I'm proud to have been associated with men whose vision has given it its present reality and whose efforts are making it a transportation landmark. This morning you begin an important interchange of knowledge and ideas, past experience and future plans. I'm here to convey to you Secretary Coleman's wishes for a productive and successful conference.

The theme of this conference is an expression of what is likely to become a matter of great interest and concern to those who live in this region: the effect of heavy axle loads on track. Certain areas in this region are already booming as a result of coal production--areas like Rock Springs and Gillette, for example. The potential for the expansion of that product is mind boggling. Now, the best way for moving coal out of this region is by rail, and the producers and the planners are thinking in terms of hundred-car unit trains, carrying 100 tons per car, and they are thinking of hundreds of such trains on a daily basis. This represents an opportunity, of course, as well as some problems for those who are involved in the rail system. To succeed, the great masses of coal that are to be transported out of this region must be moved efficiently and economically. Track and roadbed will have to be upgraded and maintained in terms of this challenge.

The alternatives, things like slurry pipelines, only become reality when trains cannot move the product. I do not think anyone needs to tell you the importance of the topic you are approaching in this conference, and I certainly will not presume to do so. In part, this message is intended to let you know that we recognize the importance of the contribution you are making, as well as to remind you that the matter does have a sense of urgency.

While my message from the Secretary is brief, this does not diminish the thrust of it--that the development and modernization of the nationwide, privately owned interstate rail system is essential to the national interest. Your conference lends momentum to that thrust.

Secretary Coleman has said that the Department of Transportation will act promptly to provide assistance in improving and modernizing the rail system and keeping it in the private sector. The 12th Annual Railroad Engineering Conference is a specific example of that intent. The Secretary feels that the kind of cooperative effort which is represented here today gives substance to the articulation of the national transportation policy. It is primarily through the efforts of an innovative, cooperative, and forward-looking private sector that we shall see a more perfect transportation system evolve. The Department of Transportation supports your efforts and will continue to participate in them.

Now, having wished you well in the work you have come here to do, it ill becomes me to stand any longer in the way of your getting it done. I would like to say welcome to Colorado and wish you success in your endeavors, and, of course, a safe journey home--no matter what mode of transportation you use.

SESSION 1

TRACK DEVELOPMENTS

Conference Coordinator Ward introduced the topic of Session I, Track Devèlopments, and noted that this session would deal with some of today's problems and some actions being undertaken to solve them. Ward introduced the Theme Address speaker for Session I, L. Stanley Crane, Executive Vice President-Operations of the Southern Railway System.

L. Stanley Crane
Executive Vice President--Operations
Southern Railway System

L. Stanley Crane's lifelong career in railroading began in 1937, when after receiving the BS degree in Engineering from George Washington University he became a laboratory assistant for Southern Railway. He was appointed Vice President--Engineering and Research, Southern Railway Company in 1965 and elected to his present position as Executive Vice President--Operations in 1970.

Crane has been active in numerous organizations concerned with rail transportation. In the AAR he is a member of the Operations-Transportation General Committee, the General Committee of the Data Systems Division, and the Research Committee. He has served as chairman of the Rail Committee of the American Railway Engineering Association and as chairman of the Rail Transportation Division of the American Society of Mechanical Engineers. An active member of the American Society of Testing and Materials since the beginning of his career, Crane has been honored with the position of Fellow and is past director and past president of the society. He is also a member of the Transportation Research Board Executive Committee of the National Research Council. The editors of Modern Railroad Magazine named him Railroad Man of the Year in 1974.

THEME ADDRESS

Figuring The Price Tag For Marketing Innovation

It is always a source of particular enjoyment to me to share ideas with people engaged in research. That was my introduction to the railroad business, and I have been involved with it in one way or another for most of my working life.

Even when I am away from research, as I now am, and largely concerned with the more immediate problems of operating a railroad, I look back to it with pleasure. Researchers are idea people - studying, observing, thinking and probing - and this makes them interesting people to be with.

I am most grateful to you for asking me to have a part in this Conference and to open the discussion on one aspect of the important considerations you will deal with during the next few days.

The effect of heavy axle loads on the track structure is becoming a matter of increasing concern to all of us in the railroad business. It is part of the price we pay for meeting customer needs in a new and effective way--and we have to know just what the price is.

Railroads made the shift to bigger, higher capacity cars largely in response to the economics of marketing freight service. Our costs of handling freight are directly related to the cost of moving the freight car. If we can handle more freight in each car, we can reduce the cost, and we can share the savings with our customers and potential customers. We can also make better use of the railroads' transportation capacity and earn a greater share of the growing transportation market.

So we made our freight cars longer and higher and heavier, and we began to reap the competitive benefits. But there is a hidden price tag for this kind of innovation – and one that we may not have taken fully into account when we priced the kind of service we were prepared to give.

There were a number of reasons for this.

The effect of heavy axle loads on the track structure does not appear for several years, and when it does appear it is difficult to evaluate. For one thing, it is hard to isolate axle loads from other maintenance factors and hard to sort out the effect of 100-ton cars from that of 50-ton and 70-ton cars. But it can be done. We are meeting today at the place where it can be done, and in my judgment we need to do it – we have no alternative.

We have made commitments to our shippers. We have given them the incentive of volume loading to reduce their shipping costs. And we are not going to be able to take back these incentives in today's competitive transportation market. Not unless government regulation should require uneconomic standards of track maintenance for the operation of 100-ton cars -- and I do not believe that this will happen.

So we need to learn what we must do in order to live with these commitments. Pueblo is the place where the lessons will be spelled out. This is not a new problem, by the way. When railroads went from the 40-ton car to the 50- and 70-ton cars, they had to wrestle with some of the same problems of roll-off and rock-off and other causes of derailments that we face with the 100-ton cars. But the problems now are of a different order of magnitude. So we've got a lot of thinking, studying, observing and probing ahead of us.

Southern had to deal with this problem somewhat sooner than most other railroads. As early as 1960 we built and began to operate our first heavy movements of "Silverside" coal gondolas, the "Big John" covered hopper cars appeared a couple of years later. In addition, we built a number of smaller but equally heavily loaded hoppers to carry alumina, salt, and cement.

We began to learn that this heavy equipment could get you into trouble on the lighter branch lines, and the unit coal trains were giving difficulties even on track maintained to the best standards of main line maintenance at that time. We're still strengthening track and bridges along the routes of unit coal trains.

We made some dynamic bolster measurements in 1961 and 1962 which convinced us that rail joints were and would continue to be our principal problem. It became evident that we were simply not going to be able to run cars that heavy on jointed track without excessive costs to maintain the track.

Nothing that has happened since has changed my belief that it is not practical to run many 100-ton cars on jointed track and still maintain consistently any degree of reliable service. Anything with a bolt or a joint in it will start to work loose after a time.

So Southern went to welded rail -- as rapidly and extensively as we could. We have something like 5,000 track-miles of ribbon rail now. When you do go to welded rail, every remaining joint you have in that welded rail becomes a problem.

That's why Southern has pioneered in the use of field welded joints and glued insulated joints -- eliminating joints wherever possible. We're using audio frequency overlays for activating crossing signals to eliminate the need for some joints. We're using welded rail through switches, and turnouts. Perhaps someday we'll come up with an all-welded turnout.

We see no likelihood that the trend toward bigger cars and heavier loads is likely to be reversed. An examination of industry statistics shows plainly the pattern that continues to develop. The average freight car load, which increased only from 40 to 44 tons between 1944 and 1960, jumped to 58 tons in 1974. The average freight car capacity -- the amount that could be loaded in a car - went from 51 tons in 1944 to 55-1/2 tons in 1960 and to 71-1/2 tons by 1974.

As engineers we must recognize that the economic factors encouraging railroads to resort to these heavier axle loads will not permit us to turn back the clock. we must be able to tell our people who are doing the transportation pricing just what the incremental costs are of maintaining track with this strength and durability.

And we have -- or will have -- the tools right here at Pueblo. By that I mean the Accelerated Service Test Track.

It is generally recognized that the maximum life of our track structure is somewhere between 600 and 800 million gross tons -- in whatever time frame this is applied. But at the 200 million to 300 million gross ton level you begin to get a pretty good idea of what is happening to rail, crossties, track, and fastenings as a result of this continued wear.

Say we have test loop track approximately eight miles in length and we operate a 10,000-ton train on it at 40 miles per hour for 16 hours a day. Every day we're going to put 800,000 gross tons of wear on that track. In 250 days we will be up to the 200 million gross ton mark. Make it two 10,000-ton trains and we can halve the test time. We can establish a base with 50-ton cars or 70-ton cars, then determine through testing how much more rapidly track deteriorates with all 100-ton cars or a predetermined mix of 100-ton cars with the lighter ones.

This is the sort of testing that the Federal Highway Administration had to do to determine the effect of heavy trucks and light passenger cars in breaking down highways and what each contributed to the cost of maintenance. Their researchers have been able to define some equivalence ratios for various weights of axle loads and thicknesses of pavement. We should be able to do something of this sort for railroad track.

Many of you, I am sure, are familiar with the study of railroad reliability conducted by Massachusetts Institute of Technology for the Federal Railroad Administration. You have seen Southern and other railroads working to put the results of this study into practice to improve the reliability of rail service. This is important to all of us because we know how reliability affects the shipper's choice of the transportation mode he will use.

Most of these MIT findings are extremely useful <u>refinements</u> of railroad operating practice to achieve greater point-to-point reliability for freight shipments.

But track is <u>fundamental</u> to railroad reliability. In fact, it is fundamental to every aspect of our business.

To sell railroad freight service, we have to make it attractive to our customers. High-capacity cars and the cost savings they make possible give us one important way of doing this.

To keep on providing that service we have to make sure that it is economically remunerative to railroads.

And track is very significantly involved in both these factors.

When you're dealing with the effect of heavy axle loads on track and the possibility of rock-off derailments and wheel lifts, there are certain things you can do in the design and maintenance of equipment. I certainly would not want to minimize the importance of what we need to learn in this area, but, basically, it comes down to track: strong track, heavy track, well-maintained track - with just as few bolted joints as possible.

That means, of course, that the problem of expansion in rail will continue to be very much with us and will have to be dealt with. That's a fertile field for research. We need to accelerate our research studies of the expansion forces in track which must be reckoned with, because we do not yet understand at all clearly the reasons for rail buckling in track. Oh, we know that the constraint of expansion in rail may break out in vertical or lateral movement or, in extreme cases, even fracture of the rail. We understand that the I-beam section characteristic of railroad rail makes it stronger against vertical movement than against lateral movement. But there are still a lot of questions to be answered about how to control this tendency to buckle.

What standards of restraint should be applied to track – especially as they relate to heavy axle loads and the heavy lateral forces they impose under certain dynamic conditions that cause track to be displaced? We need to identify what the levels of the loads are and to come up with a rational method of holding track in position. Does this mean that we need to go to a concrete crosstie because its heavier mass will not allow it to be displaced as easily? Are heavier ballast sections an answer? Do we need to resurface track more often than we do? Should we go to more rigid rail fasteners than the cut spike in the wooden tie – such as screw spikes or some sort of spring clip rail retention device to hold rail to the tie?

All these are things we really don't know yet – and need to know. Pueblo is a place where we can work out some of the answers.

This installation has an exciting potential for research in our industry. It is the first step I have seen the Federal Government make toward the kind of in-depth research into railroads that has been done so extensively for other transportation modes.

I say <u>potential</u> because the good that comes out of this research center will depend to a great extent on how we order our priorities. I have said before and will say again that we need to get down to basics in our research – to the track structure on which everything else we do depends. That is why I welcome the basic theme of this conference and am glad to be a part of it.

You are going to have a busy two days here dealing in depth with the basics of our business – the strength of track, the durability of equipment, and the effect of increasingly heavy axle loads on both. Then you're going to see some exciting vistas of the future when you tour the Transportation Test Center tomorrow afternoon. The order in which your meeting is scheduled can serve as a useful reminder to us all that certain priorities should be observed.

Linear induction motors, air-cushioned vehicles and 300 mph passenger trains are fascinating looks into the future. There is no question in my mind that we must have innovation – of all kinds – if we are to keep this industry alive and growing. But there are some basics we have to deal with first. We have to strengthen our track structure and the equipment we use on it. We have to employ more efficiently the fuel, steel, crossties, and rolling stock that have become increasingly difficult to get and costly to buy.

The events of the past decade have brought home to us the fact that marketing innovation sometimes has a hidden price tag. The volume rates and heavy loads that brought business and revenues to the railroads also brought problems in track and equipment maintenance.

I do not suggest that we are unwilling to pay the price for innovation. I do believe that we have to know what the price is. And we're counting on you to help us find out.

Howard C. Meacham
Manager, Applied Dynamics and Acoustics Section
Battelle Memorial Institute

Mr. Meacham received his B.S. and his M.S. degrees in Mechanical Engineering in 1956 from Ohio State University.

Mr. Meacham joined the Battelle staff in 1956, and is now Manager of the Applied Dynamics and Acoustics Section of the Structures and Mechanics Department, which specializes in research on problems related to vibration and noise--particularly those which are transportation and vehicle oriented. Some of his earlier studies related to the surface-compressive stress relationships in components such as bearings and steel wheels-on-rails, as well as a number of analog computer analyses of suspension systems of vehicles ranging from automobiles to large off-highway trucks to freight cars. These latter studies, directed particularly toward problems such as car rocking and rail corrugation, necessitated the development of computer programs incorporating the track structure as well as the vehicles itself, and led to Mr. Meacham's interest in the complex subject of vehicle-track dynamics.

Carrying this work forward, a series of research programs were sponsored by the Office of High-Speed Ground Transportation of the Department of Transportation, relating to the analysis, design, and construction of improved track structures. In June of 1970, Mr. Meacham became Manager of the Applied Dynamics and Acoustics Section. In addition to the heavy emphasis on rail vehicle-track technology, a substantial portion of the Section's research relates to acoustics. In addition to many studies relating to the analysis and reduction of noise from specific items of equipment, more recently the acoustic projects under Mr. Meacham's supervision involve analyses of the environmental effects of noise, or noise pollution.

During the last few years an increasing number of research studies on rail vehicle-track dynamics have been conducted, including field studies involving advanced instrumentation to measure track structure response. Recent studies have included analysis of passenger and freight truck dynamic responses, including hunting and car rocking, curving at high speeds (in connection with the Northeast Corridor improvement program), and wide gage/rail rollover problems, to name a few.

Mr. Meacham is a member of Pi Tau Sigma and Phi Eta Sigma, national engineering honoraries, and is a member of SAE.

DETERMINATION OF LOADS ON TRACK

Never before has so much track been battered so hard by so many high loads as is the case today, and I'm sure most of you are aware of that. The high loads prevalent today have spawned--and probably will continue to spawn--a number of problems, ranging from some relatively minor ones such as rail corrugation up to the potentially most serious problem of all--derailments. As a result, it has become more and more important to be able to determine what the wheel/rail loads actually are in order to apply proper design and maintenance procedures on both vehicles and track so we can live with these loads.

Ten to fifteen years ago there were very few ways available to determine wheel/rail loads, but thanks to rapid advances in technology--both of instrumentation and of computers--the situation today has been greatly changed. I would like to discuss some of the techniques we at Battelle have developed or found useful for the determination of loads on track. Both computer modeling techniques and actual field measurement techniques can be used,although the best of both worlds is to use both techniques in a complementary fashion.

Determination of Loads by Measurements on Track or Vehicle. The oldest method of determining loads is to apply instrumentation of some sort to the track. The first problem one encounters when considering a track measurement program is just where the measurements should be made. Obviously, the vertical, lateral, and longitudinal loads which are exerted on the track vary from place to place along the track, sometimes in a random fashion and other times in a predictable or known pattern. Usually only a few locations can be used because of cost considerations, and therefore the choice of these locations is extremely important. For relatively steady-state conditions such as might occur on a long stretch of tangent track where the trains generally are in a steady-state condition--whether braking, traction, or drifting--it is possible to obtain a valid load spectrum by choosing only a few track measurement locations but recording the data for a large number of trains. A sufficient data base can be obtained by this technique to ensure a valid statistical sample of loads. This is a technique which was used on our recent measurements on the Union Pacific in conjunction with the gauge widening problems this road was experiencing.

Even though the train itself is in a steady-state condition, it is well known that loads vary from one tie to the next, even in a relatively good roadbed. Thus, ideally, more than one location should be checked to ensure that valid data are obtained. It is important to realize that the track is an indeterminant structure, so the tie reactions (tie/plate loads), rail deflections, and rail-bending movements are quite dependent on the overall track stiffness in the different loading directions. With a specified wheel load applied to rail having uniform tie supports, the portion of the wheel force transmitted to individual ties is determined by the rail-bending stiffness and the effective stiffness of the track at each tie. A single tie typically carries from 40 to 60 percent of the lateral wheel/rail load and from 60 to 80 percent of the vertical load. The track stiffness also affects the magnitude of the dynamic wheel/rail forces caused by the interactions between the vehicles and track as a dynamic system.

A situation completely different from the steady-state tangent case must be faced on curved track. An example is the most recent program in which we participated on the Northeast Corridor to determine the loads exerted on track by various types of high-speed trains--including those pulled by locomotives and the self-propelled Metroliners. On this track there are many curves, and the possibility of passenger train derailment on curves--or spirals associated with these curves--is of great concern. On curved track the train dynamics may result in different loads being measured at each location chosen on the track, and the best that can be hoped for is that sufficient judgment is exercised to choose these locations wisely and that trends will be evident from measurement of the different vehicles which are consistent from one location to the next, even though conditions differ at various locations. With proper instrumentation, this appears to be the case.

Loads need not be measured only on the track; they can be measured also on the vehicle. But here again there are many problems. It is desirable to measure loads as close to the action as possible-that is, right down on the wheel/rail interface. But from a practical standpoint, with today's technology this still cannot be done well, and the closest we can approach this is to use an instrumented (strain-gauged) wheelset. However, we know that the leading axle and the trailing axle--not to mention the middle axle on a three-axle truck--will usually load the rail somewhat differently. This means that just instrumenting one wheelset is not sufficient; rather, two or three wheelsets of a truck should be instrumented. Indeed, it would be nice to know what the front and rear trucks of a single car are doing, not to mention what the trucks of different cars are doing. Obviously the problems in data acquisition and the tenfold problem of analyzing all this data can quickly get out of hand. This leads again to the age-old choice of using engineering judgment based perhaps on some preliminary abbreviated instrumented runs to narrow down the possible locations if track measurements are to be made.

Determination of Loads by Computer Simulation. That, perhaps, is enough discussion of some of the problems involved with determination of loads on track by actual measurement. Now let's look at the other way of determining loads--namely, by mathematical modeling procedures. As anyone knows who has ever given even cursory thought to the problem of modeling the dynamic action of a train, there are many, many degrees of freedom in a train when one considers the trucks and the numerous components in each truck which are free to move relative to one another. Now another problem arises: If one is to model the action of the vehicle or train by computer techniques, which of the numerous degrees of freedom are significant to the problem, and which can be left out without making the results invalid?

Since it is not possible to include all degrees of freedom--or at least certainly not economical to do so--the trend developed over the past ten years has been to develop models in which a particular dynamic mode is studied. This may be truck hunting, rock-and-roll, or longitudinal train action, but usually not more than a couple of these different dynamic phenomena are included in the same model. Furthermore, the problem is complicated by the fact that the action of the truck may be considerably affected by whether or not the train is braking, accelerating, or in a neutral condition, a fact recently being realized more and more. Therefore, there is always the legitimate question as to whether the proper conditions were simulated in the model, a question which can be answered only by some degree of validation of the model by actual field tests.

The great thing about a computer simulation which includes both the vehicle and the track in a dynamic interacting system is that one can easily specify as outputs the vertical and lateral forces between the wheel and rail--something very difficult to measure. The other advantage of the computer technique is the speed with which various parameters can be changed in the model and their effects on wheel/rail loads can be evaluated. Things such as gross weight and primary and secondary suspension characteristics can easily be simulated and run on the computer, whereas the evaluation of the same parameters in a series of field tests might take months.

RESEARCH PROGRAMS IN LOAD DETERMINATION

Now that I've told you the bad news--all the problems one gets into in trying to determine the loads on track, whether by measurement or simulation--let me tell you the good news. In several programs on a variety of problems we have been involved in over the years, we have been able to determine loads on track with what I believe is a good degree of success. I would like to discuss a few of these diverse projects briefly. We can look at these projects chronologically.

Back in 1966 we developed a simulation on 100-ton freight car rock-and-roll and validated this model by tests at the Hollidaysburg, PA test facility set up for rock-and-roll tests (Fig. 1). While we did not measure wheel/rail loads directly during the validation tests, we did measure the vehicle displacements and loads such as the side bearing loads, and we achieved excellent correlation of the computer results with the test results on the Hollidaysburg test track.

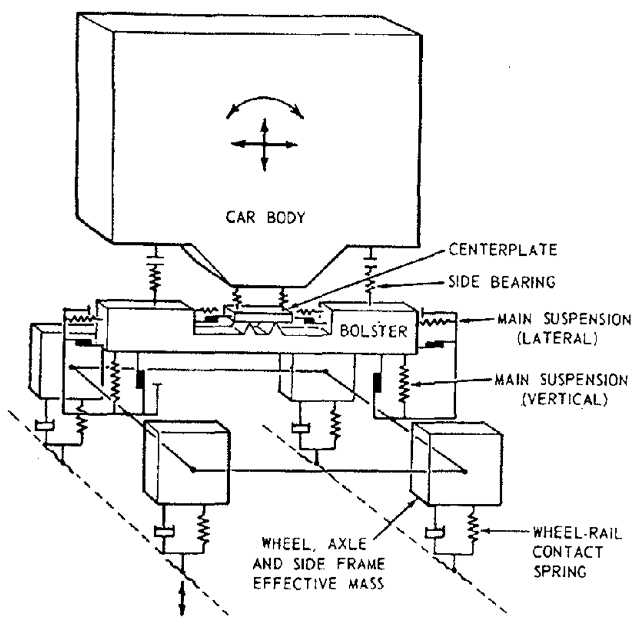

Fig. 1. Freight car model used for rock-and-roll studies.

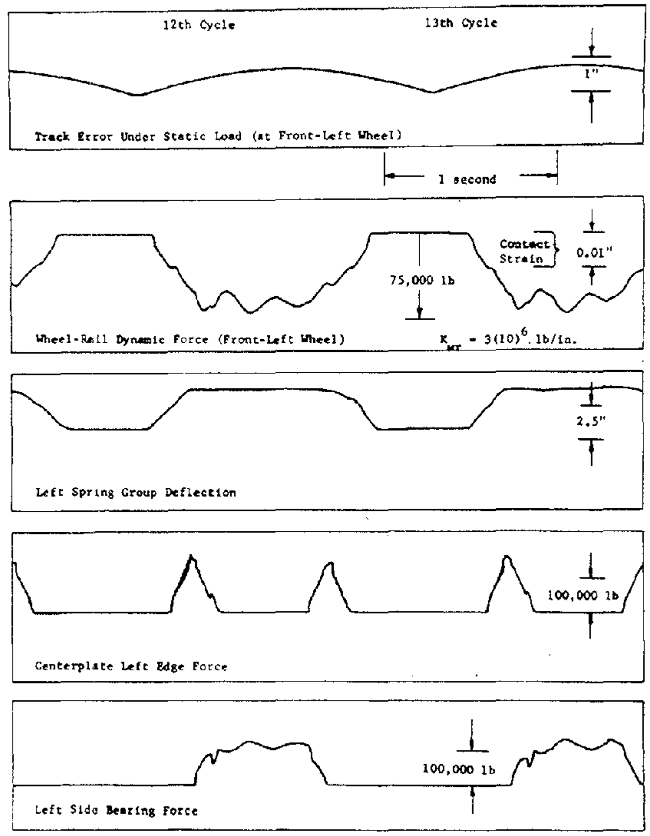

Fig. 2. Time history of several parameters during severe car rocking.

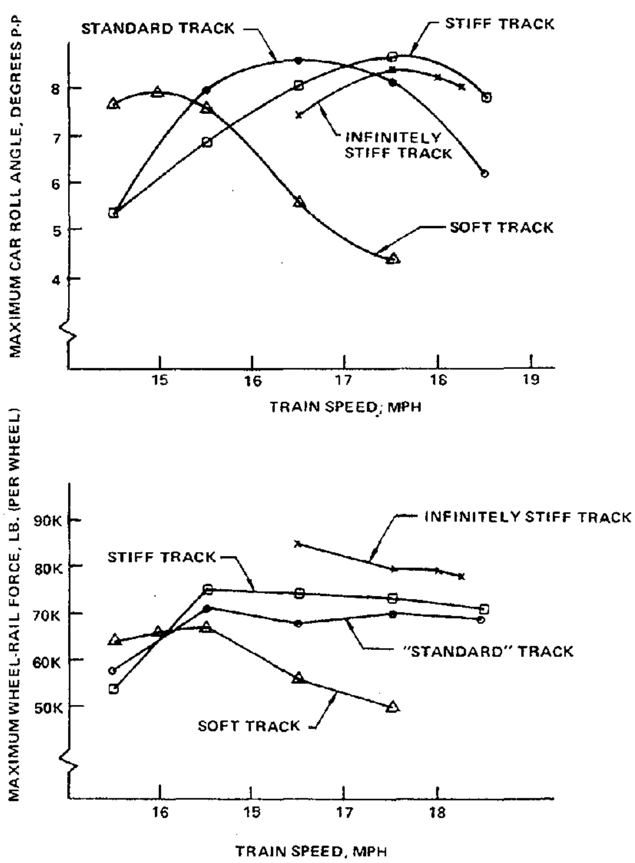

Fig. 3. Effect of track modulus on wheel/rail dynamic loads during car rocking.

Fig. 2 shows the time history of several parameters, including the wheel/rail load during the car-rocking process. Note that the load reaches double its nominal value; this is to be expected, since the one side lifts completely off the track. This project, which was sponsored and encouraged by Carl Tack of American Steel Foundries, also included studies on the effect of the track modulus, which Carl rightly believed had a large effect on the severity of car rocking.

Fig. 3 shows some of the results with different track stiffnesses. The main point is that a resilient track reduces the severity of car rocking, whereas a stiff track such as is produced in the winter with frozen roadbed conditions can greatly aggravate the car-rocking problem. This model did not include coupler forces and therefore is valid only for tangent track, but for this condition, it accurately determined the forces exerted on track for the car-rocking situation.

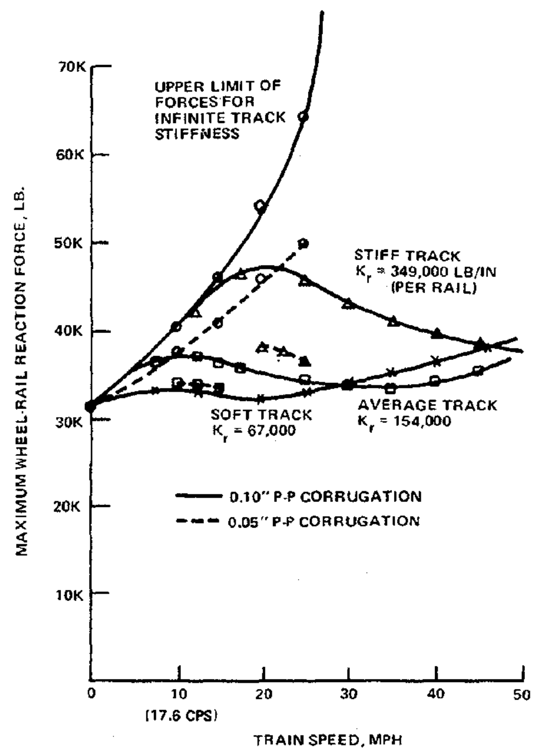

Fig. 4. Effect of track modulus on wheel/rail dynamic loads over corrugated rail.

About the same time we were conducting the car-rocking project, Leavitt Peterson of the Bessemer & Lake Erie (now with FRA) contacted me with a story about rail corrugation which was occurring prematurely on a heavy ore-carrying road in Canada. He had the same idea--namely, that the rigid track found in the winter was aggravating the wheel/rail loads, and this was accelerating the corrugation on the rail. I assured him that I felt this indeed was possible, and after a trip to look at the track we modeled both the track and the ore car as an interactive dynamic system. Fig. 4 shows the relationship of the wheel/rail load to the track stiffness--with all other parameters remaining unchanged. This is another example of what I consider to be an accurate determination of the load on track by computer simulation. Some excellent B&LE research on actual measurement of loads by instrumented wheelsets resulted from this early project.

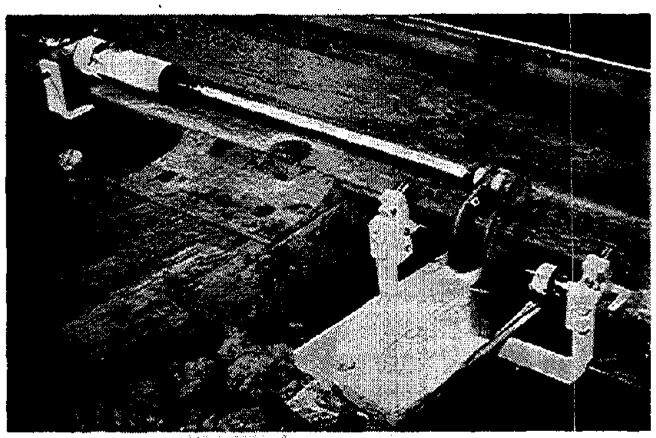

Fig. 5. Rail anchor load measurement setup.

Fig. 6. Rail anchor instrumentation setup.

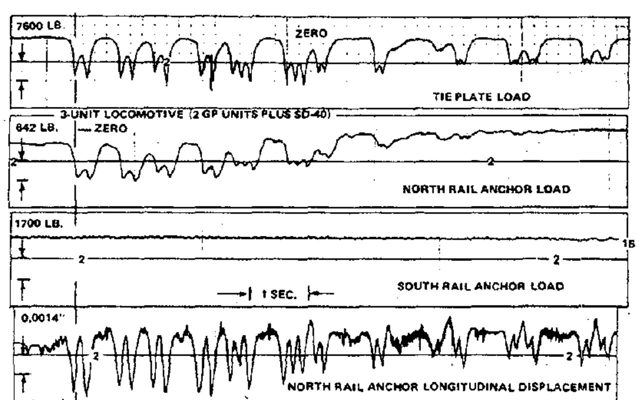

Fig. 7. Measured rail anchor longitudinal loads and displacements.

In the 1968-1969 time period we worked on two programs involving the determination of loads on track, one particularly pertinent. One involved the measurements of the longitudinal loads on track which are transferred from the rail into the ties by the rail anchors. Fig. 5 shows the rail anchor load measurement setup, and Fig. 6 illustrates how the instrumentation setup is adjusted. Fig. 7 is actual data taken on longitudinal loads measured on the Chessie System mainline near Columbus.

At this same time we were deeply involved in a study sponsored by FRA on improved track structures. We had developed computer programs to study, in particular, the effect of beam and slab track stiffness on wheel/rail loads, soil pressure, and other parameters. Fig. 8 shows the computer model representing DOT test cars, and Fig. 9 is data from the computer model. Measurements to validate computer results were made on the Penn Central near Bowie, MD., as shown in Fig. 10. Fig. 11 illustrates a tie plate load cell and pressure cell junction box. The Bowie test data, including tie plate loads and subgrade pressures, are shown in Fig. 12, and Fig. 13 shows tie plate loads at the joint and a few feet away from the joint.

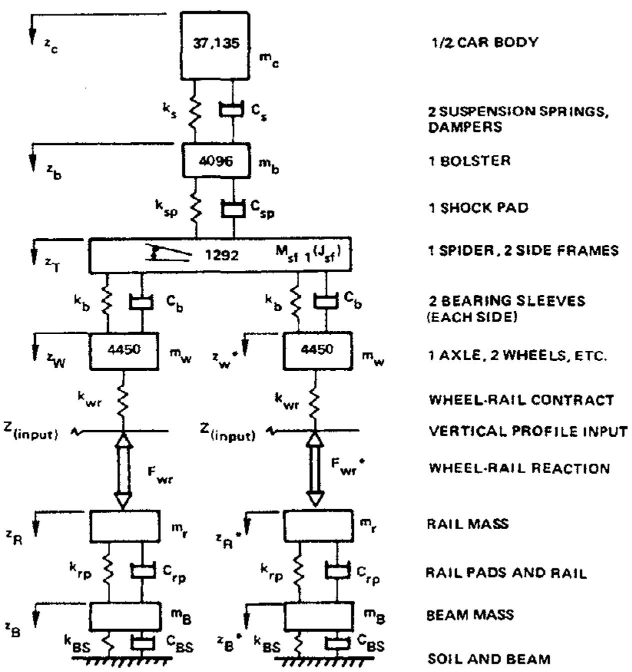

Fig. 8. Computer model for studying passenger car/track structure vertical response.

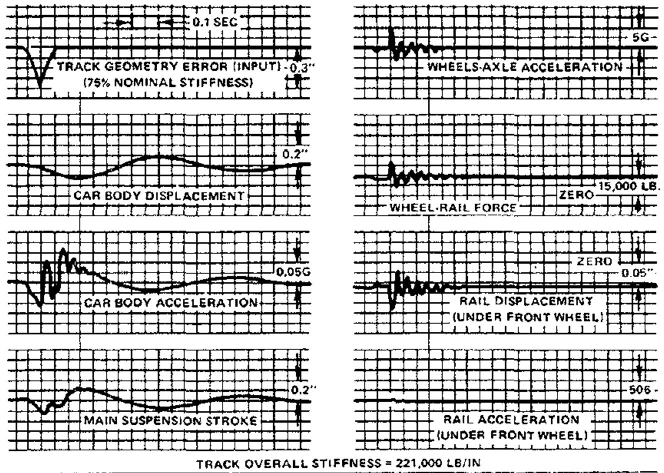

Fig. 9. Data from computer study of vehicle response to low joints.

Fig. 10. Track response measurement setup on NEC track near Bowie, MD.

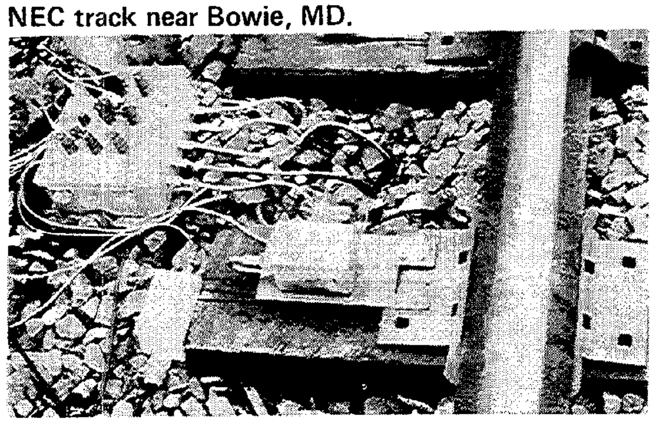

Fig. 11. Tie plate load cell and subgrade pressure cell junction box.

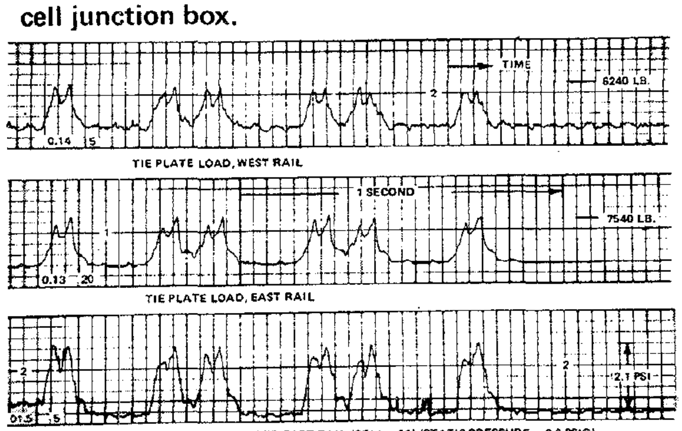

Fig. 12. Bowie test data, including tie plate loads and subgrade pressures.

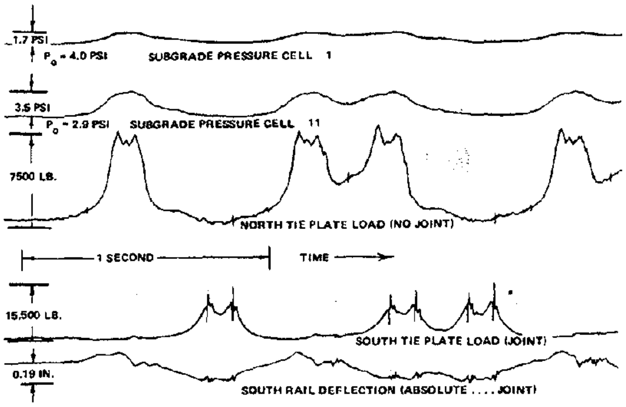

Fig. 13. Tie plate loads (at joint and away from joint).

Now I'd like to go to some of our most recent projects. As reported by Don Ahlbeck at the November 1974 meeting of the Track/Train Dynamics Program in Chicago, measurements were made on the Union Pacific track near Pocatello, ID (Fig. 14), where problems had been experienced with rapid gauge widening. For this program, which was part of Task 13A of the AAR-RPI Track/Train Dynamics Program, we designed specialized instrumentation which could be moved from one location to the next on the track, with a minimum of disturbance of the track and therefore minimum interference of revenue traffic. Fig. 15 shows the calibration of instrumentation on the Union Pacific. Our improved instrumented tie plate, similar to the one used for the Bowie runs but including improved circuitry so that we could measure rail overturning moment and vertical load, is shown in Fig. 16. We also designed a special fixture to measure the dynamic change in gauge of the track and the lateral motion of each of the two railheads relative to their bases. Fig. 17 shows the instrumentation setup on the Union Pacific. We found different types of cars gave considerably different load signatures. Fig. 18 shows some of the typical results. From this type of data it was found, for example, that the culprit was not the locomotives, as had been expected by some, nor was it even the heavily loaded cars. Rather, it was the hunting of the empty cars which was causing the large lateral gauge excursions under light load conditions. Typical data are shown in Figs. 19, 20, and 21.

Another program of this same type was carried out in the desert regions of Southern California where derailments caused concern on the SP. The test site and instrumentation are shown in Figs. 22, 23, and 24.

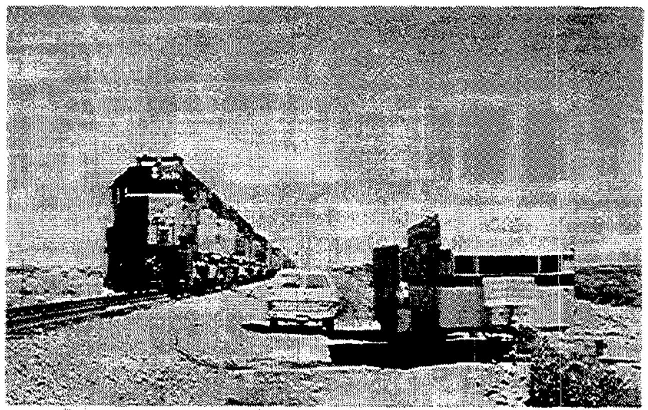

Fig. 14. UP site for wide-gauge studies near Pocatello, ID.

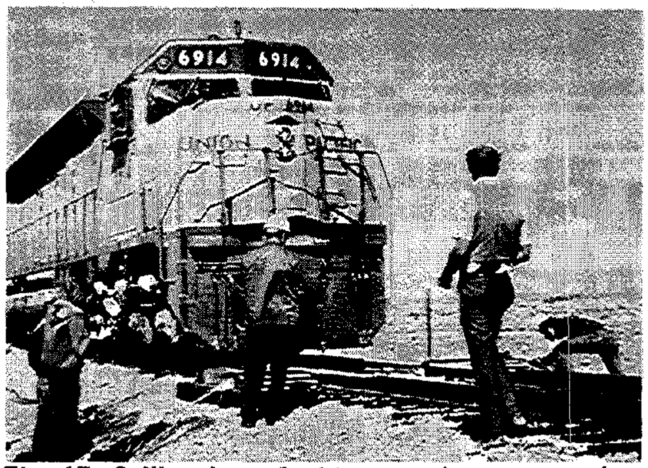

Fig. 15. Calibration of wide-gauge instrumentation on The Union Pacific.

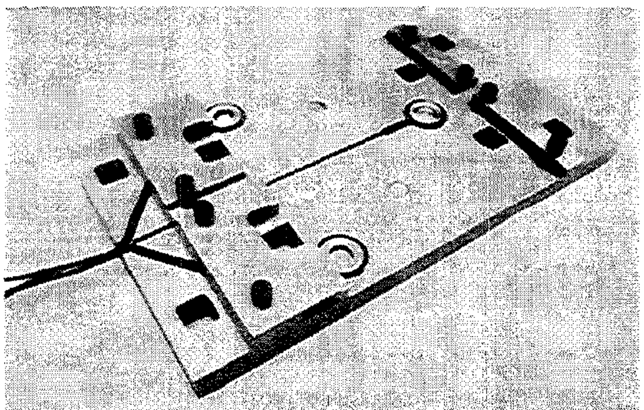

Fig. 16. Instrumented tie plate for measuring vertical load and rail overturning moment.

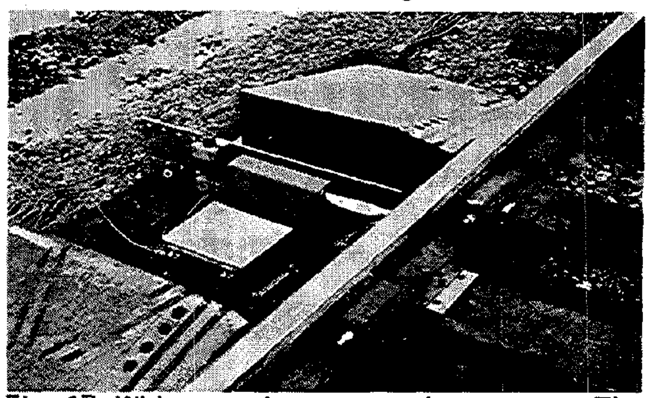

Fig. 17. Wide-gauge instrumentation setup on The Union Pacific.

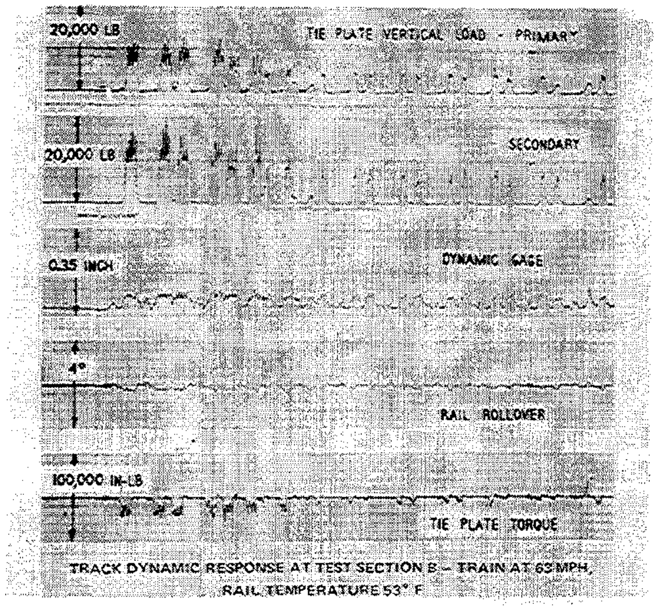

Fig. 18. Typical results showing load, overturning moment, dynamic gauge and rail rollover.

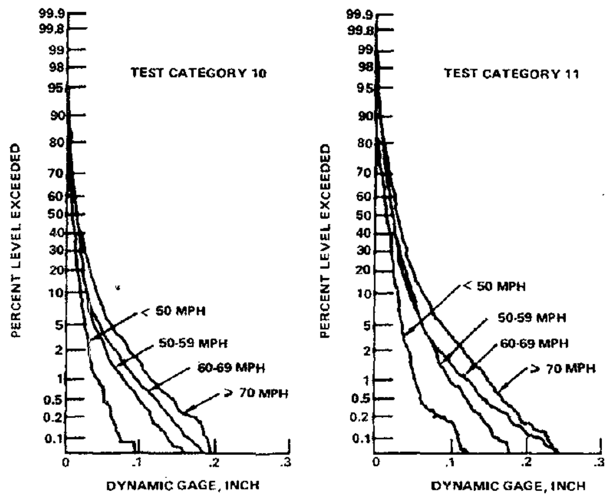

Fig. 20. Effect of train speed on dynamic gauge cumulative probability distributions, all traffic.

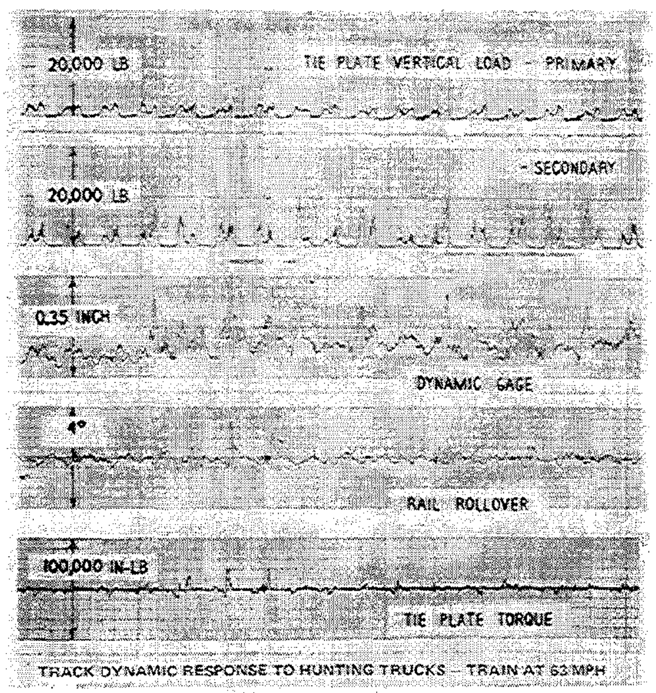

Fig. 19. Typical data illustrating dynamic gauge excursions due to truck hunting and flange impact.

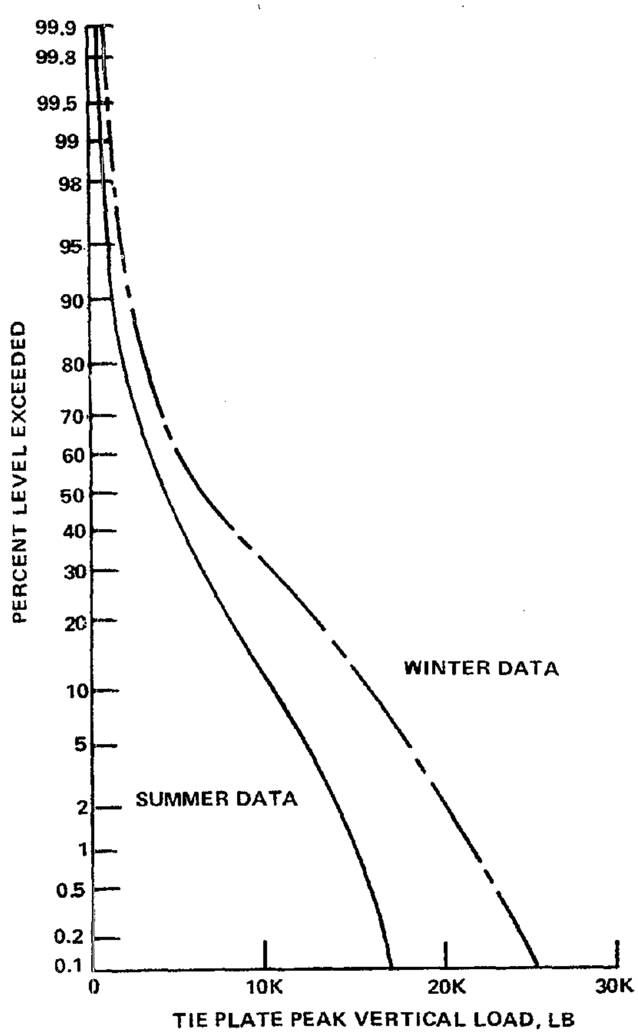

Fig. 21. Cumulative probability distribution of peak tie plate vertical load under summer and winter (frozen ballast) conditions.

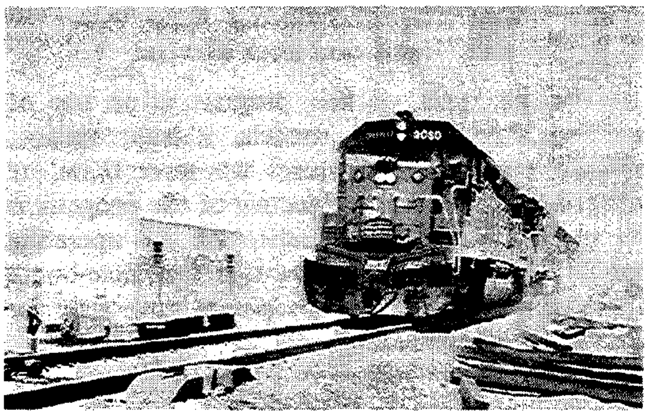

Fig. 22. Test site on Southern Pacific for wide gauge/rail rollover studies.

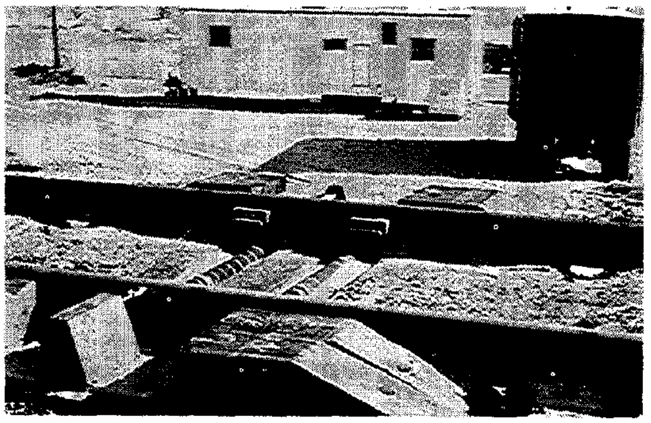

Fig. 23. Wide gauge/rail rollover instrumentation setup on SP.

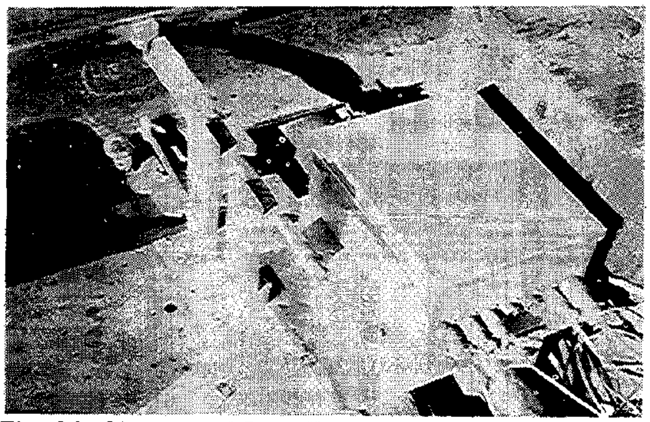

Fig. 24. Closeup of Battelle measurement setup used for measuring dynamic gauge.

On the most recent project on the Northeast Corridor, where the lateral and vertical forces of different passenger locomotives and cars on curves were of particular interest to AMTRAK, we made measurements at three different sites. The "Midway Interlock" site near Princeton, N.J. is shown in Figs. 25 and 26. At each location the conditions dictated a different instrumentation setup, yet the results led to similar conclusions at each location regarding the relative lateral and vertical forces exerted on the track by the different vehicles, including the locomotives. In particular, we were quite pleased with some new strain gauge circuits which we used for the first time for measuring vertical and lateral wheel/rail loads as shown in Figs. 27 and 28.

Fig. 25. Midway interlocking test site for AMTRAK track response measurements.

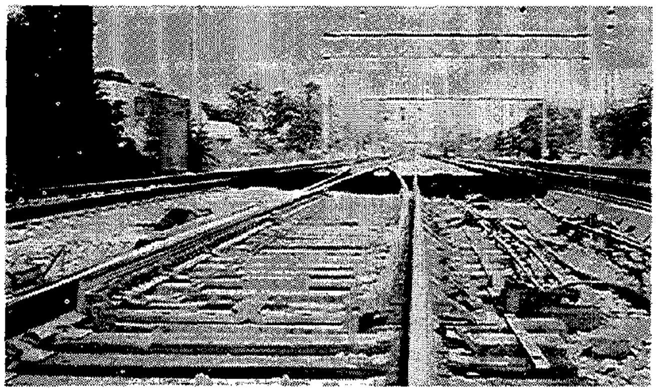

Fig. 26. Closeup of Midway interlock test site.

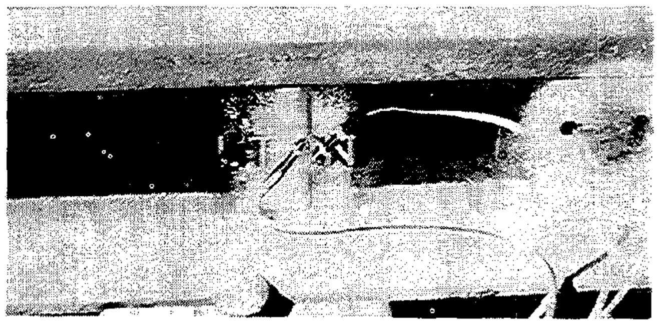

Fig. 27. Photograph of actual strain gauge installation for measuring lateral loads.

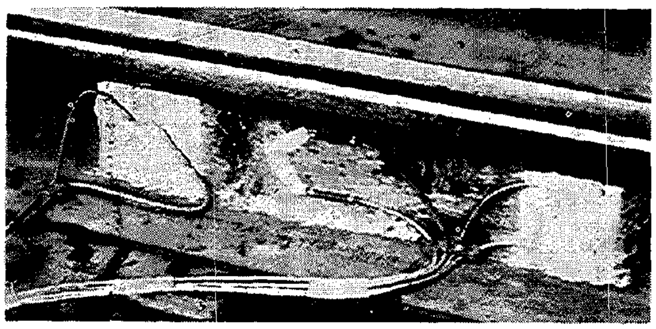

Fig. 28. Photograph of actual strain gauge installation for measuring vertical loads.

At the present time we are well underway on a project entitled Characterization of Wheel/Rail Loads, sponsored by FRA and implemented by the Transportation Systems Center (TSC). This is a part of FRA's Improved Track Structures Program. The objective of this research program is to characterize the rail loading environment for the range of vehicles, track conditions and operating parameters typical of U.S. railroad service. The quantitative description of rail loads developed from this program will be used as input for the concurrent TSC programs on rail stress analysis and rail failure prediction, and for other programs on track design and laboratory testing of track structures. The development of validated predictive technique for wheel/rail loads will be used to extend the data base obtained from the measurement phases of this program and to evaluate strategies for reducing those wheel/rail loads that cause significant track damage.

The emphasis of this program is on use of existing data, analysis models, instrumentation, and data analysis procedures whenever these are adequate. A secondary objective of the program is to provide a better evaluation of the operating limits for some of the track and vehicle-borne instrumentation that has been used previously for measuring wheel/rail loads.

As they say on the television commercials, "We've come a long way, baby", and I think this applies to the determination of loads on track. We now have at our disposal the tools with which to determine loads, whether experimentally or analytically. Now all we need to do is to apply them to help solve some of today's problems.

I. A. Reiner
Manager, Engineering Research
Chessie System, Inc.

Mr. Reiner is a graduate of the Hungarian Royal Palatinus Joseph University of Technical and Economical Sciences, Budapest. He holds degrees and diplomas in civil engineering, railroad engineering and economics obtained in 1943 and 1944.

In 1947 he joined the Hungarian State Railroad as assistant division engineer. After completing the 18-month railroad management training program and serving in the engineering, finance and planning department, he became chief engineer in 1954.

Under the sponsorship of the National Academy of Sciences, in 1957, Mr. Reiner came to the United States, where he was employed by the Baltimore and Ohio Railroad. In 1966 he was appointed manager-engineering research for C & O - B & O. In this capacity he performed numerous computer-aided systems-studies such as the economics of concrete ties vs wood ties, heat treated rails vs standard carbon rails, the development of incremental track cost models for various axle loads, traffic densities and operating speeds, etc.

Mr. Reiner is the author of several papers dealing with special technical and economical problems published here and abroad.

He is an international consultant frequently participating in various engagements here and abroad. Most of these engagements were directed toward the use of track measuring cars and maintenance programming.

Mr. Reiner was a member of the UIC and now is a member of the AREA. He is listed in the "Who's Who in Railroading" and in the "Dictionary of International Biography".

TESTING CONCRETE TIES

Causes Leading to The Search for New Track Materials. In the past fifteen years, the railroad industry has been compelled to accommodate itself to a number of environmental, technological and operational changes.

A. Uncertain Supply and Spiraling Prices of Woodties. Domestic woodtie production has always been plagued with peaks and valleys, as monitored by railroad revenue forecasting. As long as the average of highs and lows did not go much below the volume of tie replacement considered normal, there was no need for corrective action. However, the unprecedented low level of tie production beginning in 1962 (12 million ties, or 40% of the normal requirement) (References 1 and 2) and the very slow rate of increase since that time give some cause for concern. Today's production level is about two-thirds of the normal requirement.

By contrast, there is no shortage in standing timber. According to the U.S. Department of Agriculture, (Reference 3) current and prospective timber supplies appear at least sufficient to meet the projected demands for most timber products until the year 2000, if not endless. Then how can one explain the phenomenon of insufficient production level in woodties?

As you all are aware, there have been a few noticeable technological and institutional changes in the lumber industry, as well as some basic shift in the philosophy of corporate management.

First, it can be assumed that the explosive growth of pulp and paper companies during the sixties is responsible for a multitude of changes witnessed throughout the lumber industry. The paper industry began some 280 years ago, but only relatively recently has it shown a sharp upturn. Department of Agriculture figures indicate that the U.S. consumption of paper per capita was 110 lbs. in 1920, 206 lbs. in 1940, 420 lbs. in 1965, and about 500 lbs today. Its volume relative to the total wood production has increased since 1920 from 5% to nearly 40% now. By contrast, woodtie production during the same period decreased from 1.8% to 0.6% (Reference 3).

The growth of the pulp and paper industry here and in Canada has made quite an economic impact on the entire lumber industry. Among other things, the financial strength of large paper companies forced the smaller companies in the lumber business to merge. This was followed by the liquidation of hundreds of small sawmills, most of them former woodtie producers. The owners and the employees involved either left the forest product industry for good or joined the paper industry. Some of the "going out of business" was triggered by the unhealthy wage differential between woodtie producers and the other sectors of the lumber industry.

Manpower loss was not the sole problem the woodtie industry has had to face. Those who remained in the business found shortages and price increases for hardwood. This has been caused by the intensified use of hardwood by the paper companies, particularly in the North, followed by the development of suitable chemical processes for pulping hardwood. Efforts made to establish new sawmills for woodtie production have not been successful. Other attempts, such as efforts to convince the larger sawmill operators that woodtie production is profitable, also have failed. Today, the climate for diversification is even less favorable.

Second, for more than a year now, there have been signs that a basic shift is taking place in the way "corporate America" looks at itself. After more than twenty years of unprecedented growth and prosperity, corporate managers are readjusting their sights. Where once they made a cult of growth, they are now thinking small. Today, as the external environment becomes more complex, some business executives feel that the day of the conglomerate is gone, the time of daring is over, and the diversification moves they have made have proven very expensive. Hence, there is a definite trend to show preference for a single-industry rather than a multi-industry approach, and corporations are putting the emphasis on doing what they can do best and with minimal risk.

B. Increase of Wheel Loading. The trend toward larger capacity cars and heavier wheel loads is not new; it began in the early 1940's. But it was not until the mid-sixties that the increase became more noticeable. Today, the average freight car capacity of the existing fleet is over 70 tons, an increase of 55% since 1940. New cars constructed have 100-ton capacity or more.

Larger car capacity is, beyond doubt, a must in competing with other modes of transportation and a key factor in rate reduction. Large-capacity cars, in fact, helped railroad marketing in its fight to reduce rail traffic erosion. Unfortunately, this is the only opinion common to all parties involved--marketing, sales, mechanical, and engineering departments. The differences are in opinions over two basic questions: How should large capacity be obtained? What are the conditions of establishing new, realistic cost floors and rate structures?

The fallacy in today's large-capacity car design is that the systems approach is not utilized. Thus the effect of important factors such as track maintenance increments is ignored, and a sufficient number of alternative solutions is not explored.

In addition to the laboratory test results (Fig. 1), there is also an increasing amount of field observation to support the view that high wheel/rail pressures and heavy wheel loads are detrimental to the rails and tracks. It also has become evident that these effects on rails and tracks, unlike on bridges, appear only after a long time lag or large traffic volume. The usual signs of heavy wheel loads are premature shelling (Fig. 2) and corrugated rail; breakdown of switches and insulated and regular joints; shorter tie life; irregular track geometry (Fig. 3); and overloading of the subsoil, as manifested by the need to pump

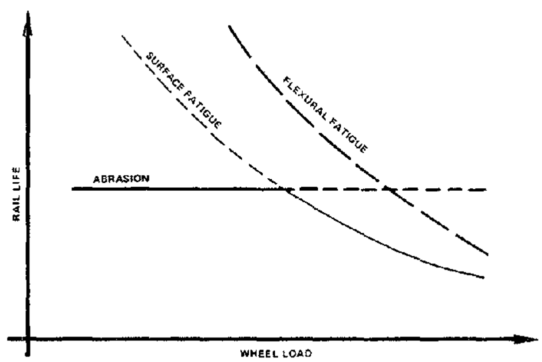

Fig. 1. Typical relationship between the magnitude of wheel loading and the number of load cycles to cause failure.

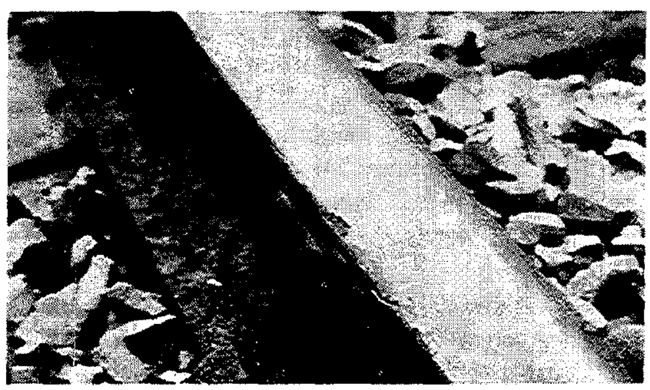

Fig. 2. Shelly rail - after less than 150 million gross tons of traffic.

ballast in territories where ballast condition was normal under lighter wheel loads.

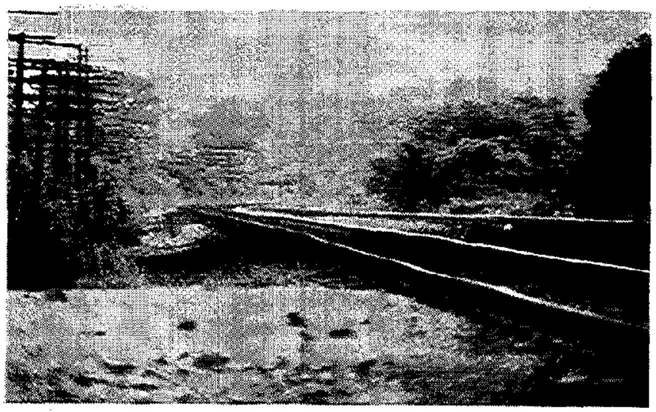

Fig. 3. Irregular track surface.

Large capacity does not necessarily mean unduly heavy wheel loads. Larger capacity can be obtained in many different ways other than just building a bigger car body and putting it on the existing trucks. The diameter and the number of wheels can also be changed. Each particular combination of capacity, wheel diameter, and number of wheels yields different costs of car construction, car operation and maintenance, and track maintenance. The one combination that results in the lowest total cost (and highest railroad profit) is the optimum car design with the most economical wheel load to carry a certain type of commodity (Fig. 4).

Assuming that an optimum capacity car has been designed for a certain commodity, the corresponding operating costs alone are insufficient for establishing new rates. Other one-time costs, such as those needed for upgrading tracks; stabilizing roadbeds; rebuilding or reinforcing bridges, track scales, or hump yard retarders; and providing clearances in tunnels, bridges, or along rock cuts should also be determined if they constitute an indispensable condition of heavy car

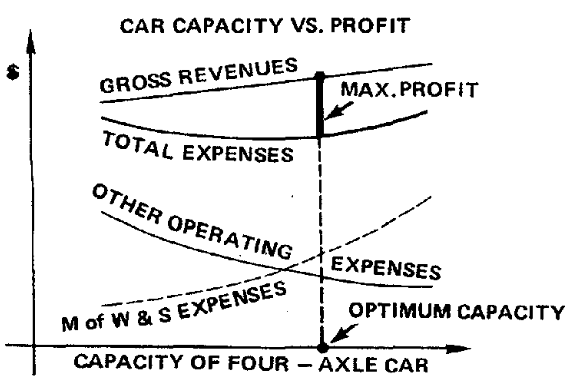

Fig. 4. The scheme of systems approach to arrive at optimum car capacity.

operations. Although these costs usually are not incremental (traffic-volume-related) costs (Reference 4), in this instance they must be included when establishing rates and cost floors. There are other requirements too. The mere possibility of rate reduction depends on sufficient traffic volume, so there is a threshold value for the degree of plant utilization (or variable cost/fixed cost ratio). Below these threshold values, the rate reduction eliminates profit.

Chessie's Participation in Concrete Tie Developments. As forecast and actual trends in woodtie production levels kept diverging, and it appeared that there was no way of changing the course of operating large-capacity cars which more and more proved to be improperly designed, Chessie launched a feasibility study to investigate the technical and economic aspects of crosstie substitutes. For a number of reasons, the examination of concrete ties received high priority. The following objectives were set:

1. To supplement the insufficient quantity of crossties so as to be able to obtain normal replacement levels.
2. To increase the supporting capability and stability of track.
3. To improve the economics of tie replacement programs and track maintenance

At the time the project began, the first objective seemed to be the most urgent one, because woodtie production was very low. Today we think the other two are equally important, in order to counterbalance the effects of heavy cars.

A. Preliminary Studies. Drawing a lesson from a number of previous not exactly successful trials regarding concrete tie performance, we felt than an in-depth study of the prospective use of concrete ties was essential. Such a study might also reveal the possible causes of the earlier failures. Factors such as the maximum proportion of wheel load on one crosstie, ballast and subsoil pressure, and vertical, lateral, and longitudinal stability of track and their correlation with tie size and spacing were analyzed. The results were essential in arriving at trade-off economics for concrete ties and woodties. We also learned immediately that one possible cause of the early failures was the weakness of fasteners.

A cash flow cost model was then derived which is applicable for a number of sets of conditions, including new track construction and replacement of existing woodties with concrete ties. The application of the model enabled us to broadly outline the conditions under which concrete ties are economically justifiable.

The analysis of the technical aspects told us that another possible cause of the early failures was the wide (30-in.) spacing of the concrete ties. We discovered some obstacles, too. For example, we learned that a large-scale installation of concrete ties requires a new type of equipment with high productivity. It also became evident that current revenue rules are unfavorable in regard to replacing woodties with concrete ties, and this made their justification difficult. This is because the IRS rule stipulates the capitalization of the entire costs of concrete ties and fasteners. (U.S. Treasury Department Internal Revenue Services--Revenue Ruling 68-418, Section 263, Capital Expenditures.)

B. Concrete Tie Test at Noble, Illinois.

B1. Laboratory Tests of Selected Ties. After reviewing the domestic and foreign literature on the subject and consulting with other railroads about their experience with concrete ties, we selected four basic types with fasteners for laboratory and field performance test. The four types of ties and fasteners are as follows:

1. Monoblock, prestressed tie--U.S. Abex-Interpace with AAR fastener (Fig. 5).
2. Monoblock, post-tensioned tie--German, B66 with British Pandrol fastener (Fig. 6).

Fig. 5. Installing the AAR fasteners on the Abex ties at Noble, Illinois.

3. Two-block, regular reinforced tie--French, RS with RN fastener (Fig. 7).

Fig. 6. B-66 ties with Pandrol fasteners at Noble, Illinois.

4. Two-block, prestressed tie--Swedish, 101 with Fist fastener.

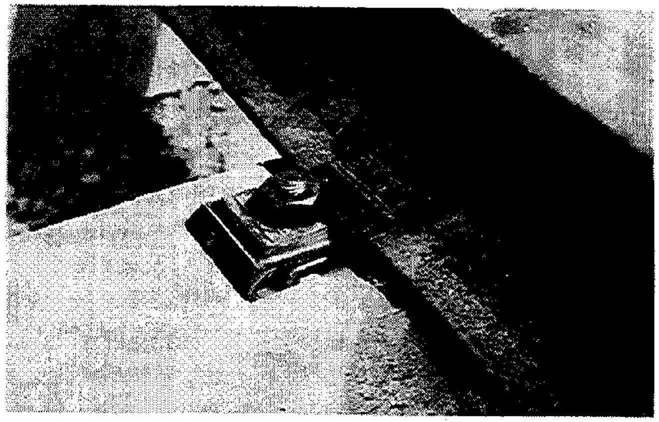

Fig. 7. RN fastener on RS two-block tie at Noble, Illinois.

The Abex-Interpace tie was designed to meet our specifications regarding size and strength. The European ties were slightly modified because of our heavier wheel loads and different standard of rail cant.

Full-scale sample ties were manufactured in this country, with the exception of the Swedish tie, which was made in Stockholm for laboratory testing. Ultimate strength of tie, positive and negative bending, lateral and longitudinal resistance of fastener (installed on the tie via 12-in.-long rails) have been determined. The impedance of the tie-fastener-rail assembly was also measured. The Abex, German and French ties were found adequate, but the Swedish tie was not, because the rail overturned on the tie during the lateral load test. This tie, therefore, was eliminated from the planned field test.

B2. Field Performance Test. The primary purpose of the field test at Noble, Ill. was to determine the durability of concrete ties under heavy main line service (25 MGT annual traffic).

Three types of concrete ties which passed the laboratory tests and a woodtie control panel with all new ties were installed on upgraded ballast, with 122-lb. welded rail. Each section was 1440 ft. long. The tie spacing was 27 in. for the Abex and French ties, 25 in. for the German ties, and 23 in. for the new woodties. The five-year observation and measurement of the test track (1968-1973) ended with the following results:

- The track depression under load and track settlement along the concrete ties are less than one-half along the new woodtie track.
- Track gauge, surface, and alignment and rail cant are significantly more uniform on the concrete ties (Fig. 8).

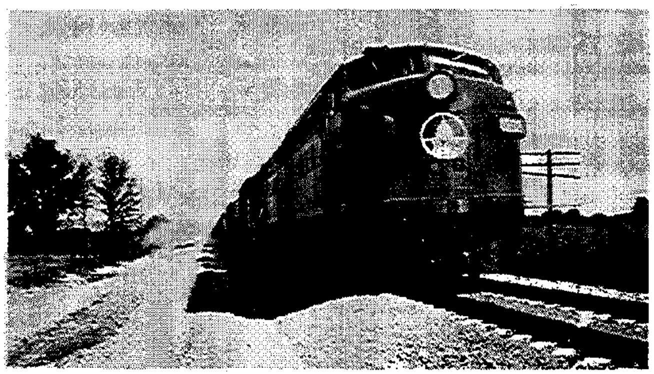

Fig. 8. The concrete tie test track at Noble, Illinois.

- Some rail seat and center cracks developed on the monoblock ties.
- The designed positive bending moment (150,000 in./lbs.) of the monoblock ties appeared low.
- The creep resistance of the Pandrol clip (2,500 lbs. for longitudinal rail movement) should be increased.
- The ballast became muddy along the French ties.
- A large percentage of tie pads were displaced or moved out entirely from under the rail on the Abex and German ties.
- A few fasteners were broken on the monoblock ties.

It was concluded that none of these types of concrete ties performed sufficiently well. We thought that the two block tie idea should be dropped and the monoblock ties and fasteners need several improvements and modifications. A detailed report is available. (G.H. Way, Progress Report on Concrete Tie Track at Noble, Illinois, 1973).

C. Further FRA-Sponsored Tests at Sabot and Lorraine, Virginia. In the early seventies Chessie entered into a contract with the Federal Railroad Administration with the aim of participating in the expanding exploration of concrete tie capabilities. The activities involved have been in three specific areas:

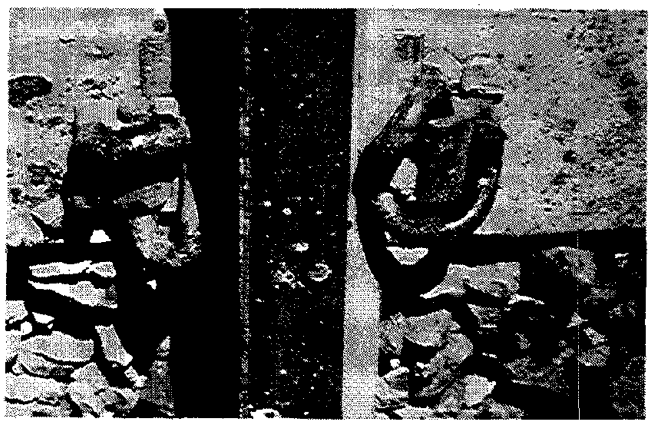

Fig. 9. British Costain concrete tie and pandrol 607-A fasteners at Lorraine, Virginia.

Fig. 10. Gerwick RT -7S concrete ties and British trec CS-5 fasteners at Lorraine, Virginia.

1. Laboratory test of improved concrete ties and fasteners--British Constain (Fig. 9,) and American Gerwick RT-7S ties (Fig. 10) with Pandrol 607-A and TREC CS-5 fasteners respectively.
2. Lateral track resistance field test of concrete ties and woodties.
3. Concrete tie in-track performance test.

The tie and fastener test was carried out during 1971 in England at the British Rail Darby Test Center under the direction of G.H. Way. The corresponding report is available on request.

C1. Lateral Track Resistance Test. Today's heavier axle loads and higher operating speeds as well as the prospective use of concrete ties with nonconventional fasteners suggest the reexamination of track stability, particularly for lateral loads. Although several field tests were conducted in Europe (by Blondel, Amman, Gruenewaldt, Martinet, Nemesdy and Schramm) and analytical work was done (by Kerr, Nemesdy and Schubert), the results of foreign tests are not necessarily applicable in this county because of the differences between U.S. and European track structures (rail weight, fastener systems, crosstie size and spacing, and ballast section). Furthermore, recent European investigations (Reference 7) were directed toward the determination of ballast resistance only, by uncoupling the rails and ties during the lateral load test. We would like to know what the total track resistance, including the stiffness of the rail, is.

Objectives. Several objectives were set to be achieved with this new lateral load test in order to answer the following questions:

1. What is the difference, if any, between the lateral resistance of concrete and woodtie tracks?
2. To what extent are tracks weakened after raising and tamping?
3. To what degree can mechanical ballast compaction restore track resistance?
4. How quickly is track regaining lateral resistance under the exposure of traffic?

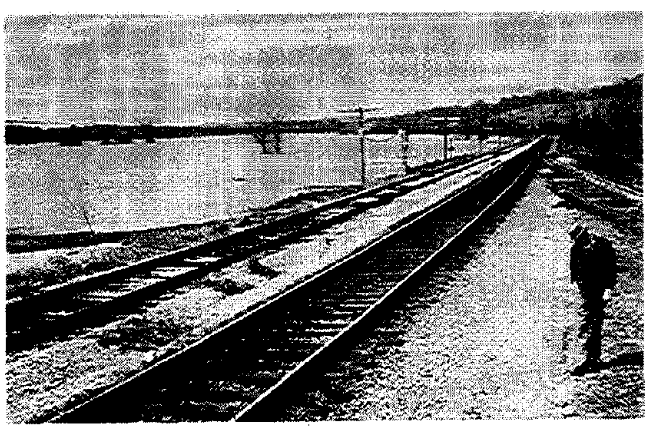

Fig. 11. Test site at Sabot, Virginia looking west.

Test Site, Layout, and Preparation of Test Panels. For the test site, the C&O mainline track was selected at Sabot, VA. (Fig. 11), twenty miles west of Richmond. This site had the following road and operating characteristics and climatic conditions:

Gradient	level
Alignment	tangent
Subgrade	clay/sand
Ballast	crushed limestone
Ties (wood control section)	7" x 9" x 8.5'
Tie spacing	20"
Rail	132 lbs. RE, jointed, rolled and laid in 1956
Fasteners	14" tie plates, cut spikes, and wooden anchors
Annual traffic	25 MGT
Operating speed	50 mph
Annual precipitation	44"
Average Temperatures:	
January	40 deg. F
July	78 deg. F
Annual	58 deg. F

PANEL NO.	TEST SEQUENCE	PANEL CODE	PANEL CONSTRUCTION	PANEL PREPARATION
			PHASE I	
1	1	A-East	22 New Wood	
2	2	B-East	16 Old Wood 6 New Wood	Two inches raise with additional ballast and tamping
3	3	C-East	17 New Concrete	
4	4	A-West	22 New Wood	
5	5	B-West	16 Old Wood 6 New Wood	Two inches raise with additional ballast tamped and mechanical ballast compaction
6	6	C-West	17 New Concrete	
			PHASE II	
1	7	A-East	22 New Wood	
2	8	B-East	16 Old Wood 6 New Wood	Two inches raise with additional ballast and 7 MGT traffic
3	9	C-East	17 New Concrete	
4	10	A-West	22 New Wood	
5	11	B-West	16 Old Wood 6 New Wood	Two inches raise with additional ballast, tamping ballast compaction and 7MGT traffic
6	12	C-West	17 New Concrete	
7	13	D-1	16 Old Wood 6 New Wood	
8	14	E-1	22 Old Wood	
9	15	D-2	16 Old Wood 6 New Wood	None (Control Panels)
10	16	E-2	22 Old Wood	

Fig. 12. Test layout and testing phases.

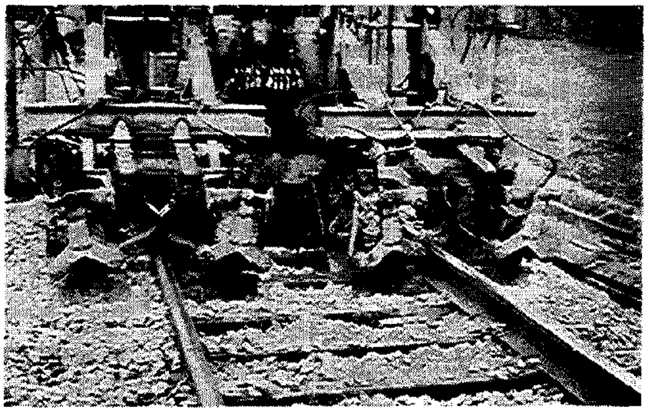

Fig. 13. Raising and tamping the test panels at Sabot, Virginia.

Fig. 14. The ballast consolidator.

The lateral test was carried out in two phases on ten individual 39-ft.-long track panels constructed and prepared for a specific task, as shown in Figs. 12, 13 and 14.

Instrumentation and Data Recording. The instrumentation and data recording were performed by a subcontractor (Reaction Instruments). The task was to obtain a continuous recording of lateral track displacements and the corresponding lateral forces. Each panel, one at a time, was instrumented as the test proceeded. For each panel 12 analog strip charts have been produced depicting the displacements of 10 ties and 1 point on the rail and the applied force (Fig. 15).

Lateral load was applied at the center of the panel through a 5-ft.-long bridle attached to the rail base. The purpose of the bridle was to split the lateral load into two components, simulating the lateral load transfer to a standard two-axle truck. The bridle was cable connected to an axial strain gauge load cell, then, in line, to a 15-in.-stroke, double-acting hydraulic cylinder. At the other end of the cable a firmly anchored bulldozer (Model D9 Caterpillar) provided the reaction force (Fig. 16).

A hydraulic system, utilizing an electrically driven gear pump (high volume, medium pressure) and a hand pump (low volume, high pressure), was used to energize the cylinder. The load was measured with the strain gauge load cell, whose output was amplified and passed through a signal-conditioning chassis which converted the load cell output into a voltage signal.

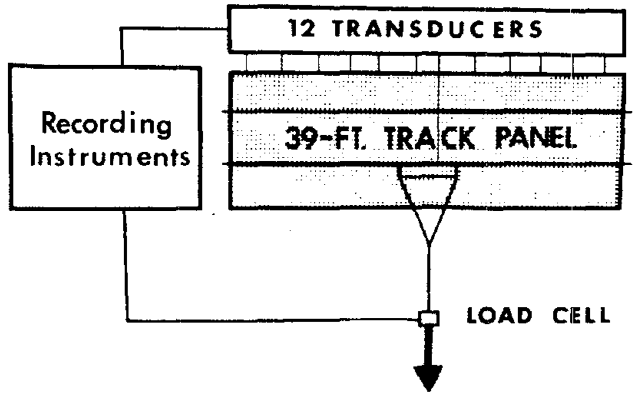

Fig. 15. Scheme of the lateral load test at Sabot, Virginia.

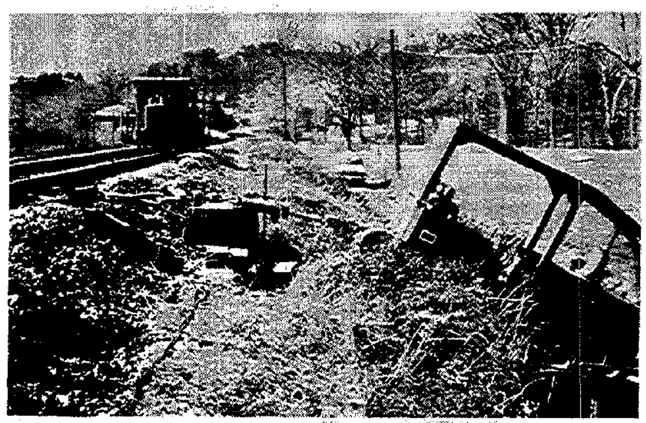

Fig. 16. Applying the lateral force to rail - shown from left to right bridle, load cell, hydraulic pump and catepillar.

The movement of the selected ties and the rail at the middle of the panel (Fig. 17) was measured by displacement transducers. All electronic signals generated by the transducers and the load cell were recorded on two (six channels each) analog strip chart recorders, whose channel sensitivities were set according to the scale factors of transducers and the load cell.

Fig. 17. Connecting the displacement transducers to the concrete ties.

Test Procedure. In operation, the hydraulic cylinder was gradually pressurized to increase the lateral load on the track panel. Load and displacements, then, were simultaneously recorded on the strip charts. After the track panel yielded (motion without force increment or with drop of force level), the cylinder was depressurized, and the test was terminated for that track panel (Fig. 18).

Fig. 18. Displaced concrete tie panel.

Evaluation of the Results. In general, the test results were in agreement with previous findings for settled tracks published by Dr. Schramm (G. Schramm, Permanent Way Technique, Darmstadt, 1961) (Reference 8). Details are given in Fig. 19. In respect to the stated objectives, we can summarize the results as follows:

1. There is not much difference in lateral track resistance between concrete and woodtie tracks (Fig. 20).

2. Tamping could weaken lateral track resistance materially, as much as 60% (Fig. 21).

3. Mechanical compaction measurably increases lateral resistance on freshly tamped track (Fig. 22).

4. The effect of about 5 MGT traffic is equivalent to mechanical ballast compaction.

Track Panel Designation		Displacement (in) at Force (K lbs) 12	15	Yield Force (K lbs)	Displacement (in) at Yield Force	Max. Force (K lbs)	Max. Displacement (in)
Phase I (Completed in April, 1975)							
East	A	1.74	–	12.00	1.74	12.25	2.29
	B	0.59	–	14.00	1.53	14.40	3.00
	C	0.70	1.88	15.10	1.88	15.00	2.12
West	A*	0.36	–	13.50	0.63	14.30	2.37
	B*	0.29	0.94	15.00	0.94	15.60	1.82
	C*	0.12	0.33	20.00	1.84	20.25	2.05
Phase II (Completed in August, 1975)							
East	A	0.20	0.47	15.25	0.77	15.25	2.10
	B	0.19	0.35	17.00	0.87	17.00	2.30
	C	0.06	0.20	18.50	1.29	19.60	2.13
	D_1	0.03	0.05	Control Wood Tie Panels		27.00	0.48
	E_1	0.02	0.03			26.00	0.10
	D_2	0.09	0.11			26.00	0.22
	E_2	0.08	0.11			28.50	0.31
West	A	0.11	0.20	16.00	0.25	16.30	1.00
	B	0.06	0.11	17.50	0.42	17.50	1.92
	C	0.29	1.10	15.40	1.28	15.40	2.80

*Track Panels Tested Immediately After Ballest Compaction

Fig. 19. Summary of results.

We have to point out, however, that these results were obtained in measuring a very small number of track panels at one particular location.

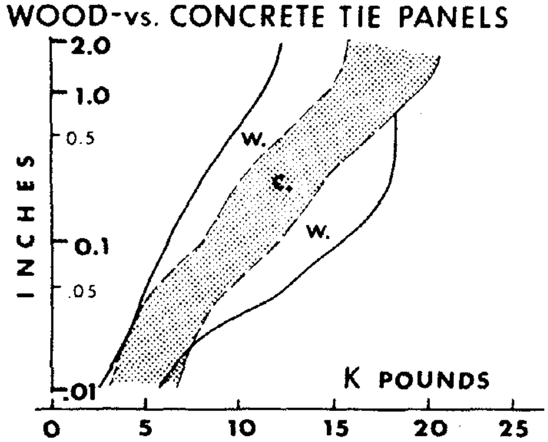

Fig. 20. The total range of force/displacement curves obtained at Sabot for wood (W) and concrete ties (C). - Results of the control wood panels are not included.

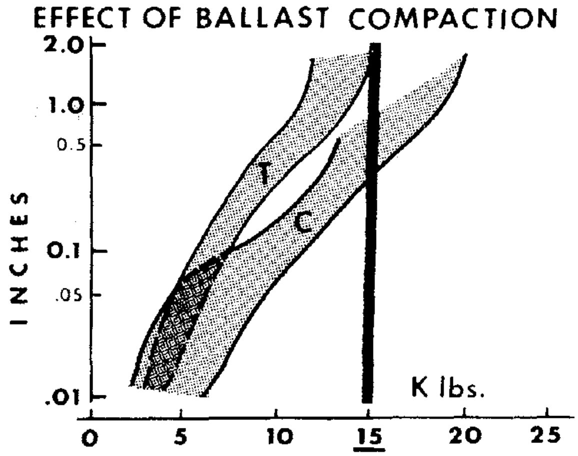

Fig. 22. Total range of force/displacement curves of panels tested immediately after tamping (T) and after tamping/compacting (C).

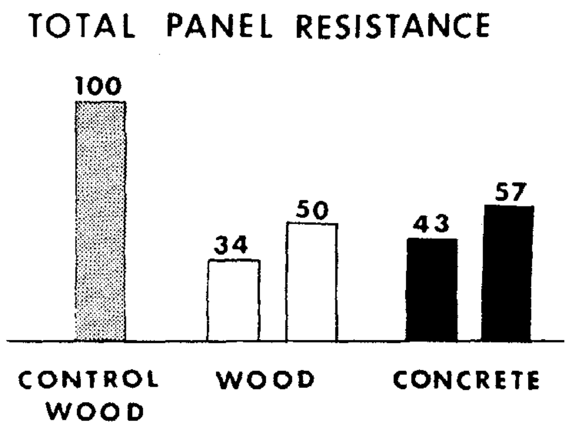

Fig. 21. Relative decrease of lateral track resistance caused by tamping on wood and concrete ties (lows and highs) as compared to the well settled control wood tie track.

Concrete Tie Track versus Woodtie Track Resistance. Concrete tie panels gave definite evidence of higher lateral resistance when measuring in three of the four track conditions prepared (Fig. 23). The results are given below.

Track Condition	Yield Force (lbs.) Wood	Concrete	Percent Change
1. Freshly tamped	13,000	15,000	+ 16%
2. Tamped and compacted	14,300	20,000	+ 40%
3. Tamped and trafficked	16,100	18,500	+ 15%
4. Tamped, compacted and trafficked	16,700	15,400	- 8%

Note: Average increase of resistance for concrete tie panels: 15.8%

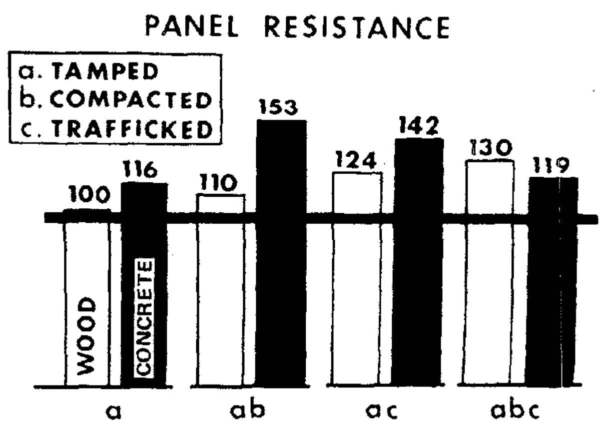

Fig. 23. The effect of various preparatory work (a, b and c) on the lateral resistance of track.

The measurements indicate that woodtie panels exhibit a steady increase of resistance, from the lowest value of freshly tamped condition to the highest value of compacted and trafficked condition. This is, in fact, what we can expect. On the other hand, the compacted and trafficked concrete tie panel has lower ultimate resistance than either the compacted only or the trafficked only. This drop in lateral resistance can perhaps be explained by the hot weather prior to the second phase of the test, which caused some loosening of ballast bond as the panel moved sideways under longitudinal compressive forces. Another possibility is that the panel moved somewhat faster than the other panels due to accelerated force application.

Differences were found in panel stiffnesses, too. It is interesting that two of the stiffest panels are woodtie panels. Accordingly, the stiffest woodtie panel has about one-third of the displacement of that of the stiffest concrete tie panel at a force level of 15,000 lbs. Another finding is that concrete tie track is weakened by tamping to a lesser extent than woodtie track.

Lateral Resistance under Various Conditions of Panel Preparation. The test verified earlier findings that there are substantial differences in lateral track resistance, not only as a function of track structure but also depending on the time and accumulated traffic since tamping operations. Raising and tamping temporarily reduce lateral track resistance. This weakening of track varies, from negligible to alarming rates. Traffic and time gradually restore lateral resistance. Recently constructed mechanical devices (ballast consolidators) are able to increase the ballast and track resistance immediately when they are used after tamping. One of these machines is now being tested for effectiveness in this country on various railroads. It was also used at our Sabot test for ballast consolidation in some track panels.

Track resistance values obtained as a function of panel preparation exhibited a wide range. In accordance with the measurements, the degree of weakening in lateral track resistance after tamping is very significant. It can be assumed that track in that state is susceptible to buckling in many cases. Ultimate lateral resistances (average for all panels) as a function of track condition are as follows:

Track Conditions	Yield Force (lbs.)	Relative Resistance
Settled (180 MGT)	35,000*	2.50
Trafficked (7 MGT)	17,000	1.23
Mechanically compacted	16,000	1.15
Freshly tamped	14,000	1.00

*Estimate

The much higher data of displacements recorded on the test panels also verify the substantial drop of track resistance after tamping and even long after that. Lateral displacement by track condition is indicated below:

Track Condition	Displacement (in.) at 12,000 lbs. Force	Relative Displacement
Settled (180 MGT)	0.05	1.0
Trafficked	0.13	2.6
Mechanically compacted	0.23	4.6
Freshly tamped	0.94	18.8

The application of ballast compaction is promising. Our measurements indicate it reduces the track displacement to one-fourth that obtained after tamping. In territories where the track is inherently unstable, we think that ballast compaction can prevent buckling. In making a decision for possible general application of ballast compactors, their trade-off economics should be investigated for various conditions. For such an analysis, of course, a wealth of data is needed.

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William W. Hay joined the faculty of the University of Illinois in 1947, after experience on U.S. railroads and as Chief Engineer of Korean Railways for the U.S. Military Government. Since 1956 he has also provided consulting services in railway engineering for U.S. state and Federal agencies; the Portugese, Rhodesian, and South African government railways; and iron ore railways in Canada. He has directed various research projects at the University of Illinois for U.S. railroads and is the author of numerous papers and articles, and two books, Railroad Engineering, and An Introduction to Transportation Engineering.

Hay received his B.S. and Mgt. E. degrees from the Carnegie Institute of Technology and M.S. and Ph.D. degrees from the University of Illinois. He has been a director of the American Railway Engineering Association and Roadmasters and Maintenance of Way Association and is a member of the American Society for Engineering Education, National Council of Engineer-Examiners, National Research Council, Illinois Professional Engineers Examining Board, and Tau Beta, Sigma Xi, and Chi Epsilon (scientific and engineering honorary fraternities).

TRACK STRUCTURES FOR HEAVY WHEEL LOADS

Back in the 1820s, a locomotive, the Stourbridge Lion, was imported from England and placed on the timber tracks of a gravity railroad, a forerunner of the Delaware and Hudson. The timber rails were too light for the wheel loads, and the Stourbridge Lion was placed in storage. A stronger design of track eventually permitted the use of locomotives and cars far heavier than the Stourbridge Lion. That incident marked the beginning of a long contest between track designers and car and locomotive builders. Each improvement in track gave the opportunity for bigger and heavier equipment. The 70-ton car created a set of problems in the 1920s not unlike those faced today with the 90- and 100-ton cars. Problems arising from the 70-ton car were overcome. Those of the 100-ton car are far from solution. Those unsolved problems can only be intensified with the advent of 125-ton cars.

Car builders and track designers have often gone their individual ways, but in recent years a recognition of the system nature of the track-train components and the dynamics involved have given hope that track and equipment designs can be more fully harmonized. It was just seven years ago that I delivered a paper to the forerunner of this conference, the Symington-Wayne Conference, when those were held in Depew, New York. That paper was entitled "The Track Structure as an Input to Car Design". Much has been accomplished in track/car analysis since that conference. Much remains to be done.

Recognition is being given to the pragmatic fact that track is a structure with a load-bearing capability dependent on the combined characteristics of foundation, superstructure, and loads to be carried. When load-carrying capability is exceeded, trouble is bound to ensue -- and has. There is ample evidence that the advent of wheel loads imposed by 90- and 100-ton cars has exceeded the load-bearing capacity of much of the track over which those operate. Improved track is needed to meet load requirements of 90- and 100-ton cars and, incidentally, of 3-axle power trucks under locomotives.

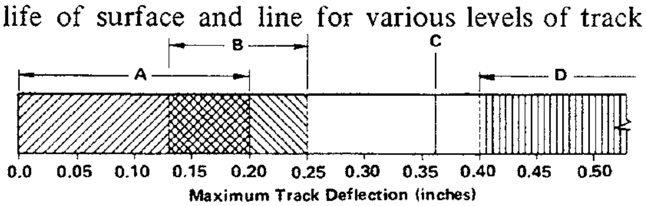
Track deterioration under heavy wheel loads appears in many forms -- loss of surface and line; conversion of subgrade and ballast sections into plastic masses that pump mud and water; wide gauge, plate cut, split, and spike-killed ties; rapid abrasive wear; battered rail ends; and the formation of corrugated and shelly rails, the last with the potential for detail fractures. This situation has not been helped by the extent of deferred maintenance on many miles of line.

Many improvements are undergoing discussion and some are even undergoing tests -- concrete ties, concrete sub-slabs, concrete pads; polymer additives, a better understanding of ballast characteristics, selection and use, and new means of stabilizing existing subgrades. It is hoped these and other possibilities will be fully developed and used. I tell my students that the best track has not yet been designed or built. Research and development will continue to give an improved track structure for the future.

But while there is hope for the future, one must be concerned with problems of the present. A realistic view of the situation should convince one that there will not be an early abandonment of thousands of miles of conventional track in favor of more exotic designs. There are no magic cure-alls in view. Improvements must be capable of incorporation into the existing track structure which will continue more or less in its present form many years to come.

Track deflection is a prime cause of track deterioration. Heavy wheel loads obviously intensify track deflection and the differential movement between track components that accelerates wear. Frequency of load application, i.e., rate of impulse, combines with deflection to hasten degradation. With a stiff track support, i.e., a higher modulus of track elasticity, not only is deflection reduced, but the individual wheel impulses can be merged to lessen their frequency. The two axles of a car truck, for example, may cause an effective single impulse because their deflection curves have merged. As with all structures, a first requisite of heavier loads is a stronger foundation.

J. R. Lundgren has prepared a diagram (Fig. 1) based on Talbot's studies that shows the effect on life of surface and line for various levels of track



Range	Track Behavior
A	Deflection range for track which will last indefinitely.
B	Normal maximum desirable deflection for heavy track to give requisite combination of flexibility and stiffness.
C	Limit of desirable deflection for track of light construction (≤ 100[lb]).
D	Weak or poorly maintained track which will deteriorate quickly.

Values of deflection are exclusive of any loosenness or play between rail and plate or plate and tie and represent deflections under load

Fig. 1. Track deflection criteria for durability.

deflection.[1] Zone A has an indefinitely long life with deflections ranging from 0.00 inches to 0.20 inches. Zone D, with deflections of 0.40 inches and over, will deteriorate very quickly. Most good track is probably in the high B range of deflections, 0.25 inches to 0.35 inches.

More can be accomplished by increasing the stiffness of support than by laying heavier rail. An increase in rail weight has a relatively insignificant effect on reducing deflections in contrast to reductions secured by increasing the modulus of track elasticity. (See Fig. 2.)

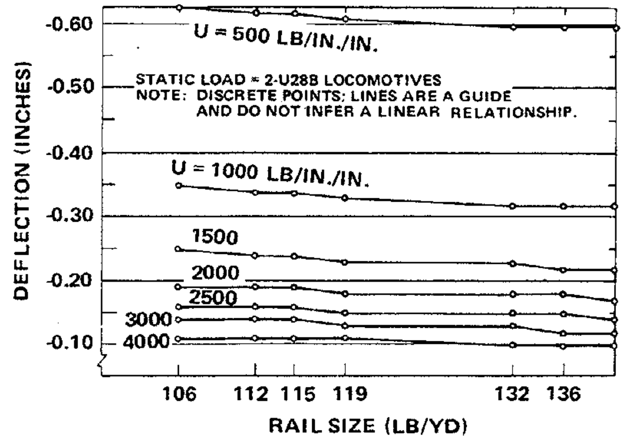


Fig. 2. Effect of rail size on deflection for various track moduli. (From M.S. Thesis by A. B. Bulter, Univ. of Illinois, "An Analysis of Bending Stresses and Deflections in Railroad Rails," 1960, Urbana, Illinois.)

Our first attention is directed therefore toward reducing track deflection by improving foundation support.

The very least one can offer toward improving track is to urge a return to good maintenance and engineering as applied to conventional track. A review of good practice offers the opportunity to present, as well, some possible improvements and new procedures and devices. Good engineering for any structure, no less the track, begins with a stable foundation; i.e., the subgrade - ballast system. For new construction this requires consideration of soil stability conditions during location; the application of soils engineering principles in selecting the subgrade soils and in the placing, compacting, moisture control, use of counterweights or buttresses; and correct design of fill widths and slopes to overcome adverse properties of soils when the choice of soils is limited. Compaction and moisture control are especially important in crowding the soil particles so closely together that excess moisture is squeezed out and the intimate contact of the particles leaves no room for moisture, while enhancing internal friction, cohesion, and shearing strength.

Soil stability is closely related to the absence of excess moisture. Good practice demands adequate drainage. Drainage requires more than the lip service it so frequently receives. Intercepting ditches are needed to prevent flow of water to the track structure. Track ditches and culverts must carry away water that does reach the track area. Water pockets in ballast and subgrade must be drained and subsurface flow and seepage intercepted and removed. All channels must be kept open and free-flowing. The intent of drainage, especially subdrainage, is to lower the water table and keep moisture away from those portions of the subgrade where the load distribution is maximum.

The ballast section cannot be considered apart from the subgrade. Experience, tests, and theory all indicate a concentration of tie load pressures immediately beneath the tie with pressure intensity decreasing as depth below the tie increases. Fig. 3 shows this relation.[2] A commonly accepted bearing capacity for subgrade soil is 20 psi.[3] If the subgrade soil has a bearing capacity of 20 psi then a depth of approximately 21 inches of ballast would be needed if 40 percent of the load is carried by one tie; over 36 inches if the entire axle load is carried by one tie. This latter situation can exist where adjacent ties do not give required

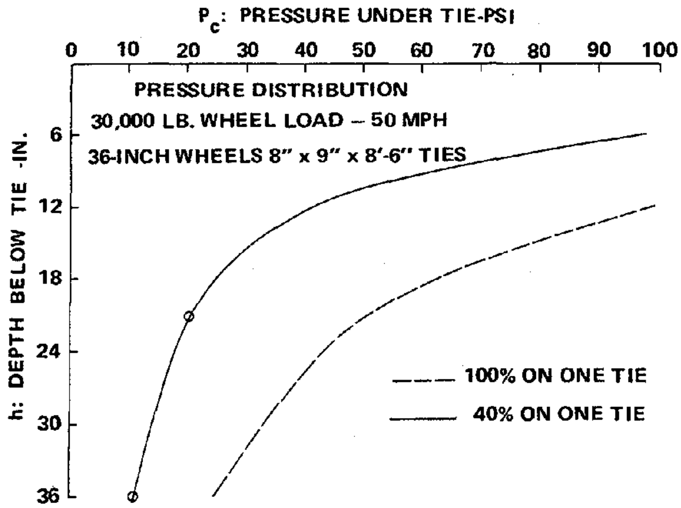


Fig. 3. Pressure distribution at various depths.

support. If, however, the subgrade has only 10 psi of bearing capacity, 37 inches of ballast will be required for a 40-percent load per tie, and considerably more than 36 inches for 100-percent load on one tie. Also note that no factor of safety has been allowed other than to account for the dynamic speed effect and to allow for the 36-inch wheel diameter. Failure to provide these depths will insure differential settlement and possible penetration of the ballast into the subgrade with loss of surface and line and, if the subgrade is of a clayey consistency and excess moisture is present, the formation of mud and pumping track.

Fig. 4 shows the effect of wheel loads on ballast depth requirements. Again, the requirement for a 30,000 lb. wheel load with soil-bearing capacities of 10 and 20 psi have been indicated and, in addition, the increased depth required for still heavier wheel loads and the lesser depths for lighter wheel loads. There is obvious need for an adequate depth of ballast, more than is generally in use.

Not all of the ballast depth need be of top grade material. Stability and anchorage of the ties demand the top 8 to 12 inches to be hard and tough with good weathering qualities, resistance to abrasion, with a high particle index and a good

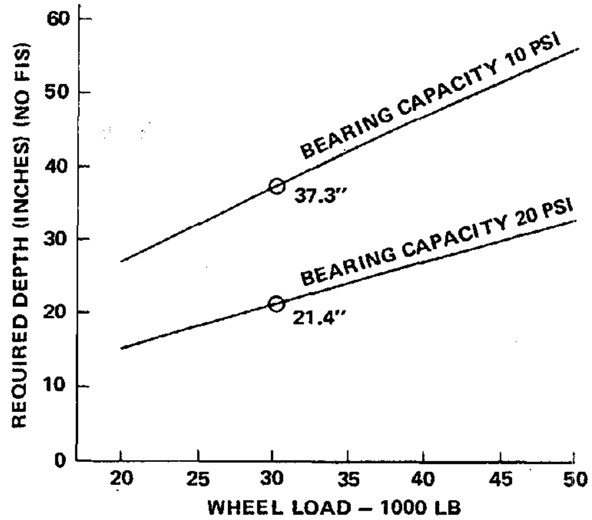


Fig. 4. Bearing capacity of soil vs. required depth of ballast.

distribution of grain size. Below this an additional 12 to 18 inches of low grade material should be used as a sub-ballast. A report on ballast research presently being conducted at the University of Illinois Urbana Campus should be available toward the end of 1976 with specific evaluations of the relative stabilities and characteristics of the more common ballast types.[4]

Mere depth of ballast is not always a complete solution, especially when the top of subgrade is composed of fine-grained soils with a high plasticity index. The area of contact between subgrade and ballast is often critical to stability. Fine-grained soils, even when containing a small amount of moisture, acquire a soft consistency as moisture is brought to the surface through capillarity and the pumping action of passing wheel loads. A slurry layer may be formed, no more than an inch or less in thickness. That softened layer can infiltrate and foul the ballast and permits ballast particles to penetrate into the subgrade to form water-laden ballast pockets with an attendant loss of track surface, line, and life.

Several means have been devised to prevent ballast and subgrade intermingling and the upward percolation of water into the ballast to form ice lenses and frost heaving in cold weather. A filter blanket of carefully graded material 8 to 10 inches thick can be placed between the subgrade and the ballast. Such a blanket may also serve as a sub-ballast. In general, the filter should have a wide distribution of grain sizes, small enough on the one hand to prevent fine-grained soils from entering the ballast section, coarse enough to maintain drainage and to prevent ballast materials from penetrating the fine-grained subgrade. Such filter materials should also be placed around subdrains to prevent clogging by infiltration of soil particles. The design of filters is given in standard works on soils engineering and in Chapter 1 in the AREA Manual.

Fig. 5 shows the required grain size distribution for one commonly accepted design of filter.

$\frac{D15_F}{D15_S} > 5$ For adequate drainage, 15% size of filter should be 5 times larger than 15% size of subgrade soil

$\frac{D15_F}{D85_S} < 5$ To prevent infiltration, 15% size of filter should not be more than 5 times the 85% size of subgrade soil

$\frac{D50_F}{D50_S} \gg 25$ 50% size of filter should not exceed 25 times larger than 50% size of subgrade soil

Uniformity coefficient of filter should be no more than 20.

Fig. 5. Filter design.

An alternative treatment is to blend hydrated lime into the upper 8 to 10 inches of the subgrade soil through a process of spreading the lime on top of the subgrade and blending it into the subgrade soil by harrowing, disking, watering, and compacting. Lime is added at a rate of 3 to 8 percent by weight. The lime performs a dehydrating action, reduces soil plasticity, and increases soil density. It will not perform well in soils with a high organic content. Cement can be used in the same manner to form a soil cement topping, using 3 to 16 percent of cement by weight in the upper 8 to 10 inches. In addition to dehydrating and reducing plasticity, some minor mechanical strength may also be gained.[5]

A recently proposed alternative treatment for the top of a subgrade is to cover it with sheets of a celanese-polypropylene material that prevents infiltration of the ballast by fine-grained subgrade particles and by moisture from capillary action. It also prevents intrusion of ballast particles into the subgrade soils. The successful use of these materials in highway work offers promise for its successful use on railroad subgrades. It has reduced the need for aggregates on haul roads by as much as 30 percent. The multiple-ply sheets have sufficient porosity to permit ballast drainage. They resist ballast particle damage by having a high-stretch capability of as much as 200 percent but are not a substitute for adequate ballast depth.

There are, of course, more radical alternatives for track support. Several have been undergoing test by the FRA and the Santa Fe in the Kansas Test Track. These include thin slabs similar to concrete highway pavements to which the rails are attached. The slabs give a more even and reduced distribution of load over the subgrade. Concrete slabs are also undergoing test by the British Railways. An alternative to the thin slab is a thick pad, beams about a foot wide and two feet deep (±), set longitudinally in the subgrade under each rail to which the rails are attached. Concrete ties and ballast coated with bituminous materials and metadiene-Styrene polymers were also included in the Kansas tests. Results from these tests are not yet fully established so one must wait for a practical evaluation.[6] Success with either slabs or pads would lead to a virtual elimination of the ballast section as we now know it. Suitable fastenings for attaching rails to the concrete seems to be as difficult a problem to solve as that of support.

These measures are primarily applicable to new construction. For most railroads the problem is one of strengthening and stabilizing an existing segment of subgrade. An initial step, too often overlooked, is the making of a sub-surface survey. The "obvious" cause of instability is not always the prime cause. The survey can include a review of construction and maintenance records, a review of local geology, or the experience of other railways and highways in the same locality. Probably much reliance should be placed on sub-surface cross sections obtained by borings made with an earth auger, 2 inches or larger. Cross sections taken every 50 to 100 feet will give an indication of the types of soils and their position and of the height of water table.

A second step is to correct any adverse drainage conditions. No subgrade can withstand heavy wheel loads if it is in a saturated condition. In addition to an adequate system of clean, free-flowing ditches and culverts, deep ditching, 6 to 8 ft. or more in depth, can be useful in lowering water tables of subgrades, especially in flat, clayey territory. Subdrains are used to drain ballast and water pockets, to intercept and carry away subsurface seepage and flow, and to drain wide, flat areas such as yards.

The injection of cement grout is an old standby that has given proven economy by reducing excess maintenance on unstable track segments. Mixes have varied from equal parts of sand and cement to one part of cement to 16 parts or more of sand. The grout carries sand into the fill to increase internal friction, seals off cracks and underground seepage, provides a certain amount of compaction, performs a dehydrating action, and even a small amount of mechanical support for load distribution. The effectiveness of grouting can be limited in fine-grained soils where it is needed most, because the spaces between clayey soil particles may not be large enough to permit the flow of sand in the slurry. Grout may have to be forced into the soil by a process of hydrostatic cracking that can render the soil less stable than before.

Hydrated lime has been found helpful in some fine-grained soil situations. The soil type and, since an ion exchange is involved, the pH value are related to its success or failure. Soils with a high organic content do not respond well to lime treatment. Lime can be introduced into an existing subgrade through slurry injection, by pouring into drilled holes, or by placing in trenches. Lime tends to reduce soil plasticity and increases workability. Lime has also reduced expansion and contraction of swelling soils. Lime treatment helps to keep moisture from reaching untreated subsoils. Strengths obtained with lime-treated soils have been highly variable. High values have been obtained, but are dependent on time and temperatures. Temperatures must be 40 degrees, preferably around 70 degrees F or more, and as much as 30 days may be required to gain full strength. As with cement grouting in fine-grained soils, the grout flows through and seals cracks and seams; injection may be accomplished through hydrocracking.[7]

Mechanical support through piles and poles driven alongside the track may be of some benefit but only if 60 percent or more of the pile or pole penetrates into firm, stable material.

If the unstable segment is not too extensive, the weak material can be removed to a depth of two to six feet or more and backfilled with select material. This of course requires putting the track out of service for a few hours or a few days. The ultimate solution may be relocation around a swamp, a patch of muskeg, an old lake bed, or other highly unstable ground.

Ballast selection and use must be directed toward achieving vertical, longitudinal, and lateral stability – in addition to drainability. While the filter blanket or lower grade of ballast will serve as subballast, the top 10 to 12 inches must be selected with care from the best materials. It is interesting to note that ballast specification tests used in the United States say very little about stability, only durability.

Stability may be attained and measured in several ways. Stability is related to the shearing strength of the material, its internal friction. The Particle Index that reflects shape, sharpness, and surface texture, in combination with hardness tests to prevent abrasion and crumbling of sharp edges, can be useful in identifying stable materials. The higher the P.I., say in the 16 to 20 range (with zero representing maximum instability), the more stable the material.[7] The ASTM has standard tests for the Particle Index for aggregates that can be adapted to ballast use. Low P.I.s are characterized by rounded, smooth particles while rough, sharp and somewhat elongated particles have high Particle Index values.[8] The Particle Index is not yet a part of ballast specifications. It can be combined with a flakiness index to reduce the breaking tendencies of long, slender particles. In general, particles with least dimensions less than 60 percent of mean size should not exceed 30 percent of the total. Ballast must be hard enough to resist shatter and abrasion and have durability to resist breakage from freeze-thaw cycles. Soft limestones that powder from abrasion and have a cementing action in the ballast should be avoided. Smaller sizes compact more readily. Larger sizes contribute to stability through their mass. A very significant factor in stability is the spread of gradation in the ballast. It should be well graded, i.e., with a wide distribution of grain sizes from fines to coarse particles. The fine particles perform a bedding and interlocking function between the larger particles. A good measure of such distribution is the void ratio; a low ratio is preferable. Stress level is of particular significance with heavy wheel loads. Some ballasts may perform well at low stress levels, pit run gravel for example, but may not do well at high stress levels. A probable listing in order of stability and resistance to heavy wheel loads would be copper, zinc, and steel mill slag followed by crushed granite, basalt, quartzite, and hard limestone, when these are well graded, have a high particle index, and a low void ratio.

Lateral and longitudinal stability follow about the same order. In addition to adequate ballast depth, cribs must be full with substantial ballast shoulders at the ends of ties. Compacting the shoulder of newly placed ballast is said to hasten consolidation and promote stability. The value of this procedure is still under debate with some railroads finding traffic compaction at moderate speed to be equally satisfactory. The AREA has listed 300 lbs. as the lateral restraining force per tie.[9] Recent tests made at the University of Illinois (Urban Campus) indicated an average force of 300 lb. ± to move an unloaded tie detached from the track. When the tie is attached to the rails with adjacent ties also attached, the force necessary to move the three ties increased approximately to 15 kips under 20 kips of vertical load and with zero inches of shoulder. With a 12-inch shoulder the lateral force required for movement was approximately 1 kip more.

The load-carrying qualities of ballast can be enhanced by coating the particles with a bituminous oil or a polymer substance. Tests made in the AAR Research Center on ballast coated with a butadiene-styrene block polymer indicated that permanent settlement was four times greater for untreated ballast than for the treated ballast.[10] Full-scale tests on these materials were conducted at the FRA-Santa Fe Kansas Test Track.

All of the foregoing is of little help unless one starts with a clean, dry ballast. This may require sledding or otherwise undercutting and removing the wet and mud-encrusted ballast already in place, to a depth of 4 to 8 inches or more below the bottom of tie, and replacing it with a clean, free-draining material.

Any discussion of stability should include the need for special attention to the support given to turnouts and railroad grade crossings. A deep, firm ballast bed with a functioning drainage system represents the acceptable minimum. Use of a stabilizing substance in the ballast material would seem warranted for heavy wheel loads.

Significance of the L/V ratio, i.e., the ratio of the lateral component of the wheel load to the vertical component, is becoming well known. An L/V ratio of 0.64 is said to be capable of overturning an unrestrained rail while one of 0.78 presents the danger of a wheel flange climbing a worn rail.[11] It is here suggested that one could turn the L/V ratio around for use as a standard of lateral stability for track. An L/V ratio of 0.40 would indicate that the track could restrain a lateral thrust with a value that is 40 percent of the vertical load. Conversely, an L/V ratio, based on wheel loading, greater than 0.40 would cause a lateral displacement. The French SNCF have used an L/V ratio of 0.70 as a standard for track having main line service capability.[12] The ratio could probably be adapted to United States track and wheel loads without too much difficulty to give a measure of the lateral strength required by track to withstand modern heavy wheel loads.

Ties perform an important load-distributing function. The longer and wider the tie and the closer the spacing the better the load distribution. Heavy wheel loads should be carried on 8" x 9" x 9-0" ties spaced 24 per 39 ft. rail. Closer spacing presents tamping problems. Wider ties, while presenting some tamping difficulties, would be recommended if these were economically available. Tie timber wider than 9 ins. is expensive and difficult to obtain. Laminated construction, whereby two or more small timbers are glued together, may be helpful. Prototypes of these ties are undergoing test. Ties composed of ground-up wood from old ties mixed with a binder, pressure molded and baked, can be cast to any desired width of base, but such new designs are unproven in service. Concrete ties have a 12-inch base, but designs previously in use have not been entirely satisfactory, especially if placed on 30-inch centers recommended for economic parity with wood ties. The new specifications developed by AAR and AREA hold promise for more success, but extensive experience with the new designs has not been fully accumulated.[13]

Where wood ties are used, the wood fibers must be protected by chemical treatment against decay and insect attack. Protection against crushing and abrasion is obtained by use of plates large enough to distribute the rail load. A minimum of 14-inches in length is recommended for 100-ton cars or greater, both on curves and tangents. For 6-degree curves or greater, an 18-inch plate has proven useful. Not only does the larger plate distribute the load, but friction between plate and tie helps in resisting lateral thrust and gauge widening. Double spiking on high-degree curves is also useful in resisting lateral forces. A 16-inch plate is now available on the market having a 1 in 30 cant. In addition to providing a greater bearing area, the higher cant is intended to improve wheel/rail relationships and reduce the incidence of shelling. Heavy locomotives on 3-axle power trucks, as well as big cars, contribute to wide gauge. Frequent respiking of wide gauge greatly reduces tie life and the ability to resist lateral thrust.

A smoother car movement with less thrust and impact is possible through revisions in track geometry. Longer spirals will reduce the lateral impact upon entering or leaving full curvature. The amount of unbalanced superelevation permitted can also be reduced. In fact, with heavy cars equilibrium superelevation should be the starting point from which adjustments can be made if needed, following field observations of the rail wear pattern.

There should be no need to suggest that track must be maintained to a high standard of excellence by a continuing policy of programmed and routine maintenance effort. The quality of work performed by maintenance forces should be fully as important as quantity. The present state of much United States track indicates there is a need to give additional emphasis to the necessity for adequate levels of maintenance. The development and widespread use of track inspection cars can be a great help in maintenance programming and in quality control.

The effect of heavy wheel loads is most often visible in its effect on rails. Battered rail ends, bolt-hole breaks and broken joint bars, head checking, spalling, shelling, corrugating, horizontal and vertical split heads, piped rails, and detail fractures are related in part to the incidence of heavy wheel loads through impact and contact stress effects. Heavy loads also accelerate abrasive wear of the railhead. The use of continuous welded rail reduces joint maintenance in all its forms, including the effects of rail-end batter, bolt-hole breaks, and broken joint bars. CWR also reduces the rock-and-roll derailing tendencies of high center of gravity cars that arise from the periodic encounter with low joints spaced approximately 19-1/2 to 39 ft. apart.

The problem of rail breakage most often arises on branch lines laid with light rails. Fig. 6 shows the effects of rail weight on bending stresses in the rail. At a speed of 50 mph and a track support modulus of 2,500 lbs. per inch, all rails in common use are within an allowable bending stress of 32,000 psi, but when the modulus is reduced to 1,000 and speed to 30 mph (a frequent branch line condition), the stresses developed in 75-lb. rail greatly exceed the allowable stress. The 90- and 110-lb. rails are not far below the allowable.

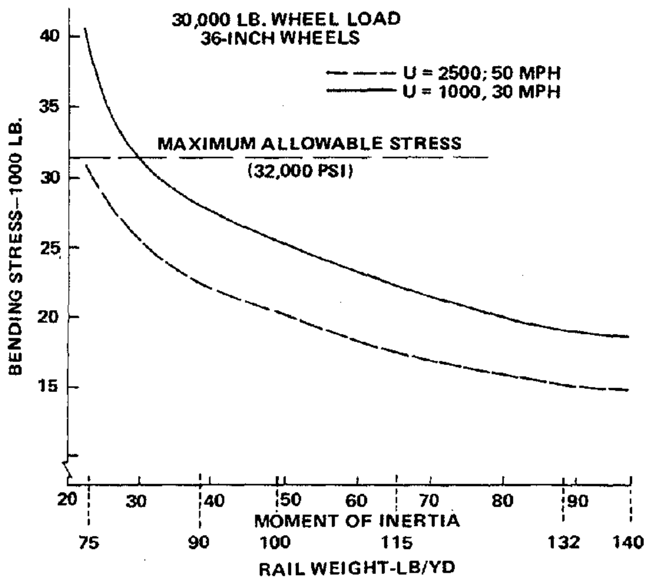


Fig. 6. Bending stress vs. moment of inertia (and weight of rail).

With a 90-lb. rail, the allowable dynamic load becomes 52,000 lbs. at 45 mph, with an equivalent static load of 37,716 lbs. Both of these are within safe limits but are based on a modulus of track elasticity of 2,500 lb./in./in. If the modulus is reduced to 1000 lb./in./in. (not uncommon on either branch or main lines), the allowable dynamic load on 90-lb. rail becomes 41,996 lbs. and the static loading only 30,000, a strictly marginal situation. Worse conditions can be anticipated when rail experiences abrasive wear (loss of section modulus) or is lighter than 90 lbs. A support modulus less than 1,000 can place new 100- to 110-lb. rails in a marginal situation.

Fig. 7-10 show the increase in broken rails on two branch lines laid with 90- and 110-lb. rail following the introduction of 100-ton cars. Broken joint bars prove to be correspondingly numerous. The conclusions are inescapable. Where heavy wheel loads are in use, rail should be 115 lbs. or heavier in order to withstand bending stresses, and the modulus of track support should be in the 2000 lb./in./in. range or higher.

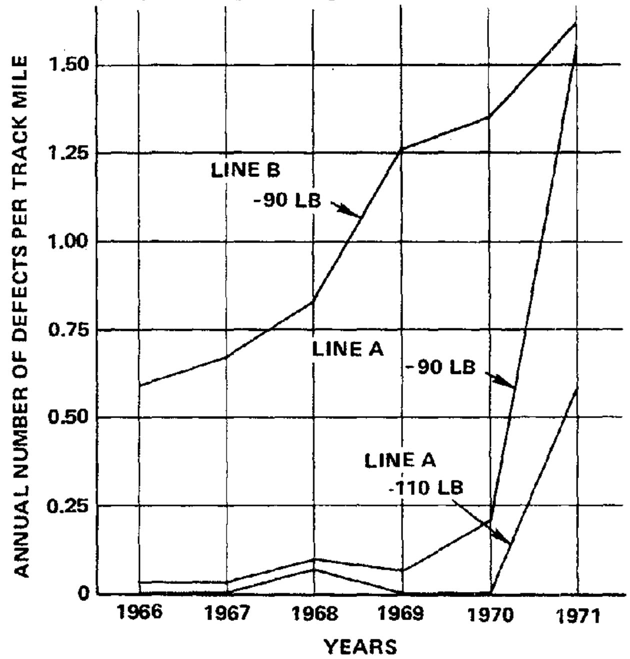


Fig. 7. Annual number of bolt-hole breaks per track mile vs. years.

Heavier rail is not a solution to problems of contact stresses, those created directly beneath the point of wheel load application. Here the problem is one of shearing and of rail steel quality. The literature, theory, and experience give ample evidence that heavy wheel loads lead to contact-stress-related defects – head checking, spalling, shelling, a hazardous group that can develop into detail fractures. Horizontal fissures and railhead mashing also occur. Corrugated rail is related to contact stresses, as are battered rail ends. The plain truth is that wheel loads of 30,000 lbs. or more on 36-inch. wheels are overstressing the rail in shear based on an allowable value of 50,000 psi.

There is good evidence that the uniformity of unit-train consist, combined with heavy wheel loads and lack of lateral play in truck and roller bearing design, contributes to the development of corrugations and shells. There is little doubt that heavy wheel loads cause contact stress defects to grow and develop, but there is still considerable

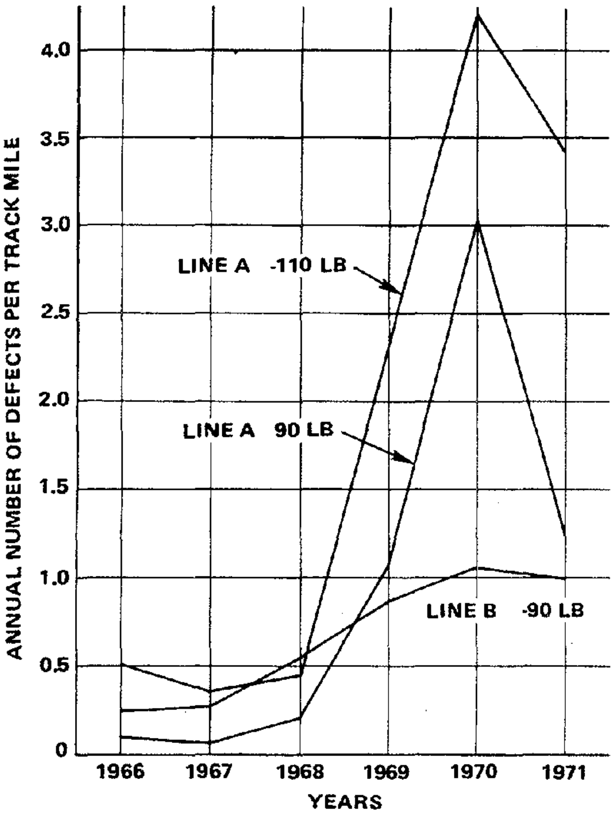


Fig. 8. Annual number of vertical split heads per track mile vs. years.

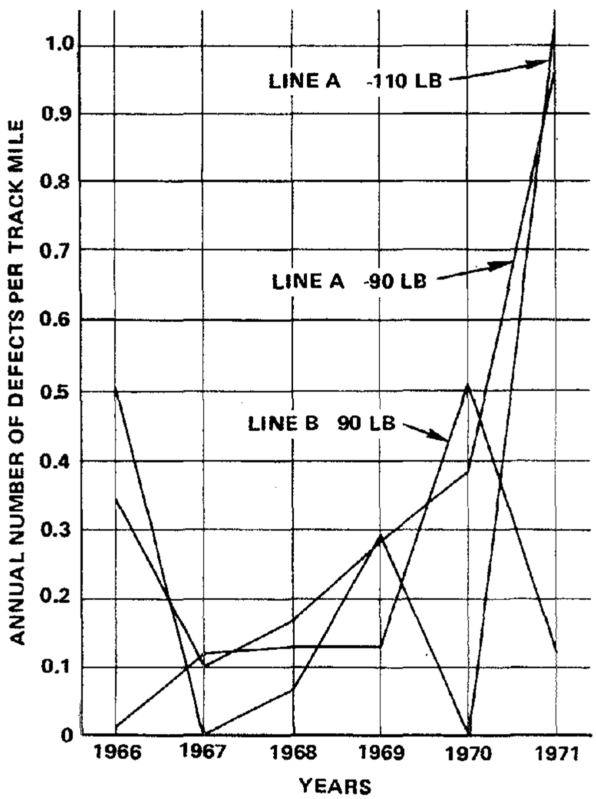


Fig. 9. Annual number of horizontal split heads per track mile vs. years.

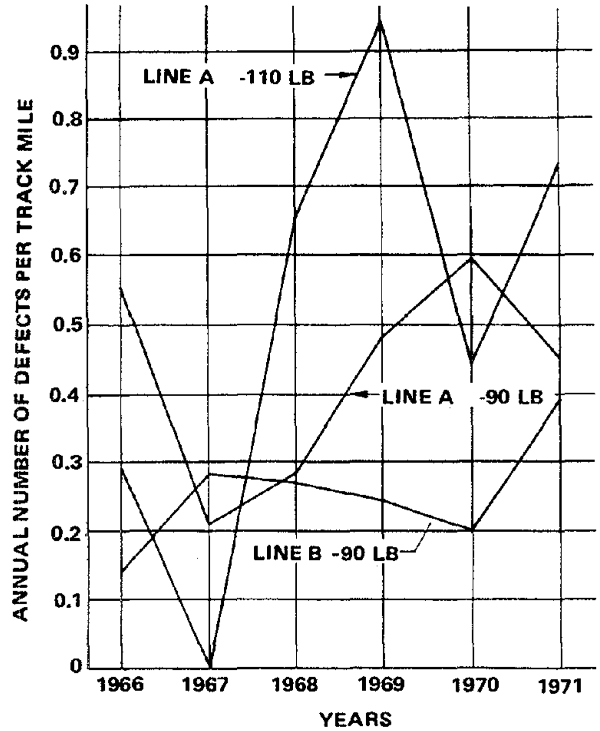


Fig. 10. Annual number of transverse defects per track mile vs. years.

mystery as to what initiates those defects in the first place. Perhaps a higher standard of purity for rail steel is needed. The growth of shelly defects can be accounted for by the accumulation of residual stresses in the gauge corner of the railhead, especially with the repetitive impact of each succeeding wheel in a long unit train. Poor track geometry permits each car to impact at a given track irregularity in exactly the same manner as the preceding car, thereby developing and accumulating high residual stresses.

The number of possibilities for corrective action are limited. Of first importance is maintaining a high level of excellence in track geometry – precision maintenance, if you will. For corrugations, the only presently known track corrective is railhead surface grinding, which removes the crests of the corrugations and retards but does not eliminate future development. Grinding is not a cure.

There is also need for a cleaner, tougher steel. The advent of vacuum-degassed rail steel may be one answer. Rail rolled with a minimum of entrapped gas has a much lower potential for the formation of hazardous defects, especially those leading to transverse failures. At the rail mill, efforts to reduce inclusions and blow holes from hot torn steel should continue. More rigid mill tolerances and the rolling of straighter rails which do not require "gag" straightening have been suggested to reduce overstressing or nicking that could lead to an intrack failure. Rails of special metallurgy have a longer life because they resist abrasion. Rails with a high silicon content or heat treated by full-section treatment or by electric induction head hardening show longer lives than plain rail. The use of jointless rails, whether by rolling degassed metal or by use of CWR, will help to reduce joint failures. One could suggest also a radical change in the rail/wheel contact geometry to give a better load distribution on the railhead. Two things work against such a solution: (1) The impossible economics of trying to change the technology of an entire industry through introducing a noncompatible element, and (2) the probability that such a radical change in design would create an entirely new and equally perplexing set of problems.

In summary, it is my opinion that any radical changes in the track structure are still far off. One must depend therefore, in adapting the present track structure to heavy wheel loads by taking advantage of a few new improvements and concentrating on excellence in maintenance. These two approaches will include:

1. Adherence to good standards of maintenance.
2. Use of a depth of ballast consistent with the bearing capacity of the subgrade soil.
3. Concentration on good drainage and keeping subgrade materials dry.
4. Giving close attention to the subgrade/ballast interface through use of lime, cement, or membrane separation and stabilization or by placing a filter layer between subgrade and ballast.
5. Selecting stable ballast materials through use of the Particle Index and grain-size distribution criteria, i.e., low void ratios.
6. Use of crossties with wider bases.
7. Use of rails consistent with the loads and speeds, preferably 132 lb. or greater, on all lines carrying heavy cars.
8. Improvement of rail steel quality and purity.
9. Use of an L/V ratio as a standard for lateral track strength.
10. Widespread use of track inspection cars for maintenance programming and quality control.
11. Use of large spirals and equilibrium superelevation.

Two other lines of action are open. The first is to limit the wheel load, either by building smaller cars or by reducing the loads placed in modern 100-tonners. The second is simply to recognize that wheel loads of 30,000 lbs. or greater are at the limit of current technology, that such loads destroy track, and that the economics of such cars must include the costs of restoring track as it wears from the passage of cars. It is a problem in comparative economics.

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Discussion Leader
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Mr. Brown received his engineering education at the University of Utah. He joined the Union Pacific Engineering Department in 1941 as a Draftsman and advanced through the organization serving as Assistant Engineer, Resident Engineer, Division Engineer and District Engineer at various points on the system. He was appointed to his present position of Chief Engineer in Omaha in 1965. Mr. Brown is past president of the American Railway Engineering Association and past chairman of the Engineering Division of the American Association of Railroads.

You see standing before you now a tired old chief engineer who has learned all of the hazards the papers have identified this morning the hard way and who is seeking frantically to do something about them. As Stan Crane has mentioned, our problems began when we started using the larger cars, particularly the 100-ton cars. Then there are a few of us who went further than that--we established tariffs for 125-ton cars and are now running a lot of those. The cars in themselves, however, did not create all of our problems; there are some of us who have also increased the operating speeds with these heavier wheel loads. If you think you have trouble with the size of the cars themselves, wait until you start running them at faster speeds.

I have 4,000 miles of main line track where we are running these 100-ton and 125-ton cars in freight trains at 70-mph speeds, and we are respacing the signals over a 300-mi. stretch of main line to run 80-mph with them. So, we have our problems--and in multiples. We started to develop these problems as the speeds were increased with the heavier cars. The first thing we experienced was a gauge problem, and when the Track/Train Dynamics Program ran the tests on our property that Howard Meacham identified, we threw in some of our ideas which included widening the gauge, putting a line kink in the track, putting the track out of cross level, and then running trains over the test section at speeds from 50 to 90 mph to measure the resultant forces upon the track structure.

We also took the opportunity to test some of our other theories on the effects of heavy wheel loads on rail in particular while the Track/Train Dynamics tests were being conducted on our property in Idaho. We felt that the 8" x 14" tie plates we were using in our high speed main line track were too small for modern high speed, heavy wheel load traffic. We also felt that the 1 in. 40 cant we have all been using in tie plates for many years was too flat for the best distribution of the wheel load upon the road, taking into consideration the average worn contour of the wheels on the locomotives and cars. We tested a number of tie plates of various sizes with 1 in. 40, 1 in. 30, 1 in. 20 and 1 in. 14 cant and have since adopted an 8-1/2" x 16" tie plate with 1 in. 30 cant that we are now installing in all of our high speed, heavy tonnage main lines. This tie plate is giving a much better distribution of the wheel load on the head of the rail, and appears to reduce the heavy contact on the gage corner of the rail head that contributes so much to the rail shelling problem.

Thus we have identified many of our problems, and now we are searching for the answers. We certainly think the high-speed test track that is proposed for the Pueblo test site will contribute a great deal in providing these answers. All of us in the industry have been grasping for solutions. When we think we have the answer to any of our problems, we try them out in the track and then find it necessary to wait for many years to determine if we were right or wrong. The test track, as we see it, will be able to put a rapid accumulation of tonnage over the track at various speeds and make it possible to identify and solve many of our problems in a couple of years that would normally require ten years to identify and solve in the real-world of railroading out on our main line tracks.

With that introduction to my own feeling about our problems, I invite the audience to address their questions to the speakers.

COMMENTS/DISCUSSION PERIOD

Delegate Comment: In view of the length of cars and the speeds involved, do you feel that a 39-ft. track section is of adequate length ot provide a fair test? I refer now to a case, for example, where the train might be moving from left to right; the leftmost section might induce a stimulus that the righthand section therefore must accept. It seems to me that this would be something of an unfair test, and one needs, in fact, longer track sections and something to take into account the wave length, if you will, of the rock-and-roll action.

Panel Response: The Sabot lateral load test is part of a test series sponsored by FRA and performed by a number of railroads at various locations. Our results are applicable only for vertically unloaded track.

Delegate Comment: I would like to know if you still consider the possibility of installing additional concrete ties on the Chessie research project? Or do you think, in fact, that you have arrived at a tie you can live with cost-wise, engineering-wise, performance-wise?

Panel Response: The possibility of installing additional concrete tie test tracks exist. However, we do not have, at the present time, any plan to do so. The performance of the concrete ties tested till now, in our opinion is not satisfactory. The price of concrete tie is usually the result of negotiation between the railroad and the tie producer, depending largely on the quantities to be purchased. It is another question, of course, how much can be paid for concrete ties economically under certain conditions. For example, our cost model told us that in medium-heavy trafficked main line track, the out of face replacement of existing wood ties with concrete ties spaced 25 inches, is justifiable if the price of concrete tie is not more than 15 to 18% higher than the price of wood tie. For newly constructed track this justifiable premium could be as high as 25 to 28%. Still higher prices can be paid for concrete ties with the adoption of new track laying methods (constructing and installing long prefabricated track panels) which offer labor savings.

Delegate Comment: Do you think any test section of concrete ties less than two miles in length, with all the hazards of curves and grades in addition to other problems such as bridges throughout the test area is adequate? Do you think you can really prove anything with less than two miles of actual in-service testing?

Panel Response: I am convinced that short test sections can do the job. In fact, under controlled conditions, one may obtain more information from several short sections where the various sections have different operating characteristics, than from one long section with the same condition along its entire length.

Delegate Comment: Do you presently have a concrete tie produced and located on your property? How do they produce the tie, by machine methods?

Panel Response: All of our test ties were machine-produced.

Delegate Comment: Then you are acquiring a tie that's manufactured by a very inefficient method. Is that correct? I mean, if a producer could produce the tie with a highly efficient machine process and supply it to you at a third of the cost, the tie would be more attractive to you wouldn't it?

Panel Response: Definitely so, providing the ties meet specifications. Assuming that a railroad makes commitments to purchase sufficiently large quantities of concrete ties over a period of time then, the tie producer can lower his selling price. This is because, in this case, the tie producer would be able to procure more sophisticated equipment, thus increase productivity and lower his own costs.

Delegate Comment: I notice the mud under the bottom of the French two-block tie in the slides. This indicates to me that it's a very bad installation ballast--wise.

Panel Response: No, there was a good sub-ballast and ballast preparation along the Noble Test track. The unusually severe deterioration of ballast and the very large amount of mud penetration into the ballast have been caused by the insufficient physical dimensions of the French ties.

Delegate Comment: Have you ever tested any tie on 24- or 22-in. centers with a ballast section that went down to a minimum of 8 in. below the base of the tie?

Panel Response: No. The smallest crosstie spacing at which we installed concrete ties was 25 inches, and the minimum depth of ballast under the tie was about 12 inches.

Delegate Comment: Let me invite you to Florida to look at 165 miles of concrete track which is performing perfectly. We have had a massive research project down there and we figure we have all the problems licked. We are not having any problems with 100-ton cars.

Panel Response: Thank you very much.

Delegate Comment: I would like to comment on the length of the panel in connection with the concrete panel testing. It appears that if one panel consists of various components, there is a bending in the track as well as a lateral resistance of the ballast. If one chooses too long of a section, like 40 ft. or longer, then one gets bending in the panel, and therefore it doesn't give quite the same resistance to the gravel. The practice on the Continent is to take about half of the length used and to distribute the load in such a way that there will be no bending of the panel, but it will move like a rigid body. Then you can really eliminate the effect of the bending, and what you measure then is just the lateral distance the ballast exerts against the rail tie structure. Therefore, although the rail section should be no longer than two miles, it appears to me that it should be possibly even shorter than the one used--not one or two ties, because then the random variation among the various ties is very large and it scatters, but about 15 ties or something like this may be sufficient. At least this is what the Europeans have relayed.

Panel Response: I agree, that the length of the panel, when applying concentrated lateral load to the panel does have influence on the results. Pulling very short panels or individual crossties uncoupled them from the rails results in lower values of lateral resistance than pulling longer panels with the rails coupled to the ties. However, our objective was to determine the total resistance of track, which is the sum of the ballast resistance and the bending resistance of the railtie assembly. We think, that total resistance is a more representative datum than ballast resistance only, because in actual service, the track is always subject to lateral bending. Nevertheless, the various components of lateral track resistance - if needed to be separately known - can be determined. One way would be to isolate these components by special test procedures and measure one component at a time. Another way is to measure total track resistance under different conditions and separate the components (ballast resistance panel bending resistance, etc.) applying statistical techniques.

Delegate Comment: Would you state an opinion on the effect of introducing on a short test sample, a few miles perhaps, the plate with a different cant? When the rail on which most wheels receive surface wear is perhaps based on a 1/40 template, locally you will redistribute the contact line using the 1 in 30 cant. But don't you think the railhead will flow or wear to readjust its contour to that suitable to the predominant worn profile as the wheels come over it, and therefore, with service wear, the contact point will reappear along one gauge corner?

Panel Response: That is possible, but I think that probably the period in which that will take place will be longer than it would be with the initial 1 in 40. In my opinion, at least, any cant that you settle on is going to be something of a compromise. Ideally, perhaps, we ought to have a plate with a different cant for each degree of curve. This, of course, would be ridiculous, and so we take something that serves as a compromise and does give us a better area of contact between the wheel and the rail.

Delegate Comment: I might add to that. We have taken some of the Canadian Pacific 1 in 20 cants and some of the 1 in 30s we are using and put them under old rail in Idaho that had been traversed, oh, probably 350 million times on 1 in 40 cant. Those rails, since we put those plates in, now have about 56 million tons over them, and they are showing the new contacts there were on the field side of the head, but we are not yet getting any gauge corner contact on them.

Delegate Comment: Does the increased efficiency of operation on your high-speed track or high-speed runs really justify the increased track and equipment maintenance?

Panel Response; We think it does. Of course, we are a long-haul railroad generally--we haul commodities 2,000 miles--and it's a competitive situation between us and trucks, airlines, or anything else. We're developing these tremendous coal deposits in Wyoming, and on the coal hauls we have as long as 600-mi. unit-train operations. We started out running these coal trains at 40 mph, and now we're trying them at 55 mph. This way you have lower car requirements; you are moving the stuff through your mechanical pipeline faster so you require fewer cars in the pool. Generally speaking, we think it pays off and we think all the railroads in the country, in the next few years, after we get over this current bottoming out in the railroad situation, will be running faster.

Mike Rougas
Chief Engineer
Bessemer and Lake Erie Railroad Company

Mike Rougas has worked in railroading for over 20 years, starting with the Union Railroad in Pittsburgh in 1953. In 1964 he became Engineer--Track of the Bessemer and Lake Erie Railroad Company, and in 1967 Assistant Chief Engineer of the Elgin, Joliet and Eastern Railway Company. He returned to the Bessemer and Lake Erie in 1970 as Chief Engineer, the position he now holds.

Rougas received the B.S. degree in Civil Engineering from the University of Pittsburgh, M.S. degree in Civil Engineering from Carnegie-Mellon University and M.B.A. degree from Duquesne University. He is a Registered Professional Engineer in several states and a member of the Board of Direction of the American Railway Engineering Association and of the Engineering Division of the AAR.

OBSERVATIONS ON THE EFFECT OF HEAVY WHEEL LOADS ON RAIL LIFE

Isolating and evaluating the effect of heavy wheel loads on rail life is somewhat difficult for reasons which include the following:

1. The typical trunk line property, with its operating problems, can hardly be considered a laboratory where all control factors are kept constant so the effects of variables can be determined.
2. Through the years, railroad recordkeeping has been oriented toward efficiency of operations, rather than determining performance of materials and maintenance practices which normally span over more than one generation.
3. Rail condemning criteria vary with service needs and the standard of maintenance experienced by the rail during its life.

It is within the limits of these constraints that I will attempt to give you our experience on the Bessemer, and then compare it with some personal observations made on other properties with greater frequency of heavy wheel loads. To make my comments more meaningful, I will tell you a few things about our property.

The Bessemer and Lake Erie is a Class I trunk line railroad in western Pennsylvania. It extends between the Pittsburgh industrial complex on the south and terminates at Conneaut, Ohio, on Lake Erie, on the north. Originally a double track railroad, it was converted to single line with CTC in 1955. The principal commodities we haul are ore and stone south, coal and coke north, and miscellaneous products in both directions. Maximum size trains are 13,000 trailing tons, or 19,000 trailing tons when using pusher locomotives at the rear end. Our maximum speed limit has been 45 mph all along. In recent years, however, we have lowered it to 35 mph for mineral trains and have found that our "unexplainable" derailments, involving a rail rollover, have been eliminated, and our explainable ones have been drastically reduced.

35% of the right-of-way is curved, with 10 degrees being the maximum curvature in main line. Along with this curvature, we have 36% of our main track on grades between 0.5% and 1.0%. All main track is laid with continuous welded rail, of which about 50% is 140-pound, 25% is cropped 152-pound, and 25% is cropped 155-pound. Almost all curves of 4 degrees and over are laid with continuous welded 140-pound curvemaster rail and anchor spiked with screw spikes. Traffic density is now 35 million gross tons per year over single line track.

Now a word about our wheel loads. During the twelve-year period beginning in 1931 (see Fig. 1), the Bessemer acquired the bulk of what was then the largest fleet of 90-ton hopers, which ultimately numbered over 6,000 cars. Of these, 5,400 are still in service. These cars have been carrying almost all of the southbound ore and stone, loaded to 90 tons until 1962 and to 100 tons since then. Their cubic capacity is 75 tons of coal. When loaded to 90 tons, they produce a static wheel load of 29,800 pounds, and when loaded to 100 tons, 32,300 pounds on nominal 33-inch wheels. It is this fleet which, mixed with other traffic, gave the Bessemer an early preview of the effect of heavy wheel loads, as compared to the rest of the industry. Recently, our hopper fleet was augmented by the purchase of 1,000 self-clearing hoppers capable of carrying 100 tons of ore and 100 tons of coal. Presently, 52% of our loads move in 100-ton cars.

Now let us see what kind of rail wear we are experiencing under the operating conditions and the track environment I have described. Shown in Fig. 2 is the cross section of 131 pound control cooled rail, laid new in 1938 in jointed tangent track near Springboro, Pennsylvania. This contour was obtained in 1964, at which time the rail had carried 420 million gross tons. I wish I could give you the breakdown of this tonnage in terms of 50, 70, 90 and 100-ton cars. Unfortunately, this breakdown cannot be obtained from our records. About the most I can tell you is that this tonnage was accumulated by some 50 and 70-ton car traffic, mostly 90-ton traffic, and some 100-ton traffic. You will notice that vertical head wear measures only about 1/16" at the center of the head. This rail was removed from track in 1965, after it had carried about 440 million gross tons. Reason for its removal was rail end batter and worn fishing surfaces. As information, this rail was cropped, turned, welded into 1/4 mile long strings, and installed at our Saxonburg lead, where it is still in service in heavy traffic.

HOPPER CARS

90-TON CAPACITY

YEAR	NUMBER PURCHASED	
1931	1,050	33 INCH WHEELS FRICTION BEARINGS
1936	2,000	
1940	1,000	LOADED TO 90 TONS UNTIL 1962
1941	650	
1942	425	LOADED TO 100 TONS SINCE 1962
1943	800	
1952	500	
	6,425	

100 TON CAPACITY SELF CLEARING HOPPERS

1970	200	36 INCH WHEELS ROLLER BEARINGS
1972	800	
	1,000	100 TONS OF ORE 100 TONS OF COAL

Fig. 1. Hopper cars.

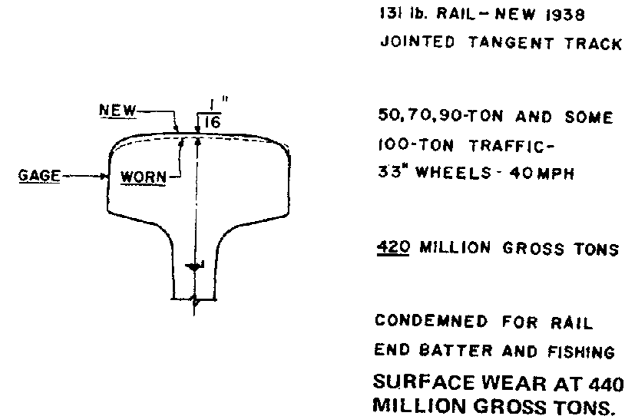

Fig. 2. Acutal control cooled rail wear experience on B&LE 1938 to date, jointed tangent track.

I will now give you our experience with tangent continuous welded rail. In 1946, a test section of continuous welded 131 pound rail was installed at River Valley, Pennsylvania, just north of where the Bessemer crosses over the Allegheny River. A recent cross section of this rail on tangent track is shown in Fig. 3. This rail, which is still in service, shows a vertical head wear of less than 1/8 inch after having carried 650 million gross tons consisting of 50, 70, 90 and 100-ton traffic at prevailing speeds of a little more than 20 mph. To my knowledge, this is the only continuous welded rail segment having accumulated 650 million gross tons of traffic with a substantial part of it in 90-ton cars.

Now, let us see how this performance compares with rail wear on newer continuous welded rail on tangent track. The left vertical axis of Fig. 4 shows rail head wear in square inches, while the right vertical axis shows the equivalent head wear in sixteenths of one inch. The horizontal axis represents total traffic in million gross tons. The cluster of points on the left represents vertical head wear on 140-pound continuous welded rail. The single point with the concentric circles on the right

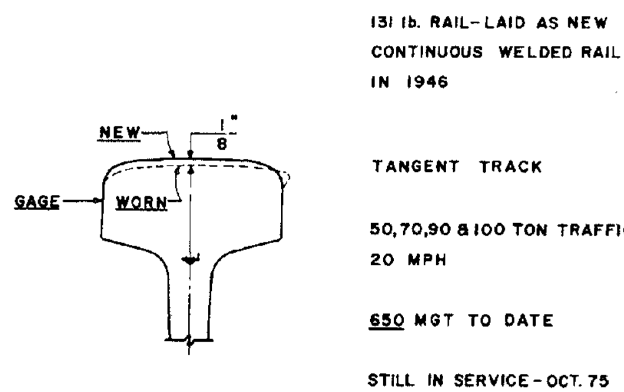

Fig. 3. Acutal control cooled rail wear experience on B&LE 1946 to date, tangent continuous welded rail.

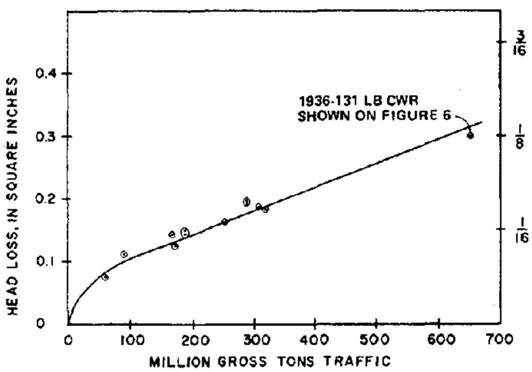

Fig. 4. Acutal head wear of control cooled continuous welded rail on tangent track on B&LE.

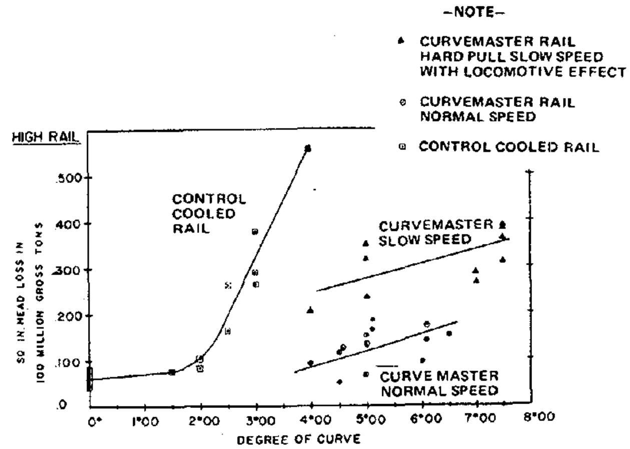

Fig. 5. Actual head wear of control cooled and curvemaster continuous welded rail on curved track on B&LE.

corresponds to the wear of the River Valley test section I have just described. Each point on this curve represents the average of several readings for each location for statistical reliability. The reason we do not have intermediate points between this cluster and the River Valley point is that our older rail has been cropped, welded, and relocated, and we do not know with any degree of certainty the total tonnage to which it has been subjected. This figure reveals a consistently low rate of wear for heavy wheel loads. Since the rate of head wear on tangent track is so small, it appears that the life of this continuous welded rail will be determined ultimately by its resistance to the development of internal defects, at an unknown total traffic level, but beyond the 650 million gross ton level it has already achieved.

I have described to you our experience on rail life on tangent track. It has been around 440 million gross tons for jointed rail, with rail end batter and worn fishing surfaces being the condemning criteria, and beyond 650 million gross tons for continuous welded rail, with the rail's fatigue limit controlling.

Now let us take a look at our experience with rail wear on curves. The vertical axis of Fig. 5 shows wear of continuous welded rail on the high side of curves in square inches per 100 million gross tons. The degree of curvature is shown along the horizontal axis. The curve on the left depicts curve wear of control cooled 140-pound rail between zero curvature (tangent track) and 4 degrees. As I told you, almost all our curves - 4 degrees and over - are laid with 140-pound continuous welded curvemaster rail. Therefore, all points plotted on the 4 degree line and to the right pertain to curvemaster rail. The high point on the 4 degree line is an exception. It represents wear of control cooled rail. It is satisfying to see that our "seat of the pants" decision eight years ago to make 4 degrees the cut-off point for control cooled rail was a wise one.

After all the curvemaster points were plotted, it became evident that they clustered in two groups. The upper group represents curves on heavy grades where heavy locomotive pull is involved. The lower cluster represents curves where no heavy pull is experienced. Again, this graph pertains to rail wear of the high rail.

I should mention that this presentation finds us in the early phases of what we intend to be a continuous and intensive maintenance research project. The initial indication is that we will need to transpose curvemaster rails on 6 degree curves after 360 million gross tons.

How does the Bessemer experience compare with other ore hauling lines? In the past 20 years several ore-hauling railroads have sprung up throughout the world with a pattern of hauling unit trains of 100-ton identical cars on 36-inch or 38-inch wheels loaded in one direction, empty in the other, on a well-maintained single line plant with prevailing speeds of 35 to 45 mph. With some exceptions, there is substantial similarity between them and the Bessemer from the standpoint of physical plant and operating and maintenance practices. Yet their rail life appears to be appreciably shorter. One carrier reports rail renewals on 3 degree continuous welded curves after less than 150 million gross tons. From my observation of the condition of rail on jointed tangent track of another carrier at the 200 million gross ton level, it does not appear that this rail will reach the 300 million level, due to plastic flow in the head. Looking for major differences between these carriers and the Bessemer, we find primarily that the Bessemer experience is based on mixed traffic, with the 90-ton friction bearing cars predominating, although the other ore carriers accumulate their tonnage through the repetitive action of identical 100-ton, roller bearing cars.

Whether any or all of these differences have a bearing on the variation in rail life will have to be determined by those whose research responsibilities have industrywide scope. It is for this reason that I am most anxious to see the upcoming full-scale FRA Facility for Accelerated Service Testing in operation. FAST, the facility's acronym, will give us a railroad plant where all other factors can be kept constant so the effect of the variables can be determined. I predict it will be a blessing for our transportation industry.

Bernard M. Monaghan
General Manager
Iron Ore Company of Canada

Bernard Michael Mohaghan is General Manager of the Iron Ore Company of Canada, which includes the heavy tonnage subsidiary, the Quebec North Shore and Labrador Railway. His career began in 1948 as Location Engineer on the Quebec North Shore and Labrador Railway and has included positions as Assistant Chief Engineer–Grade Construction, Chief Engineer–Railway Maintenance, supervisor of construction of the Northern Land Railway, manager of the Iron Ore Company Canada–Carol Lake Porject, and General Manager–Operations–Setp-Isles Division. He assumed his position as General Manager of Iron Ore Company of Canada at the beginning of 1975.

Monaghan received his B.S. degree in Civil Engineering from the University of Saskatchewan and attended the Harvard Business School.

EFFECT OF HEAVY AXLE LOADS ON RAIL AND TIES

My presentation of the effect of heavy axle loads on rail and ties is based primarily on experience with the Iron Ore Company of Canada's main line subsidiary, the Quebec North Shore and Labrador Railway, and the Carol Lake Mine Automated Railway.

The Quebec North Shore and Labrador Railway is a common carrier operating over approximately 400 mi. of main track. It provides the lifeline link between plants at Carol Lake, Newfoundland; Wabush, Newfoundland; and mines at Schefferville, Quebec, and the deep sea ports at Sept-Iles, Quebec.

Tonnages on the railway this year are down somewhat from the expected figure, due to cutbacks and strikes. However, we will handle 19.73 million short tons of product from Carol Lake, 12.23 million tons from Schefferville, 1.4 million of other shippers' ore, and 1 million tons of freight, for a total of 34.45 million net short tons. The Carol Lake Automated Railway, with its seven trains, will handle 44.128 million net short tons of crude this year, for a total of approximately 86.62 million gross short tons.

The map in Fig. 1 shows the location of mining districts, and Fig. 2 is a more detailed map of the Quebec North Shore and Labrador Railway. The railway starts at tidewater at Sept-Iles and rises to a maximum elevation of 2;066 ft. at mileage 150 before dropping to an average elevation of 1,700 ft. on the balance of the line. Maximum grade against southbound loaded ore trains is approximately 11 mi. at 0.4%. Compensated for southbound ore trains, northbound the ruling grade of 17 mi. is 1.35%. Maximum curvature is 8 deg. and approximately 40% of the total mileage is on curved track.

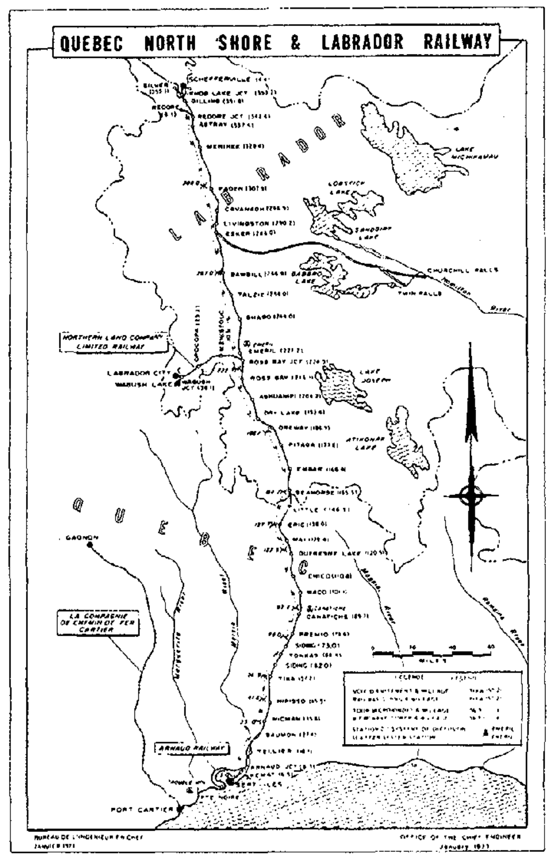

Fig. 1. Location of mining districts.

The main track of the Quebec North Shore and Labrador Railway is laid with 132-lb. R E rail with 14" and 18" tie plates on 8'6"–7"x9" hardwood ties laid on 19.5" centers. These ties are being replaced as needed with 9'0" ties. The track

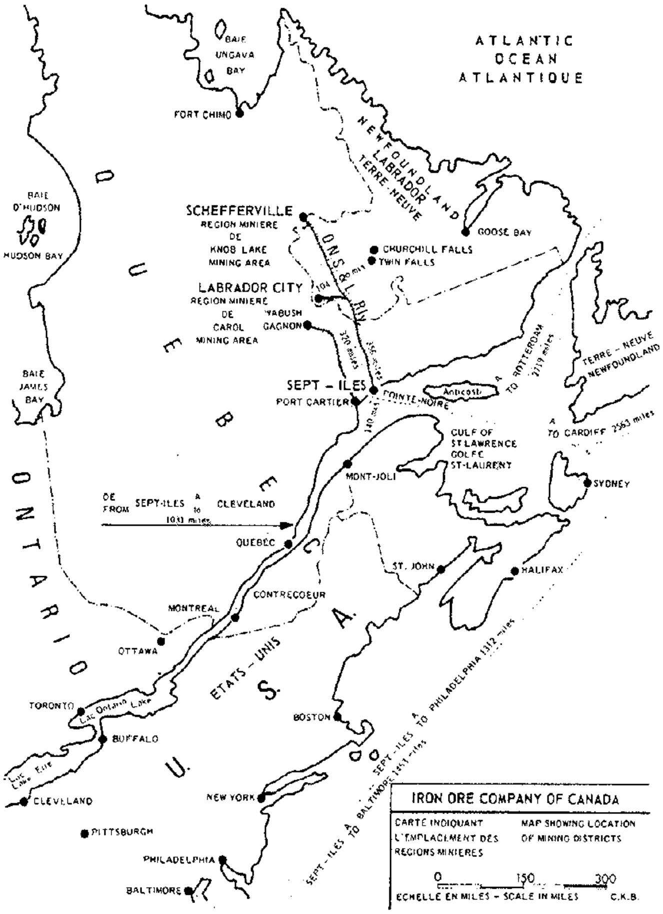

Fig. 2. Detailed map of Quebec North Shore and Labrador Railway.

structure is supported by rock ballast on a subgrade 24 ft. wide. The roadbed of the Carol Mine Automated Railway is of similar construction.

The ore trains on the Quebec North Shore and Labrador Railway run 40 mph empty and 30 mph loaded. Mine trains run 30 mph.

Ore cars are running 36 1/2" and 37" steel wheels on Quebec North Shore and Labrador Railway and 38" wheels on the automated railway, using 6 1/2"x12" roller bearings. Gross weights are up to 286,000 lbs. on the Quebec North Shore and Labrador Railway and to 333,000 lbs. on the mine railway. All cars are operating on four four-wheel sets.

Quebec North Shore and Labrador Railway ore train consists vary from 117 cars with two 3,000-hp. locomotives to 280 cars with five 3,000-hp units. Beyond 160 cars, trains are usually operated with the aid of radio control. Mine automated trains are made up of 1,500-hp. electric unit and 19-20 cars.

Fig. 3 shows two typical Quebec North Shore and Labrador Railway trains, one during summer operation and one under winter conditions. Fig. 4 shows a Carol Mine automated train.

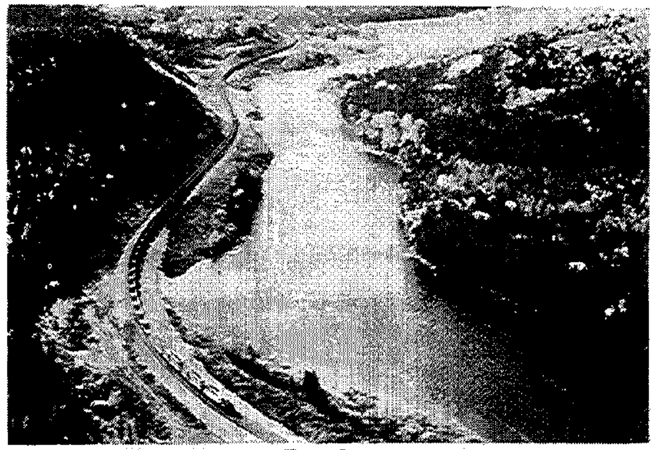

Fig. 3a.

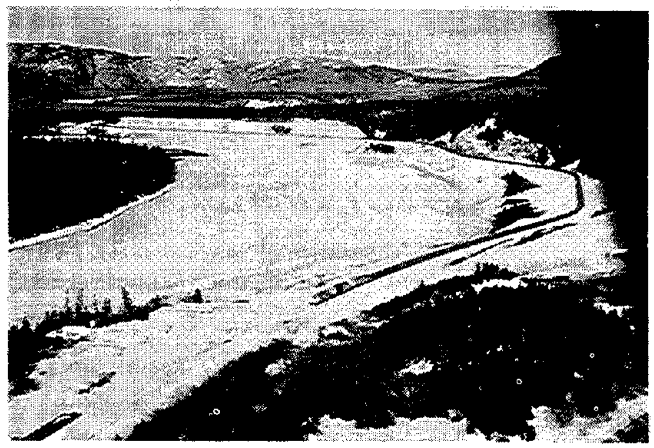

Fig. 3b.

Fig. 3. Two typical trains of our Railway.

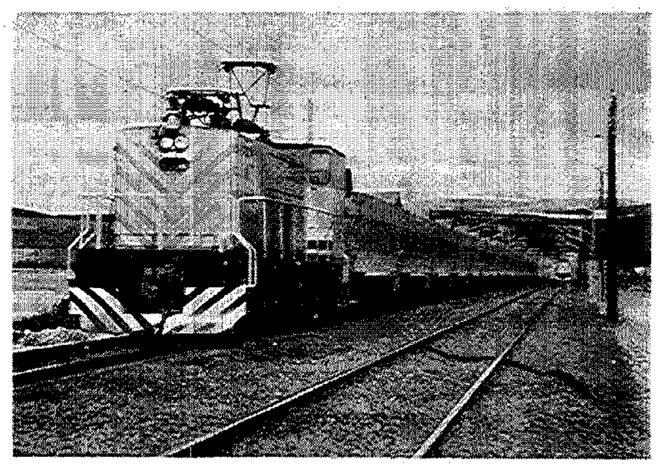

Fig. 4. Carol Mine automated train.

Iron ore is the world's most plentiful basic raw material; the earth's crust has enough of the mineral to satisfy the world's future requirements for several hundred years. Every ore has some of it. Our ore has a relatively low iron content and many richer sources of high-grade ore are available to the world's steel producers. Naturally, steel producers are going to buy their ore from optimal sources. Our rail transportation operation has been one of our major assets in keeping our expenses down to meet competition and still make a profit.

Iron ore mines are notorious for their isolated location in jungles, deserts, or far northern locations. We are no exception. Our trackage is built through rock cuts, muskeg country over permafrost where temperatures vary from 85 deg. F in the summer to colder than -50 deg. F in the winter. Winters last approximately six months a year and snowfall varies from a record high in 1968 of 246.3 in. at Sept-Iles, with most of it on the ground from freezeup to spring breakup, to somewhat lesser figures up the line. Our operating conditions are just as rough as one would wish to find.

Fig. 5 shows the accumulated gross tonnages of the Quebec North Shore and Labrador Railway since construction days in 1953.

Year	Sept-Iles to mile 224	Mile 224 to Schefferville	Mile 224 to Carol
1953 to 1973	622,048	332,732	278,316 (1961-1973)
1953 to 1974	664,786	354,001	310,785 (1961-1974)
1953 to July'75	686,252	360,932	325,320 (1961-July'75)

Fig. 5. Accumulated gross tonnages since 1953.

As to our findings on the effect of heavy axle loadings on rail, I feel that if the rail is matched with the wheel loadings, you will end up with a good rail life. This is, of course, dependent on gross tons, curvature, grade, and the maintenance you wish to give the rail and the supporting structure.

Here are a few items we have noticed as gross tonnages are increased:

1. Line surface and gauge, even on tangent track and of course on curves, must be maintained. Also elevation.
2. Corrugation develops quickly on grade and curves and must be dealt with in the early stages.
3. Joint bars must be maintained tight and rail ends built up as required.
4. Oilers are all-important on curves.

As curvature increases, the wear also increases, regardless of how well the track is maintained. We have found that heat-treated rail, even though more expensive, is a paying proposition, and we are using it on all curves over 3 deg.

Our track inspection car, which we feel is doing a very good job of keeping us informed of track condition, is run over the line every second week. Corrective action is thus taken as defects occur. Sperry rail service car No. 124, assigned to our company, makes a trip over the Quebec North Shore and Labrador Railway and the automated railway every second week.

The rail head surface is ground twice yearly, using a grinding train. Grinding smooths up the corrugations which develop considerably on curves and to some extent on tangents. This not only smooths out small irregularities in surface but also helps us to maintain better joints, where they still exist, and better gauge, line, and surface.

1,440-ft. lengths of continuous welded rail are being installed where rail replacements are made. These lengths are Thermit welded together. Factory manufactured epoxy-bonded insulated joints in the form of 13 ft. plugs are being Thermit welded into the main track where required for the signal system. There is no doubt this will add up to better track and also help to keep cost down.

Fig. 6 gives data on the rail still in service from the original laying on the Quebec North Shore and Labrador Railway. Please note that some rail on tangent and curves on the north end has been taken out, cropped and rewelded in our reclamation yard, and used on passing tracks which have been extended for longer trains.

Mile 3.3 to mile 224 220.7 miles	Mile 224 to mile 353.2 129.2 miles	Carol branch 36.1 miles
18.25 miles*	99.89 miles*	31.5 miles**

*Note: A considerable quantity used in loop tracks in Sept-Iles and Carol, New Mine Spurs, New sidings and extensions.

**Note: Miles removed in 1975.

Fig. 6. Original rail in track.

Fig. 7 indicates rail performance on the Carol Lake Automated Railway.

Average rail life is compared to degree of curvature using standard rail in Fig. 8.

1963 to July 1975	Long tons hauled:	300 million long tons
1963 to July 1975	Total traffic:	660 million short tons

All curves have been changed out at least once.

Approximately 1/2 mile of main track tangent still in service.

Fig. 7. Rail performance on Carol Lake Automated Railway.

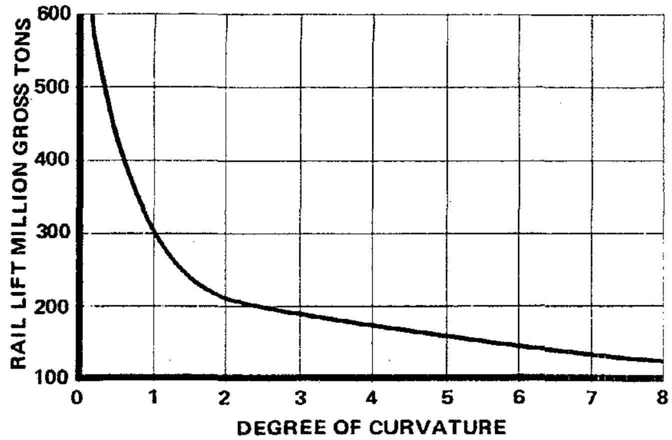

Fig. 8. Average standard rail life compared to degree of curvature.

In summary, our experience has been that if you look after the rail, you can expect good performance.

As to our findings on the effect of heavy axle loadings on ties, we have set up our main track maintenance on a five-year cycle. That is, we put the track into first-class shape every 5 years, using large mechanized gangs. Then we maintain the track between major overhauls with small mechanized setups. We are now in the fourth year of this cycle and are doing well.

Fig. 9 shows the tie removals since they were originally laid in the three high-tonnage sections of our main track system.

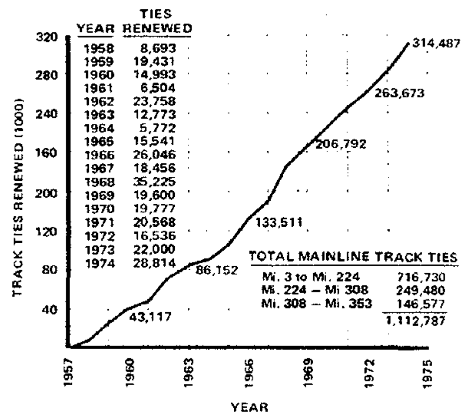

Fig. 9. Tie removals in three high-tonnage sections of main track.

Rene A. Hunziker
Vice President--Railweld Division
Holland Company

A native of Lausanne, Switzerland, Rene Hunziker was educated in the schools of his home town, graduating with the M.E. degree. He immigrated to the United States in 1954 and served in the U.S. Army as Project Manager with the Research and Engineering Agency of the Quartermaster Corps from 1956 through 1958.

Hunziker has been associated professionally since 1954 with maintenance of way machinery and CWR development through employment with Matisa. He has served as Vice President of the Railweld Division of the Holland Company since 1966.

WELDING CONTINUOUS RAIL IN-TRACK

Introduction. Continuous Welded Rail (CWR) eliminates a weak link in the structure of the track. Its most important advantages are in two areas:

1. Increased service life of track and rolling stock.
2. Reduced maintenance cost.

These remarkable benefits have the rare merit of being generated with a smaller initial investment than would be required for jointed track.

Because of these proven economies CWR has often been hailed as one of the most important advances in the history of railroad track design.

CWR is the answer to heavy and high speed traffic. It is synonymous with high quality and heavy duty track. These comments summarize the place of CWR in modern track construction and are applicable to all CWR whether produced in-plant or in-track.

Summary. Developments over the years have led to the prevailing practice of fabricating CWR at a welding plant using the electric flash butt welding process to join 39 foot rails into 1440 foot lengths, transporting and eventually installing these lengths in the field.

It has always been recognized that this method entails a major material handling problem which could be greatly reduced by producing the welds directly in the field. However, quality and cost criteria of field welds have not been met until recently.

A solution has come to us from the U.S.S.R. which has developed a highly portable electric flash butt welder. The Holland Company has successfully used this welder on various North American jobs to produce over 33,000 welds from 1972 to 1975.

This proven tool adds a new option to the fabrication methods presently available to the railway engineer and opens up many opportunities in specific applications otherwise impractical and uneconomical. Current practices should, therefore, be reviewed and reassessed in the light of this new development.

While the Soviet welder is uniquely suited for in-track work, it will perform with equal ease in a permanent set up doing the work of the conventional in-plant welders. Furthermore, closure welds (joining of 1440 foot lengths) heretofore the exclusive province of the thermite process can also be made with this machine. This versatility makes it a truly universal welder.

Brief Historical Background. The welding of rail began in North America over 50 years ago and consisted of test installations using various joining processes applied directly in the field. These pioneering efforts served the railway engineers well in proving the theory and practice of CWR even though the quality of the welds in those days left much to be desired.

It was not until the late 1930's that the welding work was taken out of the field for the first time and the predecessor to the modern rail welding plant made its first appearance.

The encouraging field experience with CWR coupled with the gradual improvements to the in-plant processes led to the acceptance of CWR as a desirable alternative to jointed track. Increasing axle loads and the advent of the automated electric flash butt welding technique added impetus to this development. By the mid-50's it became standard practice to weld all new rail on many North American railroads, and soon thereafter the welding of second hand rail released by new rail programs or track abandonments also became a viable solution.

The logical step was taken to set up welding plants near the rail producing mills in order to reduce the rail haul to a minimum. This development further promised to place the benefits of CWR within economic reach of smaller railroads willing to combine their program with others. In practice, however, the concept proved short-lived as one after the other the plants were shut down mainly due to scheduling conflicts leading to under utilization of the facilities.

The growing use of CWR, however, continued unhindered and today there are over 40 rail welding plants to serve North American requirements. Some of these facilities incorporate sophisticated rail handling systems which yield the economies commonly associated with processing large volumes of rail by high production methods.

Development of In-Track Welding. In the meantime, the thrust of the Soviet effort was taking another direction altogether. As a result of the remarkable research work of the Paton Welding Institute in Kiev, a very compact and energy saving electric flash butt welder was developed which allows direct field application of the weld. Now the machine could be brought to the work instead of the work to the machine.

This achievement offered two distinct benefits over in-plant production methods, namely:

1. Substantial reduction of capital investment.
2. Savings in transportation costs.

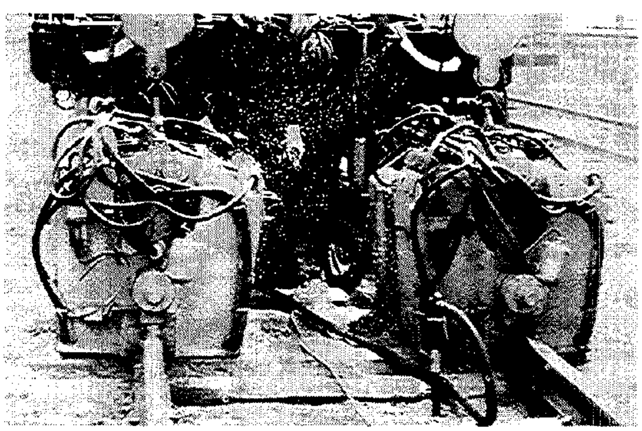

Fig. 1. Soviet dual welding line application using two in-track welders in tandem.

The success of the initial tests in the late 50's led eventually to the adoption of this system as evidenced by the 400 in-track welders the Soviets have in service today, which far outnumber their permanent plants. (Fig. 1.)

These developments attracted a great deal of interest in North America. Visitors to the U.S.S.R. returned with glowing reports. Miscellaneous papers were published and patents were issued. In 1968 several North American Maintenance of Way officers attended demonstrations of the Soviet built in-track welder on the French National Railways. Impressed by its capability and encouraged by the favorable comments of these engineers, Holland Company initiated a plan to introduce thie equipment on the American scene.

AAR Research Center Test. First, the integrity of the welds produced by the Soviet welder had to be proven beyond a doubt. This was done in a series of tests covering twelve welds made for Atchison, Topeka and Santa Fe with 100 lb. rail, shipped to France and returned to the AAR Research Center for a complete investigation. (Fig. 2.). Rolling load, slow bend and drop tests were conducted on six welds with upset metal removed around the full contour of the rail and six welds with upset removed on the rail head only. The detail of the results are documented in AAR Bulletin #626. These welds proved equal to the best welds produced by the conventional flash butt welders in use in North America.

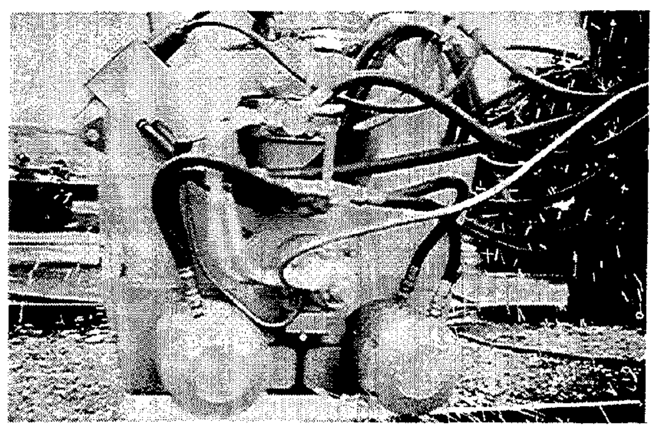

Fig. 2. U.S. test welds produced by French National Railways with American rail.

North American Debut of Soviet In-Track Welder. The AAR test results cleared the way for Holland Company to proceed in earnest with the design and construction of the necessary vehicle, power plant and other support equipment required to make the Soviet in-track welding machine operational as a moving field production line.

The first American version was completed in the spring of 1972 when it began full-fledged field testing and production. (Fig. 3.)

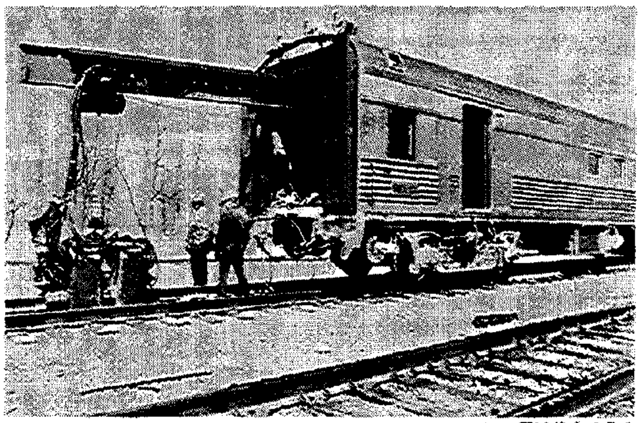

Fig. 3. First U.S. in-track welding outfit FWX-101. (Photograph courtesy of AT & SF.)

Holland's experience since then has included the following programs:

Atchison, Topeka and Santa Fe	1972–1974	16,000 welds
Belt Railway of Chicago	1974 1975	3,000 welds 2,000 welds
Chicago & Illinois Midland	1975	4,000 welds
AMAX	1975	8,000 welds

This last program launched a novel mini-plant concept using the in-track welder in a semi-permanent set up.

Applications. The applications for which the in-track welder has proven its technical suitability in actual practice include the following:

- existing track and new track
- standard and panel track construction
- opposite or alternate joints
- main lines and branch lines
- tangent and curved track
- yards, siding, and passing tracks
- transit systems, mines and industrial tracks
- new rail and second-hand rail, with or without cropping
- standard, curvemaster and heat treated rail
- CWR from one insulated joint to the next
- CWR between road crossings through towns
- closure welds, rail repairs and insulated joint inserts (See Fig. 4.)

This partial list illustrates the versatility of the Soviet welder with its outstanding capacity to handle all the jobs commonly done by both the in-plant method and the thermite process.

The benefits of CWR can now, therefore, be extended to all these areas with one machine doing the entire job at one time.

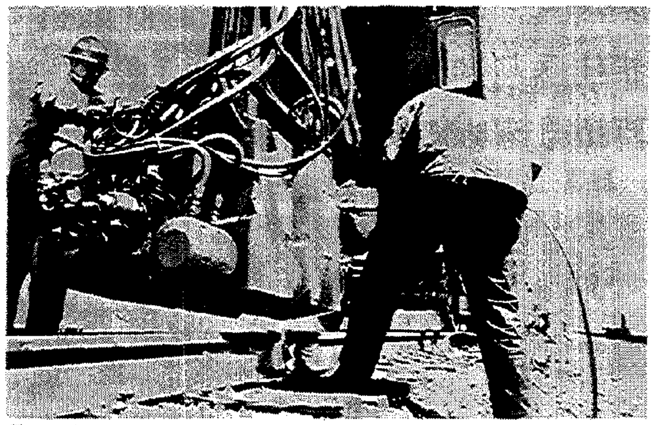

Fig. 4. In-track welder clears rail upon completion of weld. (Photograph courtesy of AT & SF.)

Welding Equipment. The welding equipment consists of the welder proper, power and control cabinets and hydraulic unit.

Mechanically the welding head is designed like a large clamp which locks the rail web between its specially designed jaws. This arrangement assures perfect angle bar alignment. (Fig. 5.)

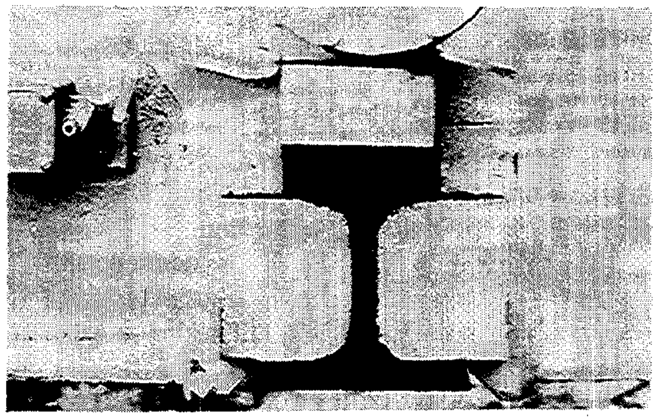

Fig. 5. Clamping jaws lock on rail web for perfect "angle bar alignment."

Electrically the welder falls into the category of the energy saving continuous flash process, requiring 3 minutes to complete a weld on 136 lb. rail. Further, the unique disposition of the welding transformer allows for a considerable decrease of the short circuit resistance of the welder.

This covers the essential features which have allowed for a substantial reduction in equipment size and yielded the further advantage of much lower power consumption than found in conventional welders.

Work Methods and Organization. The welding equipment may be mounted on a rail or highway vehicle for maximum mobility.

It may include one welding head suitable to work both running rails in one pass or two welding heads working in tandem. (Fig. 6.)

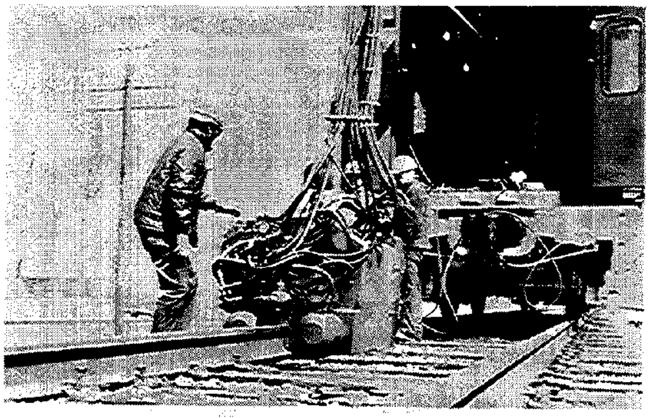

Fig. 6. In-track welder working both rails alternately. (Photograph courtesy of AT & SF.)

The rail to be welded may be positioned:

1. At standard gauge.
2. Between the gauge or on the shoulder.

While this work is normally performed at the location of actual use, special conditions may make it advisable to weld at another site for later installation.

The organization of the work includes additional auxiliary functions such as rail positioning, preparation, finishing, inspection and having the track ready for traffic. The choice of equipment and the staffing will, therefore, vary according to the specific application.

The welding function itself is handled by one man. The additional work may require as few as 3 people or as many as 14, depending on the job at hand.

Production. The in-track welder is designed to produce 12 welds per hour, yielding a potential of 96 welds in 8 hours welding time. In practice, production has reached 86 welds per 8 hour shift for work performed in siding, or yards, where no traffic delays are incurred.

In main line track, with approximately 6-1/2 hours actual welding time, the output has averaged 53 welds per day. The in-track welder can therefore be programmed to make 10,000-12,000 welds per year on a single shift basis, or approximately 40 miles of CWR. (Fig. 7.)

Fig. 7. Producing CWR with 39 foot rails cropped in-track. (Photograph courtesy of AT & SF.)

This same welder used in a semi-permanent set up has reached production of 116 welds in a nominal shift of 9 hours, with daily output averaging 80 welds per shift.

Better results are anticipated in the future to reach the full productive potential of the in-track welder.

Quality Control. Quality control takes place at two levels:

1. On the welder itself, where the welding parameters can be monitored with a recorder in order to promptly detect any machine malfunction, and
2. On the weld zone proper, with the conventional testing methods available to the railway engineers to check joint geometry and weld soundness.

Economics. The economics of in-track welding can only be stated in their broadest terms in the context of this paper.

Basically, the cost of in-track welding compares favorably with the in-plant method in those instances when:

1. There is sufficient track time (in order to assure maximum production).
2. The total haul of the new rail is reduced, or entirely eliminated as the case may be for second-hand rail. (This is especially significant if commercial freight rates are used in computing the true hauling costs.)

Each instance, therefore, has to be weighed on its own merit depending on the particular set of circumstances and specific job conditions, giving full recognition to all the cost factors.

Recent Soviet Developments. To further enhance the usefulness of their welder, the Soviet have recently perfected two very desirable additions to the system:

1. Complementary "impulse fusion" which reduces the welding time and the rail consumption by approximately one-third each.
2. Built in shear for the removal of the upset which reduces the weld finishing work and time.

These technological advancements were satisfactorily demonstrated in the U.S.S.R. to Holland Company and plans are being progressed for an early introduction of these improvements to North America.

Looking Ahead. What will the railway engineers do if and when the steel mills start production of 78 foot rails as is already the case with one Canadian mill?

It will be very costly to modify existing facilities and even more expensive to build new ones.

What about rails longer than 78 feet?

Then, the in-track welder will provide the only practical and economical alternative to the present methods.

Conclusion. Today there are 290,000 track miles in North America which are still jointed. Much of this trackage needs rehabilitation and upgrading for heavier axle loads.

The elimination of 80 million joints constitutes a compelling step in this direction.

The in-track welder provides a proven and economical solution for doing a substantial part of this work in the field.

Discussion Leader
Paul S. Settle
President
Railway Maintenance Corporation

Paul S. Settle's career began with the Pennsylvania Railroad in 1936; he started as an engineering apprentice and rose to Division Engineer at Williamsport NY and Pittsburgh. He joined Railway Maintenance Corporation in 1953, was made a Vice President in 1954, and President in 1962.

Settle received the B.S. degree in Civil Engineering from Lehigh University and attended the Executive Program of the Carnegie Institute of Technology. He is a member of the American Railway Engineering Association, the Roadmasters and Maintenance of Way Association, and Track/Train Dynamics Program Steering Committee. He has served the Railway Progress Institute as the first Chairman of the Maintenance of Way Committee, Vice Chairman in 1972, and Chairman in 1973 and 1974.

How can we quickly sum up what we have listened to this morning? I think one thing that came out of the presentations by the technical people who are operating the railroads is that if you've got the guts and the ability to take action, you can use conventional material and equipment to produce a more profitable operation. And that's the name of the R&D game--to produce a more profitable operation. That's what we are all concerned with.

Once you have the profits, you generate enough funding so that your No. 1 R&D problem can be solved a lot more easily. The No. 1 problem that always comes up first with any project is how to finance it. If the railroads become profitable they will generate these funds. Then money will be available in greater amounts to continue the work you are all doing.

Both Mr. Rougas and Mr. Monaghan indicated that they are not at all averse to using R&D in order to solve their problems, and they have been successful in doing this. I would like to congratulate them on being doers--they were able to produce an on-time performance in their presentations.

COMMENTS/DISCUSSION PERIOD

Delegate Comment: You mentioned that the 90-ton cars on the Bessemer line are friction bearing--what about the 100-ton cars?

Panel Response: The 100-ton cars are roller bearing.

Delegate Comment: I understand that in welding heat-treated rail you get a bit of a soft spot on each side of the rail. Is that true? If it is, does it represent a significant maintenance problem when the rail is in track?

Panel Response: From our experience, you do develop this problem with all rail. We have not been able to observe this to any appreciably greater extent with curved than with tangent rail. We do grind our rail on a regular basis, and perhaps this is the reason why we have not been bothered by this development.

Panel Response: We are actually using both curvemaster and heat-treated rail. And we have had no difficulty with either.

Delegate Comment: Is it correct that you run the Sperry rail service car every other week? What are your results? Do you continuously pick up defects in the rail?

Panel Response: That is correct--every two weeks. We have a rate of defect pickup which for the last year and a half has been fairly consistent by trip and by season.

Delegate Comment: You don't feel it would be possible to extend your cycle longer than every other week?

Panel Response: Well, we run the Sperry car for the purpose of picking up defective rails, both defects present when we buy the rail and defects which develop in the rail. Our experience to date has indicated that we should run this machine each two weeks. This is based on a tonnage figure, not on a time factor, as considered necessary to properly protect our operations.

Delegate Comment: Do the production figures you were using in connection with the in-track welder include cropping the rails, moving the rails, and so forth?

Panel Response: The figures which were given are actual net results for doing the entire job.

Delegate Comment: What kind of results have you gotten from not cropping rails and trying to weld the second-hand rail with the bolt holes in it?

Panel Response: The Belt Railway of Chicago followed that procedure on approximately 3,000 welds which were made late last year, and no failures have been reported to date. Care was taken, I might add, to remove the first bolt hole on the rail, depending on its condition.

Delegate Comment: You mean then you actually did crop at the first hole?

Panel Response: Yes, we had to crop some of the rail. We cropped the rail basically for two reasons. Firstly, to prevent the drilled-in signal bond connections from falling into the weld zone, and possibly detract from the quality of the welds. And secondly, to remove any cracked bolt holes as source of potential service failures. As a further consideration, we also saw to it that bolt holes would not fall into the heat affected zone of the weld.

Delegate Comment: Regarding the question about the number of roller bearing cars, I could add that out of a total of 6,000 cars, we really only have 1,000 that are roller bearing. Maybe you want to clarify the information on the 52% that you are handling in 100-ton loads now.

Panel Response: Actually, I said that 52% of our loads move in 100-ton capacity cars, loaded to 100 tons. Not all of them are standard bearing and not all of them are roller bearing. I think a more specific statement would be that about 20% of the 100-ton loads are being handled in roller bearing cars and about 80% in friction bearing cars right now. This will change, of course, to a kind of 50/50 mix shortly when we get more new cars.

Delegate Comment: You are absolutely silent about the extent that you lubricate curves on the Bessemer and Lake Erie? To what extent do you lubricate them?

Panel Response: We do perhaps lubricate them better than average using the manufacturer's recommendations. We have been lubricating most almost all of our curves and at times have been getting into a problem with excessive lubrication, but mostly I would say, our curves are lubricated properly.

Delegate Comment: But what percent of the total number of curves are lubricated--do you have some that are not? At what degree of curve do you start lubrication?

Panel Response: Approximately 4 deg. and over.

Discussion Leader Settle: One more thing before we break for lunch. The most difficult thing you people will have to do is to take your knowledge home and make use of it. I hope that when you do that, you consider that what you are turning out is going to make more money for the railroad.

Myles B. Mitchell
Director
Office of Passenger Systems Research and Development
Federal Railroad Administration

A native of the Midwest, Myles B. Mitchell received the M.S. degree from Oklahoma State Univeristy in 1951 and joined McDonnell Douglass Corporation at St. Louis as an aerodynamicist the same year. He served in various management positions there before joining The Marquardt Corporation, Van Nuys CA, where he held the positions of Assistant Manager–Applied Research, Engineering Manager, and Manager of New Products–Corporate Office.

In August 1969 Mitchell was called to Government service as Director of the Office of High Speed Ground Transportation, U.S. Department of Transportation, with primary responsibility for carrying out the High Speed Ground Transportation Act of 1965. In December 1971 he was named Chief, Test Center and Demonstrations Division of the Department, with responsibility for the newly created High Speed Ground Test Center at Pueblo CO and for the DOT's rail demonstrations program. He was made Director of the Passenger Systems Research and Development, with responsibilities for all passenger equipment, including both rail and advanced systems in April 1975.

SESSION II

SUSPENSION DEVELOPMENTS

The afternoon session (Session II) was moderated by Myles B. Mitchell, who introduced himself as Director of the Office of Passenger Systems R&D for the FRA. He made announcements regarding Conference proceedings and introduced Richard L. Lich, President, Dresser Transportation Equipment Division, who gave the theme address.

Richard L. Lich
President
Dresser Transportation Equipment Division
Dresser Industries, Inc.

Richard L. Lich is President of the Transportation Equipment Division of Dresser Industries, Inc., Depew, NY, which produces Symington, Gould, Waugh, and Hydra-Cushion products. He has been actively involved in the railroad and mass transit inductries for 25 years.

Lich received Bachelors and Masters degrees in Engineering from Washington University in St. Louis and attended the Harvard Advanced Management program. He is a Registered Professional Engineer and holds numerous U.S. and overseas patents on railroad and mass transit equipment.

He has traveled extensively and is familiar with railroad and mass transit developments in many parts of the world. Most importantly, as a long-time firm believer in the railroad industry, he is convinced that it is on the threshold of a great opportunity for service to the nation.

ADVANCING TRUCK TECHNOLOGY THOURGH A TRIPARTITE EFFORT

Good afternoon, gentlemen. The theme of the 12th Railroad Engineering Conference is "The Effect of Heavy Axle Loads on Track." The theme of this Session, specifically, is "Suspension Developments Which Minimize the Effect of Heavy Axle Loads on Track." In establishing this theme I would like to briefly put into perspective some key considerations as I view them.

The foundation of the railroad industry today is the interent efficiency of the steel wheel against the steel rail in combination with the following basic factors:

1. Long Train Consists;
2. High Rating Motive Power;
3. High Operating Speeds;
4. And High Capacity Cars.

These basic factors enable railroad systems to provide high-volume national transportation services for a wide range of lading more economically than other modes of transportation. This is particularly significant today, and will be increasingly so in the future, in view of the necessity of conserving our nation's energy supplies and promoting the development of our natural mineral and energy resources.

These basic factors, which combine to produce the great efficiency of railroad systems today, also combine to produce greatly increased forces which must be transferred between the train car bodies and the roadbed. These forces result in the following operating environment:

1. Increased vertical forces with greater dynamic complement because of larger volume cars of greater capacity and higher speeds;
2. Increased lateral forces with greater dynamic complement because of greater car capacity and higher speeds;
3. Increased roll moments because of higher car centers of gravity and greater car capacities;
4. Increased dynamic instability because of higher speeds;
5. And increased vertical and lateral forces resulting from greater train forces generated by longer train consists and higher speeds.

The function of the railroad truck is to provide a mobile combination structural and suspension system whereby all of these forces are effectively transferred between the train car bodies and the roadbed.

The transverse, vertical, and longitudinal space available for the truck system to effect the transferring of these forces is relatively fixed by the clearance diagram, the standard track gauge, and the economics of car body configuration. These increased forces must be transferred, therefore, within essentially the same historic space that was available when all of the basic factors were far less demanding.

Unfortunately, we cannot expand this space laterally by means of wider clearance diagrams and wider track gauge. We cannot expand it vertically because it would intrude on the lading compartment of the car bodies. Longitudinal expansion involves a host of clearance interrelationships that act as strong barriers.

As the basic factors have become more and more demanding, the performance of the conventional, historic freight truck occupying the historic limited space between the car body and roadbed has become increasingly marginal, resulting in accelerated wear and deterioration of the truck system and the roadbed, and as well the car body and lading. This results in a serious interrelated operational and economic problem.

What is the answer to this problem? Do we back off on the basic factors of long train consists, high operating speeds and high capacity cars? I say no! I believe the answer is a dual thrust as follows:

1. A determined tripartite research and development effort to gain the fundamental and practical understanding which will result in advanced new high-performance truck designs which can economically function in the historic space; effectively withstand the operating environment of the basic factors; and greatly reduce the deteriorating effect on the roadbed;
2. And a determined tripartite research and development effort to gain the fundamental and practical understanding which will result in an advanced roadbed design which can effectively withstand the operating environment of the basic factors.

In my comments before the 11th Railroad Engineering Conference last year in Pueblo, I stated that three different types of research and development, working in concert, are necessary for railroad technological advancement as follows:

1. Efforts to increase fundamental understanding of railroad plant and equipment relationships and performance requirements;
2. Efforts to apply such increased understanding practically in railroad operations;
3. And efforts to produce innovative hardware based on the practical application of this increased fundamental understanding.

It is logical that the first efforts be carried out principally by the Federal Railroad Administration and its counterpart, the Canadian Transportation Development Agency, in view of the magnitude of the experimental scale and the budgets that are required. It is logical that the second efforts be carried out principally by the railroads and the Association of American Railroads who have at their disposal the massive testing laboratory of the American railroad system. And it is logical that the third efforts be carried out prinicpally by the specialized individual suppliers which make up the railroad supply industry. This is what I mean by a tripartite effort.

I have been pleased to observe over the past year that this is increasingly beginning to take place in all areas of railroad technological advancement.

Session II today is symbolic of the tripartite effort in truck suspension developments which minimize the effect of heavy axle loads on track. The papers that are going to be presented cover efforts by the Federal Railroad Administration, by a major railroad and by individual suppliers. I believe it is going to be a most interesting and informative program. Thank you.

Robert Byrne
Manager--Research
Southern Pacific Transportation Company

Robert Byrne's appointment as Manager of Research for the Southern Pacific Transportation Company in 1972 followed 20 year's experience with the Association of American Railroads. He joined the association in 1953 as a chemical engineer, was appointed Director of Mechanical Research in 1964, and became Research Director in 1968. Byrne received the B.S. degree in Chemical Engineering from Lehigh University and the M.S. degree in the same field from Northwestern University.

Byrne is the author of several articles on railroad materials and components. He is a member of the American Society of Mechanical Engineers, Air Brake Association, and the Newcomen Society and is the Chairman of the AAR Committee on New Truck Designs for Freight Cars.

PROGRESS REPORT ON THE TRUCK DESIGN OPTIMIZATION PROJECT

The federally funded Truck Design Optimization Project (TDOP) being conducted by Southern Pacific Transportation Company is designed to furnish new technical and economic insights into the procurement and use of freight car trucks. With fifteen months of project work completed, a variety of outputs are emerging, including digital data tapes that may prove useful to future investigators of freight car truck dynamics.

TDOP grew out of a consideration of the recent history of freight car truck usage and anticipated future requirements that will be brought about by generally increasing traffic projections demanding more efficient and economical service. Despite truck manufacturers' and designers' contributions to freight car truck design, the dynamics of truck performance require development to enable correction of existing

Form	Consistency	Parametric Changes	Preliminary Specification Requirements	Transient Events	Periodic Regimes
Probability Density	All channels	All channels 5 DOF (bounce, pitch, yaw, lateral, roll) Lateral forces at adapters during curving	• Ride quality 5 DOF • Life factors (wear of wheels, centerplates, adapters, snubbers and lateral restraints) • Running stability • Curve negotiation lateral forces		Combine channel data to show correlations.
Power Spectral Density		VARIABLES: • Track defined • Speed • Car weight • Wheel wear • Snubber wear	VARIABLES: • Track defined • Car weight • Speed		Combine plots to establish correlation.
Time Domain				• Adapter force • Spring deflection • Side bearing closure and force • Centerplate acceleration • Carbody roll, bounce and lateral	Truck Stability • Truck rotation angle • Side frame displacement and rotation relative to truck bolster • Truck tram • Carbody lateral, roll and yaw

Fig. 1. Data reduction and analysis plan.

problems and to define future truck system needs. Furthermore, an understanding of the economics of freight car truck acquisition and use is needed to support the changing costs that will result when new and modified trucks become available.

Technical Objectives. A major part of the TDOP Phase I effort is aimed at characterizing existing trucks as a basis for equating the technical performance of modified and new designs. It is anticipated that Phase I studies will produce a truck performance specification as well as a description of the dynamic performance of existing truck types.

To accomplish these objectives, a preliminary evaluation was made of recent truck performance problems and corrections. Using simple computer approximations to supplement this evaluation, a data reduction and analysis plan was derived leading to data output formats and the selection and placement of instrumentation on the test trucks and freight car (Fig. 1).

Data Displays. Root-mean square and standard deviation vs. speed plots furnish information on the effect of speed on the magnitude and dispersion of a selected variable or combination of variables. These plots are potentially useful for establishing ride quality requirements in performance specifications but are not likely to provide insight into the dynamics of the system (Fig. 2).

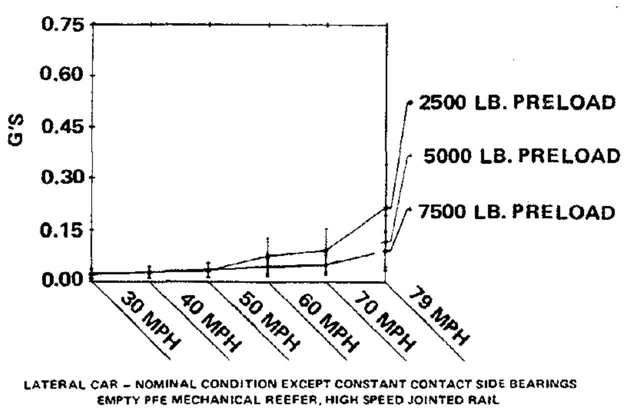

Fig. 2 Root-mean square (RMS) plot.

The histogram display permits a more detailed representation of the mean value and dispersion data applying to a specific speed. It represents one point on the plot of RMS data. While this is an uneconomical method for displaying large amounts of data, the use of histograms has the potential for permitting an evaluation of life factors associated with wear and fatigue (Fig. 3).

Power spectral density (PSD) plots (Fig. 4) have the following advantages:

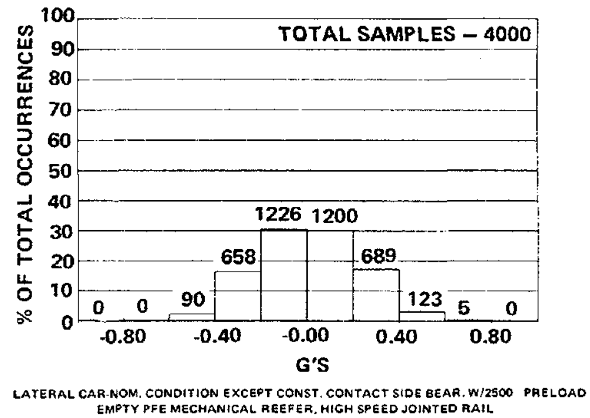

Fig. 3. Histograms for potential evaluations of wear and fatigue.

- Analyzing the effects of track irregularities (expressed as frequency) on the truck responses determined as a function of frequency

- Correlating with linearized mathematical models in the frequency domain

- Establishing cross correlations to obtain the influence of system variables

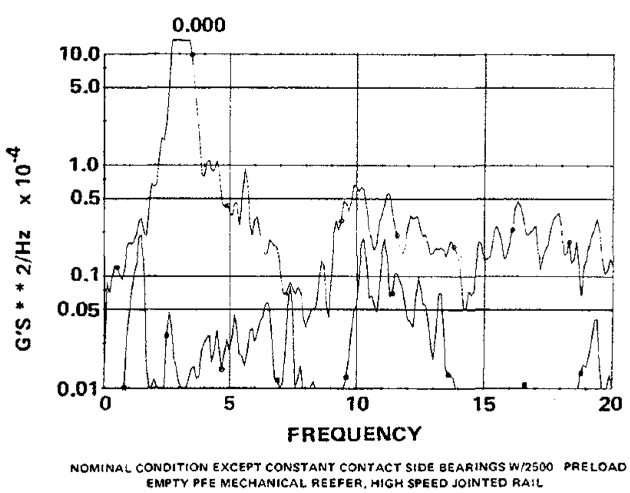

Fig. 4. Power spectral density (PDS) plot.

Time domain plots present selected sections of the test record. The type of dynamic regime being studied dictates the data variables to be displayed (Fig. 5). For example, the lateral dynamics problem of truck hunting can be studied by selecting data channels associated with truck component motions. On the other hand, the lateral dynamics of the entire car and truck system can be studied by tracing the effects of wheel inputs on the car body.

Instrumentation. Instrumentation for the freight car body (Fig. 6) and test trucks (Fig. 7)

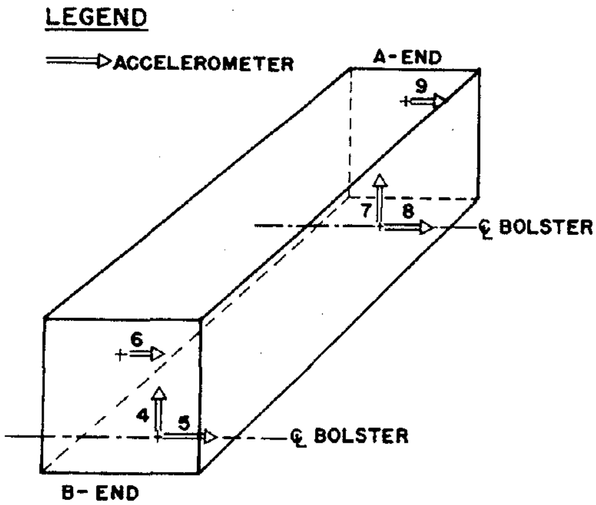

Fig. 6. Freight car body instrumentation.

was selected by considering that the forces and motions in the system would also represent variables in the mathematical models intended to explain dynamic behavior. Truck components having relative motion are instrumented with displacement transducers in such a way that both linear and angular motions are measured. Forces are measured on the roller bearing adapters and roller side bearings on the B-end truck. Accelerometers are placed on truck components and the freight car body in such a way that the path of a rail input can be traced to the car body. Accelerations on the car body are related to displacements and forces by reference to phase angles.

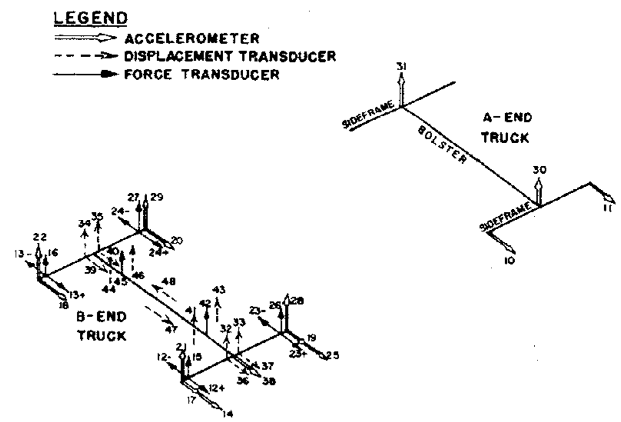

Fig. 7. Freight car truck instrumentation.

Data System. Data collection is performed utilizing a Hewlett-Packard 9601 Data Acquisition system located on board the SP-250 Instrument Car (Fig. 8). This system consists of a 2100S minicomputer with 16K word core. Peripheral equipment includes a teletype to transmit commands to and receive messages from the computer; a photo reader and high-speed punch; and a 1600 bpi magnetic tape drive writing onto 2400-ft. reels of digital magnetic tape.

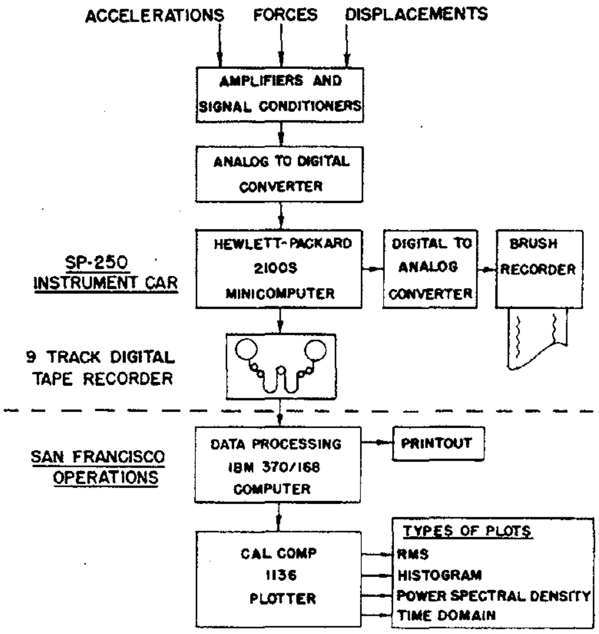

Fig. 8. Data acquisition and reduction system schematic.

The 9601 system is further augmented by analog-to-digital and digital-to-analog capabilities. After proper amplification and conditioning, sensor analog signals are converted to digital form for manipulation by the computer and ultimate storage on the magnetic tape. In addition, the computer reconverts the digital data to analog form for hard copy display on the Brush recorder. The Brush recorder is used to check data validity, verify the digitization and recording process, and obtain an immediate indication of test results. Each A/D and D/A conversion is accomplished in approximately 23 microseconds. Throughput is 9,600 samples per second, providing 200 samples per second for each of 48 data channels.

Test data are postprocessed on Southern Pacific's IBM 370/168 computer at San Francisco. Reduced and combined data are plotted using a Cal Comp 1136 plotter. Original data tapes are being furnished to the National Technical Information Service where they are available for public use.

The Cal Comp produces data displays in the form of RMS plots (Fig. 2), histograms (Fig. 3), PSD plots (Fig. 4) and time domain plots (Fig. 5). These displays were previously discussed. The time-domain plot has six channels of reduced or combined data on a single printout. Thus, dynamic relationships such as phase angles and force paths can be traced with relative ease.

Phase I Test Series. Four test series have been conducted in Phase I to technically evaluate the dynamic performance of freight car trucks (Fig. 9). Runs in each test series were made over a variety of track conditions selected to dramatize the regimes involved in truck dynamics (Fig. 10).

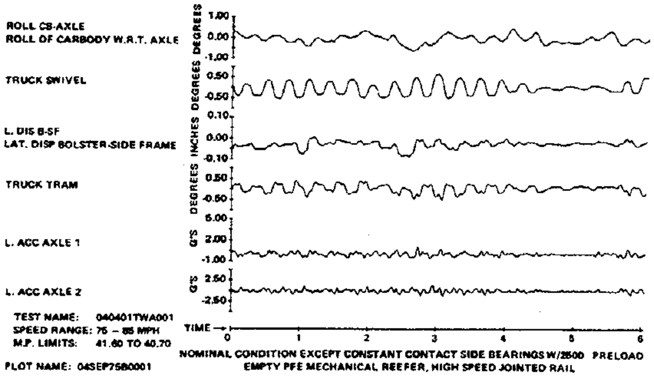

Fig. 8. Time domain plot.

Test Series 1 – Acquire Baseline Data on Conventional Truck.

Test Series 2 – Simulate Wear Ranges and Study Spring Effects.

Test Series 3 – Extend Baseline Data with Additional Car and Truck Types.

Test Series 4 – Study Effects of Modifications.

Fig. 9. TDOP Phase I schedule of completed tests.

- Suisun-Fairfield, 30 to 79 mph
  - Jointed Rail
  - Continuous Welded Rail
- Schellville Branch, 10 to 45 mph
- Niles Canyon, 25 to 35 mph
  - 1 to 9 Curves

Fig. 10. Locations for testing.

At Suisun-Fairfield, near San Francisco, the test train, consisting of an SD-40 locomotive, SP-250 instrument car, a test car, and a caboose is operated on both continuous welded rail and jointed rail tangent track over a speed range of 30 to 79 mph. On the jointed rail, 10 mph step increases in speed are made with steady speeds of 30, 40, 50, 60, 70, and 79 mph in one pass. This procedure requires two passes on the shorter CWR track.

The medium-speed tangent track located on the Schellville Branch near Suisun-Fairfield is used to study lower speed performance. On this track, 5 mph step increases in speed are made, with steady speeds at 15, 20, 25, 30, 35, 40, and 45 mph. Second passes are made at speeds where significant vehicle oscillations are observed.

Curve negotiation tests are conducted in Niles Canyon, also near San Francisco. Tests are made at an equilibrium speed of 25 mph on eleven curves ranging from one degree to nine degrees. Repeat tests are then made at an over-equilibrium speed of 35 mph.

In Test Series 1, a 70-ton capacity ASF truck was tested under a mechanical refrigerator car (SPFE-459997). The truck was tested in a new, nominal condition as defined in applicable standards of the Association of American Railroads. Test variations involved bolster-gib clearances, side bearing clearances, and wheel load (Fig. 11).

- 70-Ton Truck, Constant Force Friction Snubbing
- PFE Mechanical Refrigerator Car
- Load Variations
  - Gross Rail Load
  - 50 Percent GRL
  - Empty
- Side Bearing Variations
  - Tight, 1/8 in.
  - Nominal, 1/4 in.
  - Open, 3/8 in.
- Outer Bolster Gib Variations
  - Current Standard
  - Former Standard

Fig. 11. Test series 1 parameters.

The car and truck types used in Series 1 were used again in Test Series 2, where further changes in truck component conditions were evaluated (Fig. 12). Wheels with worn profiles were tested, as well as reduced levels of friction snubbing. Variations in springing involving the use of D-3 and D-7 spring groups also were evaluated.

- 70-Ton Truck, Constant Force Friction Snubbing
- PFE Mechanical Refrigerator Car
- Load Variations
  - Gross Rail Load
  - Empty
- Wheel Profile Variations
  - Service Worn, 285,600 miles
  - Mid-range Worn
- Snubber Capacity Variations
  - Full
  - 67 Percent
- Spring Variations
  - D-5
  - D-3
  - D-7

Fig. 12. Test series 2 parameters.

In Test Series 3, testing was extended to include Barber-type friction snubbing, other types of freight cars, and 100-ton capacity trucks (Fig. 13). Variables involved the following:

- Flexible vs. rigid car body structure
- 70 vs. 100 ton capacity cars
- High center-of-gravity cars
- 89-ft. low deck flatcar

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- Truck center distance
- Truck wheelbase
- Wheel loading

- **70-Ton PFE Mechanical Refrigerator Car**
  - **Load-Variable Force Friction-Snubbed Truck**
- **100-Ton SP 60-ft. Box Car**
  - **Load-Variable Force Friction-Snubbed Truck**
- **70-Ton Seaboard Coast Line 50-ft. Box Car**
  - **Load-Variable Force Friction-Snubbed Truck**
- **100-Ton Louisville & Nashville Covered Hopper Car**
  - **Constant Force Friction-Snubbed Truck**
- **70-Ton SP 89-ft. Stac-Pac Flat Car**
  - **Constant Force Friction-Snubbed Truck**
- **Load Variations**
  - **Gross Rail Load**
  - **Empty**

Fig. 13. Test series 3 parameters.

- **70-Ton Truck, Constant Force Friction Snubbing**
- **PFE Mechanical Refrigerator Car**
- **Centerplate Friction**
  - **Low, Molybdenum Disulfide Filled Grease**
  - **Medium, Composition Disk Insert**
  - **High, Steel on Steel**
- **Pedestal Shims for Longitudinal Control**
- **Side Frame Intertie**
  - **With and Without Elastomeric Adapter Pads**
- **Hydraulic Dampers for Control of Side Frame and Bolster Transverse Motion**
- **Constant Contact Side Bearings (Experimental Pneumatic Type)**
  - **2500 lb.**
  - **5000 lb.**
  - **7500 lb.**
- **Combination**
  - **Longitudinal Pedestal Control**
  - **High Centerplate Friction**
  - **Constant Contact Side Bearing (to 9000 lb.)**
- **Load Variations**
  - **Gross Rail Load**
  - **Empty**

Fig. 14. Test series 4 parameters.

Test Series 4 was designed to test "breadboard" modifications on the 70-ton ASF truck (Fig. 14). These modifications were recommended as the result of 1970 Southern Pacific sponsored tests of the conventional freight car truck on the Japanese National Railways' dynamic test stand. The modifications tested are intended to demonstrate performance characteristics only and are not to be construed as design innovations or recommendations. The following alterations were tested using the PFE mechanical refrigerator car:

- Increased centerplate friction
- Tighter longitudinal control of wheel sets with side frame pedestals
- Side frame intertie to maintain tram with wheel sets elastically restrained
- Independent lateral control of bolster motion in the side frame
- Constant-contact side bearings

Phase I output. A variety of outputs are being generated that are essential for an understanding of the TDOP work and the economics and technical aspects of freight car truck acquisition and usage (Fig. 15). The initial literature search has been submitted to FRA for publication. Likewise, a survey of advanced truck designs, covering truck types used throughout the world, is ready for publication. A revised edition of the Introduction and Detailed Test Plans, Series 1, 2 and 3 Tests, is scheduled for publication in the near future. Detailed Test Plans for Series 4 Tests have been submitted to FRA for publication.

- **Methodology for a Comprehensive Study of Truck Economics — Report No. FRA-OR&D 75-58**
- **Introduction and Detailed Test Plans Series 1, 2 and 3 Tests — Phase 1**
- **Detailed Test Plans — Series 4**
- **Literature Search**
- **Data Tapes — National Technical Information Service Library**
  - **PB 244292/AS**
  - **PB 244293/AS**

Fig. 15. TDOP Phase I output to October 1975.

In the economic area, a report (FRA-OR&D 75-58) is published covering the methodology for use in a comprehensive study of truck economics. In this report, the major truck operating costs are identified as maintenance and repair, freight damage payments, accident costs and train delay, and lost car day costs. Methods for evaluating investments in improved trucks are discussed, and a truck economic model is described. This model is being used in current work to derive cost data bases required for the economic analysis.

As indicated earlier, digital data tapes are being furnished to the National Technical Information Service. These may prove valuable to future investigators of freight car truck dynamics.

Summary. The TDOP effort will furnish the railroads with technical and economic information on freight car truck performance. Behavioral information is required to correct existing problems and establish future truck system needs.

The results of the Phase I effort are anticipated to include technical performance specifications and an economic methodology for use in evaluating truck selection.

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Love received his B.S. degree in Mechanical Engineering from the University of Minnesota in 1962 and held an engineering position at Remington Rand Univac before joining ASF. He is a member of SESA, ASTM, and American Society of Mechanical Engineers. As an active member of ASME, he was chairman of a Chicago subsection and secretary of the Chicago section.

IMPROVED SUSPENSION FOR 100-TON CARS ON ROUGH TRACK

Introduction. Modeling and regression techniques are moving railroad design technology into a new era. FRA funding of the cooperative projects and the development of the Pueblo High Speed Ground Test Center (now the Transportation Test Center) are opening the door for major thrusts leading to a better understanding of the complex, dynamic interaction of railroad environment and vehicle design.

Over the years, American steel Foundries has operated an extensive test facility, including a rather sophisticated test train. Among other things, this facility has permitted us to identify a deteriorating railroad environment. As track roughness increases, the demand on the suspension reaches the threshold of vehicle suspension capability, i.e., its ability to handle the energy. A feedback loop (Fig. 1) is thereby generated in which input energy beyond the reserve capacity of the vehicle suspension generates high forces in cars and track alike, which in turn causes increased roughness in track profile, which in turn places greater demands on the suspension. Higher speeds exaggerate wheel hunting, which places greater demands on control of energy, as does the harmonic roll of cars at low speeds created by the geometry and spring rate of the entire system of trucks, cars, and track.

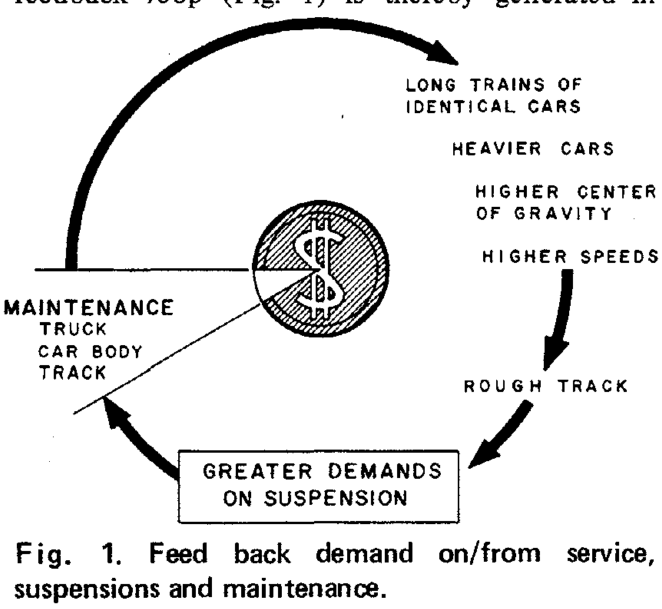


Fig. 1. Feed back demand on/from service, suspensions and maintenance.

The feedback loop established ASF's objectives for a suspension system needed to meet the demands of more severe operating conditions. Briefly, these objectives were: provide the capability to absorb and dissipate energy from all input modes over an extended life for high utilization cars, at a relatively low cost. (Unfortunately, no cost/benefit ratio could be established because no dollar figure could be attached to many of the benefits, particularly that of reduced track punishment.) Our hardware objective was defined as: a highly refined, state-of-the-art, three-piece truck, designed as a system rather than a collection of components, and tailored to today's need for a superior intensive-service 100-ton truck.

With the objectives established, it was then necessary to identify the external constraints. The truck envelope is defined by a clearance diagram relating to coupler and centerplate heights, and to car body and brake rigging clearances. The space available for the suspension components is defined by location options and the strength requirements for side frames and bolsters. Wear and fatigue life was defined, and the design made compatible with manufacturing processes and techniques.

Suspension Reserve Work Capacity. Experience with ASF's test train has allowed us to identify – and quantify – serious shortcomings in the suspensions of present 100-ton cars. Specifically, the trucks under these cars are lacking in reserve capacity under today's operating conditions. They also provide little or no control of truck hunting or car rock, shortcomings which will be discussed later.

An undercapacity suspension is depicted in Fig. 2, in which a transient input such as might arise from a switch or single low rail joint can drive the springs to solid height above some threshold speed.

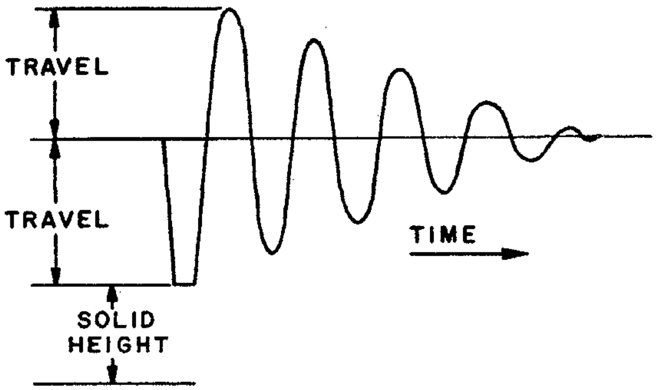


Fig. 2. Insufficient reserve capacity.

The kinetic energy applied to the system is a function of car mass and train speed squared.

$$KE = \frac{WV^2}{2g}$$

For example (and taking minor liberties with underlying assumptions), a 100-ton car moving at a speed of 50 mph might encounter a rail joint discontinuity which will impart a vertical velocity (Vv) and a vertical kinetic energy of 2,000 in.-lbs. to the car. Increasing the speed of the car to 60 mph while passing the same track joint will increase the vertical velocity to $\frac{6}{5}$ of Vv at 50 mph. Kinetic energy will increase to

$$\left(\frac{6}{5}\right)^2$$

of 2,000 in.-lbs., or 2,880 in.-lbs. Thus, a 20% increase in speed results in a 44% increase in the kinetic energy to be absorbed.

The needed reserve capacity is a function both of the reserve capacity of the springs and of the damping control of the spring action. Disturbing forces increase the total kinetic energy of a system, and control of that system requires that that energy be dissipated.

An underdamped forced vibration at resonance is depicted in Fig. 3. It represents a condition of train speed at or near the resonant speed of the truck suspension traveling over consecutive low rail joints. The amplitude is forced at each cycle to a point where springs are driven to solid height, as depicted by the clipped cycles (or peaks), and the excess energy must be absorbed in the rest of the system, i.e., truck components, car body, and track structure.

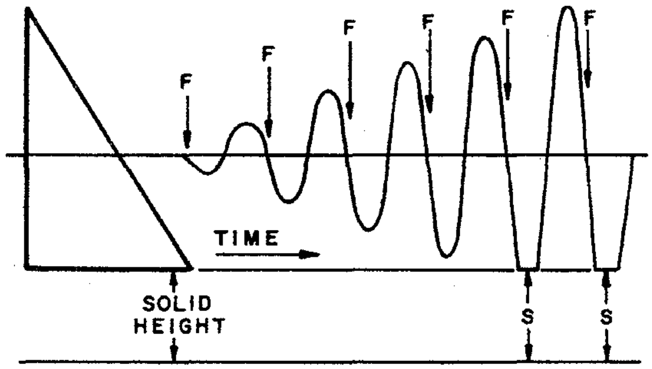


Fig. 3. Underdamped forced vibration at resonance.

Optimum Damping. Optimum reserve capacity and control would be achieved when the amount of energy forced into the system in each cycle at resonance is also dissipated in each cycle, as depicted in Fig. 4. Proper control throughout the life of the truck must be considered, however, when designing and tuning the suspension system. This requires damping somewhat above the optimum when the components are new, in order to make adequate provision for wear.

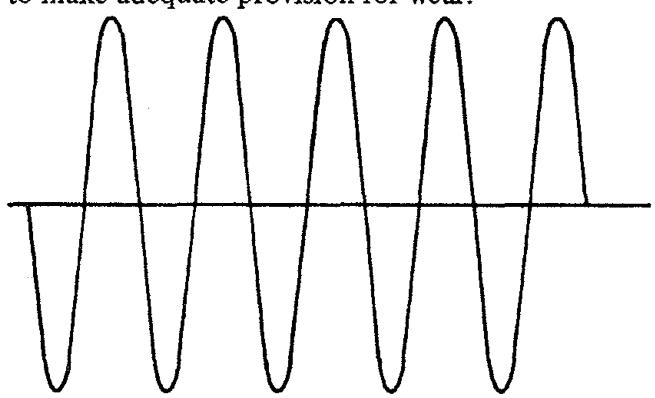


Fig. 4. Optimum damped forced vibration at resonance.

We established an optimum column load for the T-11 suspension in high-speed rough track operation, as shown in Fig. 5, and a lower threshold above which snubbing elements must be maintained for good performance on rough track. Further investigation showed very acceptable performance at a column load 15% above optimum. The design therefore utilizes the higher column load for new snubbing component conditions, passing through the optimum load at half life of the elements and finally arriving at the condemning limit at the minimum operating threshold.

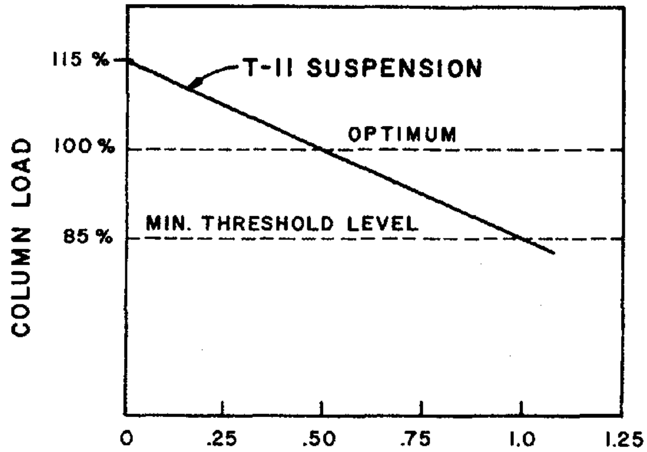


Fig. 5. Column load vs. shoe rise.

Friction damping (coulomb) is used in our improved suspension because of its low initial cost and its high degree of reliability. (Reliability was the primary consideration.) It does not, however, lend itself well to purely analytical techniques; therefore, system optimization is highly dependent on a designer's ability to experimentally evaluate proper damping levels, through repetitive use of instrumented test runs.

Design Application. Application of the basic concepts of suspensions to the design of a system for a railroad car includes consideration of the following: the static weight and peak forces imposed by the operational environment; space available for the components; suspension location options; external constraints such as coupler and centerplate heights; strength requirements of the suspension and truck components; fatigue strength imposed by materials selection; wear life; and costs.

Fig. 6 shows a comparison of our improved suspension utilizing the new D-7 spring with the best present standard suspension, which is defined herein as the conventional grouping of AAR D-5 coils and standard column loads for 100-ton equipment. The reserve work capacity is increased 31%, with a 9% lower spring rate, 15% more spring travel, and 25% more reserve spring travel.

The following oscillograms resulted from road testing and are typical excerpts from the total data package. Fig. 7 shows an undersprung, underdamped truck operating near resonance, over a rough track. Note the similarity of this travel trace and the theoretical time travel display of Fig. 3.

Springs were driven through successively greater amplitudes until solid height was reached, building the force level at the center plate in the car to nearly three times more than the static equilibrium weight (3g acceleration). The 100-ton car at that instant was in effect a 400-ton car. The

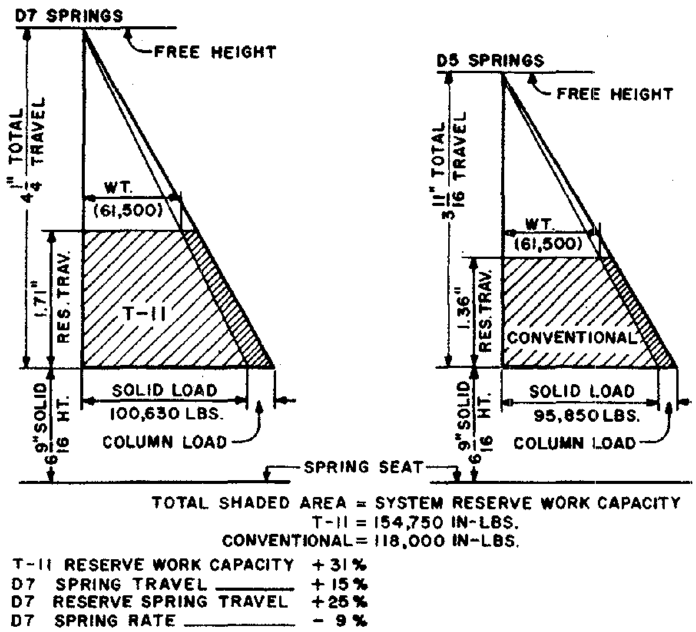


Fig. 6. Comparison of improved suspension utilizing new D-7 spring.

accelerometers were mounted on a solid steel center filler to read the direct transmission between the truck and car body. It can also be assumed, since the elastic rates of solid springs, side frames, bolsters and other components are very high, that a comparably high force is also applied to the track structure.

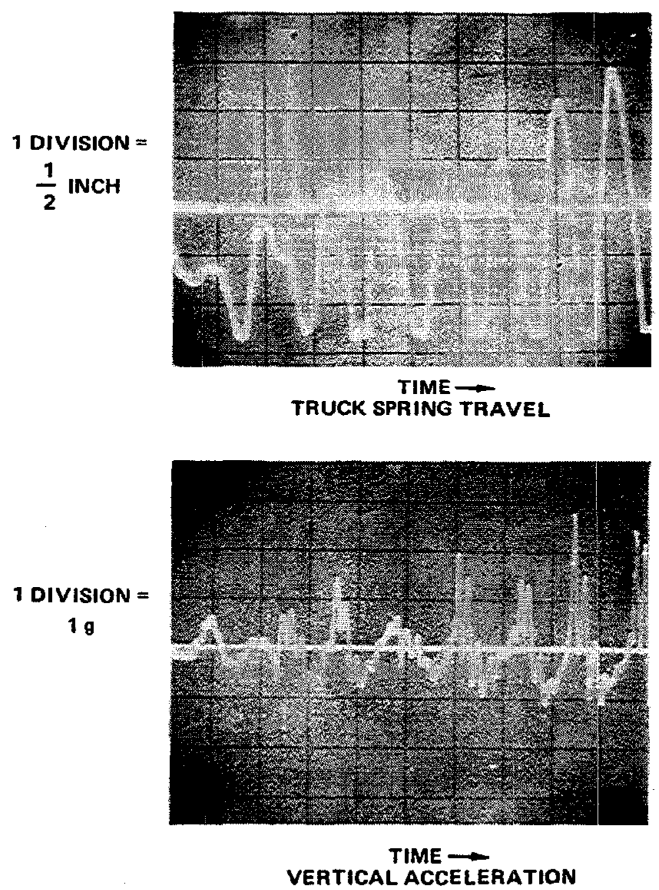


Fig. 7. 100-ton D-5 springs with standard snubbing.

Manifestations of this problem obviously will be truck component failures, failure and/or wear of body and truck centerplates, excessive settling and failure of truck springs, wheel damage from high contact stresses, car body deterioration, lading damage, and continuing degradation of track and roadbed.

Fig. 8 shows the performance of the same spring group, with the damping level increased by different control springs.

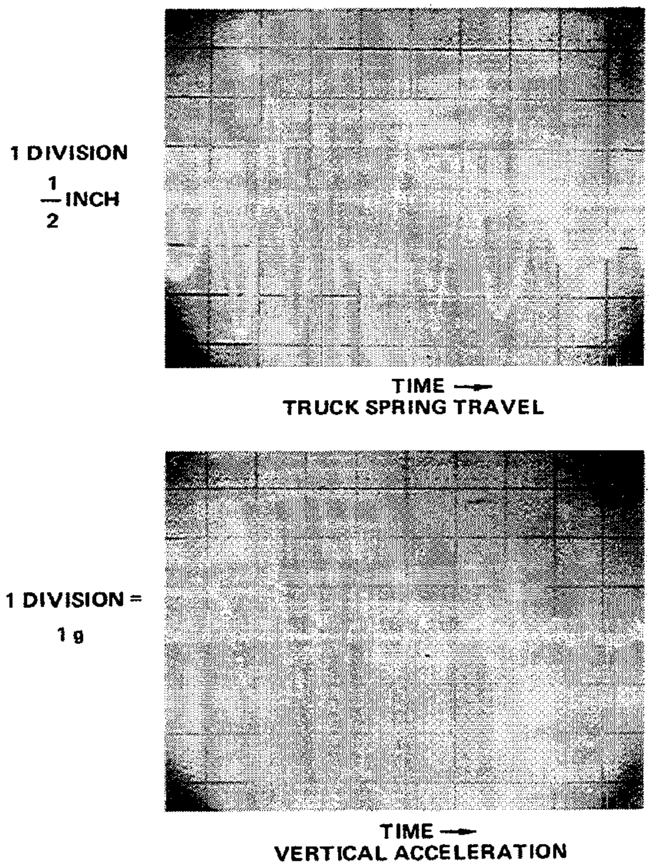


Fig. 8. 100-ton D-5 springs with improved snubbing.

The performance is that of an undersprung but adequately damped truck. Damage potential is reduced by virtue of the elimination of a number of high g occurrences. There remain, however, a number of such events wherein the g level is 3 or above – an indication that the springs are driven solid. Again, note the similarity of this travel trace with the combined theoretical time travel traces in Fig. 2 and 3.

Figs. 7 and 8 are typical examples of conventional trucks in new condition operating in today's service environment. The problem can only become worse as components wear to or beyond the threshold level of control. Higher speeds over rougher track will further accelerate the difficulties.

The results of our effort are shown in Fig. 9. These time travel and acceleration traces can be compared directly with Figs. 7 and 8. The tests were run at common speeds and with the same car on the same track segment as in the previous examples. This tuned suspension displays a greater amplitude with the longer travel, lower load-rate springs but shows no sign of springs being driven to solid height. The maximum dynamic force imparted to the car body and track structure is only 1-1/4g, as shown by the lower trace.

As mentioned, these were examples from the data package. A summary of the entire test can best be seen by applying a figure of merit we call relative Ride Quality Index (RQI). This is a measure of the ability of a suspension system to isolate lading, car body, truck components, and track structure from the damaging effects of being forced to absorb energy or to have work done upon them.

Vertical Ride Quality Summary. Ride quality indices are the result of measuring the number of occurrences of car body accelerations at discrete levels, multiplying these numbers by the squares of the acceleration levels, and summing the products. Since work is proportional to the square of the measured forces or accelerations, the result is a figure of merit for the performance of a suspension system.

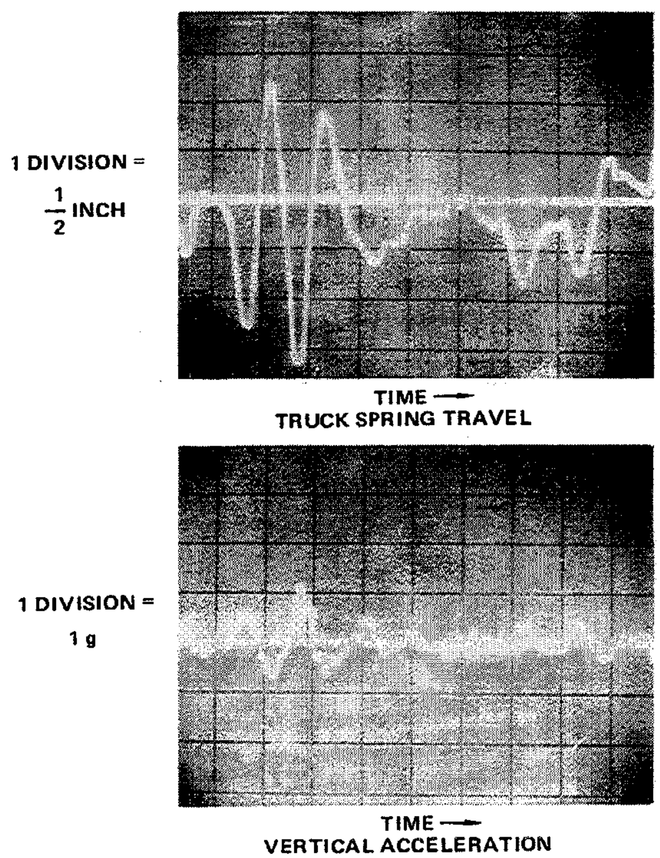


Fig. 9. T-11 suspension D-7 springs with tuned snubbing.

Acceleration Level – g's	Conventional No. of Occurrences N	Conventional g^2N	T-11 No. of Occurrences N	T-11 g^2N
.25	653	40.8125	840	52.50
.50	71	17.75	46	11.50
.75	23	12.9375	5	2.8125
1.00	14	14.00	2	2.00
1.25	6	9.375	1	1.5625
1.50	8	18.00		0.00
1.75	5	15.3125		0.00
2.00	1	4.00		
2.25	1	5.0625		
2.50	3	18.75		
2.75	0	0.00		
3.00	1	9.00		
3.25	0	0.00		
3.50	1	12.25		
3.75	0	0.00		
4.00	0	0.00		
$\sum g^2N$ (The RQI)		177.250		70.375

Fig. 10. Summary of tests comparing conventional and improved suspensions.

The tabulated data in Fig. 10 is a summary of the tests comparing the conventional and improved suspensions. The data were accumulated over a 16-mile track at speeds varying between 50 and 75 mph.

Relative RQI is the ratio obtained by dividing the test suspension RQI by the base line or conventional suspension RQI.

The improved suspension displays a 60% reduction in vertical shock and complete elimination of shocks greater than 1.25g, whereas the conventional suspension experienced shocks up to 3.5g.

Fig. 11 is a graphic depiction of the distribution of relative amounts of energy which were absorbed by the lading-car body-truck components and track structure in this test.

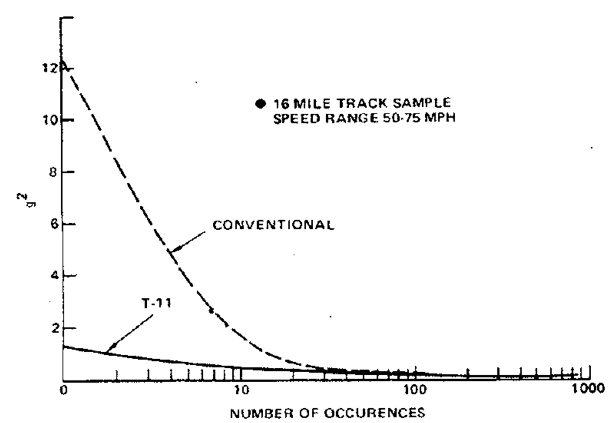


Fig. 11. Relative energy absorption by track-truck-car- body-lading system other than suspension.

Lateral Suspension and Ride Quality. The design objectives for the lateral characteristics of the system were to provide effective low-cost control of truck hunting and car rocking with reduced forces on truck components, car body, and track, and with high reliability and long life.

Car Rock Control. Every car has a natural frequency in the rocking mode which is a function of the spring rate of the overall track-to-car body suspension system and the geometry of that system. Resonance develops for sensitive loaded cars between 15 and 20 mph, and light cars exhibit a low amplitude rock on the centerplate at all speeds. These motions and resulting forces contribute to wear, damage, and derailment tendencies.

The design objectives were (in addition to providing a low-cost, dual-function control) to provide low lateral acceleration/forces, low vertical forces, and to reduce wear on truck and body components while maintaining car stability at resonance within specified limits.

Although there may be no location which is optimum in all respects for a rock control device, our studies indicate the side bearing location provides one of the best of the available tradeoffs.

Our comparative tests resulted in the lowest lateral acceleration/forces of any system tested by ASF, at the cost of slightly greater but entirely acceptable car body rock angles. The low forces are the result of utilizing the maximum deflections of the suspension system and still taking advantage of the maximum travel of the device vis-'a-vis the side bearing location.

This system with the low spring rate suspension requires more successive low rail joints to reach resonance than the conventional D-5 group -- on the order of from 13-14 versus 9-10, resulting in a better statistical chance of not operating at resonance in service.

Of benefit to the reduction of centerplate wear is the nearly complete elimination of the low-level body rock that taken place on the centerplate through conventional side bearing clearances. This is by virtue of the wide stance of the body support by the device.

Fig. 12 is one example of the many tests that were conducted.

Truck Hunting. Light car lateral ride quality rarely, if ever, presents a significant problem at speeds below 45 to 50 mph. At speeds of 50 and above, however, truck hunting enters the picture.

Wheel hunting is a phenomenon created by the tendency of coned or tapered wheels to seek an equilibrium condition while rolling along a pair of

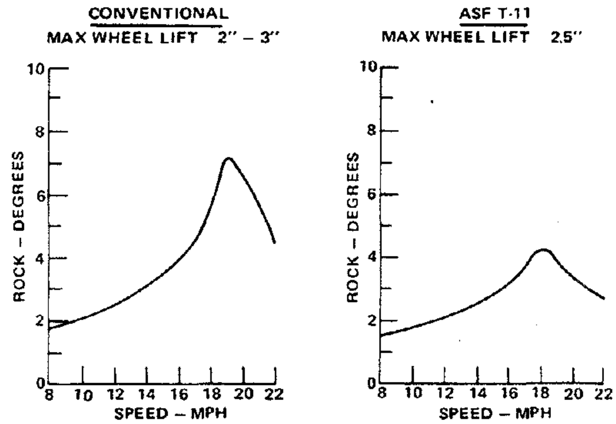


Fig. 12. Example of car rock tests.

rails. When a disturbing force of any nature causes a lateral shift of a wheel-axle set, the wheels will attempt to roll at the same angular velocity on different diameters. When this situation prevails, two simultaneous actions will occur; one or both of the wheels will be forced to slip or creep on the rail, and the axle will tend to rotate or steer away from the rail supporting the wheel rolling on the larger diameter. This steering action tends to be self-sustaining, and the wheelset, unless restrained, will hunt laterally between the rails.

In the two-axle truck, equiped with roller bearings having relatively small lateral clearances, the hunting tendency of the wheelset causes the truck frames to unsquare and the entire truck to rotate about its centerplate. As more energy is introduced to this system by higher train speeds, the friction forces inherent in the system become insufficient to adequately damp this rotational oscillation, and wheel flanges violently contact the rail head. This violent wheel hunting, accompanied by truck rotation, contributes to high flange forces against the rail and resultant wheel flange wear, derailment tendencies, and body and truck bolster centerplate wear.

Fig. 13 shows the performance of a hunting truck, illustrating the angular motion of the truck bolster with respect to the body bolster and the accompanying lateral accelerations. These data were developed on our test train with a conventional truck equipped with roller side bearings, operating above the truck hunting threshold speed. Note the corresponding lateral acceleration and rate of occurrence. (The amplitude variation on the time travel trace is the effect of primary or car body hunting.)

Displacement measurement was made between the truck and body bolsters at the side bearing location and calibrated for angular displacement. Acceleration was measured in the car body above the centerplate through a solid steel center filler.

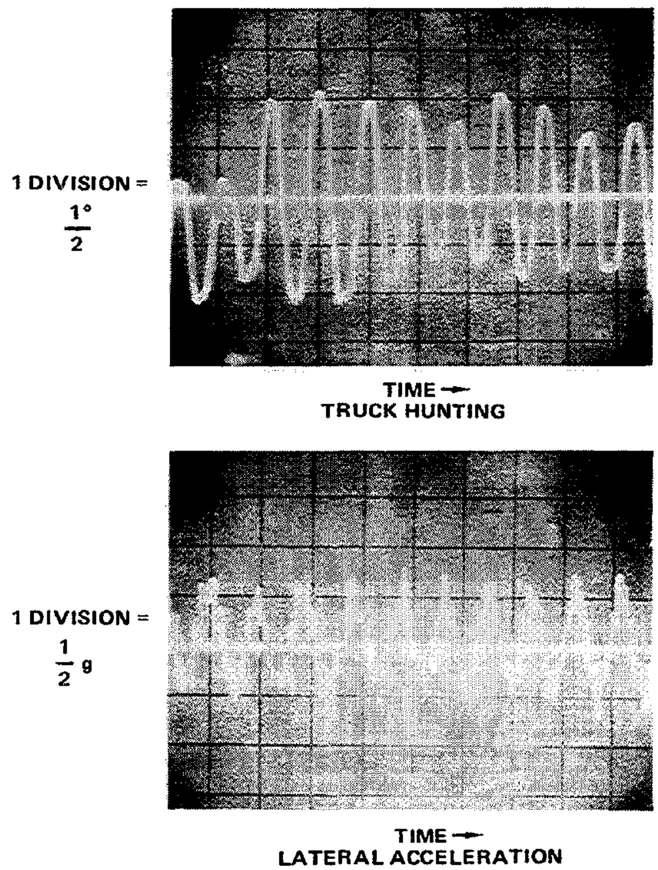


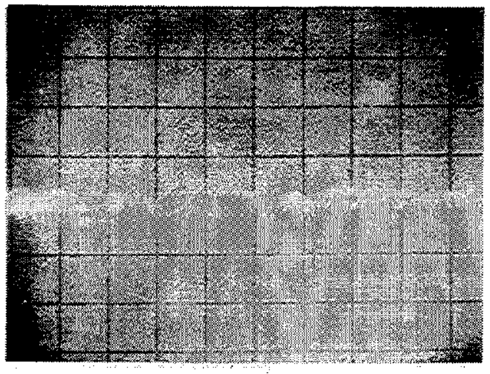
Fig. 13. Performance of conventional truck with roller side bearings.

Fig. 14 shows the performance of the T-11 Suspension System, illustrating the same displacement measurement between truck and body bolsters and corresponding lateral acceleration. The ASF Simplex side bearing provided the required control. These tests were conducted under conditions identical with those of Fig. 13, except to speeds of 80 mph. The hunting threshold speed was not reached because of the railroad speed limit of 80 mph.

Note the lateral distrubance at the beginning of the traces of both Fig. 13 and Fig. 14, which set the conventional truck into a hunting mode but had no effect on the controlled truck. This lateral disturbance is identified by a single .4g lateral acceleration in Fig. 14.

Lateral Ride Quality Summary. Using the same theory as we did for the vertical ride quality for quantifying the results, the tabulated data in Fig. 15 were used to develop lateral ride quality indices for the improved or controlled and conventional trucks.

To gather these data, both trucks were operated over a common 14-mile track in a common speed regime, varying from 50 to 70 mph.



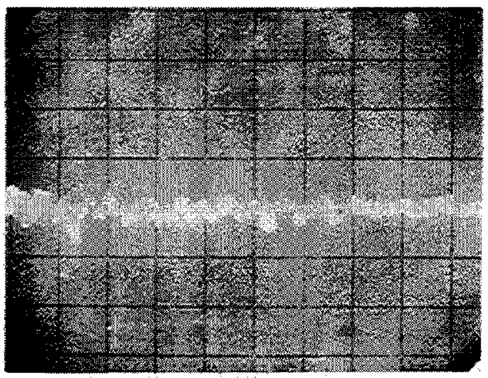


Fig. 14. Performance of T-11 suspension system with Simplex side bearings.

The controlled truck performs with a 76% reduction of damaging lateral shocks as compared to the conventional truck.

Lateral Wheel/Rail Forces. Another type of test conducted on the T-11 involved the measurement of wheel/rail forces for various operating conditions. The following data were obtained in 1974 with a calibrated wheelset at the Transportation Test Center at Pueblo.

Fig. 16 shows the positive mean lateral wheel/rail force and the relationship to speed on tangent track, and Fig. 17 shows it on curved track. The two test configurations on each graph are for a truck with conventional side bearing clearance and a truck with Simplex side bearings. The conventional truck began hunting on tangent track and continued hunting into and through the curve. Fifty mph was a self-imposed speed limit using the uncontrolled truck, since higher speeds would not have added knowledge but did entail some degree of risk.

Significant performance differences are obvious. Lateral wheel/rail forces averaged 10,200 lbs. for the conventional side bearing truck during the 50 mph run. During the 80 mph run with Simplex side bearings, the lateral wheel/rail forces

	Conventional		T-11	
Acceleration Level – g's	No. of Occurrences N	g^2N	No. of Occurrences N	g^2N
.15	2242	50.4450	1990	44.7750
.30	830	74.7000	72	6.4800
.45	367	74.3750	5	1.0125
.60	71	25.5600	3	1.0800
Σg^2N (The RQI)		225.0800		54.3475

Fig. 15. Summary of tests comparing conventional side bearings and T-11 with Simplex side bearings.

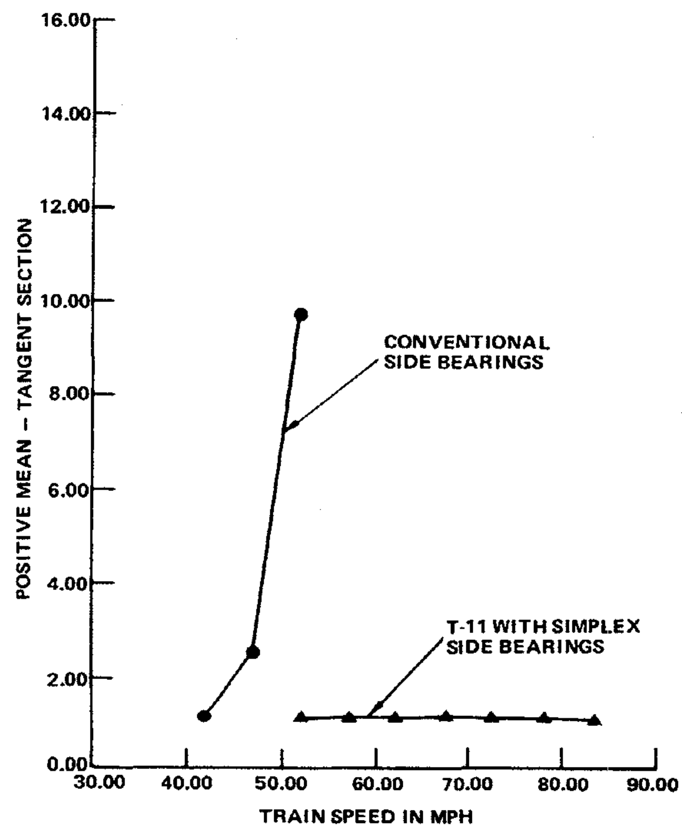


Fig. 16. Positive mean lateral wheel/rail force vs. train speed on tangent track.

averaged 1,000 lbs. on tangent track and 4,000 lbs. on curved track. The increasing force at higher speeds reflects the influence of centrifugal force.

Further interesting information from the mass of data are the maximum of peak forces and the relative number of occurrences of the force levels between the uncontrolled or conventional truck at 50 mph and the T-11 at 50 mph and 84 mph. These data, shown in Figs. 18 and 19, are for the R4 wheel, which was the outside lead wheel on the curve. Positive or plus is for flange moving toward

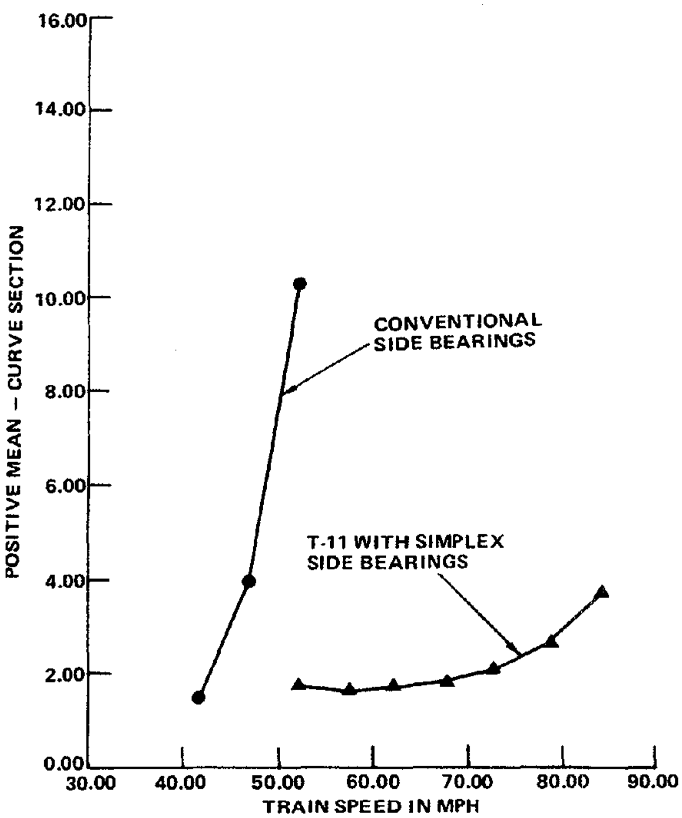


Fig. 17. Positive mean lateral wheel/rail force vs. train speed on curved track.

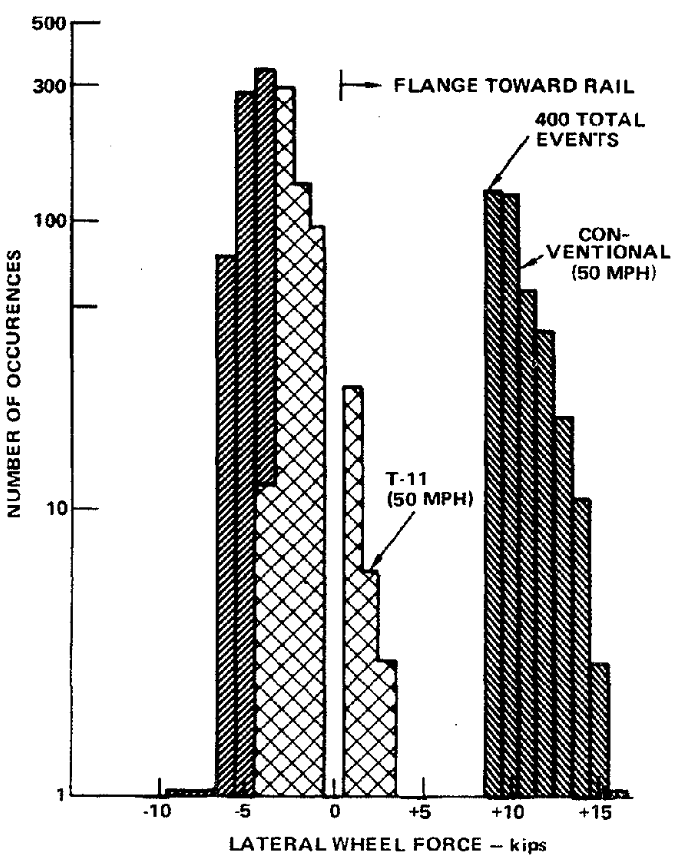


Fig. 18. Lateral wheel force vs. number of occurrences, T-11 at 50 mph and conventional truck at 50 mph.

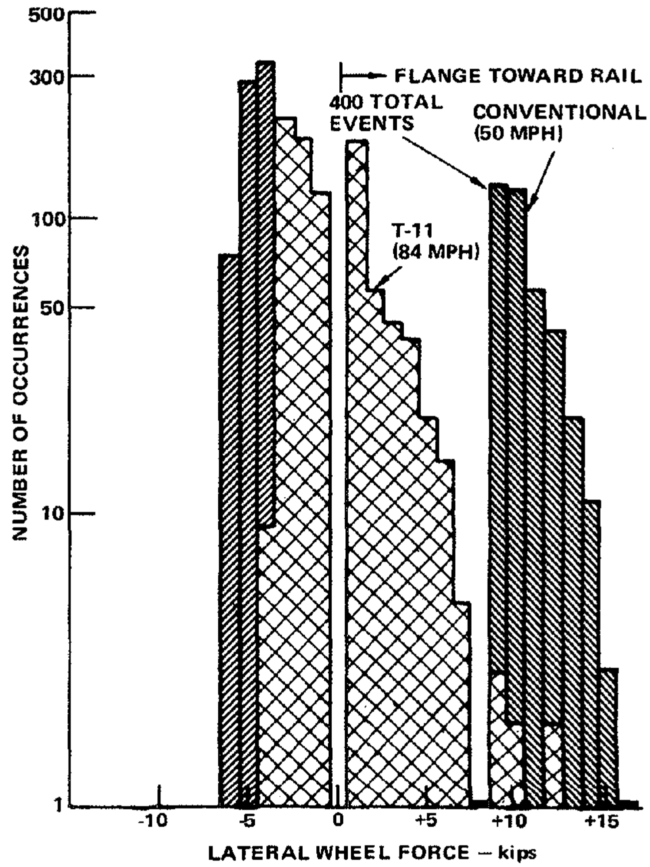


Fig. 19. Lateral wheel force vs. number of occurrences, T-11 at 84 mph and conventional truck at 50 mph.

the rail, and negative is for flange moving away from the rail. The L4 wheel data were similar.

The test track segment was 3.8 miles in length. Knowing that the conventional side bearing truck hunted throughout the test segment, approximately 400 flange contacts would be expected at 3 hertz at 50 mph. These data indicate that flange contact is made at approximately the 9,000-lb. level. Figs. 16, 17, 18, and 19 clearly show the stability afforded by the Simplex side bearing. These data also agree with the acceleration data of Figs. 13, 14, and 15 and show the Simplex side bearing can significantly reduce both lateral wheel forces and their frequency of occurrence.

Curve entry and steady state curving forces were also measured at 9 mph and 15 mph on an 8 deg. curve with no superelevation. Maximum or peak forces were approximately the same for side bearing restraint versus no restraint under light cars, indicating that curving restraint is not significantly affected by the side bearing. This is also applicable to the loaded car. A simple explanation may be due to the effect of quasisquaring or anti-lozenging control from the constant-contact side bearing, thus reducing the wheel angle of attack during curving. Future test programs are planned to determine positive explanation.

Summary. In conclusion, the benefits that can be expected for cars that operate under the various conditions discussed in this report are depicted in Fig. 20.

Reduced forces and reduced relative movement of parts in the entire system will benefit the truck components, the car body, and track and roadbed.

Service inspections of the nearly 2,000 cars in service show the system functioning as designed. Component wear is negligible after 200,000 miles. We have seen only isolated indications of solid springs, and wheel flange wear shows improvement when comparisons can be made with identical service and mileage of cars equipped with conventional trucks.

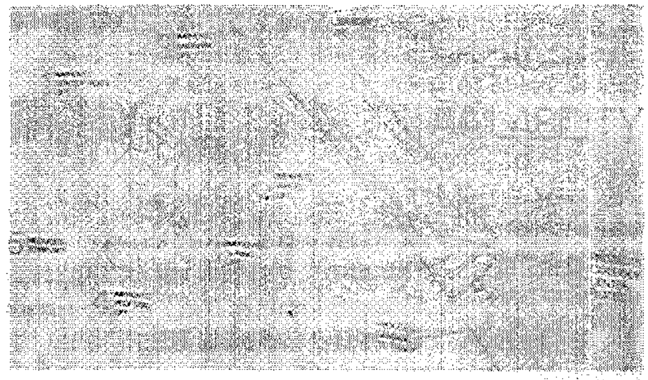


Fig. 20. Maintenance benefits.

V. Terrey Hawthorne
Director--Engineering and Quality Assurance
Dresser Transportation Equipment Division
Dresser Industries, Inc.

Terrey Hawthorne joined Dresser Transportation Equipment Division of Dresser Industries, at Depew NY in late 1974 as Director of Engineering and Quality Assurance. He completed the Pennsylvania Railroad Junior Engineer program in 1957 and subsequently served in several positions on the PRR. In 1960 he joined General Electric Company and in 1965 went with Keystone Railway Equipment Company.

Hawthorne received a B.E.E. degree from North Carolina State College in 1956 and completed several graduate courses at Syracuse University. In 1968 he attended the PMD program at Harvard University.

He is Chairman of the Cushion Unit Manufacturers Engineering Committee and a member of IEEE and American Society of Mechanical Engineers. He is a Registered Professional Engineer in Illinois, Pennsylvania, and New York.

TRUCK DESIGN - A SYSTEMS APPROACH TO SOLVING PROBLEMS

Introduction. In past Railroad Engineering Conferences there has been discussion regarding the increasing severity of problems associated with poor truck performance. In 1970 John Angold discussed the severe "parallelogramming" of trucks, which resulted in motions that cause wear of the wheels, bearing adapters, coupler shanks, and centerplates. In 1973 Leonard McLean outlined experiences Seaboard Coast Line was having related to truck performance and suggested areas to consider in future truck design. In 1974 Loren Smith demonstrated the growing "capability gap" between the truck performance provided by the present designs and the performance required in service. It would be an understatement to say there is general concurrence that present truck design does not provide adequate performance in all modern railroad operating conditions.

To establish a definition, I asked several knowledgeable railroad and supply men just what they visualized when referring to a "truck." From this investigation I found most consider the "truck" as the entire set of components from the rail to the car body centerplate and the brake pin at the live lever (Fig. 1).

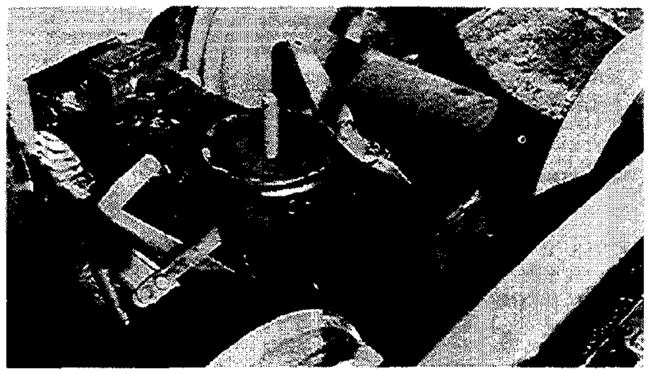


Fig. 1. Truck.

Truck Design Responsibility. A cost-effective solution to truck problems should offer an interesting challenge to the truck designer. Unfortunately, at this point, using our definition for a truck, I encountered a problem: "Who is the truck designer?" Recently I discussed the overall truck design with a car builder who indicated that in one specific instance he purchased over 20 different components from 8 companies and assembled them into a truck. He readily admitted that he was not a truck designer. He went on to express his opinion that the car builder should have more input into the truck design, since, on many occasions, dimensional constraints in the truck seriously affect the car design. An example was cited of the difficulty in holding coupler height, since truck component dimensions vary within tolerance. Thus, the car builder is not the truck designer.

Often the side frame and bolster manufacturer is considered the truck designer, but in many cases other components are modified or added to the truck, and/or the car body is considerably altered without his fully evaluating the revised truck performance. An example of this was the introduction of roller bearings and the advent of higher center-of-gravity cars. Although roller bearings provided an excellent solution to a major problem – hot boxes – it was later proven that the truck design had to be modified to accommodate the performance shifts caused by the difference in behavior between plain and roller bearings. As a result, the gib clearance was increased, as noted by the "RB" designation cast into the modified truck bolsters.

Many auxiliary components have been added to trucks in recent years to "fix" specific problem areas. Although in many cases the side frame and bolster costings were structurally modified to accommodate the new components, this was done only to satisfy the dimensional requirements of the new components. The casting vendor alone cannot be considered the "truck designer" in this case, since he simply applies his component to the system without a subsequent comprehensive evaluation of the truck performance. Likewise, the component vendor rarely can fully evaluate his device with regard to its effect on the overall truck performance and in many cases, often with justification, even disclaims responsibility for other features of the truck design. Therefore, we cannot consider either the casting or the auxiliary component supplier as the truck designer.

The wheel and axle set design has a profound effect upon truck behavior. The wheel tread contour throughout its service life and the axle diameter and tolerances are examples of areas critical to successful truck performance and reliability. Even though we accept the importance of his input we would not consider the wheel or axle designer as the truck designer.

The problem is further complicated by the AAR specifications which cover many components individually (see Fig. 2), but not the system as a whole. To obtain a complete truck or even a suspension system approval there is no formal procedure which parallels the step-by-step procedure for application for the cushion unit or draft gear approval.

It appears that we have all tended to consider the truck to be a collection of components, each of which has an independent function. We have accepted the present freight car truck design and have concentrated, each in his own field, on providing better components for this design with improved performance as a goal. This approach biases the designer, and the user away from any comprehensive design work within the existing truck system when a problem develops in a select application of the truck to an unusual type of car or operation. As a result, new problems have been solved by the application of new components which are external to the original system.

A systems approach to "total" truck design requires a clear definition of the separate subsystems. Fig. 3(a) is a Venn diagram of the car-track interface. The subsystem of the vehicle is examined in more detail in Fig. 3(b). The vehicle is considered a system comprised of three subsystems: truck, brakes and car body. In Fig. 3(c) the truck system is shown as a union of the three subsets: (1) wheels, axles, and bearings; (2) the brake components; and (3) the suspension and structural parts. The AAR committees are organized along these subsystems, with the Wheels, Axles, Bearings, and Lubrication Committee, the Brake Equipment Committee and the Car Construction Committee each contributing its input to the truck design and specification. Also, the manufacturers can be roughly categorized in this fashion.

COMPONENT	NUMBER	COMPONENT	NUMBER
AXLES	M-101	JOURNAL BOX LIDS	M-120
BOLSTERS	M-202	JOURNAL BOX SEALS	M-120
BOLSTER BOWL HORIZONTAL LINER	SECTION D	JOURNAL BOX SEALS	M-925
BOLSTER BOWL VERTICAL LINER	SECTION D	JOURNAL STOPS	M-920
BRAKE BEAMS	SECTION E	LUBRICATING DEVICES	M-918
BRAKE BEAM SAFETY SUPPORT	SECTION E	OIL, JOURNAL BOX	M-906
BRAKE JAWS	SECTION E	PEDESTAL KEY	SECTION D
BRAKE LEVERS	SECTION E	PEDESTAL ROOF LINER	SECTION D
BRAKE PINS	SECTION E	RIVETS	M-110
BRAKE RODS	SECTION E	ROLL DAMPING DEVICES	SECTION D
BRAKE SHOE KEY	SECTION E	ROLLER BEARING ADAPTORS	M-924
BRAKE SHOES	M-401	ROLLER BEARING BOLT LOCKING PLATE	SECTION D
BRAKE SHOES, HIGH FRICTION	M-926	ROLLER BEARING LUBRICANT FITTING	SECTION D
DUST GUARDS	M-903	SIDE FRAMES	M-203
GREASE, ROLLER BEARING	M-917	SPRINGS, HELICAL	M-114
JOURNAL BEARINGS	SECTION D	WHEELS, CAST	M-208
JOURNAL BEARING WEDGES	M-132	WHEELS, WROUGHT	M-107

Fig. 2. Individual truck components covered by AAR specifications.

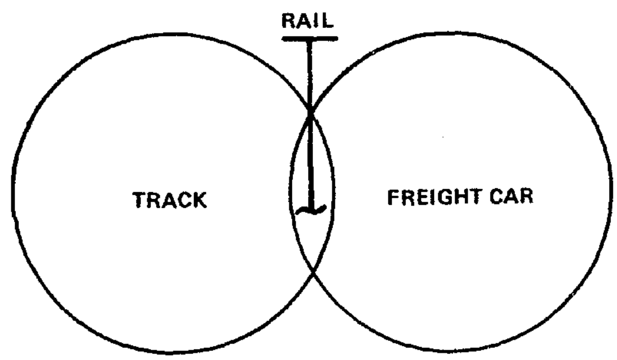


Fig. 3A. Venn diagram--track and vehicle.

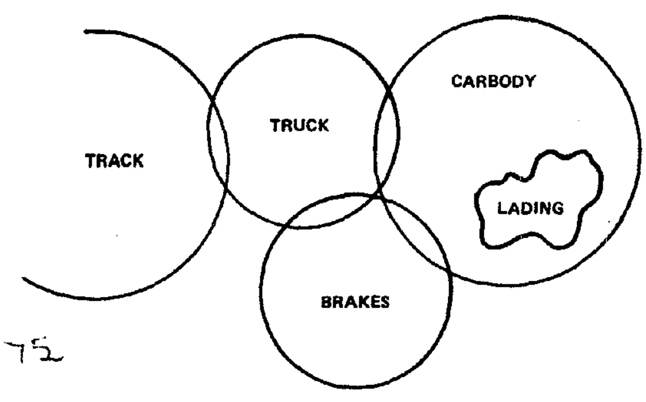


Fig. 3B. Vehicle.

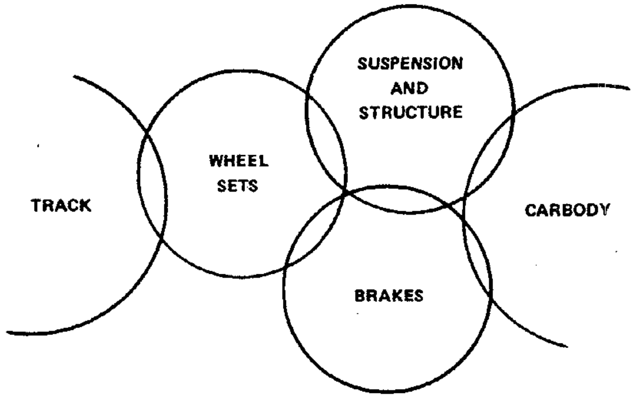


Fig. 3C. Truck.

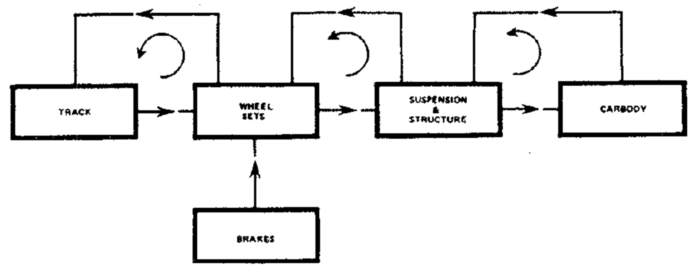


Fig. 4B. Truck subsystems.

Fig. 4(a) is a block diagram of the track, truck, and car body system. The track and the brakes apply inputs to the truck. The truck in turn passes its response to these inputs into the car body. There is a feedback path from the car body back to the truck which is filtered before returning to the track; for instance, truck hunting may excite car body yaw which can return a car body response to the truck. The feedback between the car body and brake system is ignored in this example. Fig. 4(b) divides the diagram into the truck subsystems. From a control viewpoint the track inputs drive the wheelsets, which in turn excite the suspension system rather directly, since they are tightly coupled. The suspension system provides a degree of control and passes a filtered output to the car body. Feedback paths couple the car body to the suspension system, the suspension system to the wheelsets and the wheelsets back to rail. A systems approach based upon this diagram would divide the truck design into the organizational pattern of the AAR committees and the various suppliers. The subsystem design engineers would consider the inputs imposed upon their portion of the system and design their portion of the freight car or truck to obtain the proper response. The effects of feedback both into and out of the subsystem would be evaluated and properly controlled, and a cargo carrying unit compatible with its environment would be produced.

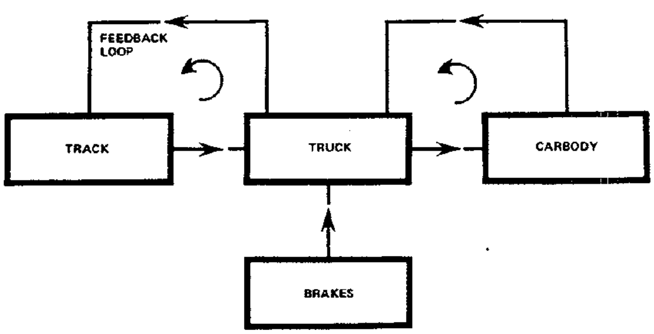


Fig. 4A. Track and subsystems of the freight car.

Unfortunately, the input parameters have not been well defined in the past, and a bias toward systems logic, which will permit a successful car to be designed, is not effective without accepted criteria for input and response and clearly defined subsystem responsibilities. As the Truck Design Optimization Project (TDOP) and Phase II of Track/Train Dynamics produce these criteria, systems approaches can be undertaken to design the complete truck to provide the required response.

Truck Problems. In solving truck problems, it is often difficult to distinguish between a problem and symptoms of the problem. An erroneous diagnosis can result in an expensive "fix" which may overlay but does not solve the problem. For example, several years ago severe premature bolster bowl and centerplate wear appeared to be a problem in 100-ton cars. As a result, a substantial amount of money was added to the cost of truck bolsters and centerplates for new materials to solve this "problems." It is now evident that the basic problem has been truck hunting and the resultant unstable performance, and the wear is simply a symptom of the problem. The application of wear plates was therefore the "solution of the symptom." Thus, we should consider the application of wear plates as an immediate remedy to alleviate the effects of symptoms of the problem and not stop working until the basic problem is solved. In order to make the truck as reliable as possible as quickly as possible we must continue to "solve symptoms," but we should recognize this as a holding action while we are developing solutions to the basic problems.

Let's examine examples of some of the problems and symptoms we have recently observed. I will separate these into two groups, performance problems and reliability "symptoms" related to performance problems, Fig. 5.

Truck Modification To Improve Reliability. As one of the organizations attacking this problem, we have applied for formal AAR approval of 5,000 car

RELIABILITY PROBLEMS	SYMPTOMS
1. LOOSENING OF COLUMN WEAR PLATES	BROKEN WEAR PLATES BROKEN WELDS LOOSE OR MISSING BOLTS
2. ATTACHMENT OF BOLSTER BOWL VERTICAL WEAR RINGS	BROKEN WELDS
3. WEAR OF SNUBBING COMPONENTS	WORN OUT COMPONENTS
4. BOLSTER RELIABILITY	FATIGUE FAILURE
PERFORMANCE PROBLEMS	**SYMPTOMS**
	RELIABILITY PROBLEMS
1. EXCESSIVE HUNTING AT MODERATE AND HIGH SPEEDS	BEARING ADAPTER WEAR CENTER PLATE WEAR BOLSTER BOWL WEAR COUPLER SHANK WEAR RAIL WEAR
	RELIABILITY PROBLEMS
2. EXCESSIVE ROLL AT LOW SPEEDS	BODY BOLSTER FAILURE WHEEL LIFT GIB WEAR SIDE BEARING WEAR

Fig. 5. Truck problems.

sets of truck components with the following new features, which address themselves to the immediate solution of reliability problems which are aggravated by severe truck hunting and harmonic roll:

1. A special attachment of the column guide wear plate, which places all the welded joints in compression to assure retention of the wear plate and avoid weldments in shear.
2. An optional design for application of an interlocking vertical wear ring in the bolster bowl, which will improve the attachment of this item by not relying upon the welds to provide shear strength.
3. Use of a wide wear plate combined with tight, rigidly-adhered-to specifications of the friction element, which will assure optimum service life of these components.

Results of the RPI-AAR Railroad Truck Safety Research and Test Project have provided valuable data regarding the environment of the truck bolster, and from this a fatigue specification will evolve to improve the reliability of this component. Using these data, combined with finite element stress analysis techniques, we will evaluate the effect of modifying sections within the bolster to locate sensitive areas and optimize the design by placing strength where needed.

Truck Redesign To Improve Performance. Thus far I have discussed reliability-enhancing improvements which can be obtained by minor modifications to the present freight car truck design. Although the long term solution to the performance problem may involve departure from the conventional three-piece truck, any major deviation from the present design will require extensive evaluation and as a result will cause undue delay in the solution of the serious performance problems. For many years the conventional truck was adequate for the then existing service requirements. Larger freight cars, faster train speeds, longer trains with increased car-to-car shocks, increased utilization, and often degrading track quality altered the inputs until the truck system could no longer provide the required response. Hopefully a relatively minor modification can be found to alter the behavior of the truck so it will provide adequate response to dampen the inputs and minimize the effects of feedback.

With regard to the performance of the truck, we are working with the Track/Train Dynamics vehicle model to evaluate possible modifications to the design and to assess the feasibility of improving the harmonic roll performance of the truck.

In the area of truck redesign to solve the hunting problem, the literature suggests two basic conceptual approaches, (1) Square Trucks and (2) Radial Trucks.

1. Square or Trammed Truck. In this case the side frames are held square to prevent "lozenging" or parallelogramming (Fig. 6). Our XL truck was an effort in this direction. The bolster was interlocked with the side frames in a manner to permit an equalizing motion over uneven track, while maintaining a tramming influence.

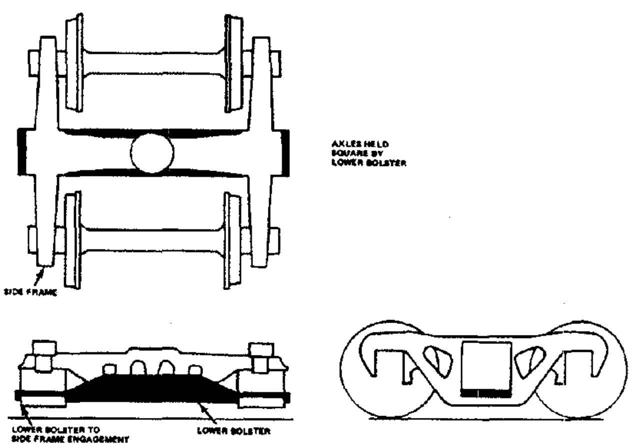


Fig. 6. Square or trammed truck.

2. Radial Truck. In a radial truck the longitudinal centerline of each axle follows the radius of the curve (see Fig. 7). At the Railroad Engineering Conference last year Herb Scheffel of South African Railways presented a paper in which he described a design of a radial truck. Recently, Dresser Transportation Equipment Division negotiated an agreement with Railway Engineering Associates for another concept

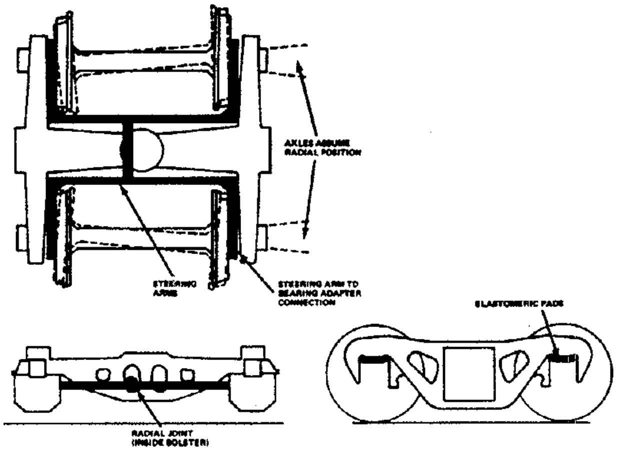


Fig. 7. Radial truck.

of a radial truck. By modifying a conventional truck (Fig. 8), with the addition of steering arms, the radial positioning of the wheel and axle sets can be achieved. If our experience with this adaptation of the concept to a conventional truck is as successful as preliminary testing indicates we intent to continue by developing a more sophisticated truck. Harold List, President of Railway Engineering Associates, will present this concept in the next address.

Next month the FRA Transportation Test Center will be making a series of tests to evaulate the performance of this radial truck concept. You will see this truck under the Dresser test car on your tour of the facility, Fig. 9.

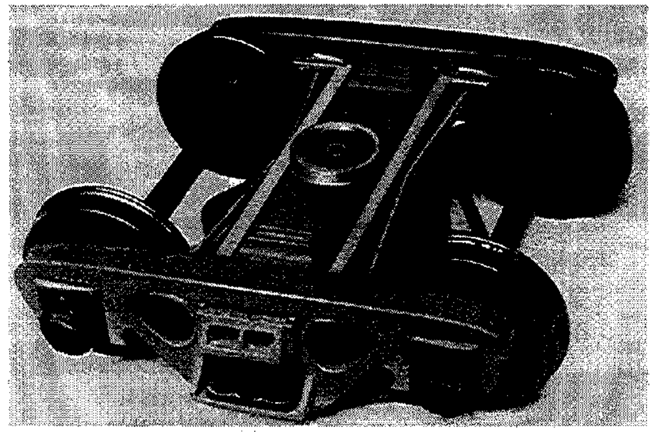


Fig. 8. Radial truck.

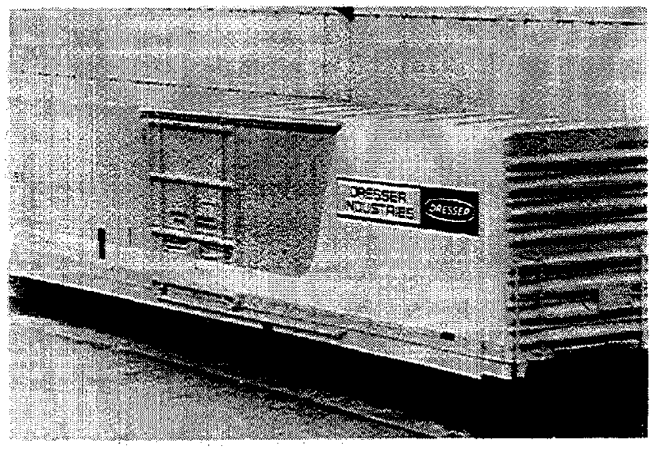


Fig. 9. Dresser test car.

Conclusion. As you can see, we are addressing ourselves toward a solution of truck problems. In doing so we intend to maintain a systems outlook in the evaluation of truck designs by studying the inter-relationship of each component in the system to the overall response.

Harold A. List
President
Railway Engineering Associates, Inc.

H. A. List is President of Railway Engineering Associates, a company organized specifically to develop railway car trucks. He received his B.S. in Mechanical Engineering from Carnegie Tech in 1946 and an M.S. in Mechanical Engineering from the University of Pennsylvania in 1950.

List served with the 765th Rainway Shop Battalion during World War II and was an engineering trainee at the Baldwin Locomotive Works immediately after the war. While teaching in the Towne School at Penn, he became interested in servo theory and the field of automatic control systems for large power plants and steel mills.

A member of ASME and ISA, he has authored several papers and holds patents in both the automatic control and railway truck fields.

DESIGN SYSTEM APPROACH TO PROBLEM SOVING

Introduction. At the April, 1971 ASME-IEEE joint meeting, a paper was presented outlining the work done by Railway Engineering Associates on the causes of, and possible solutions to, the poor tracking performance of the current three-piece truck under certain operating conditons.

Since that time considerable work has been done in the area of prototype testing, substantiation of suspected causative factors and validation of mathematical models of proposed truck design concepts, and this paper is intended as a progress report on the work to date.

Background. Specifically, the major thrust of the work that has been done by R.E.A. and Canadian National Railways has been to identify causes of freight car truck hunting at high speeds on tangent track and the high rate of wheel wear associated with operating heavy cars in curves.

The C.N.R. research department has conducted many field investigations of problems such as truck hunting, gage widening in curves, rail corrugation and wear, and wheel flange wear.

Railway engineering associates' investigations into truck mechanics with large scale models has resulted in some of the more promising configurations being designed and built for full scale prototype testing.

To date, R.E.A. and C.N.R. jointly have produced three mathematical models and tested two sets of prototype hardware which have, thus far, shown good correlation with the full scale testing results.

A simple summary of the work to date can be shown in the following three (3) slides, the first of which illustrates the problem, slightly exaggerated for clarity. Fig. 1 shows the angle of attack of the wheels of a conventional roller bearing truck going into a curve in a lozenged condition.

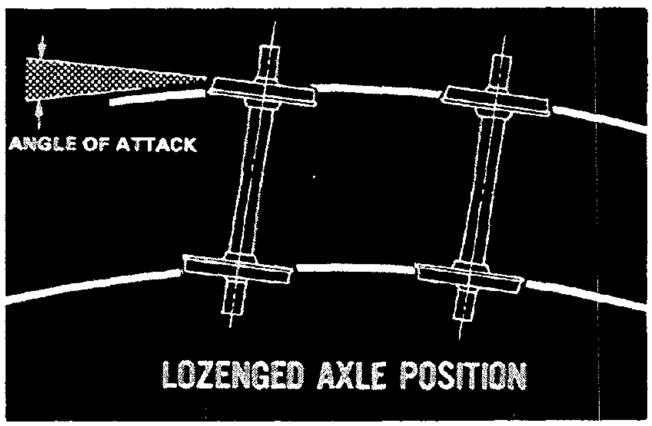


Fig. 1. Lozenged axle position.

Fig. 2 illustrates the reduction that can be achieved in the angle of attack by providing a truck with positive tramming means to prevent lozenging.

Fig. 3 shows the drastic reduction in the angle of attack that can be achieved by causing or allowing the wheel and axle sets to assume a radial position relative to the track curvature.

Obviously, the radial axle position must be our research objective and our efforts are being concentrated in arriving at a practicable design to effect implementation of the principle. (A film was run to show basic radial truck action.)

One of the constantly recurring obstacles one meets with in freight car and truck design is the

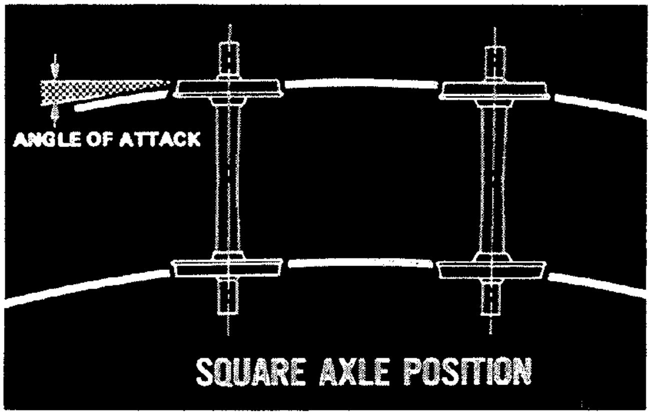


Fig. 2. Square axle position.

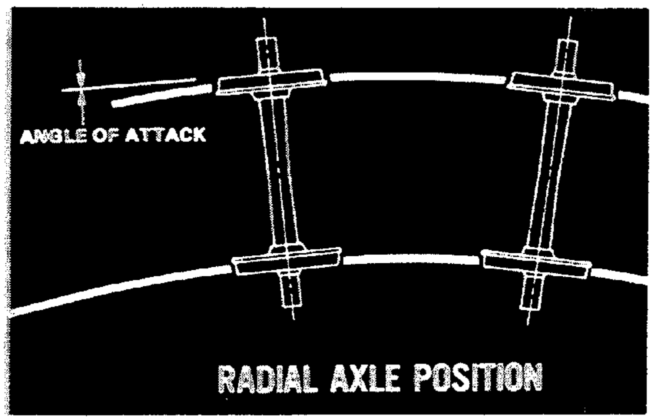


Fig. 3. Radial axle position.

very large difference in loaded and tare weight and, here again, it is a large factor in the design for radial tracking as can be seen in the computer and physical test results.

Fig. 4 is a list of the mathematical models that have been developed.

Mathematical Models:

Steering Stability. This model predicts the dynamic steering stability of a two-axle railway truck.

Flange Wear. This a model for explaining flange wear in curves as a function of wheel loading and truck geometry.

Suspension Parameters. This model provides the means for predicting the effects of various suspension parameters on the dynamic response of a railway car to irregularities of track sufface and line.

Prototype Models: Two carsets of 100-ton and one carset of 70-ton prototype trucks have been built.

Fig. 5 shows the initial 100-ton prototype, what we now refer to as the Phase III design and it represents a substantial departure from the conventional three-piece truck.

The other carset of 100-ton trucks, and the carset of 70-ton trucks have been built to incorporate the steering mechanism into a conventional truck configuration as shown in Fig. 6.

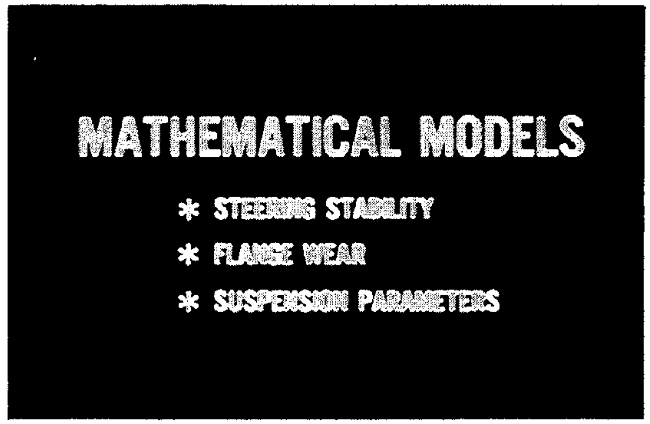


Fig. 4. Mathematical models.

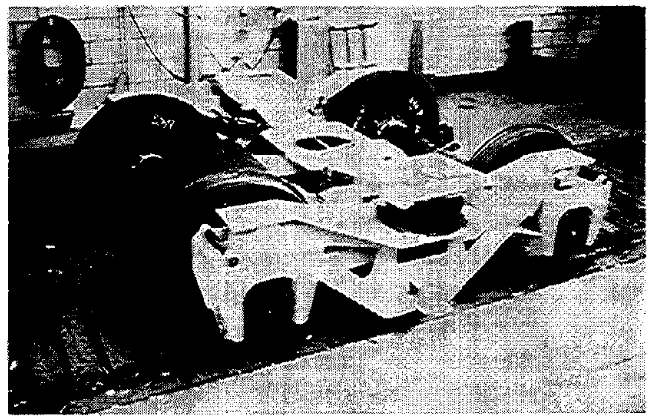


Fig. 5. Phase III radial truck.

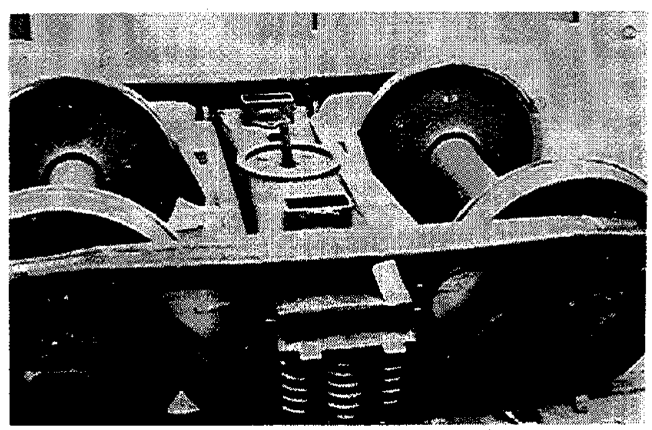


Fig. 6. Phase I radial truck.

Truck Steering Mechanics. Our early studies and the literature show clearly that the critical speed for truck hunting decreases as the wheel tread profile wears to better fit the profile of the rail head. All of our work has been based on the use of a "worn wheel" profile. Fig. 7 shows the profile used in our tests and this profile in contact with the rail is depicted in Fig. 8. For comparison, Fig. 9 shows a standard A.A.R. profile with its initial two points of contact with the rail.

Early experiments with models showed the value of steering motion of the axles. This led us to evaluate design alternatives in terms of the restraints on yaw and lateral motions between the two axles of the same truck. In general, we have found that the yaw stiffness within a truck can be reduced to give greatly improved curving if the lateral stiffness is made high enough. In addition, we have found that the deployment of the restraints on these motions is important.

While we have devised many alternative constructions which provide the desired combination and deployment of these two parameters, we have elected to evaluate only the simplest in full scale prototype form. The test work to date indicates that the simple forms utilizing a minimum of yaw freedom will be adequate for freight cars in service on most mainline trackage.

It may be necessary, however, to provide a greater degree of yawing capability in trucks that are to operate in service where there is a large percentage of curved trackage.

Theoretical considerations and test data indicate this to be the case. It further indicates that the radial principle could be used to advantage on passenger cars, rapid transit cars and locomotives where noise suppression is an important factor.

Performance. The parameters of the prototype trucks were chosen to give a critical speed of 100 mph on a worn wheel profile. The performance of the full size prototype indicates that the mathematical model of dynamic stability is essentially correct. The prototype has been run at speeds up to 77 mph with no trace of instability.

(A film was run to show high speed performance in CN Tests.)

Fig. 10 shows a comparison of predicted and measured curving behavior of the prototype truck under loaded car conditions.

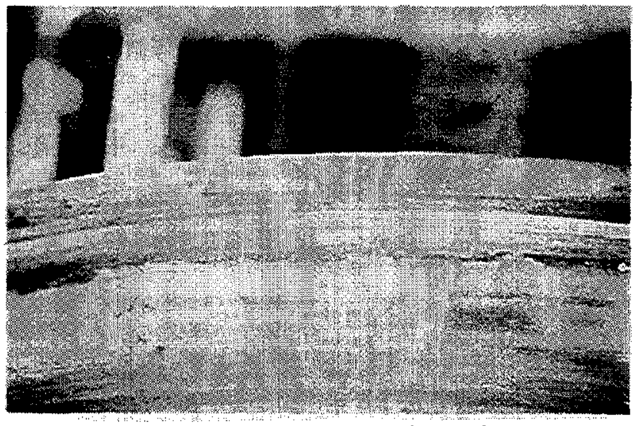


Fig. 7. Worn wheel tread.

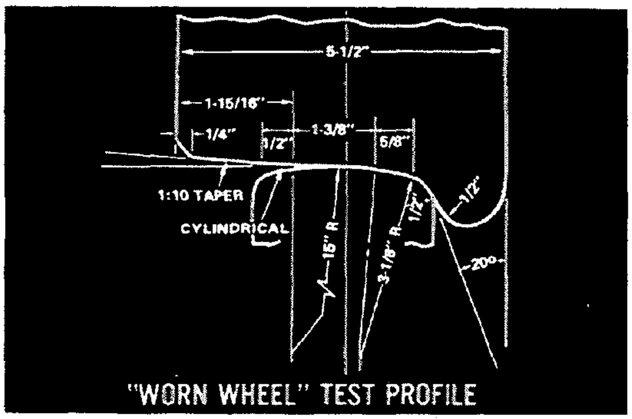


Fig. 8. Worn wheel test profile.

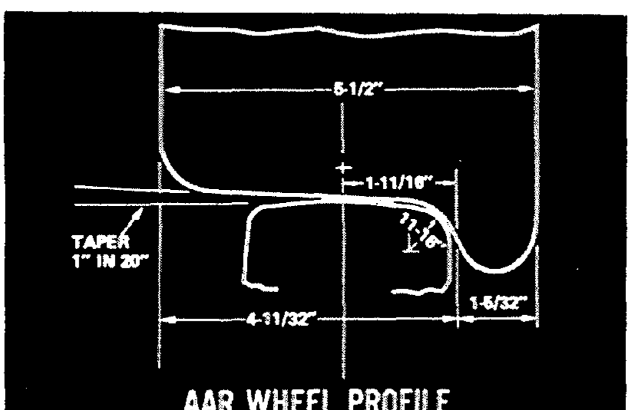


Fig. 9. A.A.R. wheel profile.

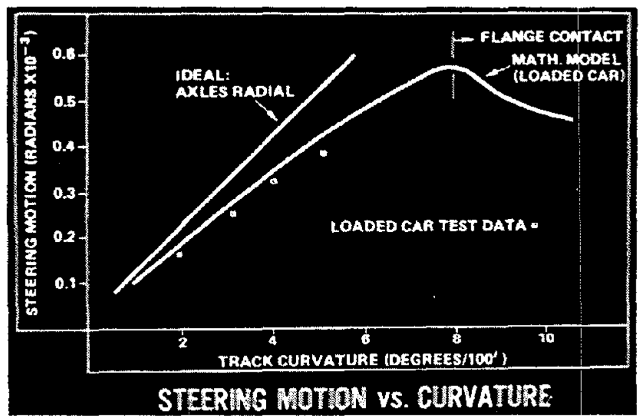


Fig. 10. Predicted and actual curving characteristics--loaded car.

Earlier tests made in the Lehigh Valley using instrumentation shown in Fig. 11, indicated radial positioning of the axles up to 12 degrees of track curvature, as depicted in Fig. 12. These tests were made with lower values for inter-axle yaw stiffness than the tests shown in Fig. 10 for loaded car conditions.

I don't propose to go into great detail in the next three slides, the first of which, Fig. 13, shows a computer-generated plot of constant damping or stiffness characteristics, using various damping factors plotted against inter-axle lateral and inter-axle yaw stiffness parameters. The second, Fig. 14, shows a plot of constant curving characteristics against the same stiffness

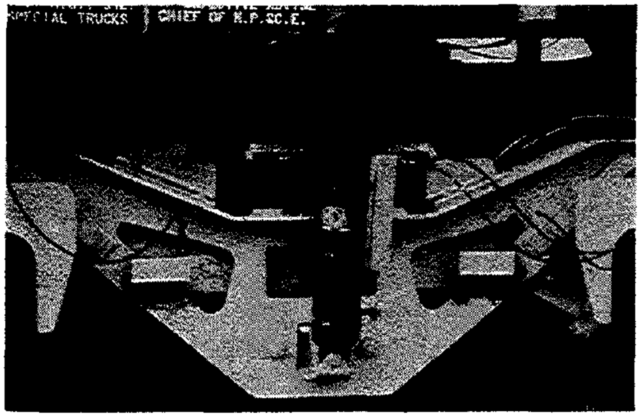


Fig. 11. Phase III radial truck test instrumentation.

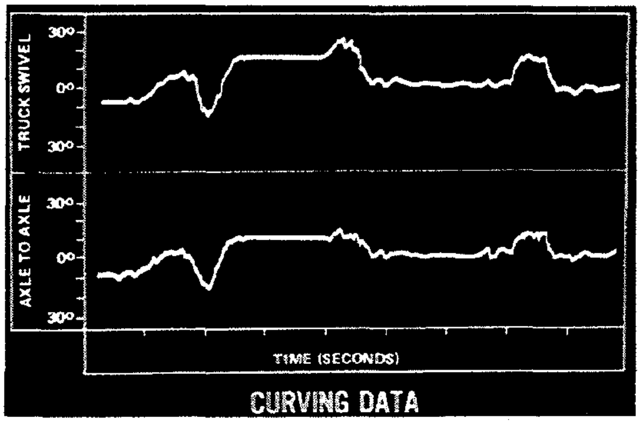


Fig. 12. Measured response of wheelsets during curving.

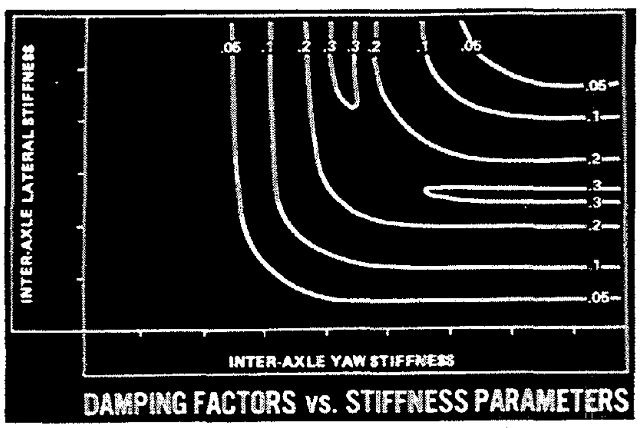


Fig. 13. Truck curving characteristics vs. inter-axle lateral stiffness.

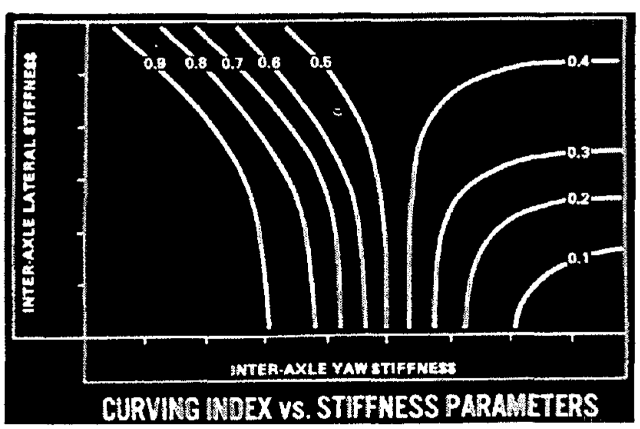


Fig. 14. Truck curving characteristics vs. inter-axle yaw stiffness.

parameters, and the third, Fig. 15, shows the combined result, plus the domains occupied by the various trucks. The conventional three piece truck is in the lower right hand corner, the square, or trammed, truck in the rectangular area above it, and the radial truck in the area near the center of the graph, which indicates the substantial improvement possible with radial steering.

Justification. The performance improvements reported previously will lead to savings in wheel wear, track wear, general car maintenance, and lower operating costs that should readily justify the additional cost of providing the steering feature.

It is not difficult to describe technical justification for the radial truck concept; the difficult part is relating economic values to the total justificationicture. We are hoping that the T.D.O.P. program will be successful in its efforts to provide data on the cost of truck ownership, or a method of determining such costs, to give perspective to the real value of engineering innovations in the truck tracking area.

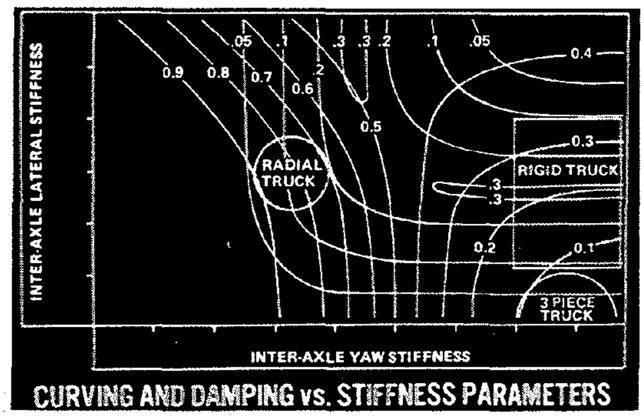


Fig. 15. Truck curving performance index.

With regard to track maintenance, we know that, particularly in curves, gage corner wear has been found to depend strictly upon wheel loading, curvature and total traffic. We also know that the curvature term is analogous to the angle of attack term affecting wheel flange and rail wear, also that with a conventional truck the angle of attack is high.

We know that gage widening occurs when the lateral force on the track fastenings exceeds their holding power. The lateral forces are a function of wheel loading and angle of attack which are both high, particularly for 100-ton cars.

Also that when gage widening occurs, the outside corner of the wheel profile can bear on the top of the rail, causing severe stress concentrations which, in turn, cause cold flow of the rail metal.

In addition, truck hunting which is common with conventional trucks on tangent track inflicts high dynamic loading on the track fasteners, causing them to loosen and, occasionally, to fail.

We do not know, however, how to relate all of this to the cost of providing a radial capability in a truck, which should significantly reduce these problems.

The same difficulty in economic justification applies to all of the following:

Truck Maintenance

Wheel Flange Wear. This has been found to depend directly on wheel loading, angle of attack and distance traveled. In the case of the conventional three-piece roller bearing truck, the angle of attack term is particularly high.

Center Bowl and Side Frame/Bearing Adapter Wear. The yaw motion of the bolster and wheelsets during truck hunting causes high wear rates for the center bowl of the bolsters and the surfaces of the side frame and the bearing adapters.

Car Maintenance

Center Plate Wear. High rates of center plate wear are caused by truck hunting and car rocking.

Impact Damage during Humping. The coupling speed in many hump yards is controlled by a computer which estimates car rollability as the car first proceeds down the hump. Variations in rollability which occur after the estimate is made introduce variations into the car's speed at coupling impact, making it impossible to attain the precise control of speed needed to keep coupling impact within reason.

Structural Fatigue. The frequency of occurrence of high values for lateral and vertical accelerations in the car body leads to fatigue failure of components and fasteners. Truck hunting and car rocking are the chief contributors to this problem.

The Proposed Solutions. Three proposals for the freight car are described in the following. The first provides the benefits of steering for existing roller bearing freight cars. The second, which requires a minor modification of the bolster and side frame castings, provides improved brake rigging as well as steering. The third provides additional improvement to tracking and ride quality, and possesses other advantages that may warrant a complete review of car design changes that could be made possible.

Type I, Retrofit of Existing Three-Piece Roller Bearing Trucks

Description. The Type I truck is obtained by adding a "steering assembly" kit to an existing roller bearing truck; see Figs. 16 and 17. This will provide the combination of lateral and yaw stiffness required for high speed stability and limited steering action.

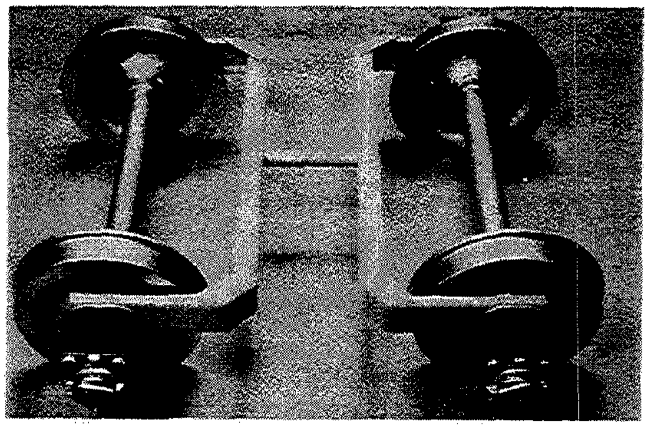


Fig. 16. Model of steering arm application.

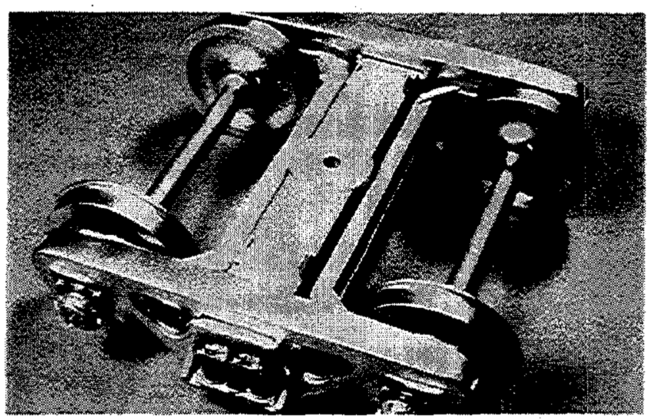


Fig. 17. Model of truck with steering arms applied.

Type II, for New Cars Built in The Near Future

Description. The Type II will modify the design of the castings for the three-piece truck to provide space so that the "steering assemblies" can provide more steering motion than in the Phase I design and be made strong enough to support the brake beams.

The incentive for improved mounting the brake beams from the steering arms is more precise location of the shoe across the face of the wheel tread, which accrues the economic benefits of longer wheel life, more efficient use of brake shoes and improved braking.

Type III, An All New Truck for The Future

This truck would be a somewhate radical departure from current practice, both in its basic configuration and in its interface with the car body.

Future Designs. While only freight car trucks are discussed here, it should be noted that passenger cars, locomotives and transit cars should be equipped with trucks having a steering feature. In the case of passenger cars, the attractions are the increased safety margin provided by the reduction in flange forces and the improvement in high speed stability which can be achieved with far less wheel and truck maintenance. In the case of locomotives, there is, in addition, the gain in adhesion associated with eliminating the lateral movement of the drive wheels across the rails. Transit vehicles can benefit from all of the foregoing, plus a major reduction in operating noise.

Conclusions. To date, there is a sufficient amount of generated analysis and test data on hand to establish that the basic steering concept is sound and that it is so far, a practicable method of controlling the tracking characteristics of a basic three-piece truck.

It is evident that what is now needed is the development of test data on a qualitative basis and this is being undertaken with tests projected that will make direct comparison of the radial trucks and conventional trucks under controlled conditions. Fig. 18 shows the Dresser Transportation Equipment Division 60' box car, DTEX-109, which you will see tomorrow with the test trucks installed for testing at the test track and scheduled to begin in a few days time.

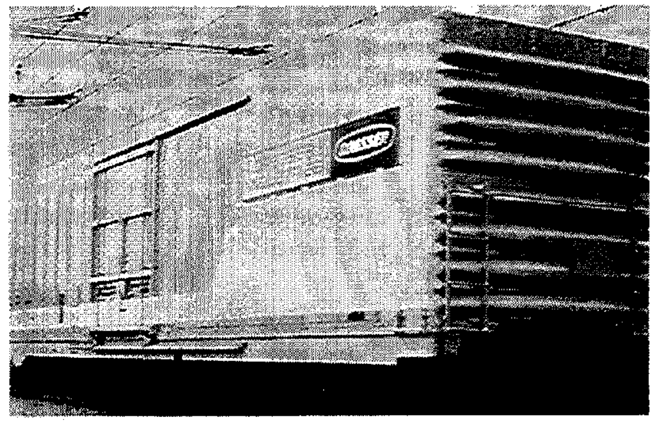


Fig. 18. Dresser test car to be used in radial truck test program.

It is anticipated that from data generated, it will be possible to determine with some degree of certainty the extent of the economics involved, at least on the same comparative basis.

Robert L. Bullock
Manager—Research and Development
Standard Car Truck Company

Robert Bullock is Manager of Research and Development for Standard Car Truck Company, Chicago. A native of the Chicago area, he received a B.S. degree in Mechanical Engineering from Illinois Institute of Technology in 1966. He is a Registered Professional Engineer in Illinois and a member of the American Society of Mechanical Engineers.

Before Bullock joined Standard Car Truck as Senior Project Engineer in 1970, he worked for Stewart-Warner Corporation as project engineer of mechanical gauges used in the heavy equipment industry. Since then, he has been engaged in truck and truck suspension design and presently is project consultant for Track/Train Dynamics Phase II validation tests.

MODIFIED THREE-PIECE TRUCK REDUCES HUNTING AND IMPROVES CURVING — STATUS REPORT

The three-piece cast steel freight car truck has two definite advantages which should not be overlooked before verdict of "guilty as charged" is returned for its inability to meet many of the demands of today's operating environment.

First, and possibly the most minor point I could make today, is that the three-piece truck is relatively easy to manufacture, which keeps it somewhat inexpensive when compared to other design approaches.

Its second advantage, and from an operational viewpoint, an important advantage is that the three-piece truck has excellent load equalization which allows it to negotiate large changes in cross elevation found in poorer turnouts and terminal yards. Heretofore, the dual demand upon the freight car truck of stability at higher speeds and improved curving ability, ideally flange free guidance, have been, to say the least, conflicting. Based on a lot of previous and on going research and experience, it has been found that whatever is done to improve stability at high speeds decreases the truck's curving ability. We are all familiar with various design approaches to achieve these two demands. High wheel set yaw constraints or more rotational resistance between the car body and truck, lower effective wheel conicity (1/40 taper and cylindrical wheel) for high speed operation. Conversely, whatever is done to improve the curving ability of the truck hurts or decreases stability, primarily higher effective conicity for better curving decreases stability at high speed.

This afternoon, I would like to bring you up to date on a design approach that its inventor, Mr. Herbert Scheffel of the South African Railroad, introduced at last year's conference. This design, referred to as the Anchor Truck, is a modification to a three piece freight car truck which makes the curving demand and high speeds stability demand compatible. That is, this truck has increasing body stability with increasing wheel profile conicity and good wheel set stability with low wheel set yaw constraints, both of which together allow guidance through curves, using the creep forces alone, allowing flange free guidance.

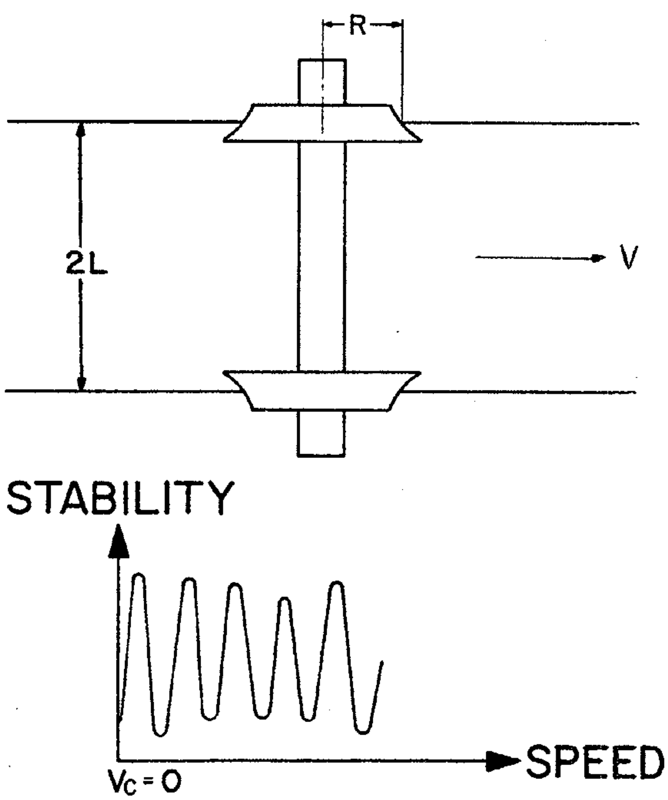
I would like to take a few minutes now to review the analysis that Mr. Scheffel conducted that finally resulted in a computer model having 17 degrees of freedom that was used to design several prototypes in both South Africa and here in the United States. The test results of these prototypes have been used to validate the computer model.

Many investigations into hunting stability of railroad cars–have been based on creep theory. Very simply, this theory states that when the wheel set is displaced from a position different from its position for pure rolling, forces are generated in the content area between the wheel and the rail. Using this theory in its linear form, the following analysis can be made:

To help clarify this analysis, I have made these following diagrams. Figure 1, shows a single wheel set rolling down rail. It is known that this system is unstable for any speed with its frequency increasing with speed.

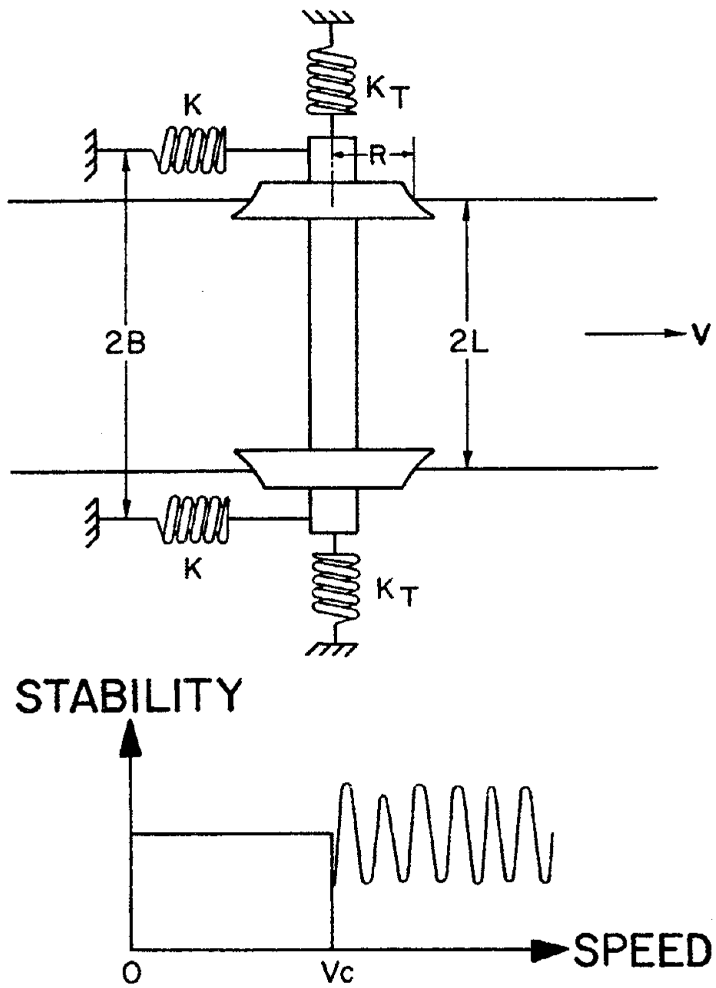
In Figure 2, a single profiled wheel set is elastically constrained to ground in the lateral and longitudinal direction. It can be shown using the linear creep theory that the wheel set now can be made stable up to a critical speed and is unstable for all higher speeds. For wheel profiles and rail profiles that are of practical interest to the railroad, the motion of the wheel sets are influenced considerably by non-linearities. However, roller rig experiments along with several previous analyses show that the critical speed is directly proportional to the square root of both the lateral and longitudinal constraints.

Figure 3 shows the profile wheel set elastically constrained to a mass. The surprising result here is that it can be shown by analysis using linearized creep theory that this system has no stability whatsoever no matter what the values of the lateral and longitudinal elastic constraints have. The conclusion one reaches for a single wheelset constrained to ground, do not aply to a single wheel set constrained to a mass having lateral freedom.



$$\text{FREQUENCY} = \text{SPEED}\sqrt{\frac{\text{EFFECTIVE CONICITY}}{R \times L}}$$

Fig. 1. Single wheel set frequency.



$$V_C = \left[\frac{LR}{\text{CONICITY}}\left[\frac{K_T}{M} + \frac{KB^2}{ML^2}\right]\right]^{\frac{1}{2}}$$

Fig. 2. Critical speed of a single wheelset.

In Figure 4, we have taken Figure 3 and added another wheel set constrained to the same mass both in longitudinally and laterally. It can now be shown both in roller rig tests and mathematical analysis, that this system can have lateral stability to a given critical speed. This critical speed is dependent on the various constraints between the wheel sets and car body. A very important point that led to the Anchor Truck development is that one way to think of wheel set stability for a vehicle is that the wheel sets obtain their stability by being suspended to each other via the vehicle's frame.

Investigating this arrangement further, the results are now similar to the case where one wheel set is constrained to ground. That is, by increasing the yaw constraints stability is improved. However, when the yaw constraint is to a vehicle's mass, the improvement is less than when the constraint is to ground. This approach, as can be seen in the next

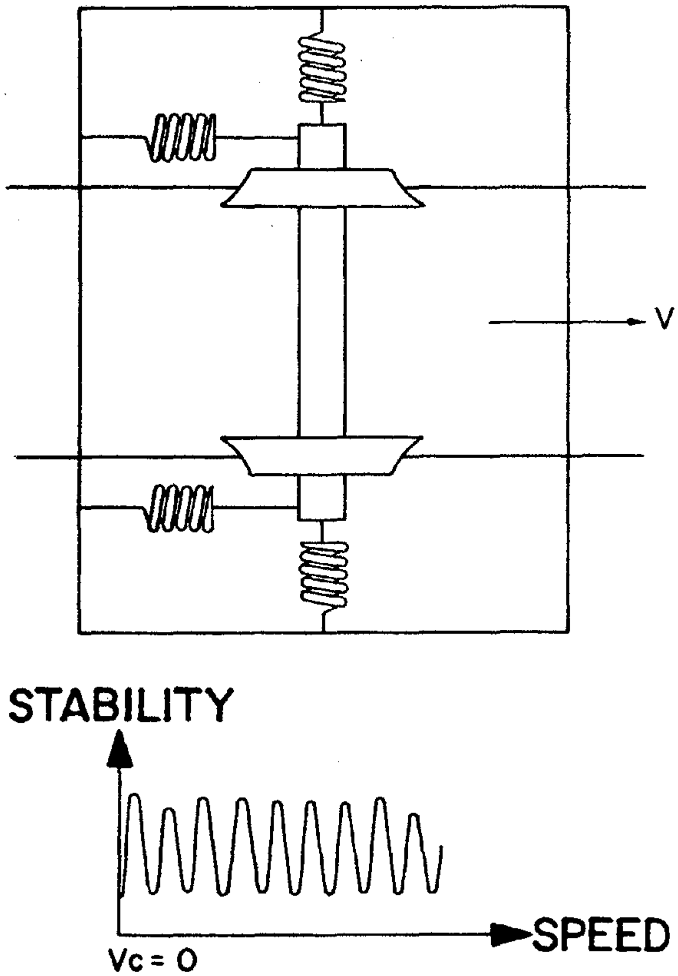


Fig. 3. A single wheelset suspended to a mass has no hunting stability.

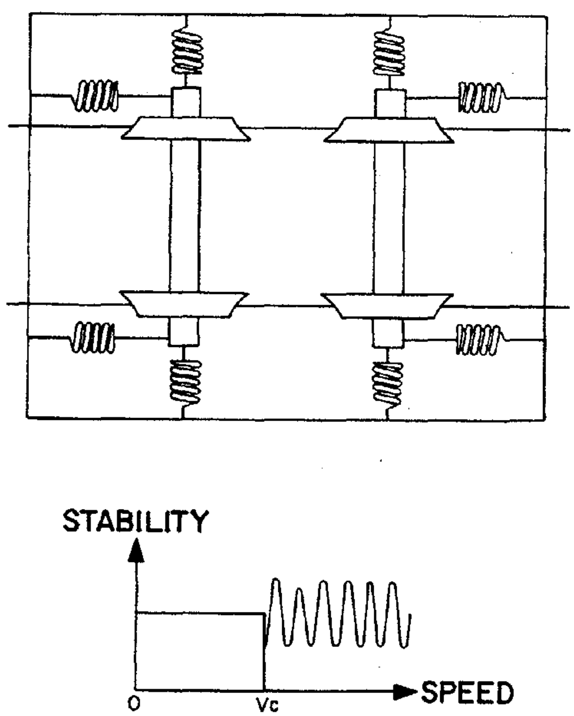


Fig. 4. A pair of wheelsets suspended to a mass gains stability.

figure, (5), leaves directly to the existing truck designs and to the proverbial stone wall of high

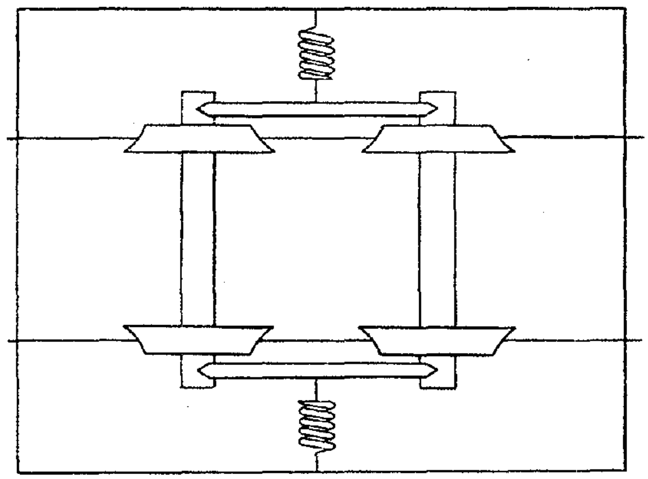


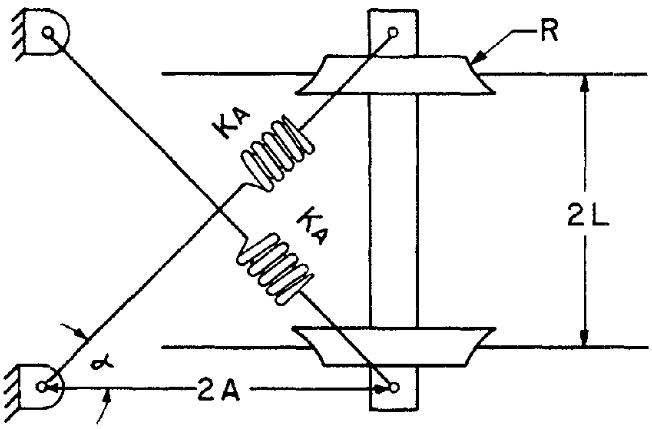
Fig. 5. Conventional truck designs have high yaw constraint between wheelsets.

yaw constraints between wheel sets for high speed stability resulting in poor curving performance. Poor curving performance causing both tread and flange wear which, in turn, decreases high speed stability.

Let's go back for a moment to the single wheel set constrained to ground and see if we can get around this proverbial stone wall. In Figure 6, by using two diagonal elastic constraints, wheel set hunting stability can be obtained just as easily as before when four elastic constraints were used.

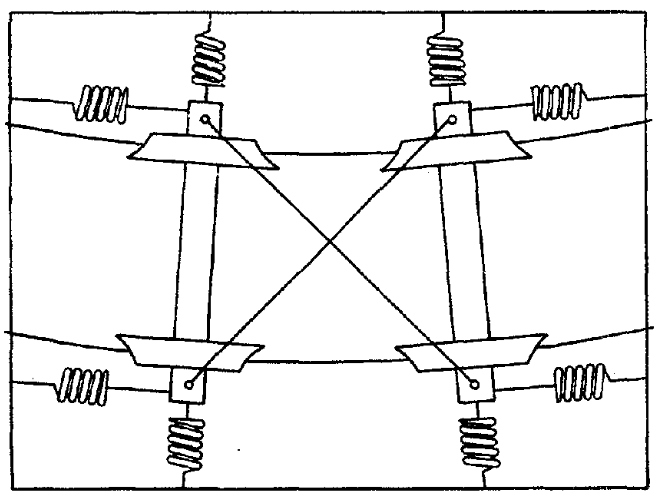
In Figure 7, the next step was taken in that the diagonal constraint was applied between two wheel sets. When the wheel sets are constrained diagonally to each other, we found from both the computer analysis and roller rig experiments a marked improvement in wheel set stability. Tests show the improvements to be on the order of two and a half times that of wheel sets constrained in parallel. Or in other words, a 250% increase in the wheel set critical speed. In a moment, I would like to show a short movie of Mr. Scheffel's roller rig test showing this improvement.

Finally, we now have good wheel set stability and in addition, by providing low yaw constraints between each wheel set and the body frame, good curving can now be realized. In practice, it has been found that when elastic yaw constraints on each wheel set have approximately the same value as the gravitational stiffness, the axles assume an approximate radial position in curves. This allows the wheels to have pure rolling in the curves with guidance supplied by the creep forces.



$$V_c = \left[\frac{LR}{\text{CONICITY}} \times \frac{K_A}{M} \cos^2\alpha \left(1 + \frac{A^2}{\ell^2}\right) \right]^{\frac{1}{2}}$$

Fig. 6. Critical speed of a single wheelset diagonally suspended to ground.



LOW WHEELSET YAW CONSTRAINT
ALLOWS WHEELSETS TO ASSUME
A RADIAL POSITION IN CURVES

Fig. 7. Diagonal constraint provides improved wheelset stability.

Lastly, body instabilities can now be suppressed to a very high speed by providing a low lateral constraint between wheel set and car body. In practice, we made this constraint the same as the longitudinal constraint between the wheel set and the side frames.

We now have a system with the following compatible design features:

1. High diagonal constraints between wheel sets for wheelsets stability.
2. Low longitudinal constraints between wheel sets and truck frame for flange free guidance in curves.
3. Low lateral constraints between wheel sets and truck frame for good body stability.

In the movie, the freight car truck model scale is 1 to 5. The effective conicity of the wheel profile is 0.22. The roller rig test of the model shows an improvement of 2.5 more times in the critical speed when the model is converted from a conventional truck to the Anchor design.

Using the kinematic frequency as a comparison, the critical speed of the model is 1/5 that of a full size truck, or in other words, the critical speed of 19 kilometers per hour is equivalent to 95 kilometers per hour or 59 miles per hour critical speed for a full size truck which has been fairly well substantiated as the typical critical speed for this design. The critical speeds for the Anchor truck model would be five times 50 kilometers per hour or 155 miles per hour. Therefore, the improvement of critical speed for the Anchor Truck over the conventional truck would be an increase of approximately 100 miles per hour.

After Movie. The arrangement of the diagonal anchors between the wheelsets would contribute nothing whatsoever to steering of the wheelsets. They are not connected to the car body mass and therefore, they are not stressed in curves. The only purpose of the cross anchors is to stabilize the wheelsets. During test of a 100 ton hopper car cyclic loadings on the order of ± 1000 pounds were measured in the cross anchors as they stabilized the wheelsets against each other at higher speed. The maximum loading in the cross anchors is approximately five tons due to maximum braking efforts. For this reason, due to the low forces in the cross anchors, they can be made of light construction.

Wheel Profile Conicity. The most exciting characteristic of the Anchor Truck is its lateral dynamic stability using high effective wheel profile conicity. It is well known that high effective conicity is desirable for good steering. In the movie, the model was stable using an effective conicity of 0.22.

Computer analysis of the Anchor Truck along with several field tests show that the wheelset hunting stability is fairly insensitive to increasing effective conicity. Also, tests have shown that body hunting stability actually improves with increasing effective conicity. Therefore, since higher effective conicity improves steering in curves and the uncompromising resulting stability of the Anchor Truck, there is no reason for using the present AAR wheel profile with its low conicity of 1/20 and resulting high contact stresses.

The new AAR wheel profile is probably the best wheel profile compromise for the present three piece freight car truck. However, the way things turn out, it has to wear. Many times 50% of the allowable flange wear has been used up in less than 15% of the expected life of the wheel. I don't have to tell you that there are many cases where the wheel is condemned with less than 100,000 miles.

Wheel life depends upon many variables. However, two factors cause new wheels to wear very quickly initially. The first obvious cause is the high hertzian contact stress of the pure 1/20 taper on curved rail head. The second cause is the two point contact in curves and the resulting sliding contact between the flange and rail.

The wheel tread wears quickly to a contour having acceptable elastic stress values and then changes realtively slowly thereafter. The flange is worn until the two point contact is minimized in curves. By this time, however, the wheel has assumed the typical hollow worn profile. Once you have hollow worn profile, it is very difficult to stabilize the freight car instabilities. The only choice left for getting stability is increasing yaw constraints and system damping. However, increasing truck yaw constraints causes increasing steering resistance and further wheel wear.

With the Anchor Truck, it is desirable to have high effective wheel profile conicity, both for stability and steering. The Anchor Truck allows for a wheel profile design that gives low Hertzian contact stresses by giving it a similar radius as the rail not unlike the worn wheel. Also, since the axles now assume a radial position in curves the tread and flange can be designed to give single point rolling contact in curves. This eliminates the sliding and flange wear.

Practically, one wheel profile cannot meet all rail conditions and have the same conicity. However, by starting with low contact stresses and resulting high effective conicities, the Anchor Truck lateral stability is not sensitive to further increases in conicity due to rail wear. In fact, the body stability may actually improve with the increasing conicity of worn rail.

For example, both for North America and South Africa, a wheel profile having an effective conicity approximately 0.20 on new rail is being used with the Anchor Truck. The same profile on worn rail tends to have increasing conicity which may go as high as 0.35.

The yaw constraint between each wheelset and truck frame is developed by a longitudinal shear spring at each journal on the order of 6,000 lb./in. This low wheelset yaw constraint together with the high effective wheel profile conicity allows flange the wheel-axle sets to assume a radial position to curves.

At last year's conference, Mr. Scheffel described the problems the South African Railroad was having with a unit train in ore service from the Northern part of South Africa down to Port Elizabeth. These were 8,000 ton trains operating over sharp curves and steep down grades. After about 10,000 miles or 9 trips, the entire area in the flange area had been eroded. After 25 trips or 30,000 miles, the wheels were condemned for thin flange.

At the same time, several carsets of the U.I.C. type bogie was put into the same service with no appreciable improvement.

One carset was equiped with the Anchor Truck and was put into the unit ore train service. After 62,000 miles of service, there was no appreciable change in wheel profile from its initial trip.

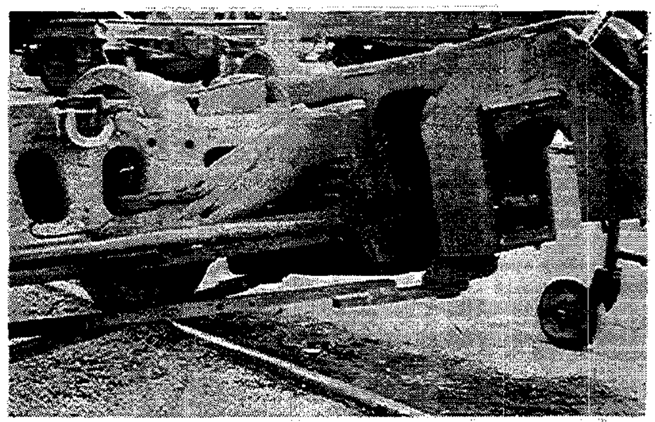


Fig. 8. Cross anchors were positioned below the truck bolster.

Based on the significant improvement wheel flange life, the Anchor Truck is being put into unit train service sometime this spring in South Africa. The annual mileage is expected to be 110,000 miles.

Using the same computer model, a 100 ton version of the Anchor Truck was designed for 4' 8-1/2" gage. One prototype carset was fabricated wherein 18 different combinations of various

Fig. 9. Test car with anchor trucks was a 100 ton coal hoper approximately three years old.

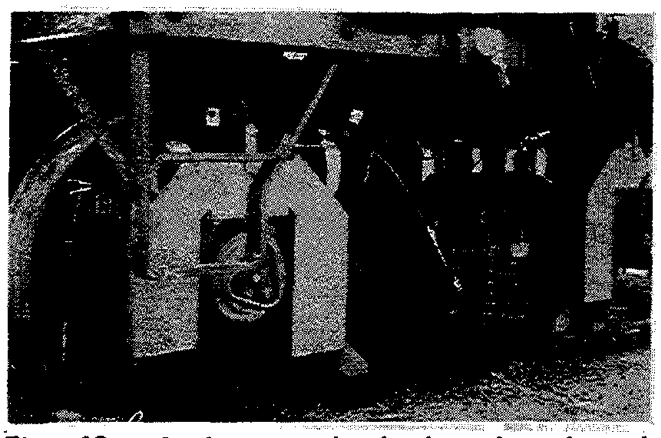


Fig. 10. Anchor truck having lateral and longtudinal shear pads at each journal.

characteristics of the Anchor Truck could be evaluated. This was done in order to validate the computer model using the typically larger car used in North America.

In order to test an existing car, the cross anchors were positioned below the bolster. This allowed the use of existing bolster, brakes, and as much of existing side frame pattern equipment as possible (Figure 8).

(Figure 9) The car tested was a 100 ton coal hopper approximately three years old with over 200,000 miles of service. Existing stabilizing wedges and springs were used. Several sets of shear pads were made up with various spring and damping characteristics. The first set of shear pads were designed to the optimum shear characteristics determined from the computer model. Incidently, over fifty computer runs were made while designing the test prototype. Each run checked the car in ten m.p.h. increments from 20 m.p.h. to a minimum of 150 m.p.h. When the model indicated changes in stability the speed increment was cut down to 2 m.p.h. The second set of shear pads were 40% stiffer and the third set was 140% stiffer than the optimum value (Figure 10).

(Figure 11) The test was conducted on the Union Pacific's California Division Main Line

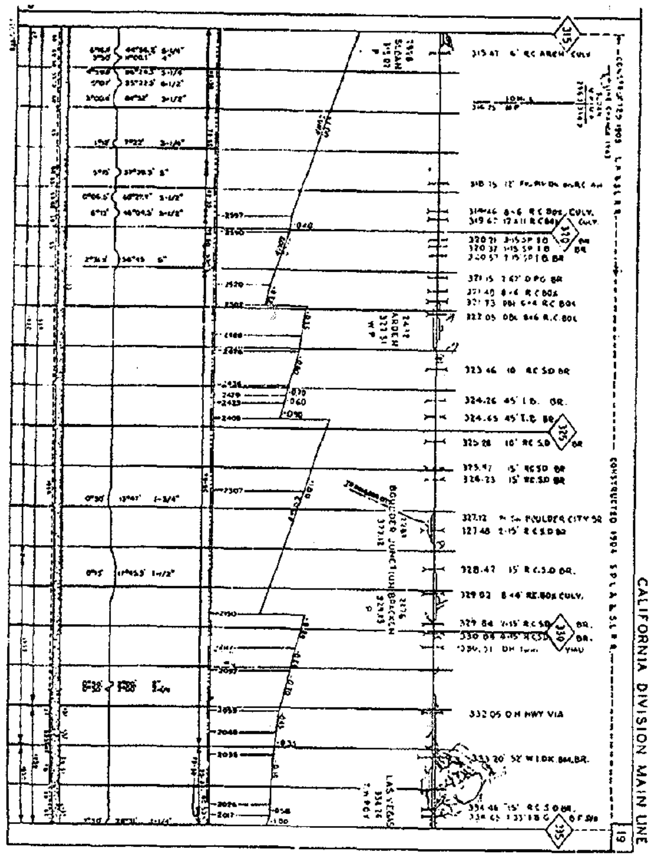
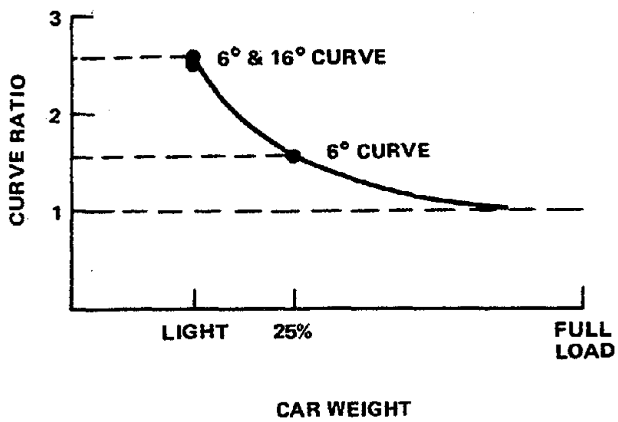


Fig. 11. Test track profile.



$$\text{CURVE RATIO} = \frac{\text{WHEELSETS RADIUS}}{\text{TRACK RADIUS}}$$

Fig. 12. Anchor curving chart.

between Las Vegas (M.P. 332) and Sloan (M.P. 315). This section of track has 10 miles of tangent high speed track with several reverse curves as you approach Sloan. The curving ability of the loaded car was evaluated between M.P. 319 and 320. This section of track had a reverse curve of 6° in both directions. One was 65-1/2° long and the other was 46° long and the train speed was held a constant 40 m.p.h. through these curves.

Fig. 13. Union Pacific test train showing research car U.P. 210, control car, and test car.

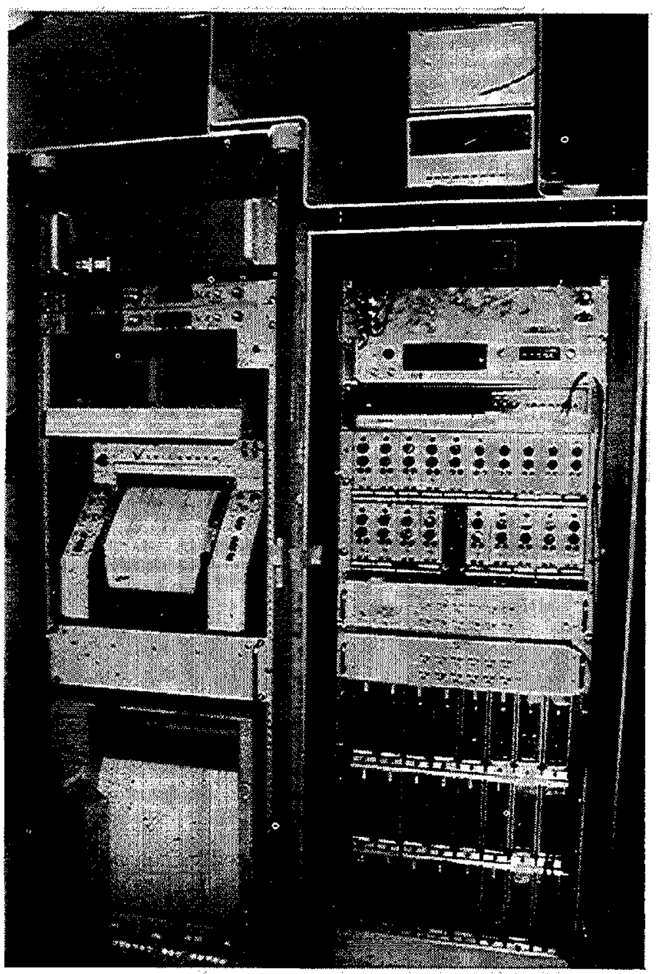


Fig. 14. Union Pacific car 210 oscillograph.

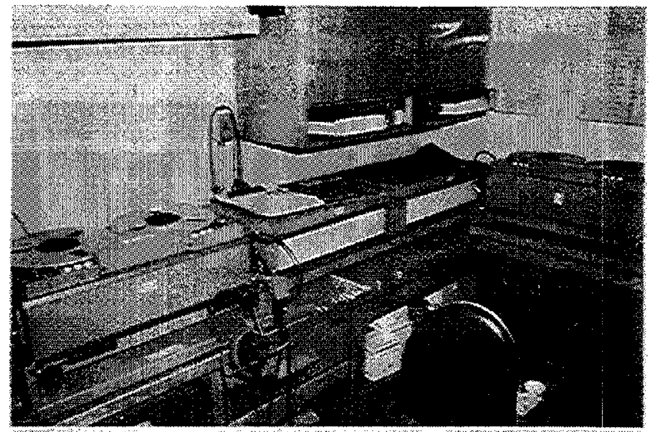


Fig. 15. Union Pacific research car 210 F.M. analog tape recorders.

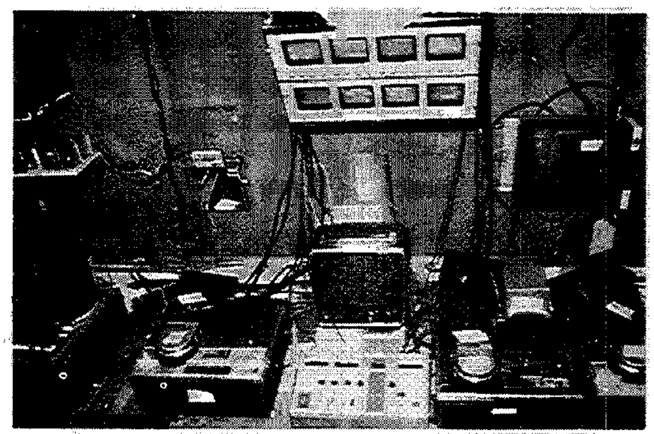


Fig. 16. Union Pacific research car 210 video tape recorders.

For good radial alignment of the wheelsets in curves, the longitudinal spring rate at each journal should approximately be the same as the gravitational stiffness. For the loaded car test, the optimum pad design had a spring rate of 6,400 lb./in. The gravitational stiffness of the loaded car using the 0.15 conicity wheels was 5,500 lb./in. This is great for the loaded car but, unfortunately, railway cars are not always loaded to capacity and sometimes even run empty.

Since we used a hopper car for test, it was not practical to partial load the car. However, by increasing the shear spring rate 140%, the steering of the loaded car was equivilent to a partially loaded car. The radius of curvature between MP

319 and 320 was 955.4 feet. Using the 140% stiffer pad, the loaded car showed good steering with the axles assuming a 1600 foot radius position in both curves. This is equivilent to a car loaded approximately 25% of capacity. The ratio of actual curve to assumed position is 1.6.

The light car axles, through the same curves, assume a radial position of 2700 feet. For all steering tests the speed was a constant 40 m.p.h.

In yard test, the light car using the optimum pad had the axles assume a radial position of 925 feet in a 16° curve. The radius of a 16° curve is 359.3 feet. For both the 6° curve and the 16° curve, the ratio of curvature to the assumed position of the wheel set is approximately the same. Specification 2.57 for the 6° curve and 2.8 for the 16° curve. The Anchor Truck test of a 100 ton design allows excellent curving of even the light and partially loaded car. Figure 12 shows how test indicates the ratio will change with car loading.

On the main line tangent track, the speed was increased from 5 m.p.h. to approximately 95 m.p.h. in 5 m.p.h. increments. Twenty-two channels of analog signals were recorded. (Fig. 12, 14, 15 and 16) Using a profiled wheel having an effective conicity of approximately 0.15, the critical truck hunting speed was never reached.

The test data was used to check the accuracy of the computer model predictions for the 100 ton hopper car. The model is now being used to finalize a production design of the Anchor Truck.

Conclusion. The Anchor Truck consists of cast steel side frames and bolsters arranged in a conventional manner. Diagonally placed anchors constrain the wheelsets to each other while at the same time, they do not interfere with the natural tendency of profiled wheels to align themselves radially on curved track. By providing a low yaw constraint on each wheelset for curving and the cross anchors providing wheelset stability, the non-compatibility between the dual requirement of steering and high speed stability has been eliminated. By doing this, the preliminary results indicated that wheel tread and flange wear can be virtually eliminated with this design.

Acknowledgments. The author wishes to thank Mr. F. D. Acord, Chief Mechanical Officer, Union Pacific Railroad, and his Research and Test Department for their sincere interest and cooperation in testing the Anchor Truck.

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Mr. Schrotberger obtained his degree in Civil Engineering at the Swiss Federal Institute of Technology, Zurich (ETHZ) in 1967. He joined the Bridge Section of the Swiss Federal Railways (SBB) at Lausanne in 1968. After working for the South African Railways in Johannesburg in 1971, he was seconded to the Office for Research and Experiments (ORE) in Utrecht, Netherlands, in 1972.

Mr. Schrotberger's work covers a variety of railway engineering problems, such as fatigue in bridge structures, chemically stabilized ballast, permissible axle loads of cars dependent on wheel diameter and speed, and outside cleaning and painting of vehicles. He has been responsible for the coordination of research and development carried out in railway research centers all over the world.

ORE ORGANIZATION AND ITS STUDIES IN TRACK, SUSPENSION AND TRACK/TRAIN INTERFACE

I first wish to thank the DOT and especially Mr. Ward for giving ORE the opportunity of speaking at your Conference. ORE is very pleased to enjoy the excellent contacts and fruitful collaboration with DOT and AAR, which can only benefit the railroads.

Introduction. After a brief introduction, I will give you as the first part of my talk a general view of ORE, the Office for Research and Experiments of the International Union of Railways. The second part covers some studies of track developments, and the third part gives you briefly some ideas about new car suspension systems. Finally, in the fourth part, I intend to summarize some of the track/train dynamic studies of ORE.

The table below should show you some major differences between U.S. and European railroads:

U.S.	Europe
High axle loads	High speeds
Mainly goods traffic	Mainly goods traffic mixed with high speed passenger traffic.
Only 4 axled cars	Mainly 2 and 4 axled cars
	Main lines with steep sloopes > 25°/00 and small radii

General view of ORE. Organization. You may ask how international an organization is ORE. I can answer this by telling you that 43 railroad administrations all over the world are members of ORE (Fig. 1). The DOT of the United States is also one of its members, through the Alaska Railroad. It should be mentioned here that AAR has signed a contract for the exchange of reports with ORE.

ORE, founded in 1950, is situated in Utrecht in the Netherlands (see Fig. 2). The principal objects of ORE are as follows:

1. To make test results obtained by the different railroads available to other member railroads.

Fig. 1. 43 world-wide ORE railroad administrations.

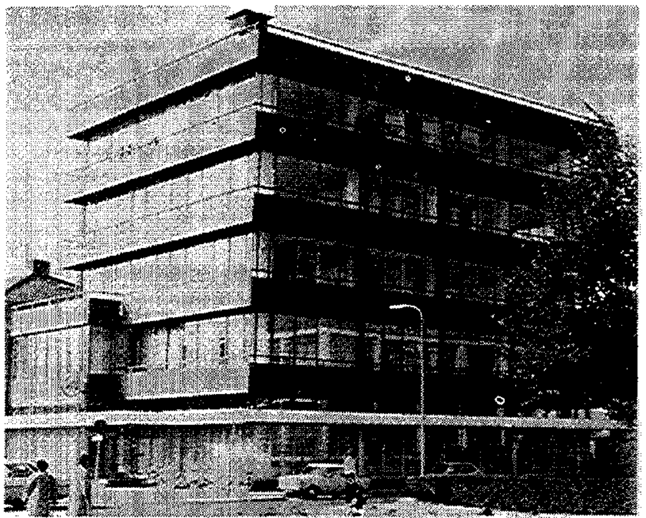


Fig. 2. ORE building--Utrecht, Netherlands.

2. To achieve common utilization of existing test facilities.
3. To make studies permitting cost to be reduced by standardization of rolling stock.

About 60 people are working full time under a director in the ORE Bureau in Utrecht (see Fig. 3), 16 of them being engineers – called technical advisers -- who are delegated by member-administrations. Policy is governed by the Control Committee, which consists mainly of about 30 Chief Officers of member-administrations and which is chaired by the President of ORE. This body meets twice a year to decide the studies to be undertaken by ORE and also to approve the technical reports.

Once the Control Committee has decided that a certain problem should be studied, the member-administrations are asked to nominate a Specialist in the particular field (Fig. 4). These Specialists and one ORE engineer make up the Committee which studies the given problem, makes tests, and prepares the technical reports. The Specialists do their work in addition to their normal duties at their own administrations, the ORE engineer being the only full-time member of the Committee.

Up to now, more than 700 reports (each in English, French and German language) have been issued by ORE (Fig. 5). They cover for example, standardization of rolling stock and components, problems of high speeds, environmental problems, automatic couplers, safety, track, signalling, etc. Nearly all reports may be bought by third parties.

Test facilities. For tests, ORE can use the facilities of its member-administrations. The exception to this rule is the Vienna Arsenal Vehicle Testing Station, which is jointly run by the

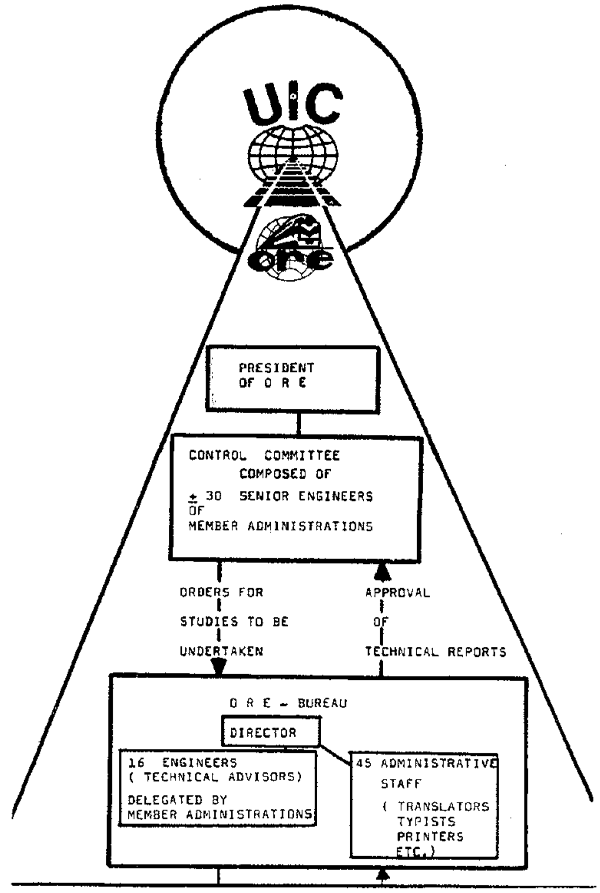


Fig. 3. Block diagram of ORE.

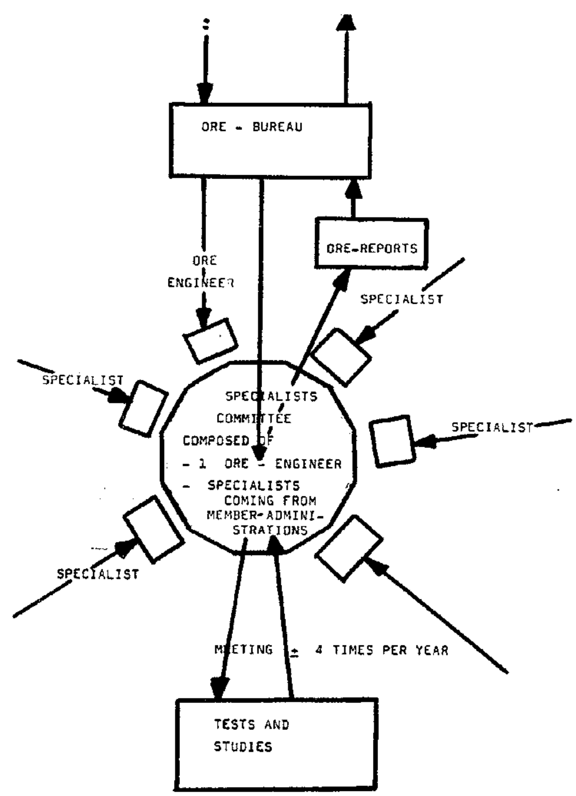


Fig. 4. Flow chart of ORE work.

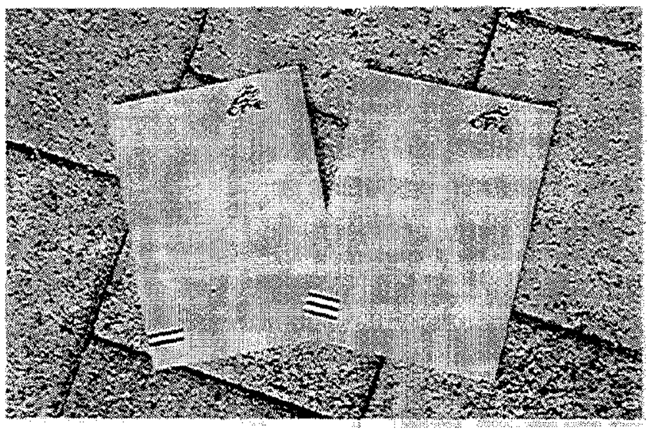


Fig. 5. Example of ORE reports.

Austrian government and ORE (Fig. 6). In this station, vehicles can be tested under accurately controlled climatic conditions. It consists of a static and a dynamic chamber, both capable of accommodating a full-size passenger coach. The chambers are fitted to provide a temperature range from -58°F to +122°F and a relative humidity of up to 95% at +75°F. The dynamic chamber is also equiped with a blower which provides an airstream permitting train speeds of up to 160 mph to be simulated. On one side, it has a solar radiation system enabling sunshine with the intensity of a July midday sun in southern Europe to be simulated (Fig. 7). It also contains a roller rig for testing brakes under controlled climatic conditions up to speeds of 175 mph. Tests made in these chambers served to check, for instance, the insulation of passenger and freight rolling stock, functioning of iced pantographs, and automatic couplers.

Studies on track developments. In the second part of my talk I would like to give you a summary of the studies in the field of track developments.

Conventional track. I begin with the study of optimisation of the conventional track for the

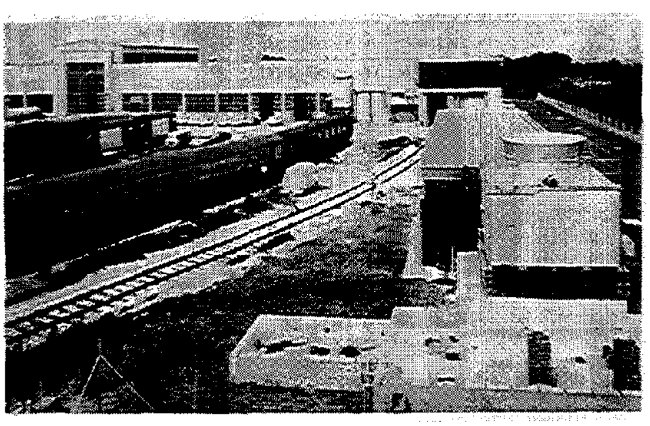


Fig. 6. Vienna arsenal vehicle testing station; general view.

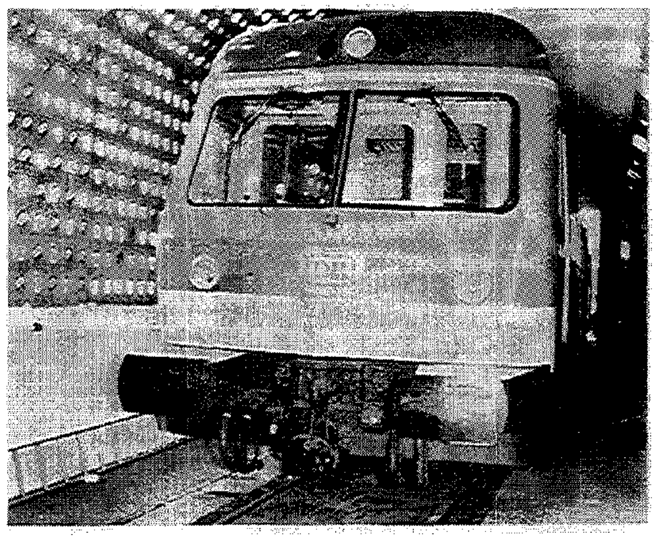


Fig. 7. Vienna arsenal vehicle testing station, dynamic chamber with solar radiation system at the left.

traffic of the future. This study is concerned with the relationship between (1) traffic and track geometry, and (2) traffic and track component characteristics.

Concerning the traffic/track geometry relationship, numerous tests have been carried out in the track laboratory at Derby in England, on the test circuit near Prague in Czechoslovadia, and on running lines.

The track test facility at Derby consists of a real track with a length of 60 ft. in a hall. The track can be loaded with a special one-axle vehicle. There, tests had been carried out varying axle load, tie spacings, rail types, and compacting methods, bearing in mind the evolution of the track geometry all the time.

The test circuit near Prague (Fig. 8) allowed tests under very heavy traffic, using different types of ties, different tie spacings and different rail types. The smaller circuit has a length of 2.5 miles and allows loading of up to 1.1 Mil. tons per day, using a 4,400-ton train running for 20 hours. Here too, attention was paid to the evolution of the track geometry. The tests on running lines were carried out on nine different track sections, varying different parameters.

I can summarize some of the findings which showed in general that the European track is nearly optimized for European traffic.

1. The track degradation follows mostly an exponential law in tonnage.
2. The initial quality of track geometry, just after maintenance work, influences heavily the evolution of track degradation.

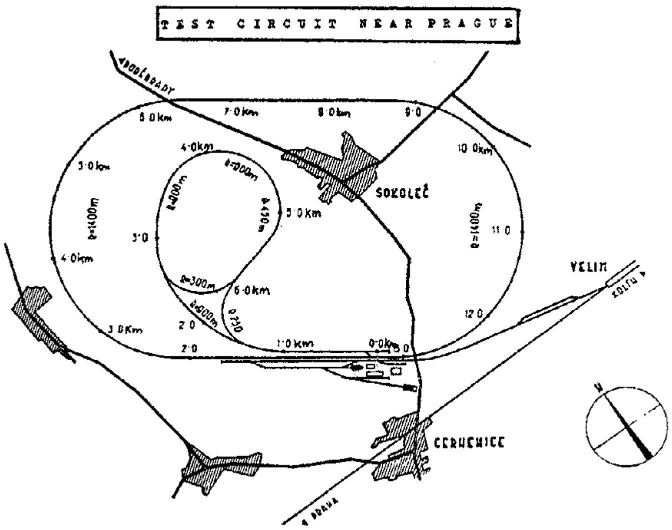


Fig. 8. Test circuit, near Prague.

3. It is mostly the number of heavy axle loads and not their mean value which dominates the deterioration of track.
4. It seems that the deterioration of track depends linearly on the mean axle load.
5. Compacting of the space between ties and the shoulders in the case of concrete ties gives good results from the point of view of the track stability, but attention should be paid when doing it with wooden tie track.
6. The type of rail had a very small influence on track degradation.
7. Reduction of tie spacing seemed to be valuable when initial spacing was great and heavy axle loads were used.

Concerning the relationship between traffic and track components, statistics of five Administrations on rail ruptures had been analyzed to determine the influence of rail profile, steel, joints, subsoil, traffic, and environment.

Recommendations for the choice of the rail profile, depending on the daily traffic, could be given. Heavy axle loads have of course a bad influence on rail fatigue. It could be seen that wear-resisting steel for rails should be used on long sections when increasing the mean axle load and decreasing the wheel diameter. The welded track is, of course, recommended. The problems of welded joints seem only to be a question of work supervision; the greater the traffic, the worse the working conditions. Inhomogeneity and a frost-sensitive subsoil have a bad influence on rail fatigue. Finally, the climatic influence showed an increase of rail ruptures in winter.

It should be mentioned here that attention is now given within ORE to the stabilization of ballast track by means of synthetic dispersions. This seems to improve the track quality and safety, especially of welded track. Special attention is now given to application methods. Several tests are being planned, among them a test under heavy loads on the Prague test circuit. The extremely high prices of these products may of course limit an economic application.

Slab track. I am coming now to the studies carried out in the field of slab track with the main aim being to reduce vibrations, noise, and maintenance costs to a minimum. This study includes also the rail fastening.

First experience was obtained on the Radcliffe test track (Fig. 9), where six types of slab track were tested, each of them 240 ft. long and loaded by 4 Mil. tons a year. After four years of operation, no significant problems have occurred.

In addition to experience being compiled from main line installations in France, England, Germany, and Switzerland, ORE has sponsored tests at high speeds in Germany, up to 160 mph, and tests on sharp curves without cant on the Prague test circuit. On the latter, three types of track were built, using Dutch fastenings (see Figs. 10 and 11) because of its advantage in changing the

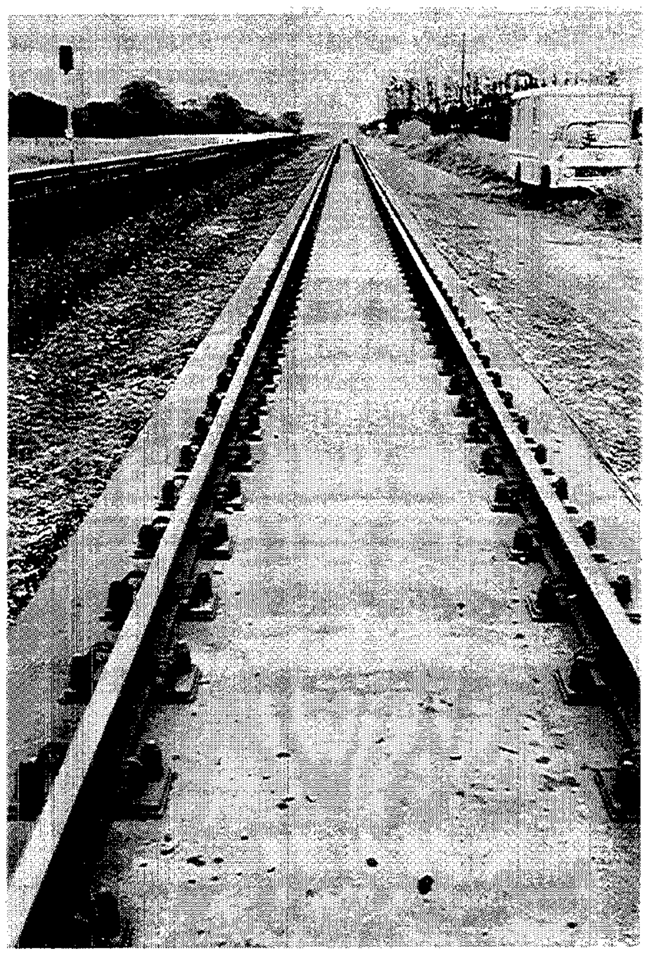


Fig. 9. Radcliffe test slab track.

lateral position of the track ± 1/4 in. by simple means. The tests are still under way; up to now about 100 Mil. tons have run over the slabs.

Car suspension studies. In the third part of my talk I would like to say some words about modern car suspension systems. Such a study within ORE is based on two-axled cars, but attention is also given to fitting them to truck cars.

The main object is the design of new suspension systems to improve the riding stability of torsionally stiff cars as well as that of empty cars. A so-called progressive suspension system shall be developed to fit cars in service and also new cars. The bases for developing such suspension systems were taken from the recommendations of the Committee dealing with the prevention of derailment of cars on distorted track, the work of which I shall mention later. Four different types have been proposed and are under test now. An improvement of the safety against derailment can be seen.

Another study deals with the influence of the transverse play of the suspension systems on the riding quality. This study is dealing with two-axled cars only and is therefore of little interest to you.

Studies on track/train interaction. The fourth part of my lecture concerns the studies of ORE on track/train interaction.

Mathematical models. The mathematical formulation of the dynamic performance of railway vehicles is the work of a Committee which has made use of a large number of parameters relating to:

1. The position of the track.
2. The geometry and the transmission of forces in the wheel/rail contact zone.
3. The dynamic properties of the railway vehicles.

The representation of track irregularities in the form of power spectral densities is used (Fig. 12). Spectrum envelopes have been prepared by different Administrations, and it appears that they can be effectively used for certain comparisons and calculations.

After establishing a simple mathematical model of a test vehicle, the theoretical performance was compared with the actual performance of the vehicle on a test track. The results showed that the vertical acceleration and the purely vertical body-wheelset displacement can be predicted fairly well by means of the mathematical model and the techniques used. Certain difficulties arose, such as nonlinearities of suspension components, particularly the damper. Laterally, the predictions were rather poor because there were three critical unknowns in the lateral equations:

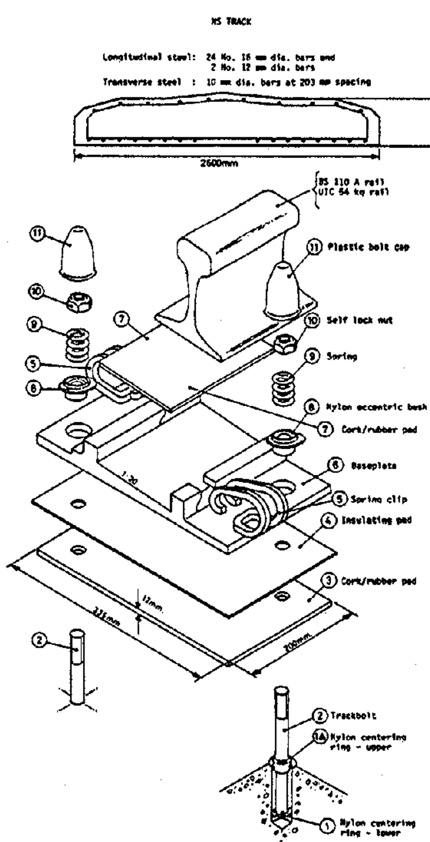


Fig. 10. NS rail fastening.

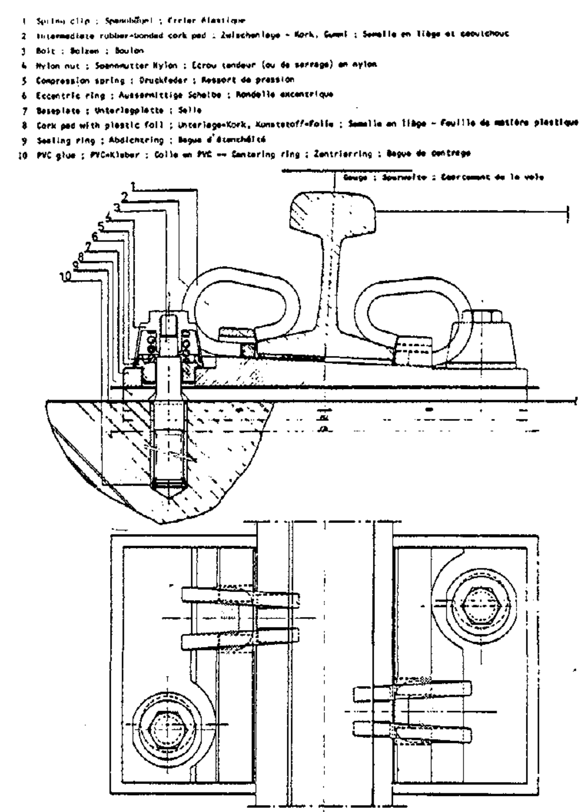


Fig. 11. NS rail fastening.

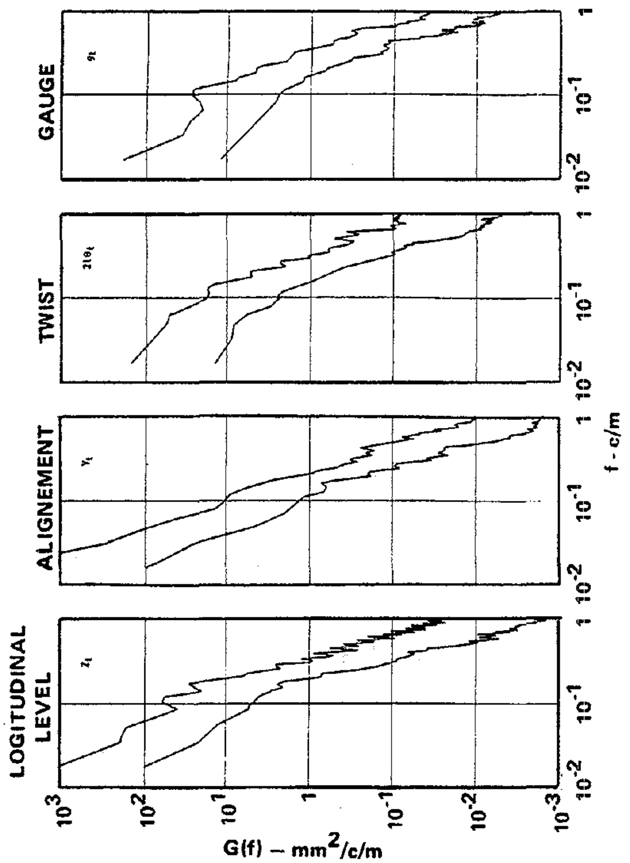


Fig. 12. Example of power spectral densities.

1. The conicity varies with the lateral movement of the wheelset and linearization is difficult. The conicity is defined as tan $y = \frac{r_1 - r_2}{2y}$ where r_1, r_2 are wheel radii in contact point, and y is lateral displacement of wheelset.
2. The wheel/rail creep coefficients can only be calculated for certain ideal conditions (depending on weather).
3. The true lateral input spectrum is not known.

Up to now it could be shown that equivalent conicity is influenced by the track gauge, rail cant, initial contact angle between rail and wheel, and the conformability of wheel and rail profiles.

The vehicle parameters considered are: (1) the geometric one, (2) the inertia, and (3) the suspension, and the methods for their determination are now given in a report.

The equations of the motion of a railway vehicle have now been derived. Since an understanding of the dynamic behavior of a wheelset is fundamental, a start was made by developing the equations for this. An isolated wheelset connected through its suspension to an infinite mass moving along the track center line was used. Then the equations were extended to the cases of a rigid frame, supported on two wheelsets, next to that of a truck vehicle, and finally to that of the articulated vehicle.

The future will show where simplifications can be adopted and how the models adopted agree with the performance during line tests.

Interaction locomotive/track. I would like to describe some aspects of a recently concluded study concerning the constructional arrangements for improving the riding stability and the guiding quality of locomotives.

With the general development of modern high-powered truck locomotives entailing high axle loads and high speeds, the study of the riding stability and guiding quality has become a matter of importance when considering the permited speed in curves, on straight track and through points, and crossings, and also the associated stresses exerted on the track. The recommendations made are based on tests with locomotives at maximum speeds up to 125 mph. This Committee has also developed methods of measurement on the wheel to determine the forces between wheel and rail.

Before I give you some recommendations, some explanations of the terms used are necessary. The lateral forces Y occurring between the wheel and the rail during the running of a railway vehicle (see Fig. 13) can be divided into:

$$Y = \Delta Y_{qst} \pm \Delta Y_{dyn}$$

where

ΔY_{qst} is the quasi-static wheel force variation which remains constant over a given time.

ΔY_{dyn} is the dynamic wheel force variation, caused, for example, by accelerations origination from track irregularities.

Conclusions of the whole study may be summarized as follows:

1. The quasi-static Y forces on curves with radii below about 2,600 ft. can be kept low if the following measures are adopted:

   1. Reducing the angle of attack of the leading wheelset when the driving axles are ridgily guided in the x direction, by means of a transverse coupling between the trucks (Fig. 14).

   2. Choice of a short wheelbase for the trucks.

   3. Create a possibility of adjusting the wheelsets radially on a track curve. This

solution exists for cars; it still needs to be found for tractive units, and in such a way that the riding stability on straight track does not suffer.

2. The dynamic Y forces can be reduced if the following measures are taken:

   1. The transverse suspension parameters should be chosen so that no resonance phenomena occur.
   2. The transverse suspension system between the body and the truck should be soft so that the body center of gravity follows the transverse shock displacements of the truck as little and as slowly as possible.
   3. Installing of a device allowing a decrease of the centering moment with an increase of the angle of rotation of the truck.
   4. A transverse suspension (as shown on Fig. 14) between the wheelset and truck mass reduces the dynamic Y forces considerably during sudden changes of the direction due to irregularities in the alignment of the track and when running through points and crossings. This is recommended for all wheelsets of BB locomotives and for the end wheelsets of CC locomotive trucks.
   5. The use of a short wheelbase for the truck also reduces the dynamic Y forces.

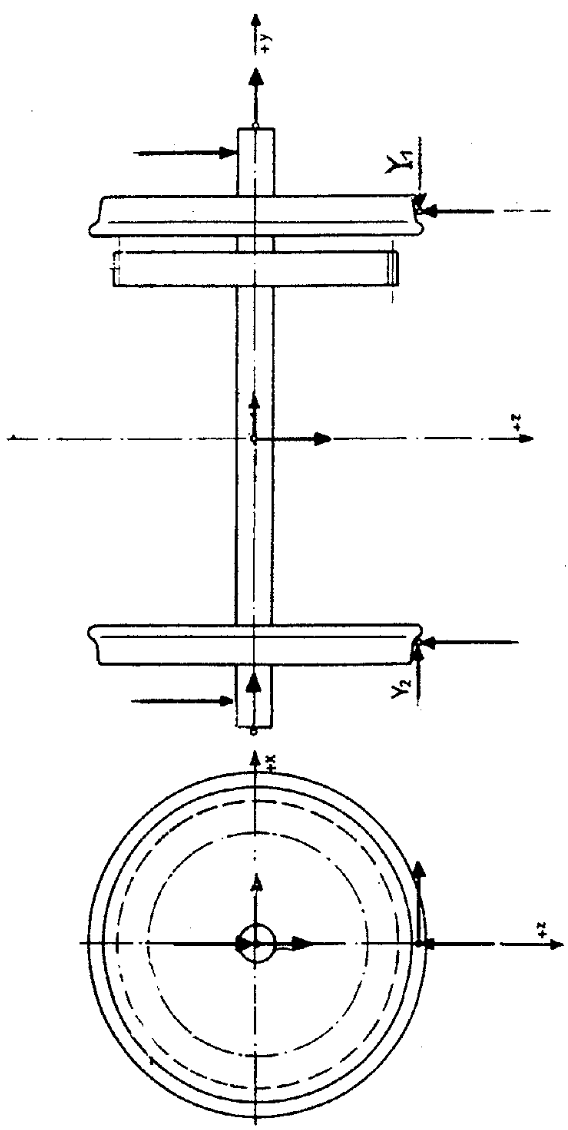


Fig. 13. Notation of wheelset forces used in the test.

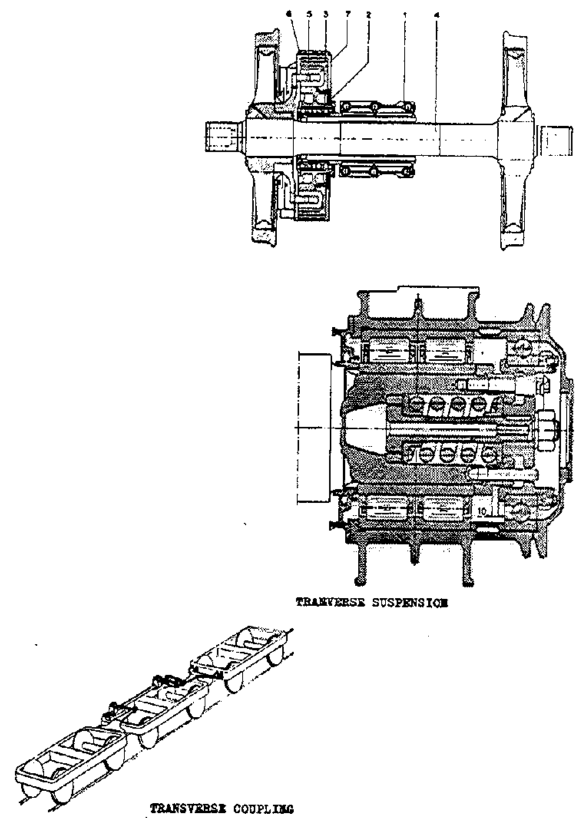


Fig. 14. Scheme of transverse coupling and transverse suspension of locomotives.

Derailment studies. Another ORE study deals with the prevention of derailment of cars on distorted track. This study has so far enabled recommendations to be given for the safe negotiation of track twists which are to be observed in the design of truck cars.

Statistical analysis of a track twist survey, which covered a total of 4,600 miles of track of five European railways, allowed that, for the design and testing of vehicles, the limiting track twists for truck cars be recommended.

The decisive factor in assessing the safety against derailment in track twists is the quotient of guiding forces Y and vertical wheel loading Q. To determine such permissible values for Y and Q as well as ratio Y/Q is the task of another Committee. This Committee, which has recently started its work, has also to determine Y and Q force limits with reference to rail stresses, track displacement, and the vibratory behavior of the track as a function of the speed.

Axle loads as a function of speed and wheel diameter. One Committee is engaged in specifying maximum permissible axle loads for cars as a function of speed and wheel diameter. This study was subdivided into three groups.

The first concerns maximum axle loads for modern cars at speeds up to 75 mph. The idea was to compare the aggressiveness of modern truck cars in relation to old cars.

Two criteria were adopted, based on the standard deviation of the statistical distribution of wheel loads; the one determining the damage to the rail and the other the maximal wheel load.

These criteria are defined as follows:

$$\alpha = \left[\frac{\frac{Q_N}{D}}{\frac{Q_o}{D_o}}\right]^2 (1 + 3\theta + 3s^2)$$

where it is admitted that the damage to the rail is proportional to ΣQ^3 (including costs)

$$\beta = \frac{\frac{Q}{D}}{\frac{Q_N}{D}} (1 + \theta + 2s)$$

It is admitted that the maintenance cost is proportional to Q^3

where Q_N is the nominal wheel load

D is the wheel diameter

Q_o is the reference wheel load

D_o is the reference wheel diameter

θ is the relative overload of the wheel due to the deficiency or excess of cant

$$\frac{\Delta Q_I}{Q_N} = \frac{2Ih}{e^2}$$

I = cant deficiency
h = height of the center of gravity above rail level
e = track gauge (4 ft. 8-1/2 in.)

s is the relative standard deviation $\frac{\sigma(\Delta Q)}{Q_N}$

$\sigma(\Delta Q)$ is the standard deviation of the random dynamic loads for one wheel

The result of this first part was that modern cars with 22-tons axle loads at 75 mph do not affect the rail more than old cars with 22-tons axle loads at 50 mph.

The second part of the study was to define the admissible axle load for cars with normal wheels of 36 in. diameter at speeds exceeding 75 mph.

Here, tests had been carried out up to 100 mph with 22-tons axle loads using a Y 25 truck car. The same criteria, β and α, were adopted. In addition, a riding quality factor was introduced, the so-called Wz factor, which is defined by:

$$Wz = 0.896 \sqrt[10]{(b^3/f) \times F(f)}$$

in which

b is the maximum value of the acceleration (in cm/sec^2)
f is the frequency (in Hz) recorded
F(f) is a function of the frequency f, defined as follows:

1. Index relating to the track quality in the vertical direction:

   f between 0.5 and 5.9 Hz . . . $F(f) = 0.325f^2$

   f between 5.9 and 20 Hz . . . $F(f) = 400/f^2$

   f higher than 20 Hz . . . $F(f) = 1$

2. Index relating to the track quality in the horizontal direction:

   f between 0.5 and 5.4 Hz . . . $F(f) = 0.8 f^2$
   f between 5.4 and 26 Hz . . . $F(f) = 650/f^2$
   f higher than 26 Hz . . . $F(f) = 1$

Wz = 1.0 - Very good
= 1.5-Nearly very good
= 2.0 - Good
= 2.5 - Nearly good
= 3.0 - Satisfactory (desirable limit for passenger coaches)
= 3.5 - Still satisfactory
= 4.0 - Suitable for operational purposes (desirable limit for goods wagons)
= 4.5 - Not suitable for operational purposes
= 5.0 - Operationally dangerous

The results of this second stage were as follows:

1. From the point of view of vertical and transverse forces exerted on the track, the running of such a car with 20 tons per axle could be admitted at 95 mph, without any special restrictions, on those lines where the quality of the track enables such speeds to be attained by passenger trains.
2. However, as a sustained hunting movement appeared from 80 mph onwards even when unloaded, the Wz coefficient was not satisfactory; the body accelerations observed would not be acceptable, especailly in the case of perishable or fragile goods, for which these very speeds could be interesting.

It could be seen that the axle load and not the speed had a major influence on the α factor (see Fig. 15).

The third part of the study of axle loads deals with the permissible axle load as a function of speed and wheel diameter.

Owing to gauge limitations it was necessary for the European railways to develop cars with small wheels for the transport of lorries. UIC leaflet 510-2 gives the following permissible axle loads as a function of the wheel diameter for speeds up to 75 mph:

Wheel Diameter		Axle loads	
in mm	inches	metic tonnes	tons
920/840	36/33	20	22
840/760	33/30	18	20
760/680	30/27	16	18
680/630	27/25	14	15

For diameters below 25 in., the Committee has to check if the following propositions of maximum axle loads are realistic:

Wheel diameter		Axle loads	
in mm	inches	metic tonnes	tons
630/550	22 /22	12	13
550/470	22 /18.5	10	11
470/390	18.5/15	8	9
390/330	15 /13	7	8

Tests have been made with cars equipped with such very small wheels (Figs. 16 and 17). Vertical and lateral forces have been measured on the track while the test train passed over it in 1974 and in summer 1975, these forces being also measured on the cars. Analysis of the signals are still under way. Here too, the α and β criteria will be used.

Interaction axle loads/bridges. Finally, a study deals with the estimate of load or traffic spectra on different lines. These spectra will then be used to estimate bridge fatigue life due to service loads of existing and new bridges.

This work is fully under way, using two different methods:

1. The spectra are based on trains, determined from the indications about the rolling stock fleet, and the composition of the traffic is based on statistics of the line. This part is nearly finished.
2. The spectra are based on recorded axle loads of different lines during several days. All that data may now be analyzed by a computer, a special program digitalizing the recorded axle loads and axle spacings.

Conclusions. Let me make some conclusions: As stated before, the studies undertaken by ORE cover most of the fields of railway engineering. But the psychological and human aspects of the work

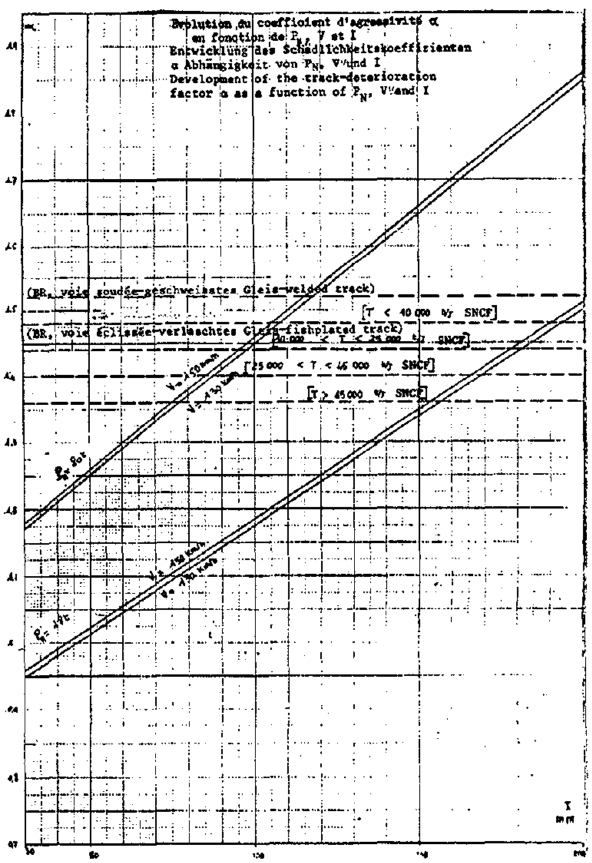


Fig. 15. Track deterioration factor as a function of axle-load, speed and cant deficiency.

done during the last 25 years should also be mentioned.

Without attempting to forecast the future, it might be considered that the close collaboration of more than 40 administrations, and the joint work of nearly 450 Specialists, already represents a very significant success and should be seen in this context as a promise for the international future of the railways.

Summary. The ORE is the Office for Research and Experiments of the International Union of Railways, with headquarters in Utrecht, Netherlands, founded in 1950. 43 Administrations all over the world are members of ORE, among them also the DOT through the Alaska Railroad. The studies carried out cover most of the railroad engineering problems, especially Track/Train Dynamics, the automatic coupler and standardization of railroad material. For testing ORE can use the test facilities of its member-administrations.

The only test facility of ORE is the Vienna Arsenal Vehicle Testing station, where whole vehicles can be tested under various controlled climatic conditions, such as wind, snow, rain, sun, cold, and heat.

Studies of track developments deal with the conventional ballasted track, a track type which seems to be nearly optimized in Europe, as well as with slab track, where experience is being gained now from several test tracks in Europe.

The studies of new car suspension systems cover, up to now, only the two-axled cars.

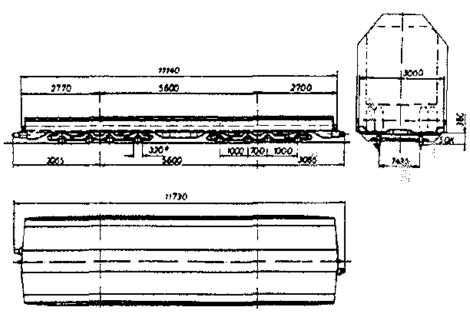


Fig. 16. SGP wagon (wheeldiameter 13-15 in.).

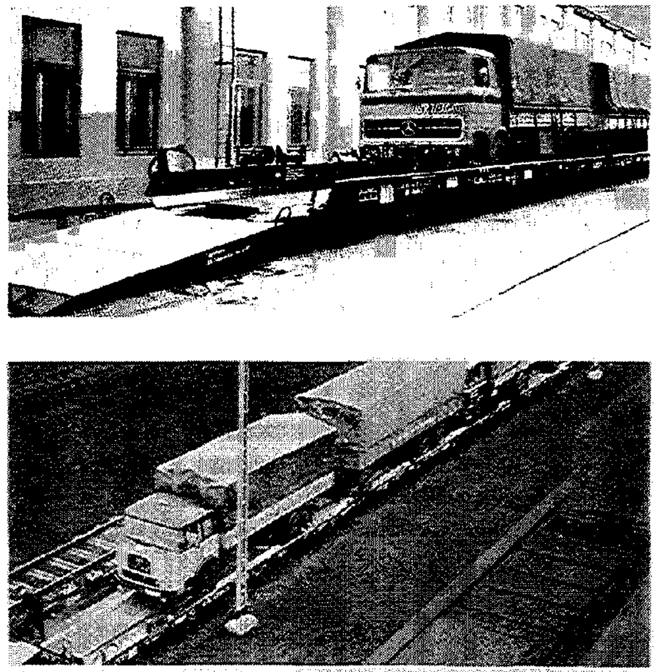


Fig. 17. Loaded SGP wagon.

Studies of Track/Train Dynamics are very widespread. The mathematical formulation of the dynamic performance of railway vehicles with the help of track, geometry of the wheel/rail contact zone, and vehicle parameters is on the way.

Recommendations concerning constructional arrangements for improving the riding stability and guiding quality of locomotives have been made. In the field of derailment studies, values of limiting track twist for the design of truck cars could be given. The study of determining maximum axle loads as a function of speed and wheel diameter has been carried out up to 100 mph for cars with 36 in. wheels and up to 80 mph for 14 in. wheels.

Finally, the effect of service loads as regards the fatigue of bridges and bridge components is being studied by producing load spectra.

In addition to all the studies, the close collaboration of most of the 43 member-administrations already represents a significant success and is a promise for the international future of the railroads.

Discussion Leader
William J. Ruprecht
Director-Engineering
Shippers Car Line Division
ACF Industries, Inc.

William J. Ruprecht is Director of Engineering for the Shippers Car Line Division of ACF Industries, Inc., St. Charles MO. He joined ACF in 1954 as Senior Metallurgist and progressed through various technical and management assignments to his present position. He previously was Assistant Director of Engineering for the Amcar Division of ACF Industries.

Ruprecht was graduated from the University of Missouri with B.S. and M.S. degrees in Metallurgical Engineering. He has been associated with various trade and professional groups and is presently a member of the American Society of Mechanical Engineers, the American Society of Metals, the British Iron and Steel Institute, and the British Institute of Metals.

COMMENTS ON SUSPENSION AND COSTS

I think all of us appreciate the discussion and the papers that were presented in this session. I would like to review them and talk a little bit about one person's view of the cost impact as related to improved trucks to try to add some financial yardsticks to the overall picture.

Before I do that, I think it's well to understand with what background and what qualifications I'll be framing some of these costs. Our fleet differs from each of yours, and you've got to extrapolate and interpret the ideas. Shippers Car Line is a lessor of primarily tank and covered hopper cars. We have some flatcars and some boxcars, but primarily it's tank and hopper. We have about 37,000 cars; 50% of them are 100 ton. Over a year's time we run from between 400 to 750 million miles. We have detailed mileage and cost figures going back about eight years and covering in detail about 3 to 3.2 billion miles. Our cars run from about 4,000 to 100,000 miles per year.

We have heard from the FRA/SP cooperative program, truck designers, and manufacturers. They discussed improved performance, improved ride if you wish, in trucks, under the general heading of suspension development. Now, we all know improvements are needed today. A number of years ago this type of improvement had very limited possibilities and a restrictive cost picture. No railroad, no owner could afford to pay much for this improvement at that time. But, gentlemen, each year that goes by, what we can afford to pay for, is increasing. Let me give you some percentages and figures from our data; than look at yours, and see possibly how closely it parallels ours.

The service maintenance costs in 1975 are going to be up about 40% over costs in 1974. This is no projection; this is actual fact as it is occurring every month in our fleet. The proportion of that cost related to trucks is actually up 50% over last year. Now, either last year was possibly an unusual year, or this year is possibly unusual because of last year. Parts were scarce and less maintenance was done by railroads and Rip tracks last year. More is being done this year because of the availability of parts.

Between 1973 and 1975 you will find your fleet maintenance cost is up about 50%. This is your AAR billing. You can control your in-house work, you can shut down your own repair tracks insofar as you own cars or your rebuilds are concerned, but you get billed just as we do every month because your cars are running. Take a look at what you've got there; it's about 50% higher right now. The truck area itself (based on 100-ton cars) has expanded about 62%. If you want to go further and get shocked, look at 1970 versus 1975. Same thing; it's up about 100%.

Light car hunting is certainly biting into us. We need improvement in this area—we need it now. The three component truck, I look on as a woman of ill-repute. She is with us and she will be there a long time. She needs improvement and she is aging, but the truck manufacturers have to know what amount of improvement we can afford.

I'll give you some of our experience in this area of cost. We have a number of unit trains and I rode one recently. The loaded train, averaging between 40 and 50 mph, ran smooth as a dollar. This has got about 400,000 miles on it right now. But it was "light" going back home. Those sons of guns were rolling it like mad and our trucks were jumping all over the rail and wearing out wheels, trucks, centerplates, bolsters, side frames, the whole bit. It's common practice from what I can see to go about 50 to 65 mph light on the return trip.

Our maintenance costs for these cars on a per mile basis is two times what it is on any other similar cars of identical design. In our fleet, cars X, Y, and Z can be identical, but for different services.

We need, although not immediately, an exotic new design. There are some limitations or restraints on dollars. But we do need now a modification to the present three component truck that will allow component wear, perhaps 80-90% of replacement point, without causing or allowing light car truck hunting at 40, 50 or 60 mph. A number of things can be done now to stabilize the truck in this area. We have heard some discussion on this, and I think this is money in the bank.

With that, I would like to go over briefly each of the presentations and then open the discussion for questions. Bob Byrne from the TDOP program discussed how they are approaching truck design optimization, and it's a very sound approach. First the project is determining how the present truck is operating and how this relates to the cost picture. Then Phase II of the program will be designating specifications for a new truck.

Bob Love talked about improved suspension in the 100-ton truck, including vertical, rock-and-roll, and hunting solutions.

Terrey Hawthorne brought up an interesting point. We have been into it many times and I have had many fights with truck manufacturers over who really is the designer and responsible agent for the car truck. That's a real moot point today, because there are so many modifications.

To give just one example, it's a tradeoff in dollars. We perhaps have one of the largest (or had one, because we are changing them over to roller bearing) one of the largest 100-ton friction bearing fleets, aside from perhaps two railroads, in the business. As car builders our wheel cost on the friction bearing 100-ton car is 14% less, across the board, mile for mile, than for a roller bearing car. That's not saying we want to go back to friction bearing cars, but we are paying a tradeoff in the deal; we are paying 14% more for wheels for a roller bearing car. It's a tighter, stiffer truck.

Harold List came into the picture with a new design, utilizing the radial truck aproach to stabilization of the car.

A question comes up about how much can be afforded in this area. Perhaps to throw it into some perspective, it would be better if we could eliminate entirely the thin flange wheel from the car owner standpoint, or from the car standpoint alone. What's it really worth to us? I think if you are keeping track of your miles, you will find that a thin-flange wheel will range from about 70,000 to 160,000 miles--than boom, it will go out. We have had them go out at 50,000 miles, but that's rare. We have also had them go out at 200,000, but that's where they were in several services.

If you think about thin rim or high flange, you are going to roll at about 140,000 to 250,000 miles plus. We have them up to 300,000 miles. Generally, you get your thin flange at about half wheel life. With 100-ton cars, if the wheel cost is $200 a wheel (plus all the labor to do it, but let's just take the wheel) and you have lost half your life in the wheel, you have $800 half life the 8th year, if you are going AAR 20,000 miles a year. But the accountant will tell you that you haven't got $800, you have the present worth of that today, and that's about half at 10% interest, so you are out $400 right there to start with. Of course, that doesn't deal with rail, with the problem of the bad order car, with the out-of-service time, the crowding of the Rip tracks, and so forth. That all has to be factored in.

Bob Bullock came through with the cross anchor arrangement and another approach to stabilization of the truck and improving the ride.

The last paper, Mr. Schrotberger's, was a very fine discussion on ORE and their work to date. I think here again Europe is showing us the way–they have been doing basic railroad research many, many years and were government oriented before USA got into the act.

With that, I would like to open the session to questions.

COMMENTS/DISCUSSION PERIOD

Delegate Comment: I think the attention of the papers this afternoon seem to be mainly on what we would call the cross-braced bogie, and there has been no consideration of a bogie or truck with a primary suspension, i.e. one which has elastic connection between a truck frame and wheelset in the lateral, the longitudinal, and the vertical direction. That seems a little strange, and it highlights what I think is a very big omission in the discussion today. That is, in looking at dynamic loads applied to the truck, axle load is important but so is the unsprung mass and its effect on the total dynamic course. So, in addition to improving the design of the truck in the lateral sense, one should be improving it, I think, in the vertical sense. One should be reducing the unsprung mass. I would have thought in those circumstances it would be worth considering a bogie which had a proper primary suspension rather than just a pad of rubber between the wheel and the wheelset and the truck frame.

That's a general point, and I would be interested in knowing whether any of the people who were talking about a cross brace this afternoon had done any comparative calculations between that kind of configuration and a more conventional primary suspension truck in which many of the same objectives would be met, but some additional ones would be met as well.

Panel Response: As regaras a primary suspension to reduce the vertical unsprung weight, some benefit is derived from the rubber pad between the side frame and the wheelset which is divorcing the side frame from the unsprung weight. You could use more, particularly on jointed track. Of course, if we can persuade people to weld the track, that might not be quite such a severe problem.

One of the problems with the primary suspension is achieving the lateral stiffness which is needed even if you have a high value yaw stiffness. If you wish to reduce yaw stiffness in the interest of curving, the lateral stiffness has to go higher yet. This raises the very serious question of how you interconnect the two axles effectively and maintain a high value of lateral stiffness. Then linkage for doing this gets to be more complex if you try to reduce the unsprung mass to the minimum possible value.

I think the economics of primary suspensions scare many of us in the United States, forcing many railroads to arrive at a solution. There have been two or three designs in this area. Unfortunately they never had a chance to be proven because the cost factors were so high. There was just no economic judgment to go all the way and sell it.

Delegate Comment: We, the track people, find that the track is weakest in the lateral direction. Most of the problems that we seem to be getting are due to forces applied to the track in the lateral direction. We would welcome any improvement in the vertical direction, but primarily the lateral forces are the ones we are fighting.

Second, as a layman, I think I can comprehend the 1.25g dynamic force level attained between 50 and 75 mph on the 16-mi. long stretch of track, but I find it rather difficult to understand that the standard truck with a D-5 Spring arrangement had only six such occurrences in the same stretch. Perhaps my misunderstanding may be in the condition of the track that I am envisioning, since the title of your speech referred to rough track. Could you give me some description of what the 16 miles of track looked like?

Panel Response: I can't give you a description of the track; we didn't identify it except from the measurement of the input to the truck. In other words, the truck was the test parameter for the track, you might say, but we did not go back and identify that track condition. When we search out a test segment of track, we want one that is going to give us a ride quality whereby we can see an improvement--in order words, we want to go to the boundary conditions of the suspension. This happened to be main line track, and we found the proper test conditions. We do this every year, and each year we find the conditions continually growing worse. But as far as identifying the magnitude of that particular bump, I can't give you a quantity--it was bad enough to drive the conventional suspension to solid.

I might mention that we have been working with the Track/Train Dynamics computer program and we applied three successive 1-in. bumps and ran a series of speed over that condition and with various conditions of springing and snubbing. In one instance we drove the spring solid to the same acceleration levels that we measured here at the centerplate and achieved a centerplate separation on the fifth bounch of the car which kind of interested me, but it was not necessarily totally unusual.

Delegate Comment: Can you tell me if that was a single occurrence on the T-11 suspension? Was that a turnout or a crossover?

Delegate Comment: That single occurrence was at a road crossing. It was going into the part of the crossing that was soft and then had a transition into a solid crossing. Now in both cases, measurements for the D-5 and the D-7, or the T-11, were taken at the same instance on the track and the D-5 are picked up I think five or six high acceleration counts as opposed to one for the T-11, which would account for the number of counts in the T-11. It damped out after the one single impact.

Delegate Comment: Do you have a target data for offering your anchor truck for sale to railroads?

Panel Response: No. This is an ongoing research project jointly between ourselves and the South African Railroad. They are slightly ahead of us in that they are going ahead with production of 1,800 cars sets, so we made the decision to wait and see how that works out and how that truck is sensitive to manufacturing variations. It's a little bit harder to design for our axle capacities--we have to be more careful in designing our side frames and bolsters, and it is going to take more effort than they had to put into it. For example, we are completely changing the cores in the bolster so a static and dynamic stress analysis is being conducted. These things have to be worked out before we can start tooling.

Delegate Comment: Would it be within two years?

Panel Response: I think so.

Delegate Comment: I would like to ask the same question about the radial truck. When will it be available for sale to the railraods?

Panel Response: If the tests which we will be running in December, 1975 are successful, we are aiming to have prototype construction in about 12 months. That would be very limited quantities.

Delegate Comment: Is there are trend toward higher axle loading on European railroads?

Panel Response: You certainly know the official limit of the 22-ton axle loads, with the exception of some special ORE lines or some special lines in Britain, for instance, where 28 tons are admitted. There is a study underway now to raise the axle load from 22 to 24 tons, which seems only a 10% increase. There are two problems here. First is the increase to 24 tons, and second the eventual introduction of the automatic coupler, will probably give an additional load of 500 kilos per axle. The raise from 22- to 24-tons axle loads was criticized by several Administrations because they are afraid that the profit gained from the higher axle loads and the better use of cars will be paid by higher maintenance costs. The results of this study is not yet available; tests will start next year on the Prague test circle to show what the deterioration of track is with 24-ton axle loads and especially how the rail is affected by it.

Delegate Comment: I understand that Soviet railroads are contemplating raising their axle loading. Do you know anything about that?

Panel Response: We have not enough information from them; they are not a member of ORE, thus limiting the amount of information we receive.

Delegate Comment: I have two questions. The first relates to whether or not ORE has run experiments on track at higher superelevation or cant deficiencies with these 125-mph locomotives. The second question is that you mentioned the effect of having lateral stiffness or a lateral flexibility on three-axle locomotive trucks to the outboard axles, but you didn't say anything about the center axle. Is anything being done there?

Panel Response: I can answer your second question first. The proposal was only to do it on the end wheelsets, the first and the third, not on the second. The second axle is coupled to the frame. Yes. And as to the first question I can say yes, but without giving you any figures. These tests have been made more than five years ago. Interestingly, track tests are now under way, running with a speed of 40 mph over track curves of 1,000 ft. radius without cant.

Delegate Comment: Do you know what that would mean in cant deficiency? Would it be equivalent to 3 inches cant deficiency--or 6 inches?

Panel Response: Running without any superelevation on that curve at 40 mph, that would be the equivalent cant deficiency of 4.5 inches in that curve. This has been put down to zero to get a maximum lateral force on the track.

Delegate Comment: So the experiments have really been run under what are considered to be very good rail holddown conditions?

Panel Response: Yes.

Delegate Comment: How could a U.S. railroad obtain an ORE report from ORE if they desired to get one?

Panel Response: Thank you for asking this question. We have received quite a lot of requests from U.S. railroads to get our reports. You should first go to the AAR because with the contract we have now with AAR, AAR gets all the reports we distribute. You can first screen the reports in Chicago and see if you think the report is valuable to your administration. You should then order the report through the AAR and the AAR will ask ORE to send the reports.

EVENING SESSION

Moderator Robert E. Parsons: It is indeed my pleasure to welcome this group for the second time today. We are delighted at the turnout we have had, and I believe the discussions were just great, reflecting the caliber of the speakers and the conferees as well. I would like to thank Ed Ward, our Conference Coordinator who has worked so hard behind the scenes to arrange what was a busy, but I think a very fruitful first day. I believe our second day will be even more so.

Those of you who have been delegates at previous Conferences have, I am sure, missed the presence of one of the dynamic personalities behind these Conferences. The conferees have owed so much to the diligence and skill of Jack Loftis. Our friend is missed, and I would like to again note with deep sorrow his passing. He worked hard to make the FRA transition to sponsorship of the Conference a smooth one.

The key point in the transition is the atmosphere of government/industry cooperation which is so vital to the success of all of our efforts. This atmosphere puts the Transportation Test Center in a dynamic position in the country and makes it possible for you in the industry to use it to get this rail industry back where it belongs.

This evening I would like to thank the industry representatives who have been so gracious in assisting us to create the congenial atmosphere necessary to promote this government/industry cooperation. The reception that we just left was hosted by the Dresser Transportation Equipment Division and they are inviting us to be their guests at breakfast tomorrow. We certainly appreciate it. Our host for this evening's dinner is ASF, and we likewise certainly appreciate all they have done. It is events such as these that help accelerate the exchange of information that generates new friendships and stimulates new ideas. This is what it's going to take to get the railroad back in the transportation picture.

I would like to introduce the gentlemen at the front table:

Paul Garin, Assistant Vice President, Research, Southern Pacific Transportation Company.

Bill Ruprecht, Director of Engineering, Shippers Car Line Division, ACF Industries.

Carl Sundburg, President, American Steel Foundries.

The Mayor of Pueblo, Melvin Takaki.

Bruce Flohr, Deputy Administrator, FRA.

Jack Stauffer, Director, Transportation Test Center.

Dick Lich, President, Dresser Transportation Equipment Division of Dresser Industries.

Bob Brown, Chief Engineer, Union Pacific Railroad.

Paul Settle, President, Railway Maintenance Corporation.

We are very privileged to have the Mayor of Pueblo, Mayor Takaki, here with us this evening. May I present the Honorable Melvin Takaki, with an address to the delegates.

Mayor Takaki: Thank you. Distinguished guests. It is certainly a privilege for me to welcome all of you here to our city. I must apologize for the bad weather we have had for the past few days, and if you think I'm apologizing now, we have a storm warning coming in from the north. With the dry climate that we have right now, the lack of rainfall and snowfall, you are really going to know what a wind storm is, I'll tell you. I think it may come after you leave, and I hope so.

It is kind of ironic – as I came before you this evening I was just planning on having dinner, but I was asked to welcome the group, and certainly it's my delight. But yesterday I had addressed the Vice President of the United States in a Public Domestic Policy Forum in Denver in terms of actual energy problems. The things that I was concerned with and the policy statements that I submitted had to do with the conservation of energy and the development of new resources, and how that development is important to us. Equally important I had also mentioned the development of the transportation industry and the need for new efforts in that line and for the coordination of all different modes of transportation.

This community is certainly proud of the Transportation Test Center we have here. We have been very interested in transportation for a long period of time. And as I thought about that and about the great group of people that have assembled here this evening, I realized that it is imperative that all of us become involved in the political process. I think you are all technically oriented. It is very important that you railroad people get politically involved in order to balance other interests out and take your rightful place in the transportation picture of the future. Without your involvement, the other forces are very strong. We need this balance. We need the balance of all modes of industry – a great new alliance, not only with the railroad people but with air and bus and anything else that moves. It takes an alliance between government and industry, and I am certainly glad to see that all of you are here making that effort.

Certainly anything we can do to make your stay more hospitable while you are here, we would be pleased to do. Thank you very much.

Moderator Parsons: Thank you very much. Dick Lich, would you like to say a few words.

Richard Lich: Dresser Industries is certainly pleased that so many of you are in attendance at the 12th Railroad Engineering Conference. We believe in these conferences, and we are convinced that they can be an increasingly significant and positive factor in progress for the railroad industry.

And it is certainly most fitting that these conferences be held in Pueblo, the site of the Department of Transportation's Transportation Test Center. The Test Center is a unique asset of the United States, indeed, of Northern America and other parts of the world, because of the common standards which are established here. There are no other facilities like it anywhere. But, like any asset, it must be utilized in order to produce benefits. We would certainly urge that railroads and railroad suppliers give the greatest consideration to fully utilizing the test center in their research and development programs. This will further support what I talked about today, the tripartite approach to research and development, which we believe will produce the greatest advances in railroad technology.

Dresser Industries is proud to be here in Pueblo with all of you. Thank you.

Moderator Parsons: Thank you Dick. Carl Sundburg would like to say a few words.

Carl Sundburg: I simply want to say that I think it was with a great deal of foresight and certainly of great interest to see that people like Dick Lich and the FRA people managed to have this conference here in Pueblo and to turn it over to the FRA to the point where others of us can participate. Free exchange of information and the opportunity to do such things as we did yesterday and what we are hearing today and tomorrow are all very significant in overcoming the problems that we all are facing every day. We have enjoyed it very much, and we are looking forward to coming out here again and participating to the fullest. Thank you.

Moderator Parsons: Thank you Carl. As all of you know, we were scheduled to have two other distinguished guests with us this evening, but as we mentioned this morning, pressing legislative business kept both Congressman Rooney and my boss, FRA Administrator Hall, from being with us. Mr. Hall, however, has sent us his number two man. Some of us are pretty lucky if we have a boss we get along with, and I am in the very envious position where I have two guys I think are great to work for. Both of them are topnotch and represent the type of leadership we need on the federal side to work with the type of leadership we have on the industry side to do the job that has to be done to accomplish the goals of improving rail safety and ensuring economic revival in the private sector, the free enterprise system.

This may be the first meeting with the new Deputy Administrator, but I am sure it won't be the last. You may be interested to know that Bruce Flohr received his B.S. in Industrial Engineering at Stanford and his Masters in Industrial Administrator at Purdue. He was with Southern Pacific for ten years, one of their division superintendents, before joining FRA. It's my great privilege to introduce a great boss, Bruce Flohr.

Bruce M. Flohr
Deputy Administrator
Federal Railroad Administration

KEYNOTE ADDRESSS

It is a pleasure for me to be here with you this evening. Tonight I would like to talk about the rail transportation challenge and what the U.S. Department of Transportation is planning to do to meet that challenge.

Before I begin, however, I would like to read the telegram that Congressman Rooney sent to you, the delegates. "I'm disapointed that I'm unable to participate in the 12th Annual Railroad Engineering Conference at the University of Southern Colorado. Unfortunately, the House of Representatives has scheduled for today, Wednesday, and/or Thursday the emergency Rail Transportation Improvement and Employment Act, and I am scheduled to manage this bill on the House floor. Nevertheless, I hope you will convey my warm regards to everyone in attendance as well as my best wishes for a most productive conference." Signed, Fred B. Rooney, Chairman of the Subcommittee on Transportation and Commerce.

Congressman Rooney wanted to attend this conference and have the chance to meet with each of you to share his views on current transportation industry problems. It is significant that we do have the attention of Congress now, finally, after so many years. Rail transportation is recognized as a major problem area that must have positive action now. Congress realizes it, and Congress and the Administration are willing to accept the challenge to produce that positive action.

William T. Coleman, Jr., the Secretary of Transportation, has recently issued a National Transportation Policy Statement which I believe is the most comprehensive overall policy statement issued by any cabinet officer. We, in the Department, are fortunate to have a Secretary who is willing to state what he thinks, to give direction, not only to the people within his Department, but direction to the Congress, direction to the general public, and, of course, direction to the various industries involved in transportation.

Many have said that possibly this is politically unwise, because any time you put something in writing people have a chance to take shots at it, to criticize it. This is true, but now is the time that we must keep transportation problems before the nation. We want this attention and we must keep this problem before the general public so they can be informed and advise their Congressmen.

Secretary Coleman begins his report with the following statement: "The development and modernization of a nationwide privately owned intrastate rail freight system, preferably providing at least two competing lines between major industrial points, is essential to the national interest." The current Administration is going to make every effort possible to keep the railroad industry in the private sector.

There are five basic areas in which the Secretary proposes to initiate this program. First of all, we will provide assistance to the industry in restructuring the system along more rational and efficient lines. When thinking about the word "restructuring" it has some bad connotations for some people, but let's look at it a little more closely. We do not want your – and my – taxpayer dollars to go for the perpetuation of duplicate rail facilities. They are not necessary; and we don't want to spend our money that way. At the same time, however, there are many restructuring projects currently on industry drawing boards that only a question of money is keeping from being initiated. These are projects that the railroad industry wants to do itself; things that the government will not dictate, that the carriers will willingly and cooperatively get together and accomplish.

For example, in Spokane, Washington, there was the old Great Northern and the old Northern Pacific. They merged into the Burlington Northern. What did this do for the City of Spokane? It permitted the Great Northern to move their facilities out of the downtown area, an area that had high development potential but was intersected by many surface grade crossings which were continual public image problems because trains blocking the tracks prevented motorists from crossing them. They moved the rail system over into the Northern Pacific area, a viaduct situation completely separate from any street crossing. This released the Great Northern property, most of which was used for the construction of the World's Fair, Expo '74. At the same time, the Burlington Northern was able to initiate many operating savings when they moved to new terminal areas.

At the present time in St. Louis, 19 carriers have gotten together and have come up with a proposal that will make it possible to remove numerous duplicate and antiquated rail facilities from the east bank of the Mississippi River. This is something that the City of St. Louis very much desires, it is something that the carriers all want, but there is a high capital cost problem. The carriers joined together to come up with a feasible system, and with the cooperation of the FRA this plan is moving forward. Certainly the monies necessary to complete many parts of the project may come from what is now in the pending legislation before Congress.

The Seaboard Coast Line has already approached the Federal Railroad Administration with examples of three different cities where they have duplicate terminal operations; the old Atlantic Coast Line and the Seaboard Airline separate yard facilities. By building new terminals in each city, they would not only be able to eliminate antiquated yards, but they would be able to initiate many operating efficiencies with a new, modern switching area.

The second point that the Secretary brings up is that we must have reform in the economic regulatory structure. There are really two areas that should be addressed in this field. One is the problem of modification in service. An example is the Rock Island – it took the ICC 12 years to come up with a no-decision situation followed by the bankruptcy of the Rock Island. This country can no longer tolerate this approach to the protection of the service to the shipper. We have to have an agency that is more responsive to these needs. All sectors should be heard and appreciated as to their various needs. Still, we must have action taken in a much more reasonable time frame than was the case with the Rock Island.

With this economic regulatory reform, we must have revision in the rate structure area – more flexibility in the rate-making field. We must look at the rate bureau and determine whether it is any longer a necessity. The whole area of competition has changed radically in the past 10 to 20 years, and we have to look at not only whether rates will possibly go up where necessary, but equally important, whether rates will be going down. We cannot allow this process to be hindered unnecessarily by the regulatory practices currently in existence.

Third, remedy must be had for the inequity of Government subsidy to the major competitors of the rail industry. We have to wake up and realize that the water carriers are totally subsidized; they pay not one cent for the locks, for the dredging, for the navigational aids. These are very expensive, and yet no user charges are paid. Truckers do pay some user charges, but certainly nothing commensurate with what is necessary to truly bring about a fair and equitable balance among the competing modes of transportation. Your Secretary (and my boss) is firmly committed to the principle that we must correct the situation of subsidy payment to only certain sectors of the transportation industry. If they are all treated equally, then we really will have a fair competition amongst the various modes.

Fourth, we must encourage continued development of more efficient labor and management practices. I get a little upset at times because one of the images the general public has of the railroad industry is that it is mismanaged. The comment is often heard: "Is this any way to run a railraod?" It really means something to many members of the general public who do not truly appreciate the problems within the industry. Well, my first reply is that the railroad industry and its management have some of the most capable people in any area of transportation. How in the world else could this industry have survived so long in such an inequitable position in the marketplace? And yet the rail industry has survived.

So, first impression wrong – second impression work rule reform. People ask why in the world doesn't the railroad correct the work rule problem, the featherbedding issue. Well, the railroad industry has taken many steps to correct the work rule problem. The industry has a long way to go, but make it a point whenever you talk to someone representing the general public that much has already been accomplished. Take the issue of firemen; this has been resolved, and it's been resolved not only to the economic benefit of the employees that were affected, but there was no long-term strike situation, which not only the industry but also the users of the industry could not endure. The steel mills, the auto assembly plants, the grain movers, they could not stand a long-term strike, and the industry had to correct this labor problem area.

As another example, up until four years ago there were still some states that had on their books the law that required three brakemen on every train. People overlook this fact. One state was Arkansas, and only five years before that another state was Oregon. The industry has gone out to these states, and, with the cooperation of labor, through job protection provisions, this labor-management problem has been resolved to the satisfaction of everyone.

Right now in St. Louis there is a joint labor-management project to change to work rules to bring about benefits to both. This is a joint effort between the Federal Railroad Administration, the Missouri Pacific line, the affected labor organizations, and the Association of American Railroads. What they have done in this group is to establish a total of 18 experiments that are now in the process of being performed. Five of these experiments are already concluded. These situations address problems such as the deadheading of motive power from one terminal area to another terminal area and doing it without a full complement of crew to handle the movement. Prior to the test, the Missouri Pacific moved cars in only one direction with one seniority grouping, and returned without cars. Now many of the seniority barriers have been eliminated, and there is cross movement of cars by the same crew.

Critical to this whole operation is the cooperation of management and the guarantee to the individual working out in the yards that if he suffers a loss in pay, he will be fully reimbursed. Out of the ten experiments in operation and the five that have been completed, a grand total of only $630 has been paid out in lost wages to all those adversely affected. At the same time, car movements were improved by at least 10 hours per car on a volume of about 200 cars per week. It is estimated that within the whole terminal there has been at least an average improvement of four hours in the movement of cars. Not only has this been a benefit to the carrier, but it has been a benefit to the shipper, because the shipper is getting better service. And certainly it has broken down one of the basic apprehensions of labor regarding the protection of employees. Certainly this is only a start, but it is significant, and we hope it will spread throughout the industry. Don't ever let the industry be sold short with criticisms that they are not addressing themselves to the work rule problem.

Finally, the transportation policy addresses itself to the light-density branch line situation. State and local governments or shippers must assume the responsibility for light-density branch lines outside the interstate freight system, with some transitional Federal economic assistance. If the local communities are not willing to bear a portion of the burden, then the Federal Government cannot be expected to subsidize such operations idefinitely. We have to look critically at many of these light-density branch lines.

Secretary Coleman addresses every one of the major problems within the rail industry. In the rail passenger area, he says that a determination is needed on whether rail passenger service provided by Amtrak without Federal subsidy can compete with other passenger modes. If it cannot, a basic policy decision is needed on whether national priorities justify long-term Federal subsidy and, if so, at what level. In other words, what the Secretary is asking is whether, if Amtrak cannot be self-suporting, you and I, as taxpayers, want to see a nickel per mile paid for every passenger that is hauled on an Amtrak train? It is a political decision. Do you want your Congressman to vote in favor of continued Amtrak subsidization?

There is current related legislation in Congress addressing the basic problems of regulatory reform, interim money for rebuilding so that the carriers can get back to a normalized maintenance situation, and the necessity for restructuring, because we do not want the perpetuation of duplicate facilities. Along with this, the Federal Government, the FRA, is totally committed to providing a leadership role in the area of research and development. We realize that substantial capital costs are necessary in order to construct test facilities to provide the basis for industry decisions. We now have the Transportation Test Center here in Pueblo. We hope to have an interim facility for accelerated service testing known as IFAST in operation by the end of next summer, and I'm hoping that as you get a better appreciation of what your individual needs are, you will give us the guidance so that we can schedule as many tests as possible to take full advantage of the facility.

We want to take the role of assisting the industry. No longer can the FRA be considered as a policeman for the industry. We have a safety role, and we will not deny the importance of this role. So much of the testing that is going on here at Pueblo and will be going on in the next several years addresses this very basic need of providing safety in our rail transportation product. But equally important, we are here to help the private-sector railroad industry return to economic viability. We look towards the time when such a rail industry will provide the kind of service that our shipping public needs, wants, and will be willing to use on a long-term basis.

I thank you very much for your kind attention.

Moderator Parsons: We certainly appreciate your remarks, and I think all of us in the research end of the business apreciate the support you and Administrator Hall are willing to give us in terms of providing the facilities to meet those needs.

Leavitt A. Peterson
Director – Office of Rail Safety Research
Federal Railroad Administration

Leavitt Peterson's FRA association began in 1974 when he left the position of Director – Applied Research of the Bessemer & Lake Erie Railroad Company in Pittsburgh to become Chief – Rail Systems Division of the Office of Research, Development and Demonstration of FRA. He began his railroad career with the Elgin, Joliet & Eastern Railway Company in Chicago in 1957 as an industrial engineering assistant and subsequently accepted several intermediate positions prior to joining the Bessemer as Manager – Operations Research in 1964. He received the B.S. degree in General Engineering at the University of Illinois and the M.S. in Industrial Engineering from Illinois Institute of Technology, and is a graduate of the Transportation Management Program of Stanford University.

Peterson has been active for a number of years in investigations and presentations on wheel/rail interactions, including publications for the American Society of Mechanical Engineers, Dresser Annual Railroading Engineering Conference, and the AAR/RPI/FRA/TDA Track/Train Dynamics Program. As Director of the Office of Rail Safety Research, he is responsible for FRA research work to improve the safety of track and rail vehicles and for the development of appropriate inspection technology and testing facilities.

SESSION III

TRACK/TRAIN INTERACTION

This is Session III of the 12th Annual Railroad Engineering Conference, and the subject is "Track/Train Interaction." The theme of this session is really only a formal recognition of what has been frequently expressed in the first two sessions and what we all realize – that separate concerns for track and the vehicle must be united in a broader systems approach aimed at optimizing the interaction between the two.

A Track/Train Interaction systems approach is the means for making necessary trade-offs to yield payoffs in both safety and economics. While the existing facilities and future plans for the Transportation Test Center here at Pueblo, and which you will tour as a part of Session IV, are intended to provide assistance in understanding the complexities of this highly interactive system, the backbone of meaningful R&D is a thorough and comprehensive research activity. The joint Government/industry effort represented by the AAR/RPI/FRA/TDA Track Train Dynamics Program has proven very effective in providing the means to focus these efforts to produce tangible achievements – and many of you in attendance, including most of the speakers in this session have been more than casually connected with this cooperative program.

The previous two sessions have laid the groundwork for this afternoon's presentation. At the risk of being presumptuous, I would suggest that there were a couple of searching questions posed in Sessions I and II respectively, that have a counterpart in this Session. Namely:

1. Who is the track designer?
2. Who is the car or truck designer?

In this context, Session III asks:

Who is the track/train interaction designer? In the presentations which will follow, a number of approaches to dealing with elements of this question will be explored.

Our theme address speaker is one of those who for a number of years has been actively engaged in dealing with the implications of this question and, more importantly, in ensuring positive accomplishments in this area. He is Paul Garin, Assistant Vice President for Research of the Southern Pacific.

Paul V. Garin
Assistant Vice President – Research
Southern Pacific Transportation Company

Paul Garin has been with the Southern Pacific for 40 years, moving up through successively more responsible jobs to his present position as Assistant Vice President-Research, which he attained in 1970. He is a co-inventor holding patents on various railroad car devices and has written approximately 50 papers for technical and railroad publications.

Garin has visited the U.S.S.R. several times as a railroad delegate on technical exchange missions and has been honored by the Pan American Railway Congress with several awards. He is a Fellow in the American Society of Mechanical Engineers, Honorary member of the American Society for Testing Materials and member of Tau Beta Pi and Eta Kappa Nu Engineering Societies. He received the B.S. degree in Electrical Engineering from the University of California.

Garin is a co-inventor on patents for "Apparatus for Shipping Automobiles," "Apparatus for Opening and Closing Door Pivotally Attached to a Railway Car," and "Door Raising and Lowering Means for Railway Cars."

THEME ADDRESSS

Yesterday we heard a number of interesting and informative papers on heavy loading on track, new truck designs, and vehicle suspension. The theme of this morning's session, Session III, brings these subjects together. The dynamic interaction between equipment with steel wheels moving coupled-in trains on steel rails is, in the final analysis, what rail transportation is all about. We must look at this as a total dynamic system in order to exploit fully our mode of transportation.

During the past few years, there has been a growing awareness of the importance of the dynamic environment of railroad operations, including the interaction between locomotive and cars as individual units, and the dynamics of this same equipment when coupled in trains with its corresponding effect on track, roadbed, and structures. The international government-industry program on Track/Train Dynamics has done much to bring home the importance of this interaction to those concerned with equipment design, train operation, and maintenance of way. The need for accelerated dynamics testing, both by simulator and test track loops, is nowadays being given the attention it deserves. The freight car Truck Design Optimization Project, TDOP, discussed at yesterday's session is another excellent example of modern thinking in this field.

In the past, equipment designers were principally concerned with the static characteristics of cars and locomotives as individual units. These static guidelines, such as axle loads and clearances, with appropriate built-in safety factors, were considered adequate in years gone by. Dynamic environment had never been fully investigated for use in a systems approach to equipment design. The general trend was to get the most in the car within the maximum allowable axle load and clearances for operational reasons, without investigating the resulting dynamic performance and interaction in train service. One exception was the counterbalancing of driving wheels on steam locomotives, where the dynamic effects on the track and locomotive ride could not be overlooked.

The introduction of larger cars, higher centers of gravity, more powerful locomotives, faster or longer trains, and unit trains has emphasized the need for and importance of this research work.

Now we are necessarily and rightfully concerned with how railroad equipment performs dynamically, how it interacts with other equipment in the train in an integrated manner, and what effect moving equipment has on the track and structures.

Other factors, such as establishing speed zones in areas of short reversing tangents; placement of long, light cars in trains; train handling; and alignment control on locomotives, all enter into this complex railroad operating environment. Such parameters as L/V ratio, harmonic roll, longitudinal train action, and lateral train stability are subjects of extensive study in the T/TD program. Information in these areas is currently being disseminated. Mathematical modeling has been accepted as a valuable method of predicting train performance, determining optimum operation, and analyzing derailments.

Economic trade-offs resulting from present and projected operations must be recognized and understood. Excessive costs can result from undesirable train/track interaction, including potential derailments, personal injuries, loss and damage to lading, equipment wear and damage, and accelerated deterioration of the track structure. In short, we must have a better understanding of the dynamic environment in which trains operate, including the forces developed by locomotives and cars in trains, the stability and reaction of track structures, and the manner in which the engineer handles the train. We must understand what we are dealing with today in order to move ahead in the future. We must handle traffic with larger equipment, faster speeds, longer trains. We can't go backward, we must go forward. This requires understanding of what must be done to optimize the conditions under which we operate. In the U.S.S.R., where maximum traffic density is the goal, trains operate at about 50 mph in parade fashion – no overtaking. Shipper demands differ, of course. However, they are considering increase in axle loading.

Dynamic testing of new and existing freight car designs must be expanded. Computerized analysis of designs should be more widely used. Before hundreds of car designs are built, we should develop dynamic characteristics, not only for design and maintenance reasons, but to find out how the design responds to force inputs it will encounter in actual train service. The facilities at Pueblo, both the dynamic tester and proposed "FAST" track, can be used for this purpose.

Papers and discussions at this session will touch on the aspects of these challenges, related both to present and future railroading.

It is particularly gratifying to see the ORE represented in this session. This Office of Research and Tests of the UIC is today a most significant voice in railroad research. Their accomplishments are well recognized throughout the railroad world. The paper presented at yesterday's session (which was scheduled for this session) gave us an insight into the work of the ORE, now celebrating its 25th anniversary. From personal experience, I can recommend their technical reports which they will make available through the AAR.

Research overseas has realized the advantages of dynamic testing. For example, Alan Wickens and his associates in BR Research at Derby have done outstanding work in this field. SNCF has an excellent facility at Vitry. JNR developed their high-speed trains using their dynamic test facility at Kunitachi, which incidentally was used to evaluate the AAR truck – the work that led to the TDOP project discussed yesterday by Mr. Byrne.

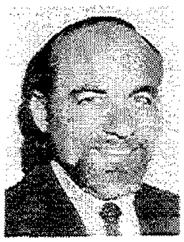
Test loops have been used extensively. We learned of one at Prague yesterday. The Soviets have a multitrack test loop at their research laboratory near Moscow and keep about 100 cars committed to this testing.

The JNR has a derailment track at Karakachi where fully instrumented cars are released on the grade up to the point of derailment (sometimes beyond) to determine their P/Q derailment ratio. So if we can it L/V and the ORE calls it Y/Q, we are all seeking the same basic knowledge. Trucks with primary suspension mentioned by Mr. Wickens may be economically difficult to justify under our conditions, but we should certainly include this design in our considerations on a comparative performance study. European research can furnish information on this type of truck.

I was interested in the discussion on wheel tread geometry, conicity, and so on yesterday. Apparently, after all these years, we do not have basic agreement on the optimum wheel contour. This is a worldwide problem. Much work has been done abroad. The JNR has developed a matched wheel and rail contour for their high-speed equipment. In our own work, we have found dramatic improvement in lateral ride quality on the same car by changing the wheel tread contour.

What is the optimum wheel tread contour? If tapered, what should be the degree of taper – 1:20, 1:40, or even 1:5, as we heard discussed yesterday? Cylindrical wheels and circular are treads have also been tried. Computer programs should help us predict the relationships of wheel/rail geometry.

There are many common problems and challenges facing the railroads, both here and abroad. We all have much to learn from each other. A case in point is the possible trend toward higher axle loading in Europe, which undoubtedly can benefit from our domestic experience. Likewise, the significant work being done overseas by ORE and others in many fields of railroad technology can be of much benefit to us. I have long advocated international cooperation on railroad technology and greater sharing of knowledge for mutual benefit. It goes without saying that the FRA railroad engineering conference is an excellent forum for bringing together and sharing the results of our ongoing research on an international level. We can all look forward to the continuation of this productive association in future years as activities at the Transportation Test Center expand.



Dean de Benedet
Director of Operations
Colorado Springs Facility
Wyle Laboratories

Dean de Benedet is Director of Operations of the Wyle Laboratories Scientific Services and Systems Group Colorado Springs Facility. He joined Wyle Laboratories in 1970 as a Senior Project Engineer and progressed through various technical and management assignments to his present position.

Mr. de Benedet was graduated from San Diego State College in 1963 with a BS in Mechanical Engineering. He has completed post graduate courses in Engineering Mechanics at both San Diego State College and the University of California at Los Angeles.

RAIL DYNAMICS SIMULATOR

Thank you Mr. Peterson . . .

I am very pleased to have the opportunity to introduce this conference to the railcar testing facility recently activated at the Rail Dynamics Laboratory located at the Department of Transportation Test Center in Pueblo, Colorado. This testing facility was designed and constructed for the purpose of assisting government and industry in the evaluation and characterization of the dynamic behavior of railcars equipped with two axle trucks. Included in this mornings presentation is a series of twelve figures which depict the configuration of the testing facility, known as the Vertical Shaker System, and a brief description of the system performance capabilities. In addition, I have included a brief summary of a current test program being conducted by Wyle Laboratories, under contract to the Federal Railroad Administration, of an 89 foot flatcar with various payload configurations.

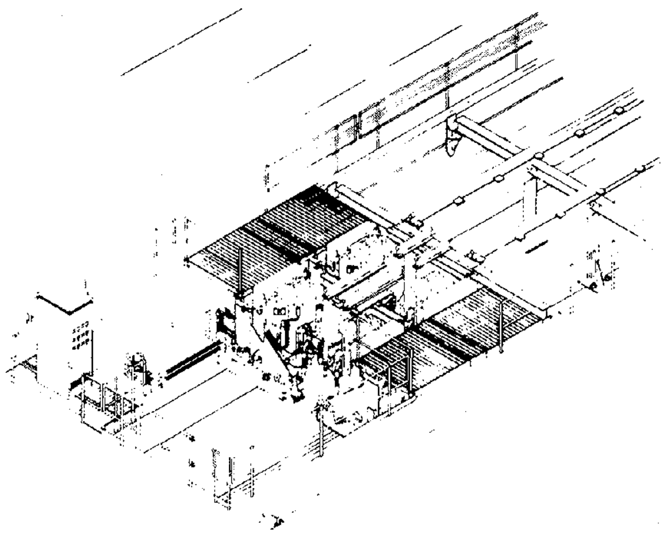


Fig. 1. General configuration of vertical shaker system installed in the rail dynamics laboratory, Transportation Test Center, Pueblo, Colorado.

Figure 1 illustrates the general configuration of the mechanical portion of the Vertical Shaker System installed within the test pit at the Rail Dynamics Laboratory. This equipment consists of four "Excitation Modules" that allow for the independent vertical excitation of each wheel of a two axle truck. This independent motion capability, therefore allows a user to specify any combination of inputs that include:

- vertical translation of the entire truck assembly;
- roll motion of both forward and rear axle sets; or
- pitch motion between the forward and rear axle sets.

Each of the Excitation Modules is equipped with a servo controlled hydraulic actuator designed to support wheel loads up to and including 40,000 lbs. In addition, a 100% transient overload capacity has been designed into the actuator system in order to accommodate load transfers across the axle sets that may occur during severe roll oscillations of a high center of gravity car. The static wheel loads are taken out through a low frequency suspension or bias system acting in parallel with the actuator assembly, thereby maximizing the amount of actuator force available for wheel excitation. Further, the excitation modules can be configured for any increment of axle spacing between 54.0 inches and 108.0 inches any any increment of gauge from standard to 5 feet 6 inches.

The displacement capabilities versus frequency for each excitation module are shown in figure 2.

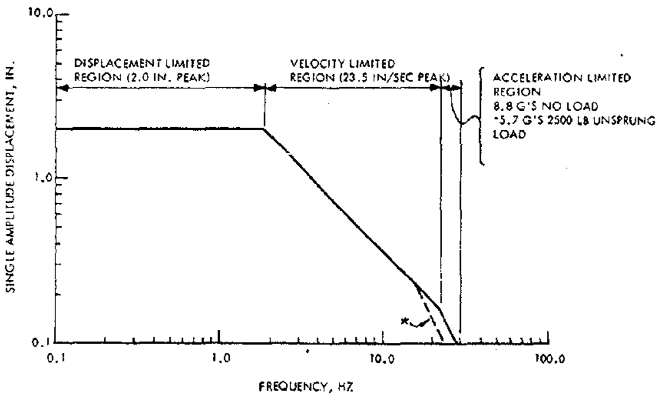


Fig. 2. Vertical shaker system displacement vs frequency capacity.

As illustrated a constant displacement of 2 inches peak can be maintained from DC to 1.87 Hz at which point flow saturation of the hydraulic power supply is reached and excitation is limited to 23.5 in/sec up to a frequency of 23 Hz. At 23 Hz the system becomes force limited as a result of the hydraulic power supply source pressure of 3000 psig and actuator piston area of 20.0 in.^{2}. The 8.8 g acceleration limit indicated corresponds to an unloaded excitation module. Also, included is an example of the acceleration limit associated with a 2500 lb unsprung weight which can be considered typical of 25% of the weight of many freight trucks.

Figure 3 is a transformation of the displacement versus frequency capabilities shown in figure 2 to a displacement versus simulated vehicle speed as a function of various vertical profile wavelengths. This figure illustrates that the system capacity is such that vertical irregularities (profiles) can be introduced which exceed, by an adequate margin, what a vehicle would experience traveling over the road. This will allow users to evaluate the full extent of vehicle/truck nonlinearities in a laboratory environment. As in figure 2, the broken lines again reflect the limitations resulting from unsprung weight of the truck.

The hardware system previously described is coupled to an analog servo control, digital computer and data acquisition system designed to provide:

- closed loop control from DC to 30 Hz;
- sinusoidal sweep, or discrete frequency signal
- generation;
- analog to digital conversion and acquisition of 128 data channels; and
- post test processing of acquired data.

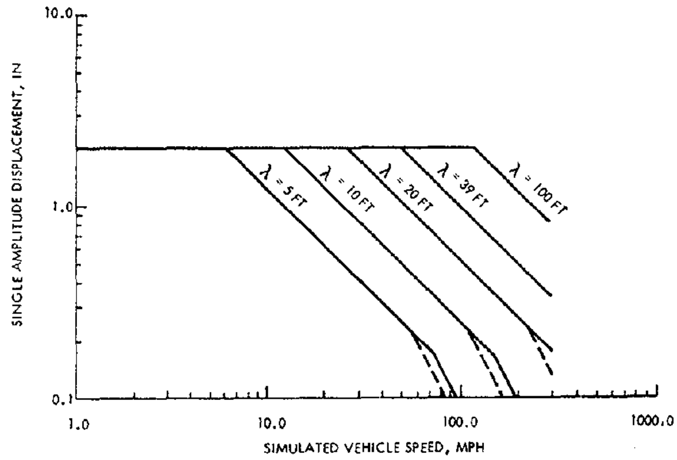


Fig. 3. Vertical shaker system displacement capacity vs simulated vehicle speed as a function of wavelength.

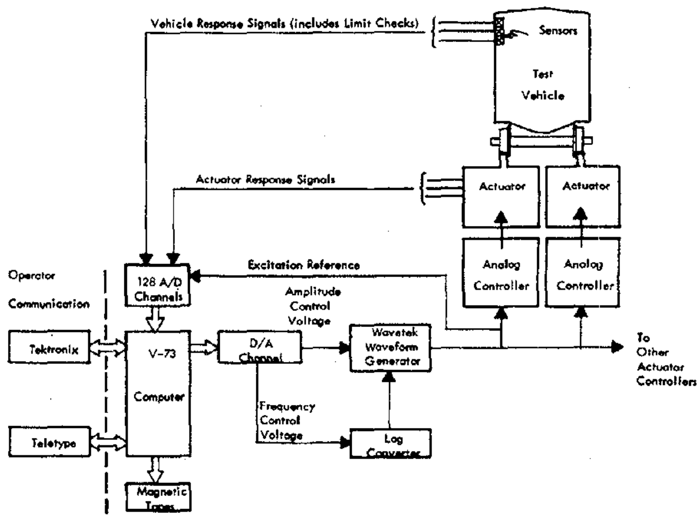


Fig. 4. Block diagram of vertical shaker system sinusoidal sweep test configuration.

A block diagram of the sinusoidal sweep configuration is presented in figure 4. It should be noted that signals proportional to the desired amplitude of excitation and sweep rate are generated by the digital computer which in turn are fed into the analog portion of the control circuitry. The operator may specify an input spectrum made up of as many as 1024 discrete amplitude/frequency points via the teletype. This spectrum can take any shape within the actuator limits specified in figure 2. For sinusoidal sweep testing the phase relationship between any pair of excitation modules is restricted to 0° (in-phase) or 180° (out-of-phase). This restriction is due to the limited time available within the digital computer as signal generation and acquisition are performed in parallel. Current plans for expansion of the system allows the introduction of variable phase differences between excitation modules. As stated previously, there are 128 channels of data acquisition capability available, portions of which are assigned to control - leaving 96 total channels for the user. Provision has been made in the operating software for preprogramming of limiting values for selected response channels to insure that over test conditions are not encountered. The limit check can be triggered on the first, second, third etc. exceedance at the users option.

For the purpose of conducting discrete frequency dwell and decay type excitation the control system is reconfigured by a patch panel as shown in figure 5. In this case signals are generated

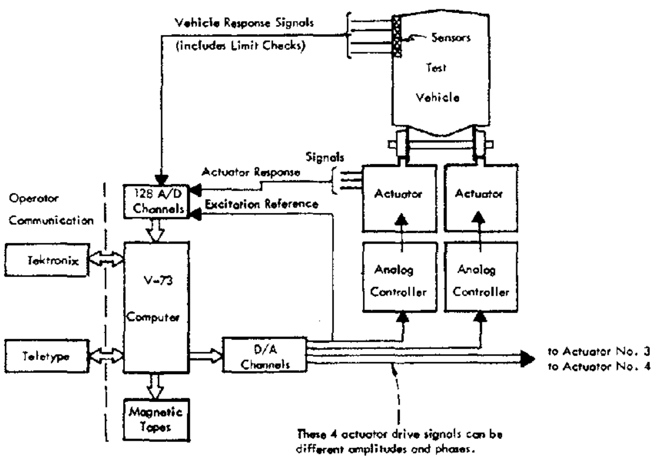


Fig. 5. Block diagram of vertical shaker system sinusoidal dwell test configuration.

completely by the digital computer and total independent drive of each excitation module is affected, thereby allowing the operator to increment or decrement the amplitude and phase at each wheel in any prescribed fashion. The operating software included with this configuration allows for the instantaneous termination of the input waveform in order to acquire damping characteristics of the particular response mode being excited.

The Vertical Shaker System was activated on the 4th of October, 1975, and the first characterization program is of an 89 foot flatcar with various payload configurations. This program is being conducted by Wyle Laboratories under contract to the Federal Railroad Administration and in cooperation with Trailer Train and Trail Mobile. Three payload configurations are being studied as shown in figure 6. Configuration 1 is the

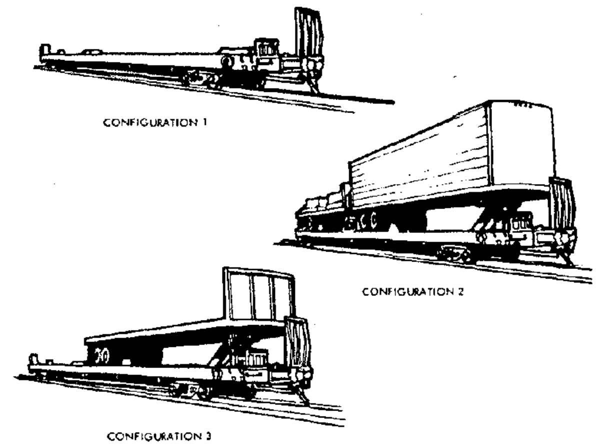


Fig. 6

evaluation of the unloaded flatcar and is now in process. This configuration will undergo a series of demonstration tests this afternoon at the test center; and I encourage you to participate in this demonstration. Configuration 2 includes two vehicles - a van and trailer each loaded with 50,000 lb of ballast. Configuration 3 is an evaluation of a single empty platform trailer loaded on the flatcar. The primary purposes of evaluating these three particular configurations is to provide test data to assist in the validation of analytical models currently being developed and to support future testing activities associated with the follow on trailer-on-flatcar program.

The response measurement locations associated with the truck and vehicle are shown in figures 7 and 8, respectively. The instrumentation consists of accelerometers, displacement transducers, and force transducers. Transducers are located such

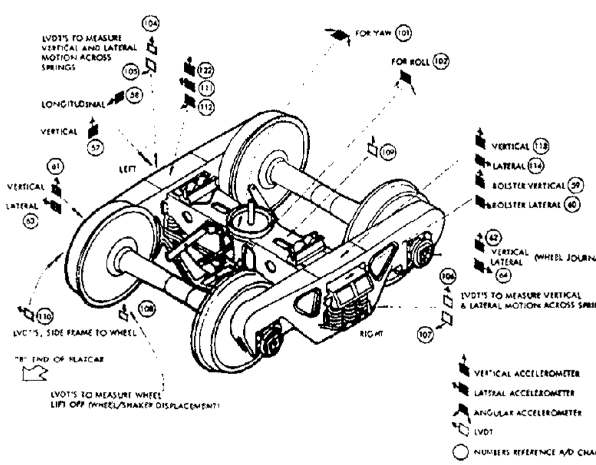


Fig. 7. Demonstration test plan complete truck instrumentation B-End truck.

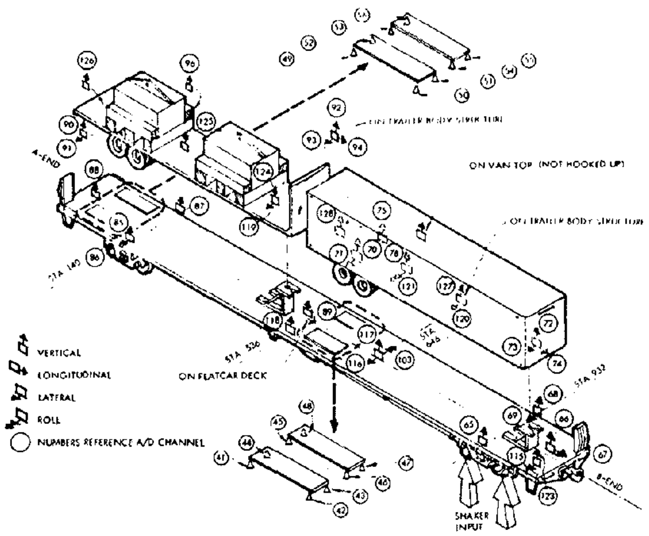


Fig. 8. Demonstration test program complete instrumentation for flatcar and trailer bodies 89'4" trailer train flatcar with trailers.

that rigid body motions and at least the first two flexible modes of the flatcar and trailers can be determined.

I noted previously that analytical models are being developed in conjunction with the experimental program. The general form of the analytical model under development is as shown in figure 9. The model includes nonlinear elements in

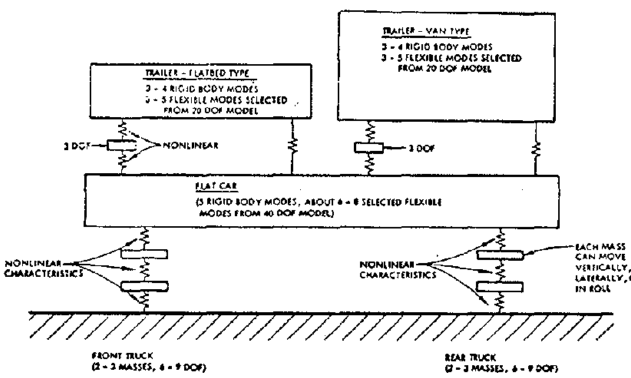


Fig. 9 Proposed TOFC model

both the flatcar truck and trailer tandems. The chassis of the flatcar as well as the van and trailer are constructed using finite element techniques and then the significant normal modes coupled into the composite model. We had hoped, for the purposes of this conference, to have some comparisons of results obtained analytically and those from laboratory testing available for your review. However, as I mentioned previously the Vertical Shaker System was activated on the 4th of October and we have not had sufficient time to prepare these comparisons. We have had the opportunity, however, to perform some validation work on a model of a loaded 100-ton hopper equiped with ASF ride control truck which includes similar types of linear and nonlinear elements as that used for the flatcar model.

The basic configuration and associated degrees of freedom of the 100-ton hopper car model is illustrated in figure 10. Experimental data used for

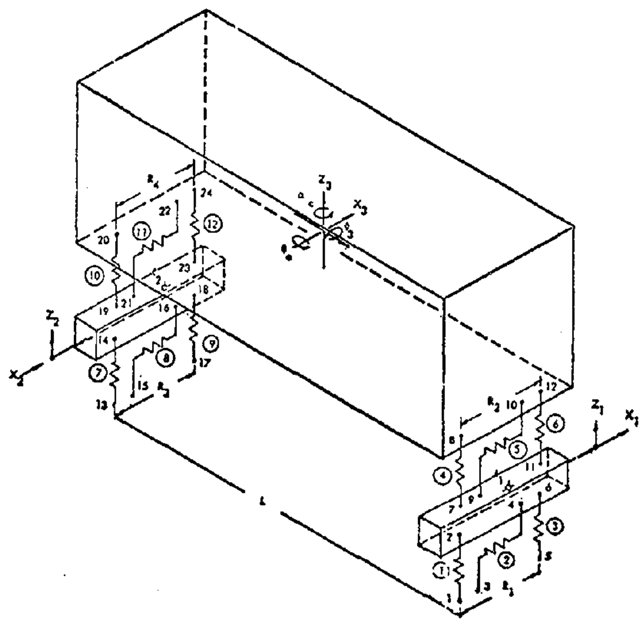


Fig. 10. Railroad car three dimensional model 11 DOF.

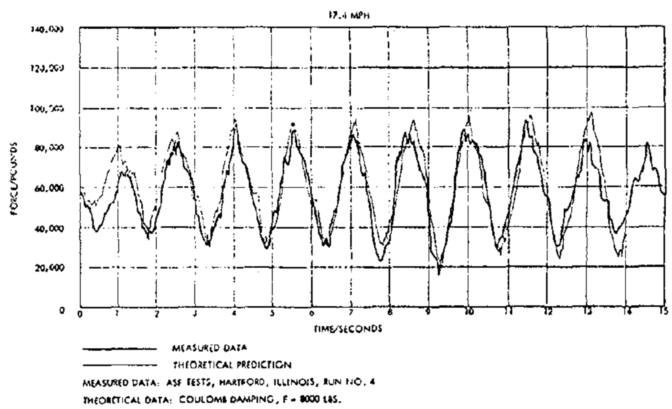


Fig. 11 Vertical force on side frame vs time

validation purposes was provided to us by ASF. Figure 11 provides a comparison of side frame force as a function of time. In this particular case coulomb damping was used in the math model which we find provides a better high frequency resolution in the response. Figure 12 presents another set of comparative measurements. In this case car body roll angle versus time are shown. We are very pleased with this effort to date and are

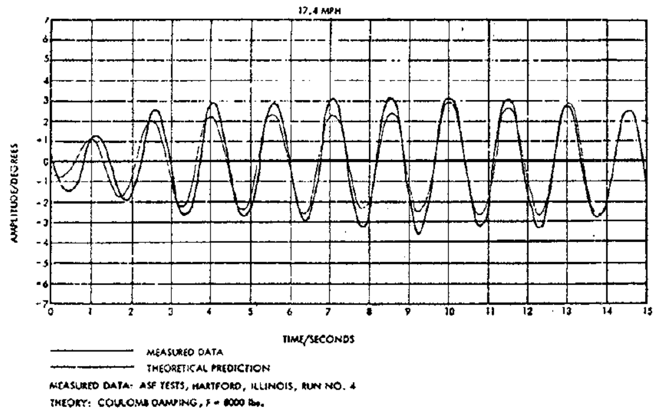


Fig. 12 Car Body roll angle vs. time

confident that a similar predictive tool will be available with the flatcar model.

The current schedule for completion of the testing and modeling efforts is mid January 1976, at which time a report will be issued documenting both experimental and analytical results. We look forward to the review of this documentation by attendees of this conference and will be interested in any comments you may have regarding approach, findings and conclusions presented.

Thank you.

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Prior to joining Electro-Motive Division in 1969, Vijay K. Garg taught at both graduate and undergraduate levels. He has been awarded the following degrees: B.S., Civil and Structural Engineering, Banaras University; M.S., Structural Mechanics, University of California, Berkeley; and Ph.D., Civil and Structural Engineering, Illinois Institute of Technology.

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As a truck and underframe engineer at Electro-Motive Division of General Motors Corporation, Kenneth Mels has participated in numerous studies and projects, with emphasis on stress analysis of a variety of locomotive components, including engines, underframes, and trucks. He has also been involved in numerous wheel-to-rail creep and adhesion studies.

Mels received the B.S. and M.S. degrees in Mechanical Engineering from Illinois Institute of Technology and joined Electro-Motive Division in July 1962. He is presently in charge of the truck design and underframe structural design engineering group at EMD. Mels is a member of Pi Tau Sigma and Tau Beta Pi and the author of a number of technical papers.

COMPARATIVE STUDY OF LOCOMOTIVE LATERAL STABILITY MODELS

Introduction. In railway vehicles, severe lateral oscillations may develop at certain operating speeds, resulting in an unstable condition. An unstable condition imposes constraints on a high-speed operation. In addition, the dynamic forces between wheel and rail resulting from this condition may contribute to high lateral rail forces and rapid wear of vehicle components and track structure.

In the past, numerous mathematical models have been used to study lateral dynamic stability of railway vehicles on tangent track. Due to the complexity of the vehicle system, attempts have been made to simplify a model by using fewer degrees of freedom and ignoring system nonlinearities. Simpler component models with fewer degrees of freedom representing either a single wheel-axle or a truck frame assembly have been used to predict unstable condition resulting from axle or truck frame motion. Also, more complete models have been employed to investigate the overall vehicle stability. In these models, a railway vehicle has been considered as a system consisting of various components.

To the best of the authors' knowledge, none of the previous investigations have studied the total vehicle model where both lateral and vertical motions are considered together. In the present investigation, a mathematical model with 39 degrees of freedom (DOF) is used to study lateral stability of a six-axle locomotive. In this model the effect of coupling between vertical and lateral motions is included.

In this paper a comparative study is made between the present investigation of the 39 DOF model with other simplified models. The results of the study of a typical six-axle locomotive obtained from the 39 DOF model are presented. The merits and demerits of the various models are discussed.

Description of Models. The simplest of the models consists of a single wheel-axle set whose dynamic behavior is qualitatively similar to that of a complete truck. This type of model is often used to investigate the secondary hunting phenomenon characterized by the lack of dynamic stability of a wheel-axle assembly. A wheel-axle assembly model would consist of two wheels rigidly connected to an axle. The wheel-axle assembly is considered to be isolated from the truck frame by primary suspension elements. The suspension elements are usually assumed to be linear springs and viscous dampers, which are connected in parallel. Two degrees of freedom corresponding to lateral and yaw motions of the wheel-axle set are considered. This type of model has been used by Wickens [1]*; Boocock [2], Law [3], and Law and Brand [4].

A somewhat more complex model is that of a single truck assembly in which the truck frame is assumed to be rigid, and the connection between the wheel-axle sets and the truck frame is assumed to be either a rigid attachment or through a primary suspension system. In the case of a rigid attachment, the model is defined by two degrees of freedom corresponding to the lateral and yaw motions of the truck assembly. In the latter case, the primary suspension elements allow relative motion between the wheel-axle sets and the truck frame. Therefore, in addition to the assigned degrees of freedom of the truck frame, each wheel-axle set also has separate degrees of freedom. Secondary suspension elements are provided between the truck and the carbody to control the relative motion between them. The total degrees of freedom used in this type of model may vary from two to nine, depending upon the number of wheel-axle sets and their method of connection to the truck frame. Truck models have been used by Wickens [1], Clark and Law [5], Cooperider [6], Matsudaira [7], and Newland [8].

The next class of mathematical model consists of a rigid carbody connected to two rigid trucks through a secondary suspension system. The connection between the truck frame and the wheel-axle sets may be either through rigid attachments or through primary suspension elements. In most of these models, coupling between the vertical and lateral modes of oscillation is neglected. The carbody is assigned three degrees of freedom corresponding to roll yaw, and lateral motion. Each truck frame is provided with degrees of freedom in lateral, yaw, and roll directions. Each wheel-axle is assigned two degrees of freedom corresponding to lateral and yaw motions. Most of the studies have considered suspension elements to be linear. This type of model has been used by Wickens [9], Hobbs [10], and Garg and Mels [11].

The 39 DOF Vehicle Model. Unlike previous investigations, in the present study the track structure is assumed to be flexible. A dynamic coupling between vertical and lateral modes of oscillation for the vehicle is assumed, and so vertical and lateral dynamics of the vehicle need not be treated separately. A kinematic model for a six-axle locomotive system consisting of a carbody, two truck frames, and six wheel-axle sets is developed (Fig. 1). The wheel-axle sets and truck frames are connected by a primary suspension system consisting of linear springs and viscous damping elements. Another set of linear springs and viscous dampers, referred to as the secondary suspension system, is provided between the carbody and each truck frame.

In the analysis, all displacements are assumed to be small, and any free lateral clearance between the wheel-axle sets and truck frames is neglected. Furthermore, nonlinearities arising from suspension elements are also disregarded.

In the model, the carbody and truck frames are assumed to be rigid. The carbody is assigned five degrees of freedom corresponding to vertical, lateral, roll, yaw, and pitch motions. Each truck frame is given five degrees of freedom similar to the carbody. Each wheel-axle set is provided with degrees of freedom in the vertical, lateral, roll, and yaw directions. Thus the described model has a total of 39 degrees of freedom.

We define the sets of generalized displacements q_j and $\bar{q}_j^{\alpha}$ corresponding to each of the degrees of freedom for the carbody and truck frames and

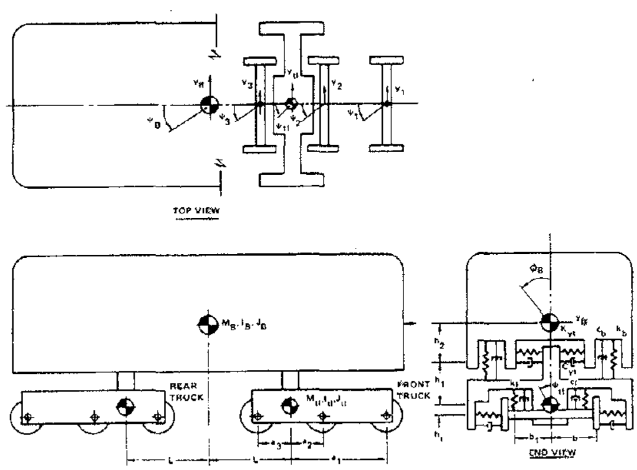

Fig. 1. Vehicle model -- 6 axle locomotive.

*Number in [ ] refers to references.

establish the relative displacement vectors u_i^α of secondary suspension systems.

$$u_i^\alpha = T_{ij}^\alpha \bar{q}_j^\alpha - \bar{T}_{ij}^\alpha q_j \quad ; \quad i = 1, 2, \cdots, 6 \qquad [1]$$

where $\alpha = 1, 2$ corresponds to the leading and trailing truck frame, respectively, and $j = 1, 2, \cdots, 5$ denotes five degrees of freedom for the carbody and truck frame. T_{ij}^α and $\bar{T}_{ij}^\alpha$ are transfer function matrices and are derived from the geometry of secondary suspension elements.

Next, we can define the relative displacement vectors v_i^n of the primary suspension of the nth wheel-axle set in terms of generalized displacements $\bar{\bar{q}}_k^n$ and $\bar{q}_j^\alpha$

$$v_i^n = S_{ik}^n \bar{\bar{q}}_k^n - \bar{S}_{ij}^n \bar{q}_j^\alpha \quad ; \quad k = 1, 2, \cdots, 4 \qquad [2]$$

where $n = 1, 2, 3$ represents the wheel-axle set in the leading truck frame, and $n = 4, 5, 6$ corresponds to the wheel-axle set in the trailing truck frame. S_{ik}^n and $\bar{S}_{ij}^n$ are the transfer matrices associated with the primary suspension system of the nth wheel-axle set. Elements of three matrices are obtained from the geometry of primary suspension systems.

Finally, we establish the relative displacement vector w_r^n of the track suspension as

$$w_r^n = -A_{rk}^n \bar{\bar{q}}_k^n \quad ; \quad r = 1, 2 \cdots, 4 \qquad [3]$$

where A_{rk}^n is the transfer matrix for the track as a third suspension system.

The forces used in the formulation of equations of motion are either internal or external to the system. Internal forces either act between physical components or they are derivable from a potential. Thus, gravity is considered to be an internal force. The component of the gravity force which is not equilibrated by forces resulting from constraints is of special importance in a lateral stability analysis. In a locomotive system, the unequilibrated force develops when the center of a wheel-axle set is slightly raised as it moves laterally in either direction. The effect is analogous to the restoring force in a pendulum and has become known as the "gravitational stiffness." The expressions of the gravitational stiffness similar to those given by Wicken [12] and Joly [13] have been used in the analysis. All internal forces are a function of the generalized displacements or their time derivatives.

Using u_i^α, v_i^n and w_r^n from equations (1), (2) and (3), expressions for potential and dissipating energies of the system are obtained. The kinetic energy of the system is derived using generalized masses and generalized displacements. Lagrange-Hamilton's principle is then applied to the energy expressions to obtain the following equations of motion for the system:

$$M_{lm} \ddot{q}_m + C_{lm} \dot{q}_m + K_{lm} q_m = Q_m \quad ;$$
$$l, m = 1, 2, \cdots, 39 \qquad [4]$$

where M_{lm}, C_{lm} and K_{lm} are the mass, damping, and stiffness matrices for the system. M_{lm} is a diagonal matrix and its elements refer either to mass or moment of inertia of various components in the locomotive system. C_{lm} and K_{lm} are symmetric and positive – definite matrices. q_m is a set of the generalized displacements with subsets q_j, $\bar{q}_j^\alpha$ and $\bar{\bar{q}}_k^n$. The generalized external forces exerted upon the locomotive system by rails are represented by Q_m. These generalized forces result from the tangential and lateral frictional forces at the contact between the wheels and the rails due to relative creep motions. The friction-creep relationships are nonlinear, as are the resulting frictional forces. Since lateral instability often occurs at a small creep level, the friction-creep relationship is assumed to be linear. The expressions for Q_m based on a linear assumption are

$$Q_m = D_{mk} \bar{\bar{q}}_k^n + E_{mk} \dot{\bar{\bar{q}}}_k^n \qquad [5]$$

where D_{mk} and E_{mk} are friction coefficient matrices, whose elements depend upon tangential and lateral creep coefficients, the shape of wheel tread and rail head profiles, and the speed of the locomotive. D_{mk} is a nonsymmetric matrix. The generalized force vector Q_m possesses non-zero elements for degrees of freedom associated with lateral and yaw motions of wheel-axle sets.

The frictional forces can either conserve or dissipate energy. They are capable of adding energy to the system. This happens because wheel friction can transfer energy from the propulsive mode to the lateral mode of motion.

It may be observed that when equation (5) is substituted in (4), the symmetry condition for K_{lm} no longer holds. This will result in a complex eigen value problem, unlike the symmetric K_{lm}, which gives all real eigen values. 135

The equations of motion in (4) represent a set of 39 homogeneous differential equations of second-order with constant coefficients. The general solution of (4) is of the form

$$q_m(t) = \sum_{p=1}^{39} B_p\, y_p^m\, e^{-\xi pt} \operatorname{Sin}(\omega_p t + \phi_p^m) \quad [6]$$

where B_p is an arbitrary constant. ω_p, ϕ_p^m and ξ_p are the frequency, phase angle, and effective damping associated with the pth mode of oscillation. y_p^m is a modal vector corresponding to the pth mode.

Information of modal damping and modal vector is obtained from the solution of the complex eigen value problem associated with (4). If ξ_p in (6) is negative, the oscillation associated with the pth mode will grow exponentially, and the motion becomes unstable. The values of ξ_p will vary with locomotive speed. Any locomotive speed at which the effective damping, ξ_p vanishes is called the critical speed. Once the critical speed has been exceeded, unstable conditions will persist. Unlike a resonance problem, in this there is no higher speed range where normal motion returns.

Discussion of Results. A six-axle locomotive was analyzed to compare the results of four different mathematical models. Model 1 refers to a wheel-axle set model with 2 degrees of freedom; model 2 represents a truck model with 9 degrees of freedom; model 3 is a lateral vehicle model with 2 degrees of freedom; and model 4 refers to a vehicle model with 39 degrees of freedom. For each model the critical speed of the locomotive is obtained for various effective wheel tapers ranging from 1 in. 5 to 1 in. 40.

A plot of the reciprocal of effective wheel taper versus critical speed is given in Fig. 2. The plot shows that the critical speed of locomotive increases with a decrease in the effective wheel taper. All four models predict instability of motion due to lateral oscillation of the wheel-axle set. The critical speed predicted by the wheel-axle set model is the lowest. As the dynamic interaction between various components is increased by introducing more degrees of freedom in the model, the predicted critical speed of the locomotive increases. The critical speeds obtained from models 1, 2, and 3 are relatively close (a variation within 5%). This shows that the lateral dynamic coupling of axles to truck frames and truck frames to carbody has a relatively small influence on the predicted critical speed for the locomotive. However, the results from the 39 DOF model are 15 to 20% higher than those obtained for the 21 DOF model. This higher predicted critical speed for the locomotive is attributed to the dynamic coupling which exists between the lateral and vertical modes of oscillation. The lateral vehicle model (model 3) predicts a lower bound for the critical speed and hence, from a designer's point of view, it is a conservative value.

In Fig. 3, a plot of the critical speed versus critical frequency is given. From this plot it is evident that whereas the trend of predicted critical frequencies from models 2, 3, and 4 is similar, model 1 is different. It is felt that this difference is due to the absence of system damping between various components, which has not been included in the wheel-axle set model. Some of the test results available to data indicate that the predicted critical frequencies by the 39 DOF model compare reasonably well within the speed of 80 to 100 mph.

Further comparison between models was made by evaluating critical speeds of the locomotive with a worn 1 in 20 wheel profile. In the analysis, it was assumed that the nominal wheel taper of each wheel-axle set is the same, and all the wheels are

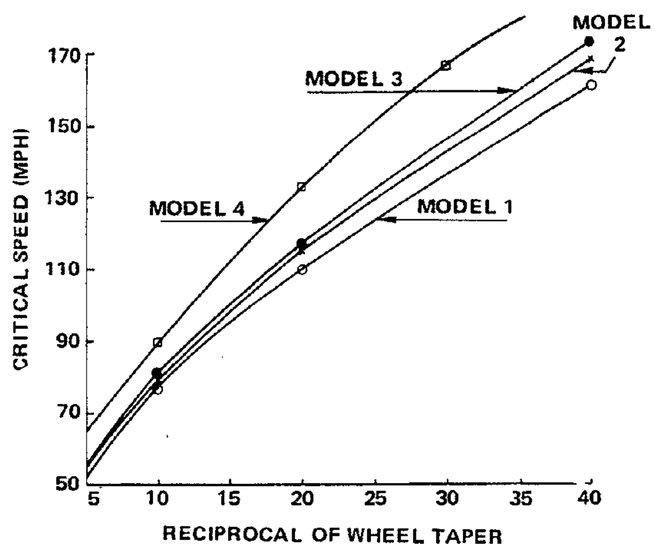

Fig. 2. Reciprocal of wheel taper wheel taper vs. critical speed.

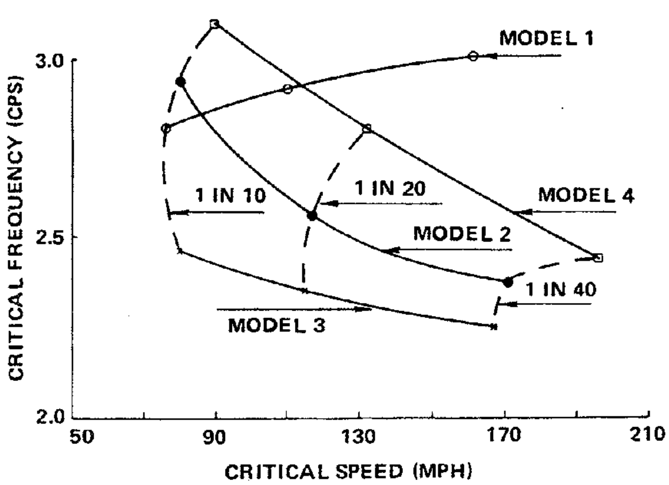

Fig. 3. Critical speed vs. critical frequency.

worn to the same degree. The radii of wheel tread profile and rail head are assumed to be 20 in. and 8 in., respectively. Critical speed of the locomotive is evaluated by each model. The results are shown in Fig. 4. The wheel-axle set model gives a lower critical speed. The predicted critical speeds by the truck and lateral vehicle models are fairly close, but they are 10 to 12 mph lower then the one obtained from the 39 DOF model.

In actual practice, wheels of a locomotive never wear out to the same degree at each wheel. Therefore, to simulate a condition close to field operation, the nominal wheel taper of the lead wheel-axle set in each truck was changed to 1 in 10, while maintaining 1 in 20 nominal wheel taper for the four remaining wheel-axle sets. It is interesting to observe (Fig. 5) that the resulting critical speed of the locomotive is reduced and occurs between the critical speeds that would be obtained if all the wheel-axle sets are either of 1 in 20 or 1 in 10 nominal taper. Thus, it may be emphasized that even a single severely worn wheel-axle set in the locomotive will significantly reduce critical speed of a rail vehicle.

Conclusions. The thrust of this paper was to compare the results of various stability models and study the effect of vertical and lateral dynamic coupling on the calculated critical speed of a locomotive. In all the cases analyzed, it was found that the result of the critical speed obtained from the truck (model 2) or lateral vehicle model (model 3) are on the conservative side as compared to those obtained from the 39 DOF model. Because of the assigned degrees of freedom, the truck model is only capable of predicting the secondary hunting phenomenon, which refers to the dynamically unstable condition initiated either by the wheel-axle set or truck frame motion. Information about the primary hunting condition (carbody oscillation) cannot be obtained from this model. The lateral vehicle model can be used to study both primary and secondary hunting conditions. In general, the critical speed of secondary hunting predicted by the truck and lateral vehicle models are in good agreement. The computer (CPU) time required for the solution of a typical three-axle locomotive model for these two models varies in the ratio of 1:12 (i.e., 10.22 CPU sec. for the truck model versus 113 CPU sec. for the lateral vehicle model).

Similar to the 21 DOF model, model 4 can be used to study the primary as well as secondary hunting conditions of a locomotive. As the model takes into account the dynamic coupling between vertical and lateral modes of oscillation, it provides information which can be further used to study the

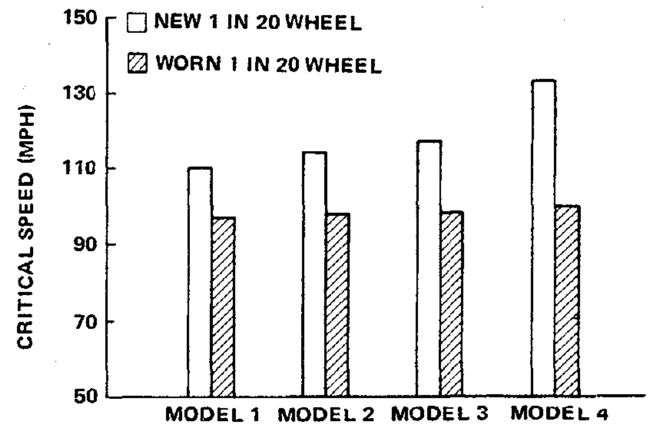

Fig. 4. Comparison of critical speed for new and worn 1 in 20 wheel taper.

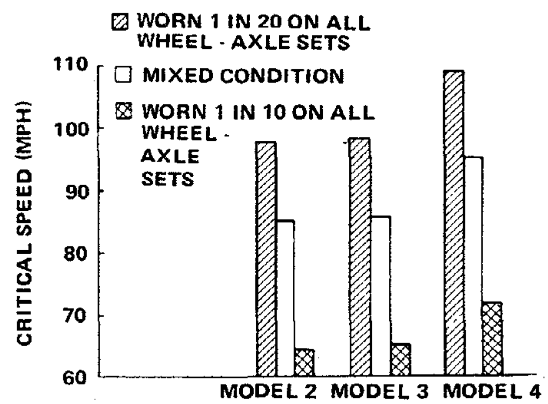

Fig. 5. Comparison of critical speed for worn wheel with different nominal taper of axles.

response of a locomotive to rail irregularities. In general, the predicted critical speed of the locomotive is 10 to 12% higher than those obtained from models 2 and 3. The difference in the critical speed indicates the influence of the dynamic coupling between lateral and vertical modes, which had been neglected in the previous investigations. The computer time required by this model for a three-axle locomotive analysis is about eight times that required by the lateral vehicle model (i.e., 113 CPU sec. for the lateral vehicle model versus 803 CPU sec. for model 4).

The results obtained from the single wheel-axle set model compare reasonably well with the truck and lateral vehicle models, but it will not be able to reflect the effect of such important parameters as truck wheelbase on the critical speed.

In general, it may be concluded that the truck model (9 DOF) should be used when the interest is only in a secondary hunting analysis. However, the use of the lateral vehicle model (21 DOF) is suggested whenever both primary and secondary hunting characteristics of a locomotive are desired.

Although the 39 DOF model requires more computer time, the additional data provided by this model could be utilized in designing the vertical suspension elements. Also, with a little additional effort using modal superposition techniques, the model may easily be extended to study response of a locomotive to vertical and/or lateral rail inputs.

Acknowledgment. The authors express their sincere appreciation to P. W. Hartmann, Research Engineer, AAR, Chicago, for his kind help during the computation phase of this study.

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Notations

B_p = Constant.

C_{lm} = System damping matrix.

D_{mk} = Friction coefficient matrix.

E_{mk} = Friction coefficient matrix.

K_{lm} = System stiffness matrix.

M_{lm} = System mass matrix.

Q_m = Generalized external forces.

q_j = jth generalized displacement for the carbody.

$\bar{q}_j^{\alpha}$ = jth generalized displacements for the truck frame number a.

$\bar{\bar{q}}_k^{n}$ = kth generalized displacements for the nth wheel-axle set.

q_m = generalized displacements.

u_i^{α} = relative displacement for secondary suspension elements.

v_i^n = relative displacement for primary suspension elements.

w_r^n = relative displacements for track suspension elements.

y_p^m = modal vector.

ω_p = model frequency.

ϕ_p^m = phase angle.

ξ_p = effective model damping.

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Mr. Selzer received his B.A. degree in Economics from the State University of New York at Buffalo and his M.A. and PhD. from West Virginia University. His work experience includes two years with Amtrak's corporate planning division and one year with the Federal Railroad Administration. While with Amtrak he was involved in demand forecasting, pricing policy, and route cost analysis. His present duties include financial and cost analysis, econometric research, and internal technical consulting.

ECONOMIC FACTORS INFLUENCING THE DECISION TO USE JUMBO RAIL CARS

It is a pleasure for me to be here today to share my views on car size with you. In preparing this speech I, being an economist, elected to discuss the motivation behind using jumbo rail cars instead of talking about the engineering aspects of the problem. Economics is more concerned with human behavior and the efficient utilization of resources than with physical relationships and the properties of materials.

There are several major differences in our disciplines which are important to recognize. Engineering is an applied laboratory science, and theories on engineering relationships can be tested in controlled experiments. Fairly accurate and detailed conclusions can be drawn based on these tests. The science of economics, on the other hand, is somewhat less precise. We have no laboratories in which to test our theories. Instead we must be generalists. We attempt to explain behavior on the basis of empirical observation, without the benefits of isolation. It is within this context that I will attempt to explain the motivation which, I think, lies behind railroad management's use of jumbo cars.

As you are quite aware, engineers have studied the impact of heavy wheel loadings on track wear for over 40 years. The conclusions reached by engineers such as Robey, Magee, Code, and Reiner, to mention a few, all supported the hypothesis that very heavy axle loading leads to a disproportionate degree of track wear. Yet, railroad management has continued to use larger and larger cars, in spite of these admonitions. Average freight car capacity has continually crept upwards, year after year. In 1929, average capacity per car was 46.3 tons, by 1951 it had risen to 52.9 tons, and in 1960 it averaged 55.4 tons. In 1974 a new record was reached with an average of 71.5 tons.

From an economist's point of view, the important question concerns why management continues to purchase and use these jumbo cars in the face of very pervasive evidence to the contrary provided by the engineering profession. Is management deliberately trying to destroy the right-of-way? Does management doubt the findings presented?

The answer, I believe, can be found largely in the way the problem is viewed. To the engineer the problem is, relatively speaking, straightforward, in spite of its complexities. Briefly, it has been viewed as follows: Given the physical characteristics of the wheel and track, the contact area must sustain a great deal of compressive stress which apparently increases disproportionately as weight is increased. These high compressive stresses from wheel loads result in internal compression, tension, and shearing stresses within the railhead. These stresses at any one point vary in magnitude as the wheel rolls along and also reverse in direction to some extent, or fully, thus giving a condition of stress change or reversal that is conducive to the development of progressive or fatigue failures.

Management, on the other hand, lives in a much more complex world. They must deal with many phenomena which occur simultaneously and are often difficult to identify. They live in the world of demand functions and cost functions, of accounting principles and regulation. They must deal with factors such as marginal revenue functions, intermodal and intramodal competition, demand and cross elasticities, economies of scale, the cost of capital, rates of return, and even subjective utility functions.

Decision making within this maze of variables, constraints, and uncertainties is indeed a difficult task. Although managerial decisions are often found to be irrational ex post, they are, in my opinion, usually quite proper and rational ex ante. My point is that mangement makes the best decisions they are capable of making, given the information they have at their disposal and given the nature of railroading.

Faced with the decision of whether to use jumbo freight cars, management must assess all relevant factors, give each a weight, balance them, apply a conjuctural adjustment, and derive a net conclusion on which policy is to be based. Among the most important factors which must be considered and dealt with is competition. Intermodal and intramodal competition for transportation is often substantial. The slightest advantage may be sufficient to shift demand from one carrier to another. Consequently, of all the economic factors I've mentioned, competition ranks as one of the major variables in management decision making. The use of jumbo cars is intricately tied to competing.

The large heavy cars offer, or appear to offer, significant advantages which weigh heavily in the calculus of operating a railroad. Principle among these advantages are economies of scale. The term economies of scale, or returns to scale, implies that in the area of operation, the long-run average cost curve slopes downward. Increasing plant size serves to reduce average cost under these conditions. The economies associated with jumbo cars are well known. Anticipated cost savings are based on cars handled. For example, maintenance costs on such items as wheels, couplers, air hoses, and air brakes are relatively constant on a per car basis.

Crew costs are not affected by car size. Neither are inspection costs, humping, and classification costs. Acquisition costs do not increase in direct proportion to car capacity. Larger cars offer more efficiency in movement by offering greater net weight to tare weight ratios. This in turn implies lower locomotive requirements, fewer trains, and greater utilization of the potential cube.

Perhaps these economies are more apparent than real. As Harry Meislahn alertly pointed out at this Conference a few years ago, the measurement of efficiency on the basis of cost per car may not be appropriate. ICC cost formulas, as well as many internal railroad cost formulas, use cost per car as the principle measure of efficiency. Such use may be appropriate over time to measure productivity changes given a car size, but when car size becomes variable, such comparisons may not be appropriate. Measurement with a rubber yardstick leads to distortion. As Mr. Meislahn suggests, efficiency should more appropriately be measured by one or more measures of output which are independent of car size. Average costs on these bases may not be as favorable.

Another factor which has not received adequate attention is the diseconomies of scale associated with the use of oversized cars. Rights-of-way historically have been built with clearances adequate for conventional-sized cars. Larger cars often require additional switching and rerouting, with all the associated costs, such as switching crew time and engine time, additional fuel and mileage costs, larger numbers of routing clerks, and even relaying of tracks at loading facilities to accommodate the higher cars.

A second major consideration supporting management's use of jumbo cars and associated with the economies of scale is the competitive rates which are made possible by the use of the jumbo cars. Based on ICC regulatory requirements, the savings realized from the use of the large cars can be passed forward to the shippers. This is one of the few opportunities in railroading where price competition is permissible. Lower rates mean greater volume and higher revenues. In fact, the whole theory of railroading is based on the principle of mass movement: The full utilization of the potential cube. Such special rates can only be justified if it can be demonstrated to the ICC that cost savings in fact exist. The demonstrated savings must however be based on ICC accounting.

Thus management is almost forced into using ICC accounting standards if it wishes to compete effectively, and such acceptance makes the jumbo cars extremely attractive. Thus ICC accounting becomes the only game in town, and failure to play may imply an institutional inability to win. Not competing on common ground could mean loss markets and short-run bankruptcy. Therefore, seeing through any artificial savings or recognizing additional cost on an individual independent basis may be counterproductive. This provides an example of the famous "Prisoner's Dilemma" game, in which the solution is cooperative action. In this case it would be recognition of the facts and fallacies by all competitors, followed by the appropriate actions.

Under independent decision making observing of real long-run economic costs, including those associated with track deterioration, and basing rates on these costs may paradoxically be the wrong things to do. Inability or unwillingness to compete in the short run may make the long run irrelevant. The inability to produce a short-run profit has serious consequences in terms of attracting capital and thus may ultimately jeoparidze the jobs of current management. This situation may most appropriately be termed the Good Samaritan paradox. Doing the right thing for a railroad may be the wrong thing for its management.

What we have then is a propensity for short-run profit maximization, which is tantamount to utility maximization. Under these conditions it is quite clear that present management cannot obligate itself to future management in terms of turning over an optimally used physical plant. Given the uncertainty of the amount and degree of damage, it is rational to maximize near-term profits and to heavily discount in uncertain future. To put it another way, the costs to present management of not using jumbo cars may be very high, while the discounted benefits of not using them, regardless of the magnitude of the benefits, would effectively be zero. Conversely, the benefits to present management of using these cars tends to be relatively high, while the discounted future costs, regardless of their magnitude, would also effectively be zero.

Thus, strangely enough, it appears that the decision on whether or not to use jumbo cars may be independent of whether they in fact cause a disproportionate amount of track damage. Should this hypothesis be correct, the solution to the problem must go beyond determining the exact relationship between car weight and rail damage. What must be achieved is the elimination of the institutional and economic incentives which result in distorted decision making.

First, however, it must be indisputably proven that the 100- and 125-ton cars create a disproportionate amount of damage on adequately built track. The efforts of A.R.E.A. in that area are well recognized. They are, however, being supplemented by other sources.

For example, the Federal Railroad Administration is currently concluding a study on the variability of maintenance-of-way costs. It may please you to know that the conclusions reached tend to support the findings of those who oppose the use of jumbo cars. These findings, unfortunately, are not fully available at this time. We do, however, expect the study to be released in the next few months, at which time the technical details will become available.

Similar research is being conducted by the Canadian National Railroad. Their evaluation is being conducted in an atmosphere which should minimize the institutional distortions which are present in our own environment. It is anticipated that they will place greater emphasis on long-run economic costs and reach a decision which considers all the previous work accomplished in this vital area. It is hoped that their evaluation will be available to us and will provide the evidence necessary to warrant a policy decision on the subject.

It if can be demonstrated that damage from jumbo cars is excessive, major changes in the nature of regulation will be in order. The first of these changes will require the ICC to place more emphasis on future costs than on past costs. Long-run marginal cost has received little if any attention in determining rates. From an economic viewpoint, long-run marginal costs must be met if an enterprise is to remain viable in the long run.

Secondly, productivity measurements will have to be reevaluated. The per car standard may have to be discarded, if it can be shown to be biased.

Finally regulation must be relaxed to some degree. Management must be permitted to lead, to make creative decisions, and to have a greater role in establishing rates in accordance with the way they, rather than the ICC, perceive their costs to vary.

The railroads must also make major institutional changes. They must show greater cooperation in these areas of mutual concern, for only through a united effort can they protect their individual interests.

M. Noyszewski
Bridge Engineer
Illinois Central Gulf Railroad

Upon graduation from the University of Illinois in 1956 with a B.S. degree in Civil Engineering, Noyszewski entered the Bridge Department of the Illinois Central Railroad. Advancing through positions of increasing responsibility, he was appointed Bridge Engineer of the merged Illinois Central Gulf Railroad in 1972.

Concurrently Noyszewski earned an M.S. degree in Civil Engineering from the Illinois Institute of Technology. He is a Registered Structural Engineer and a Registered Professional Engineer. He has served as Chairman of the American Railway Engineering Association Committee 30, Impact and Bridge Stresses, and on the AAR Engineering Division's Committee on Bridge Structures. He is also a member of American Society of Civil Engineers (ASCE), National Society of Professional Engineers (NSPE), and other professional organizations.

EFFECT OF HEAVY AXLE LOADS ON BRIDGES

It is a pleasure and a privilege to share with you some thoughts about the effects of heavy axle loads on bridges. For the past few years there has been increasing dialogue on the detrimental effects of heavy axles on track, but too little attention has been given to bridges. A chain is only as strong as its weakest link, and in a rail system the weakest link is often an obsolete bridge.

It is unfortunate that bridge engineers are not prophets. It we were, we could predict if a structure will still be needed 100 years from now, or only ten, and during its life we could predict the magnitude and frequency of loading, operating speeds, maintenance problems, and the effect of service disruptions. Today, I will not make any predictions as to future equipment trends and the design of new bridges. I will confine my remarks to the question of existing bridges, and how we can carry modern equipment on bridges built many years ago for much lighter loading.

The 1975 edition of AAR's Yearbook of Railroad Facts estimates that there were 200,000 miles of railroad lines in the United States on December 31, 1974, with a total of 326,000 miles of track. There are no comparable totals for bridges, but from my own studies I have estimated the bridge total at 3,500 miles, with a replacement value at current prices in excess of $10 billion. On a length, not price, basis, the steel and timber bridges are about equal, and concrete represents only 10% of the total.

As we continue to defer reconstruction of our bridges, we will be forced to impose more speed and weight restrictions in an effort to prolong the life of bridges designed to earlier and much lighter standards. Designers of locomotives and cars must become aware of our problems, or they will find that their modern equipment will be prohibited on many lines. For example, a heavy-duty flatcar with 6-ft. axle spacing has a 25% higher permissible loading on our timber trestles than a similar car with only 5-ft. spacing. The increased loading is 37% for 6-ft. versus 4'6" axle spacing. This extremely important effect of axle spacing will be explained later.

Loading History. Let us review the evolution of the design loadings to discover how these weak links have developed.

The first locomotive to operate in America was the "Stourbridge Lion," brought from England by Horatio Allen and placed in service in 1829. His "The Best Friend of Charleston" was the first locomotive built in the United States. Both weighed only 7 tons.

By 1837 the Baltimore and Ohio Railroad locomotives weighed 12 tons, and the engineers were faced with the rebuilding or reinforcing of their structures. The locomotive weights continued to increase to 23 tons in 1844 and 30 tons in 1854. The weight of locomotive and tender exceeded 1,000,000 lbs. by 1940, and individual driving axles weighed between 75,000 to 80,000 lbs.

The driving of the golden spike at Promontory Summit on May 10, 1869, occurred just 40 years after the first appearance of the steam locomotive. While the nation's attention was focused on the transcontinentals, some men went quietly about their business of spanning small streams and mighty rivers. James Buchanan Eads, one of the pioneer geniuses of civil engineering, started the famous bridge across the Mississippi River at St. Louis, Mo. on August 20, 1867, two years before the famed spike ceremony. It was opened to rail traffic on July 2, 1874 and today is still in service.

Monumental railroad construction continued, and a high point was reached during the decade of the 1880s, when some 70,000 route miles were laid. In 1882, the greatest single year of railway building in the U.S., 11,569 miles of track were completed.

Fig. 1 is a composite photograph of the Santa Fe's famed Canyon Diablo Bridge in Arizona. The upper portion shows the 1891 construction, which was replaced in 1920. The lower portion shows the current arch, placed in service in 1946 and designed for E72 loading.

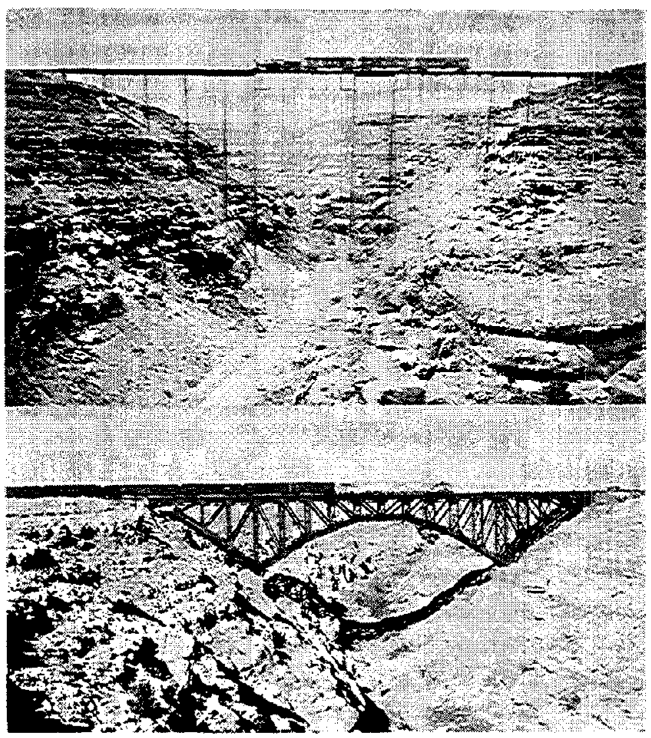


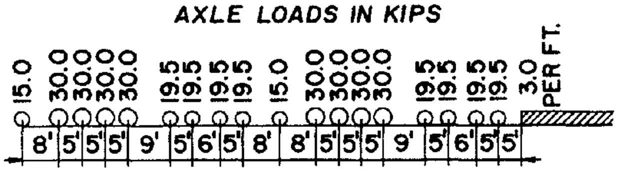
Fig. 1. AT & SF Canyon Diablo Bridge, Arizona.

The rapid increase in engine loading and the multitude of design specifications used by various railroad companies demanded some standardization. About 1880, Theodore Cooper, a prominent consulting engineer, recommended a system of axle loads representing the heaviest doubleheaded locomotive of that time, followed by a uniform load. The total weight of each locomotive and tender was 213,000 lbs., with four driving axles spaced 5 ft. apart, each weighing 30,000 lbs., followed by a uniform load of 3,000 lbs. per ft. This became known as Cooper E30 loading (Fig. 2). By 1895 this loading was increased to E44. In 1906 the American Railway Engineering Association (AREA) adopted the Cooper E50 as the recommended design load. In the E50 loading the driving axles were 50,000 and the trailing load 5,000 lbs. per ft., and the pilot axle and tender axles were increased by the 50/30 ratio.

COOPERS E30 DESIGN LOAD

(RECOMMENDED ABOUT 1880)

AXLE LOADS IN KIPS



WEIGHT OF ONE LOCOMOTIVE = 135,000 LBS.
WEIGHT OF ONE TENDER = 78,000 LBS.
TOTAL = 213,000 LBS.

TRAIN LOAD $= \frac{\text{AXLE LOAD}}{10} =$ *3,000 LBS. PER FT.*

Fig. 2. Cooper E Loading.

In 1920 an increase was made to E60, in 1935 to E72, and to E80 in 1967. For comparison the current AAR Mechanical Divisions Car Construction Rules permit E60 for cars (Fig. 3). Please note that a span built in 1900 for a Cooper E44 loading will now be subjected to E60 loading, or a 36% increase in live load. We should be grateful that engineers of that time were very conservative and specified high impact percentages. Thus with a speed reduction, and assuming a minimal loss of section due to corrosion, we may be able to handle these heavier cars.

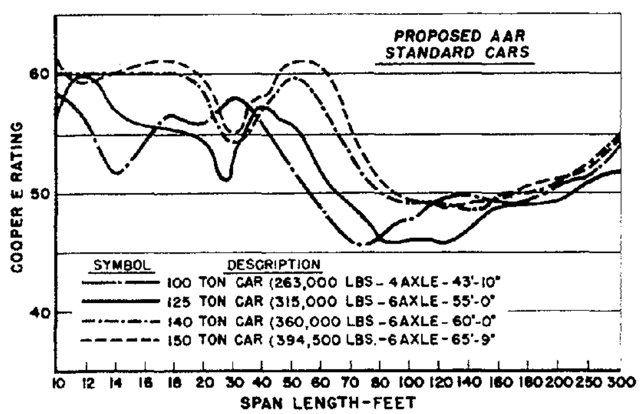


Fig. 3. Cooper E Rating of cars.

Railroad bridge engineers have been criticized for continuing to use the Cooper loading system for many years after the steam locomotives have disappeared. However, much of our data on existing bridges, including stress sheets and rating computations, are based on the Cooper loading. Furthermore, with the ready access to the AAR's car rating program and to its publication "Moment and Shear Tables for Heavy Duty Cars on Bridges," most of the equipment has now been rated for the Cooper loading. We are able to compare the effects of the cars and locomotives with the carrying capacity of the bridges, and hence the Cooper loading has become an accepted system of measurement.

Fig. 4 shows car rating curves for three cars. For equipment in use today, the car length is not a factor in the rating of cars for spans up to about 50 ft., as the governing moment is produced by the two trucks of adjacent cars. Thus to reduce the rating of a car, we should increase the distance between the axles of adjacent cars.

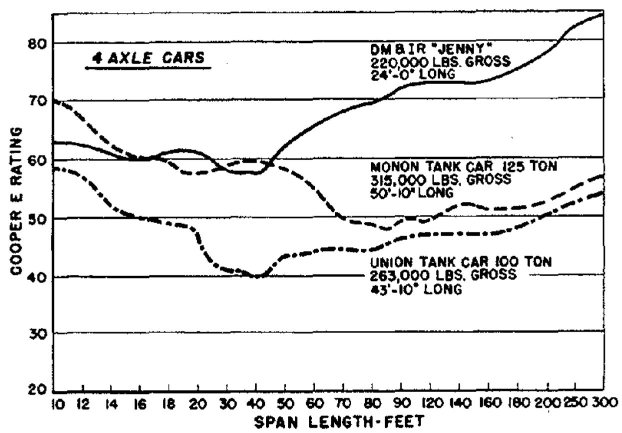


Fig. 4. Cooper E rating of 4 axle cars.

I have chosen the Union Tank car because it has the same 43'10" coupled length as the 100-ton car in Fig. 3, except that it has a 14-ft. distance between end axles of adjacent cars, compared to the customary 6-1/2 to 7-1/2 ft. Gross weight, axle spacing in the trucks, and coupled length are identical, but the E rating is much lower for intermediate span lengths. For a 40-ft. span, the Union Tank car rates only E40, as opposed to the E56 for the conventional 100-ton car, a striking difference of 40%.

Frequencly the weakest truss member is a floor beam hanger, and many fatigue failures have been recorded in AREA Proceedings. For this reason, particular attention must be given to maintain low E ratings for the 40 to 60-ft. spans. This span range is representative of the effects experienced in floor beam hangers.

Design of New Bridges. In designing a new bridge, it is proportioned for the following loads and forces:

1. Dead Load: The estimated weight of the structural members, plus that of the track, ballast, and any other portions of the structure supported thereby.
2. Live Load: The current recommended live load for each track is Cooper E80.
3. Impact: The dynamic effect of rolling loads as determined by appropriate formulas and taken as a percentage of the live load. Impact usually consists of vertical and roll effects. For steel bridges the vertical effect increases with speed up to 60 mph. The roll effect is substantially the same for all speeds. Impact is not considered for timber trestles, because of the inherent ability of timber to absorb momentary overloads.
4. Centrifugal Force.
5. Other Lateral Forces:
   a. Wind on loaded bridge.
   b. Wind on unloaded bridge.
   c. Other forces from equipment.
6. Longitudinal Force: Tests conducted by the AAR show that the maximum longitudinal force from starting or stopping of trains is 15% of the live load. Where the rail is continuous and the bridge short, practically all of the longitudinal force is transferred to the adjacent enbankment.
7. Seismic Forces: At the present time AREA has no seismic requirements, although some railroads in earthquake-prone areas have evolved their own requirements.

Fatigue. The ASTM defines fatigue as "The process of progressive localized permanent structural change occuring in a material subjected to conditions which produce fluctuating stresses and strains at some point or points and which may culminate in cracks or complete fracture after a sufficient number of fluctuations."

After a century of study, a majority of our structures still fail in fatigue. Fatigue generally develops late in the life of a structure. They develop at relatively low nominal stresses, are of a progressive nature, start at small flaws or stress concentrations, and propagate slowly. However, when such cracks propagate to a critical size they may quickly lead to catastrophic brittle failure.

Since 1910, the AREA has required increasing the area of members subject to reversal of stress. In 1969, the specifications were revised, as fatigue under some conditions will reduce the left of members and their connections, even if all stress is of the same sign. Reversal of stress is not necessary to cause fatigue failure.

I am sure that most of you are familiar with traditional representations of the relationship between maximum stress and cycles to failure (Wőehler or S-N curve), or the Modified Goodman Diagram. I have briefly touched on the subject of fatigue because, as high-speed unit train operations become increasingly popular, we can expect to see an alarming increase in fatigue failures.

Existing Bridges. The steel bridges constructed in the first part of this century usually were proportioned for an allowable stress of around 16,000 psi and utilized steels with a minimum yield of 30,000 psi. Hence the apparent safety factor was a conservative 1.88.

In investigating a bridge for the passage of an infrequent special load we are not concerned with fatigue, and are sometimes willing to reduce the safety factor to as low as 1.25. I am referring now to the occasional shipments of heavy-duty equipment moving under closely supervised conditions, sometimes in a "Special Train." Under such conditions we are not concerned with excessive wind on the train or bridge, nor longitudinal forces. We are able to control the speed at which the shipment crosses the weak bridge and take advantage of the reduced impact at lower speed (Fig. 5). Thus a bridge designed in 1900 for a Cooper E44 loading could safely support perhaps an E80 load at 10 mph. Please note on Fig. 6 that for an 80-ft. open deck, deck plate girder span, the impact reduces from 40.5% at 60 mph to 18.1% at 10 mph. For a 150-ft. truss span the corresponding reduction is from 26.6% to 9.8%.

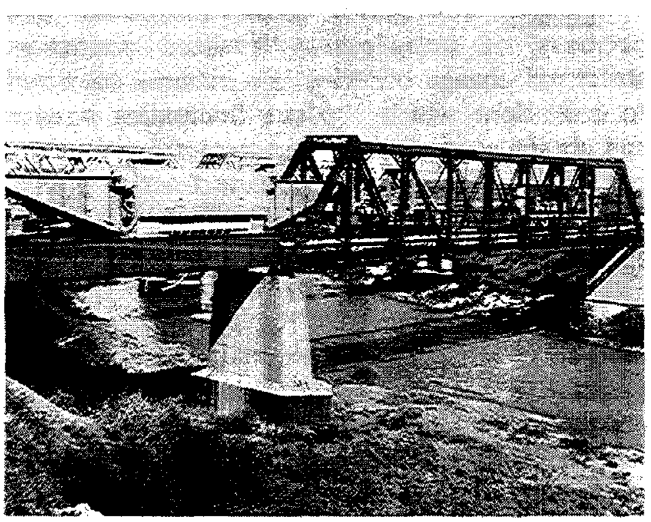


Fig. 5. Bridge testing of "Special Train."

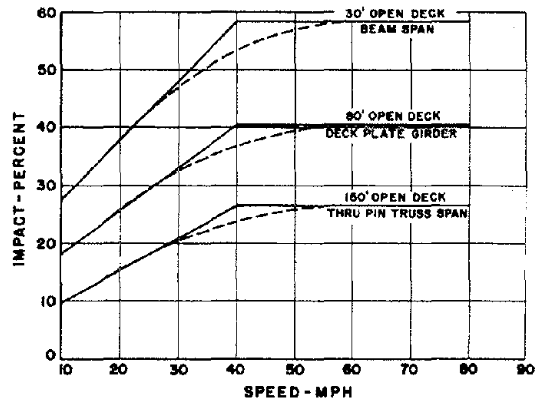


Fig. 6. Typical impact curves.

When permitting a load to cross at such a low safety factor, we must be absolutely certain that we have completely investigated the bridge in question. In rating each member, not only corrosion losses must be considered, but also any damage to the member that may have occurred. We often find that truss members have been kinked by pulpwood or other shifted loads. Sometimes the damages occurred in much more spectacular fashion, but were expertly repaired. The casual observer may not ever notice where the repairs have been made. In February 1963 the third unit of a westbound freight derailed in a curve ahead of a truss span and slammed into the end post, twisting the post and the bottom chord (Fig. 7). We were able to temporarily support the truss and, some weeks later, splice in a new section of the bottom chord and replace the entire end post.

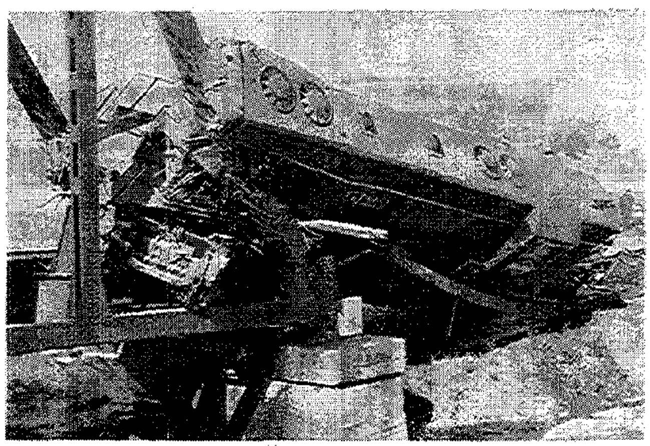


Fig. 7. Bridge with twisted end post and bottom chord.

In March 1918 a 238-ft. swing bridge built in 1897 was almost dropped into the Wabash River. An elaborate wooden twoer was used in raising the span together with the engine. Repairs were completed, and the span is still in service today.

In evaluating the carrying capacity of such bridges we must not overlook their history. Severely damaged members have been repaired, but smaller kinks remain to this day. We do not know how much internal damage has occurred to the other members, and I certainly would not reduce the safety factor to only 1.25 for such a span.

When investigating the normal (daily) maximum permissible loading on a weak bridge, we must be extremely careful of fatigue and use conservative allowable stresses. For example, five 150-ft. deck truss spans fabricated in 1887-89 and originally used in the north approach of our Cairo bridge were removed from that bridge and erected in 1911 on an important main line in Iowa.

The original specifications for the deck trusses permitted the use of either open hearth or Bessemer steel, and records were unavailable as to the actual composition of the steel used. The bottom chords rated only E35 at 15 mph, with wind and braking forces. The bridge was at the bottom of a sag with long 1/2% grades. Table 1 was prepared for the guidance of the Operating Department to ensure that the bridge would not be overloaded.

I am sure that you will appreciate the difficulty that our operating people had in verifying that a car of a known weight was of the presecibed length. This is perhaps better shown in Fig. 8. Please note t.. a 100-ton car with a gross weight of 263,000 had to be 55'0" long or longer if coupled to a similar car, to avoid exceeding the allowable stress of 18.0 ksi (incidentally, this was live load + dead load plus impact at 15 mph, with no other forces considered). You can also see that several 44-ft.-long 100-ton cars would produce a 20% overstress.

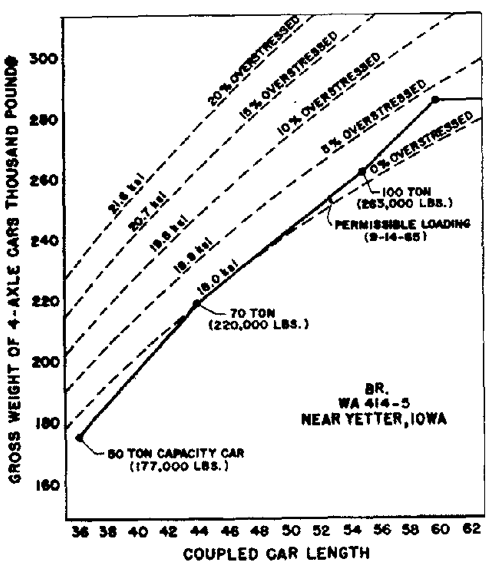


Fig. 8. Permissible loading--bridge WA 414-5.

Table 1

Nominal Capacity, Tons	Maximum Gross Weight, Pounds	Axles Per Car	Minimum Permissible Coupled Length: Continuous Train of Coupled Cars	Minimum Permissible Coupled Length: Single Loaded Car Coupled to Adjacent Empty Cars
40	142,000	4	29'0"	20'0"
50	177,000	4	36'0"	20'0"
70	220,000	4	44'0"	25'0"
100	240,000*	4	49'0"	30'0"
100	263,000	4	55'0"	30'0"
100	286,000	4	60'0"	30'0"
125	315,000	4	Not permitted	35'0"
150	350,000*	6	75'0"	40'0"
150	395,000	6	Not permitted	50'0"
150	414,000	6	Not permitted	53'0"

*Represents car not loaded to maximum capacity

I have devoted most of my time today to steel bridges, but we must not overlook timber trestles. They comprise approximately 1,600 miles, or 45% of all bridge lineal footage, and are found not only on forgotten branch lines but also on many main line tracks. Because their span length varies between 12 and 15 ft., they are especially affected by short axle spacing.

Fig. 9 shows the allowable axle loads over ICG's open deck timber trestles. These trestles have 14-ft. panels and four 7" x 16" stringers under each rail. The effective stringer span length is assumed to be 13.5 ft. For conventional four-axle cars, only two axles produce maximum moment, hence a high allowable axle load (18,000 lbs. for 5'8" axle spacing). However, for cars with six or more axles, three axles can be positioned for maximum banding on the 13.5-ft. span, and the lower curve governs. The allowable axle loads are 60,600 for 4'6", 66,500 for 5'0", 73,700 for 5'6", and 82,700 for 6'0".

This illustration is very important because many of the older heavy-duty cars have some 4'6" axle spacing, significantly reducing the maximum

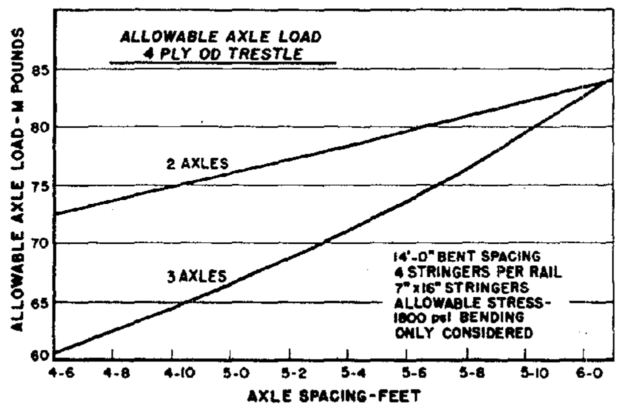


Fig. 9. Allowable axle loading--open deck timber trestles.

gross weight that can be carried on such equipment. For example WECX 201 (Schnable +2), a 12-axle car, has 5'0" axle spacing except for the four inside axles, where the spacing is reduced to only 4'6". Since the car is designed for equal axle loading, the 4'6" spacing reduces the permissible weight on this car over timber trestles by 10%.

Similar reduced axle spacings affect the loading on GEX 40003-40004, PC (F38) and (F40), D&H 16157, N&W 202906-8 and others.

Summary. Gentlemen, I leave you with a few suggestions:

1. Don't think of the spectacular, long-span, modern steel bridges. That is not where the problems are found.

2. Remember that with capital scarcity it will be many years before all of the frail 1880-1900 steel spans are replaced. The weak spans will continue to control the line capacity.

3. Don't rush to design conventional cars for the E60 currently permitted by the Mechanical Division of the AAR, if you want unrestricted operation over essentially the entire rail network.

4. Remember the timber trestles and the effect of reduced axle spacing. Avoid axle spacing less than 5'8" for heavy-duty equipment.

F. E. King
Senior Technical Advisor
Canadian National Railways

F. E. King, Senior Technical Advisor, Canadian National Railways, began his railroad career as a machinist apprentice at Stratford, Ontario in 1937. He was graduated from the University of Toronto as a Bachelor of Applied Science in Mechanical Engineering in 1949. After graduation, he served in various capacities as a Material Inspector, Supervisor of Stationary Steam Plants, Research Engineer, and Manager of Canadian National's Technical Research Center.

In his present position, King is engaged in major technical studies in research and development, as well as serving as Coordinator for the Track/Train Dynamics Program for Canadian National. He is a member of the American society of Mechanical Engineers, the Air Brake Association, and the Canadian Air Brake Club.

RAIL WEAR AND CORRUGATION PROBLEMS REALTED TO UNIT TRAIN OPERATIONS: CAUSES AND REMEDIAL ACTION

Introduction. The use of unit trains to haul bulk commodities such as coal, oil, potash, sulfur, and ores offers many opportunities for providing low-cost transportation from the resource areas where these products are found to seaports and other major distribution centers. For example, car equipment can be designed and built specifically for the proposed service, achieving significant reductions in tare weight and equipment cost. Vehicles capable of carrying 100 tons of product can be built to have approximately the same tare weight as those carrying 50 tons. Other things being equal, the 100-ton capacity vehicle will always be selected for this service. Since all the cars in the train can be identical and designed for a single service, important savings can arise from simplified equipment maintenance requirements, from rapid loading and unloading, from short turnaround capability, and from reduction or virtual elimination of switching and other yard costs.

Unfortunately, many of our resources are located in mountainous terrain where sharp curves are frequently encountered. In some cases existing railway lines were not built to carry the 100-ton vehicles which are now in service The combination of curved track, heavy vehicles and large annual tonnages can result in greatly accelerated rail wear in curves. The severity of this rail wear problem is often much greater than would be anticipated based only on consideration of the increased annual tonnage over the line.

This paper is a case study of rail wear problems on Canadian National's main line through the Rocky Mountains. Three rail wear problems occur simultaneously in curves on this line:

1. Gauge face wear on the high rail.
2. Rail head flow on the low rail.
3. Corrugations having a wavelength varying from 8 to 30 inches on the low rail.

The causes of rail wear and the proposed remedial action apply specifically to this section of our main line, although they are generally applicable wherever similar conditions exist. Causes of rail wear and corrugation other than those described above also exist and may govern rail life on lines where the mix of vehicle loadings, the maintenance and operating practices, or the terrain are different from that on the territory studied. It is hoped that this paper will be useful to other railroads who have similar problems or who may be contemplating similar operations in taking appropriate action to minimize or avoid punitive rail wear.

The Role of Unit Trains. In the case studied, there is no doubt that the rail wear and corrugation problems are the direct results of unit train operations. Prior to the introduction of unit trains hauling coal, sulfur, and potash, there was no really serious problem on this portion of our main line.

In spite of all the obvious advantages of unit train operation cited in the introduction, rail maintenance problems will impose a limitation on the use of such trains unless the wear mechanisms are understood and appropriate action is taken to diminish or eliminate them. It was for this reason that Canadian National Railways carried out extensive field testing on this line in June 1974.

Much of the information given here is found in my report "Tests on B.C. South line, Clearwater Subdivision."

The report confirmed that the damage to the rail was indeed attributable to the operation of unit trains. However, the rail damage is not caused by unit trains, per se, but arises from the use of fully loaded 100-ton cars in these trains. Freight cars in general service are usually loaded well below their maximum permissible gross rail load, while cars in unit trains service operate either fully loaded or empty.

Table 1 shows a traffic split by vehicle type, excluding tonnage generated by empty cars, passenger trains, and locomotives. Note that on average 100-ton cars are loaded to 94% of their rated capacity, while 50- and 70-ton cars are loaded to 74% of their rated capacity.

Examination of the train consists confirmed that nearly all of the 100-ton cars moved in unit trains. Since the basic design of freight car trucks of all capacities is the same, the rail wear problems are more accurately attributable to 100-ton carloadings. It is apparent that in going to 100-ton carloadings we have unwittingly stepped over a threshold and are now suffering punitive rail damages on lines where sharp curves are frequently encountered. The role of the unit train has been to bring this problem into sharper focus. The need for a program of remedial action to improve existing services and for the exercise of caution in designing new services is now apparent.

Understanding the Problem. For any remedial action to be effective, the causes of the problem and the nature of the remedial action must be understood. This understanding is necessary not only for the researchers and other engineering personnel who design and carry out the remedial action programs, but also for those whose involvement in the program is more remote. I refer specifically to field supervisors in the operating department who are responsible for equipment, track, and train operations, to transportation planners and marketing personnel who design new train services; to vehicle and track designers; and to senior officers of the company who must aprove the expenditures and ensure that the maximum return on investment of plant and equipment is achieved. I will, therefore, attempt to deal with the subject by posing the following questions, and then attempting to answer them in a manner that can be understood by people of varying educational backgrounds and work experience:

How severe is the problem?

What are the causes of accelerated rail wear?

What remedial action is required, and by whom?

How severe is the problem? The problem occurs to some extent on all curves but seems to be more troublesome on the sharper curves of 4 deg. and up. I have made rather extensive analyses deriving data from traffic splits by gross car loadings from 1967 to 1974, from rail replacement data from 1964 to 1974, and from annual gross tonnages back to 1960. There is a clearly discernible trend of severely escalating replacement rates as the percentage of fully loaded 100-ton cars increases. It appears that we can anticipate replacing about one-third of the track in these curves annually if the present traffic patterns and replacement rates are sustained.

On our own railway this problem also occurs in other locations where substantial numbers of heavy vehicles operate on curved track.

Curve wear, head flow, and rail corrugations also occur on other railways in Canada and elsewhere in the world. In planning new unit train movements, it is important to consider the percentage of curved trackage over which the trains will operate. Otherwise, greatly accelerated rail wear may occur, and the rates charged by the railroad may not be fully compensatory.

What are the Causes of Accelerated Rail Wear? The accelerated rail wear, although closely linked with unit train operation, is in fact a direct consequence of overloading of the rail by the fully loaded 100-ton cars used in these trains. The rail wear takes three forms:

1. Gauge face wear on the high rail.
2. Head flow on the low rail.
3. Corrugations on the low rail.

Each of these is caused by a different mechanism or mechanisms and will be treated separately, although head flow and corrugation often occur on the same rail. Before discussing these causes, I would like to point out again that this accelerated wear is much greater than would

Table 1
1972 Traffic carried by vehicles of 50-,70-and 100-ton capacity

Car Capacity (Nominal)	Number of Cars	Gross Tonnage	Gross Tons/Car Actual	Gross Tons/Car Gross Limit	% Car Capacity
40-50 tons	126,487	8,257,315	65.3	88.5	74
70 tons	41,595	3,374,630	81.1	110.0	74
100 tons	59,261	7,306,415	123.3	131.5	94
TOTAL	227,343	18,938,360			

be expected due to tonnage increases alone. Nor is overspeeding or underspeeding in curves a necessary condition, since this wear will take place when the vehicles are passing through the curve at the designed speed. However, overspeeding or underspeeding does aggravate the condition and should not be permitted.

Gauge Face Wear. Gauge face wear on the side of the high rail of curved track is caused by vehicle tracking problems. Existing vehicles, both locomotives and cars, do not track very well in curves. This subject has been studied by members of Canadian National's Technical Research Branch as well as by many others. A brief exposition of this tracking problem is given here.

Assuming that the rate of abrasive wear depends linearly on the work done by the friction force between the rail and the wheel flange, then the rate of wear on the gauge face of the rail can be approximated by the following formula:

$W_f = K \cdot \mu_f \cdot \tan\beta \cdot \alpha \cdot F_f \cdot$ where

$W_f =$ Rate of wear.

$K =$ Proportionality constant.

$\mu_f =$ The coefficient of friction between the flange and the gauge face of the rail at the point B as shown in Fig. 1.

$\beta =$ The angle of the tangent to the flange at the contact point between the wheel and the rail, measured from the horizontal position, also as shown in Fig. 1.

$\alpha =$ The angle of attack between the flange of the wheel and the gauge face of the rail as shown in Fig. 2.

$F_f =$ Flange force, which is equal to the sum of two components as given in the equation.

$$F_f = 2\mu eN + H$$

The term $2\mu eN$ is the lateral component of the tread creep force required to slide the wheels laterally in the curve. The term H is the lateral thrust due to unbalanced centrifugal forces, alignment irregularities, dynamic effects such as vehicle rocking, and the interaxle forces on the truck. The forces are illustrated in Fig. 3.

It is apparent that reduction in gauge face wear as well as wheel flange wear can be effected by reducing the magnitude of these four parameters, μ_f, β, α and F_f. The value of the coefficient of friction, μ_f, can be reduced by judicious use of track-mounted wheel flange oilers. The angle β cannot readily be changed, since this angle should not be appreciably less than 70 deg. because of the danger of the wheel flange climbing the rail. Also it should not be greater than about 80 deg. because of the danger of derailment at switch points. The angle of attack α and the flange force can be reduced by proper combination of sufficient wheel tread conicity and flangeway clearance to help the wheelset to steer itself around the curve. The interaction of these two parameters is quite complex and requires further explanation.

The standard AAR new wheel profile has two major defects in its curving capability. These are insufficient conicity and two-point contact in curves. The conicity of 1 in 20 or 0.05 limits the ability of a single wheelset to negotiate curves without flanging to those which are less than 2.4 deg. The two-point contact allows the flange of the

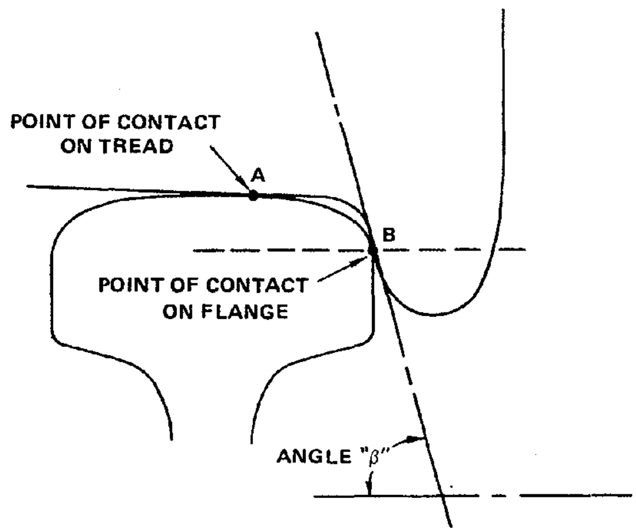


Fig. 1. Wheel-rail contact.

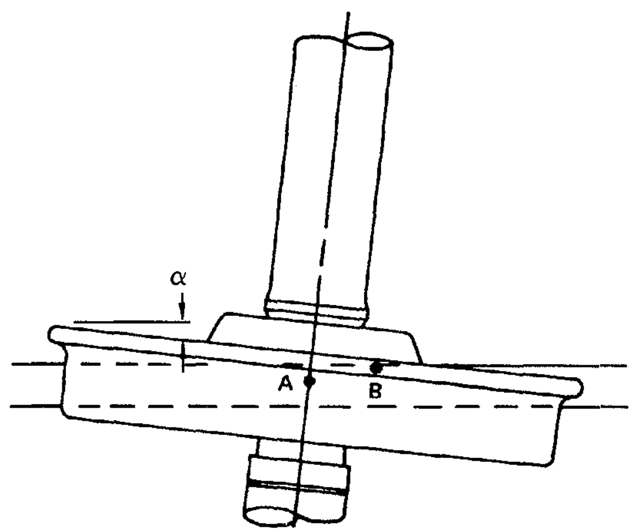


Fig. 2. Angle of attack between wheel flange and rail.

wheel to scrub the side of the rail in curves. Both the defects can be minimized by special profiles having conicities three to five times greater than the new AAR profile and shaped to avoid two-point contact. Fig. 4 shows a new AAR wheel profile with contact at both the flange and crown of a new 132 lb. rail. Fig. 5 shows a special experimental profile designed to give sufficient conicity to allow a single wheelset to pass through most main line curves without flanging and two-point contact.

Special wheel profiles alone on standard freight car trucks will not make the wheelsets negotiate curves in a flange-free condition, because this truck does not have the ability to align the wheelsets radially in the curve. The special profile will, however, reduce wheel flange and rail gauge face wear. Canadian National is presently setting up a test to evaluate the comparative wear characteristics of new AAR profiles against the experimented profile shown in Fig. 5.

Since the flange force $F_f = 2\ \mu_e\ N + H$, as shown in Fig. 3, it can be reduced by diminishing either or both of these components. The term 2 $\mu_e N$ is the lateral component of the tread creep forces required to slide the wheels laterally in a

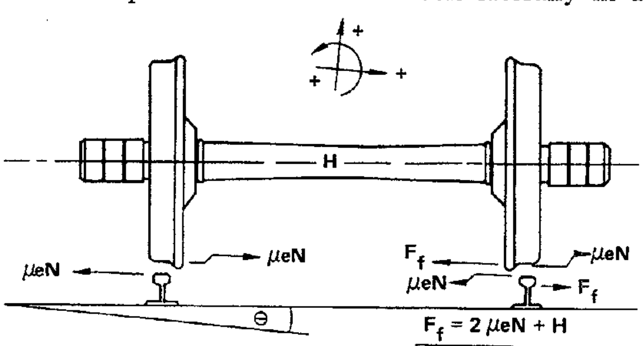


Fig. 3. Lateral forces on curves.

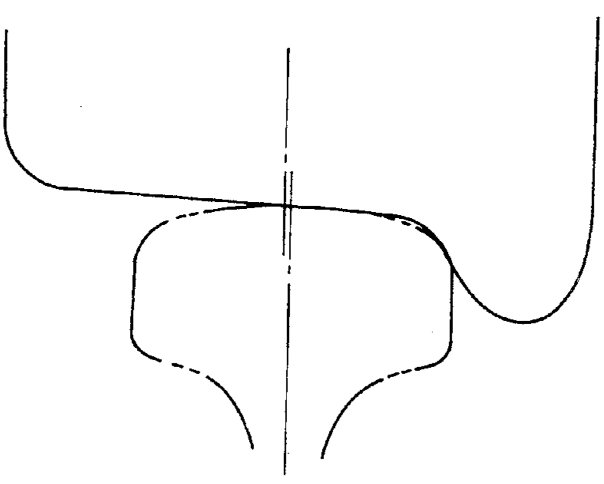


Fig. 4. New AAR wheel profile.

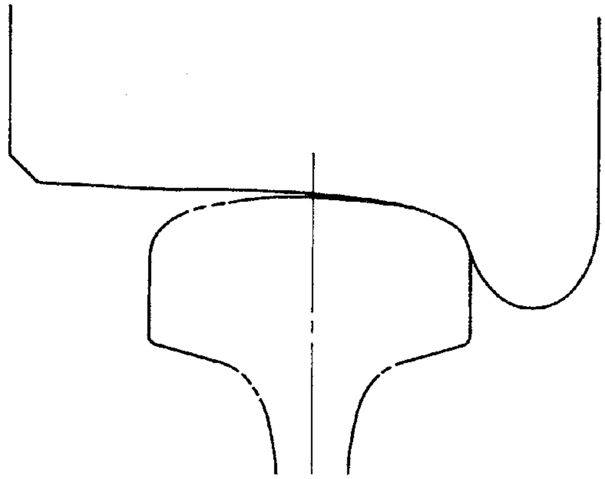


Fig. 5. Experimental wheel profile.

curve, where μ_e is the effective lateral tread coefficient of adhesion and N is the wheel load normal to the rail. The coefficient μ_e depends on the angle of attack a and can vary from zero at zero angle of attack to μ, the limit of wheel rail adhesion at an angle of attack of about 1 deg., as shown in Fig. 6. Thus, if the wheelset has sufficient tread conicity and the ability to align itself radially in the curve, this term, $\mu_e N$, will become zero. With existing AAR profiles and standard three-piece trucks, the angle of attack, a, often exceeds 1 deg. and the value μ_e approaches μ, the limit of wheel-rail adhesion. This gives rise to very high values of tread creep force, which can range from 9,800 to 23,000 lbs. for typical values of μ between 0.15 and 0.35 and a wheel load of 32,875 lbs. (100-ton vehicle). This can be considered to be a major component of flange force, and the importance of achieving a minimized angle of attack, a, through the combined use of profiled wheels and improved truck design with radial curving ability can hardly be overemphasized.

The other component of flange force, H, the lateral thrust, is due to unbalanced centrifugal forces, alignment and cross level irregularities, dynamic effects such as car rocking, and interaxle forces on the trucks. This component can be reduced by diminishing or eliminating these conditions. Interaxle forces arise because the existing freight car truck does not permit the axles to align themselves radially in a curve, preventing the wheel flange from assuming a zero angle of attack. In addition, the clearances between the major components of the truck permit the truck side frames to lozenge, further increasing the angle of attack. Fig. 7 shows the angle of attack of the leading wheel in curves for the three configurations, lozenged, square, and radial.

From the above discussion, it can be seen that there will be minimal lateral force in curves if four conditions are met simultaneously:

1. The vehicles pass through the curve at the exact speed for which the curve is banked.
2. The curve has no alignment and cross-level irregularities.
3. The wheel treads have sufficient conicity and flangeway clearance to steer the wheelsets in the curve without flanging.
4. Vehicle trucks allow the axles to align radially under the action of tread creep forces.

Since the force on the flange increases with increasing angle of attack and since flange wear on the wheel, gauge face wear on the rail, and curving resistance all increase directly with the angle of attack, the benefits to .be derived from a truck design which permits radial action are obvious. A prototype truck with radial curving capability is currently under test at the Technical Research Center.

In summary, gauge face wear on rails in curves can be reduced and controlled within economic limits by a five-pronged effort to achieve the following:

1. Close maintenance of alignment, superrelevation, surface and cross level, and gauge in curves.
2. Operation at equilibrium speed in curves.
3. The use of specially profiled wheel treads with increased conicity for curve negotiation.
4. New or modified truck designs to permit the axles to align themselves radially in curves.
5. Judicious use of track-mounted rail lubricators in curves.

Rail Head Flow. Rail head flow is found on the low rail in curves on the B.C. South Line and is caused by excessive pressure at the point of contact between the wheel and the rail. The mechanism, which is believed responsible for this excessive contact pressure, is described in this section.

Both the wheel and the rail have curved surfaces at their point of contact. For a fully loaded 100-ton vehicle, the contact area is an ellipse, usually with the long axis lying along the rail. This contact area is very small -- less than one quarter of a square inch, assuming a new coned

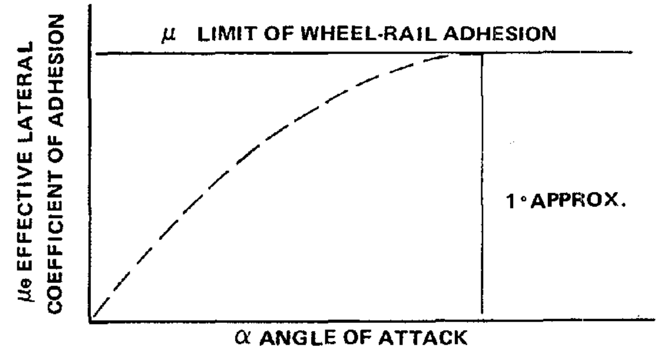


Fig. 6. Effective lateral coefficient of adhesion vs. angle of attack.

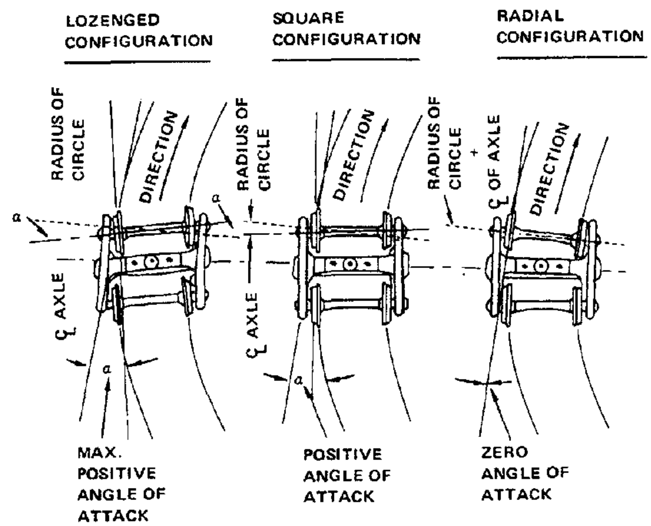


Fig. 7. Angle of attack of leading wheel in curves.

wheel, 36 ins. in diameter, resting on a rail with a crown radius of 10 ins. Thus, the whole weight of the vehicle is supported on an area of less than two square ins. This works out ot an average contact pressure of 131,000 pounds per square inch. However, the maximum stress at the center of the contact area is about 1.7 times greater than this average stress, or about 220,000 pounds per square inch.

Materials subject to this type of three-dimensional stressing can support higher stresses than is possible under uniaxial loading in pure tension or compression. There is a limit, however, and when this limit is exceeded, the material yields. A piece of rail steel having a yield strength of 75,000 psi in tension was found to yield locally under an applied load of 220,000 pounds per square inch, indicating that the head of a new 132-lb. rail would yield under the loading imposed by new wheels on a fully loaded 100-ton car. In practice both the rail head and wheel tread will yield slightly. The wheel tread will become slightly hollow and the rail head will flatten slightly, increasing the surface of contact. This yielding will continue until the contact pressure is below the yield point. This is one reason why the rate of wear on a new coned wheel running on a new crowned rail is usually found to be quite high in the initial stages. The other reason is that work hardening takes place during the wearing-in process, increasing the hardness and strength of the rail. The combined effect of these two processes is to increase the ability of the wheel to withstand higher contact loads before yielding.

Unfortunately, a set of conditions occurs in curved track on our B.C. South Line which does not permit this stabilization of yielding on the rail head. Excessively high head flow and associated corrugation are still occurring on the low rail in curves. This will continue to occur until corrective action is taken. Moreover, this mechanism is present not only on our lines but occurs on railways all over the world where similar conditions are present.

The first satisfactory solution for contact stresses occurring between two elastic bodies having curved surfaces was provided by Hertz in 1881. For a steel wheel on a steel rail, the maximum compressive stress can be approximated using the following formula:

$$q_o = 2.36 \times 10^4 \; \Sigma \; \left(\frac{1}{R}\right)^{2/3} (P)^{1/3}$$

where q_o = The maximum compressive stress in pounds per square inch.

p = The imposed wheel load in pounds.

$$\Sigma \frac{1}{R} = \frac{1}{R_1} + \frac{1}{R'_1} + \frac{1}{R_2} + \frac{1}{R'_2}$$

R_1 = The radius transverse to the tread in inches.

R'_1 = The radius of the wheel in inches.

R_2 = The crown radius of the rail head in inches.

R'_2 = The track curvature in the vertical direction. Since there is virtually no vertical curvature, R'_2 approaches ∞. Therefore, $\frac{1}{R'_2}$ can be assessed to be always equal to zero, and this term can be eliminated from the calculation of maximum contact stress.

The above formula can be used to assess the relative importance of these variables in generating the maximum contact stress, q_o. Table 2 shows the effect of varying the design parameters, P, R_1, R'_1 and R_2.

1. The effect of the applied load -- P. The formula states that the maximum compressive stress varies as the cube root of the applied load, that is, as $(P)^{1/3}$. Thus if the wheel load is doubled, the maximum compressive stress is $(2)^{1/3}$, or 1.26 times the original maximum compressive stress. Minor variations in wheel loadings will therefore have a very small effect on the maximum compressive stress.

Table 2
Effect of design parameters on q_o

Change No.	Design Parameters P (pounds)	R_1 (INS.)	R'_1 (INS.)	R_2 (INS.)	$P^{1/3}$	$(\frac{1}{R})^{2/3}$	q_o (PSI X 1000)	Change in q_o (%)	Parameters Varied
0	32875		18.0	10	32.03	0.29	219	0	None
1	27500				30.18	0.29	206	- 6	P
2			16.5		32.03	0.30	227	+ 4	R'_1
3	27500		16.5		30.18	0.30	210	- 4	P and R'_1
4				+14	32.03	0.25	190	- 13	R_2
5		- 15.0			32.03	0.20	151	- 31	R_1
6		+ 6.0			32.03	0.47	355	+ 62	R_1
7		+ 2.0			32.03	0.75	570	+ 160	R_1
8		- 15.0		+15.0	32.03	0.15	110	- 50	R_1 and R_2

This is shown in Table 2, change No. 1. In this table, the standard for comparison (change No. 0) is a fully loaded, 100-ton capacity vehicle mounted on new AAR profile wheels of 36-in. diameter. The wheels run on a new 132-lb. rail section with a head radius of 10 ins. under these conditions, the maximum contact stress is estimated to be 219,000 psi. If the load in the vehicle is reduced to that of a 70-ton capacity vehicle, the value of P reduces from 32,875 lbs. to 27,500 lbs., and a maximum contact stress is reduced to 206,000 psi, a reduction of only 6%. For this reason, reduction of loads carried by 100-ton vehicles will be relatively ineffective in reducing the incidence of head flow on the rail, although other track maintenance problems, such as rail spreading, spike bending, tie cutting, flexural fatigue, and maintenance of line and surface, should be alleviated. Remedial action for the head flow problem must involve the other term in the equation,

2. The effect of change of wheel radius -- R'_1. In general, the smaller the wheel, the larger is the maximum contact stress. However,

within the limits of normal railway practice, the size of the wheel has a very marginal effect on the maximum contact stress. Change No. 2 shows that if 33-in. diameter wheels were used on the 100-ton vehicle rather than the standard 36-in. diameter wheel, the value q_o increases only 4%. Therefore, increasing the wheel diameter is not an effective solution for the head flow problem.

3. The effect of change in rail head radius--R_2, the radius of the rail head, is a function of rail design, for example a new 132-lb. rail has a crown radius of 10 ins. Change No. 4 shows that an initial improvement of 13% is possible by providing a new rail with a 14-in. crown radius. However, in service, the rail head will tend to wear or flow to accommodate the average wear pattern of the wheel treads, except on the low rail in sharp curves where the rail head will be flattened and flowed regardless of the initial rail profile. Therefore, a change in rail head curvature will not remedy the head flow condition in curves.
4. The effect of change in the radius transverse to the tread -- R_1. As shown in Fig. 8, the value R_1, the radius across the tread, may be infinitely large, negative or positive, depending on the wear condition of the wheel. When the wheel is new, the coned surface is a straight line in the contact plane, and R_1 becomes infinite and $1/R_1$ becomes zero. This is the comparison case, Change No. 0 and q_o equals 219,000 psi, as shown in the table. When the wheel is worn, the central portion of the tread hollows out to approximately 15-in. radius. By convention, this radius is considered negative for purposes of calculation. This condition is shown as Change No. 5, giving a value of q_o equal to 151,000 psi, or a reduction of 31% over the standard for comparison.

At the edge of the worn tread, a reverse curvature of 2 - 6 ins. may develop, as shown in Fig. 8. With reverse curvatures of 6.0 and 2.0 ins., values of q_o of 355,000 and 570,000 psi, respectively, are developed, giving increases of 62 and 160%. These are shown in the Table as Changes Nos. 6 and 7. It is this reverse curvature on the edge of the wheel tread which is responsible for the head flow problems encountered on our lines.

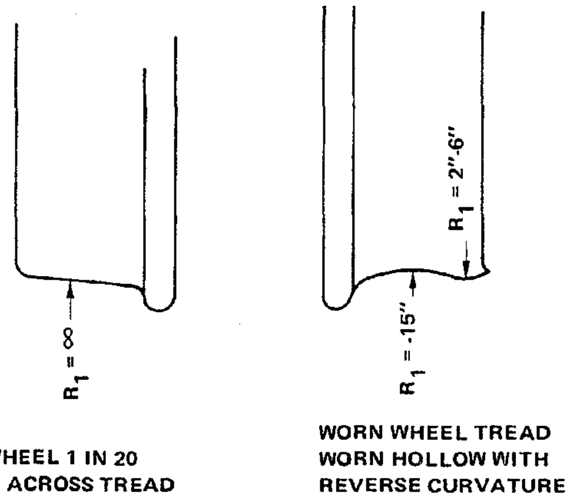


Fig. 8. R_1 transverse tread radius.

Provided that sufficient flangeway clearance exists between the wheels and the rails, the outer convex portion of the wheel tread can ride up on the rail head (Fig. 9). It can be shown that under conditions which are not considered condemnable, the outside edge of the wheel can be 0.6 ins. inside the field side of the rail, and the center of the reverse curvature may therefore contact the rail 1 in. or more from the field side. This condition is essentially point loading and generates maximum contact stresses several times that developed for a new coned wheel. This point loading is illustrated in Fig. 10 for actual sections of worn wheel and rail.

Fig. 11 shows graphically the effect of transverse tread radius on maximum wheel rail contact stresses for a rully loaded 100-ton car on 36-in. diameter wheels and a fully loaded 70-ton car on 33-in. diameter wheels. Note that the effect of reduction of gross rail load on the maximum contact stress is rather small, being about 3%. Even if the 70-ton car were fitted with 36-in. diameter wheels, the reduction would not exceed 6%. Thus, the remedial action to eliminate head flow must involve the elimination of this reverse curvature condition on the worn wheel tread.

The formation of Rail Corrugations. Rail corrugations are found on the top surface of the low rail in curves and can cause serious maintenance problems. If these corrugations are not removed, they can quickly become so severe that rail grinding becomes impractical and the rail must be scrapped. The standard procedure is to use a rail grinding train, which removes the corrugation in one to three passes of the machine. Generally it is considered uneconomic to grind out corrugation deeper than 40 thousandths of an inch. Since the rail grinding train moves at about 1.5 mph, rail grinding maintenance results in loss of track capacity.

The causes of rail head corrugations are quite complex and have been studied by railway administrations all over the world. A number of variables are obviously involved, such as rail head lubrication, metallurgy, vehicle design, and operating speeds. The wavelength of corrugations measured on the B.C. South Line ranges from 8 to 24 ins., and wide variation exists in wavelengths on the same rail. I am unable to explain this variation in wavelength using generally accepted vibrational properties of the vehicles and track.

However, corrugations cannot form if the rail head surface is sufficiently strong to resist plastic deformation from the wheels of the vehicles passing over it. A mechanism has been described which is believed responsible for the head flow condition on this line. This mechanism is the reverse curvature on the outside of the wheel tread, which can generate a very high contact stress between the wheel tread and the rail head whenever sufficient flangeway clearance is present to allow this portion of the wheel tread to ride on the rail. The presence of rail corrugations can be detected by the track recorder car. A study of these records shows that corrugations are associated with wide gauge conditions in curves.

The control of rail head corrugation can, therefore, be most effectively achieved by the reduction or elimination of the incidence of rail head flow. Corrugations are not a serious problem on the B.C. South Line, where rail head flow does not exist.

<u>What Remedial Action is Required and by Whom?</u> Remedial action is required to correct gauge face wear caused by two vehicle tracking deficiencies. The first deficiency arises from the fact that wheel profiles do not have sufficient conicity to steer the wheelset around most curves without flanging. The second tracking deficiency is that the vehicle trucks are not designed to allow the axles to align themselves radially in a curve. To correct for head flow and corrugation, it is necessary to prevent the reverse curvature on the outside of the wheel tread from running on the top of the rail. This can be done by reducing or eliminating the reverse curvature on the tread of the wheel and by close control of excessive flangeway clearance, whether due to gauge face

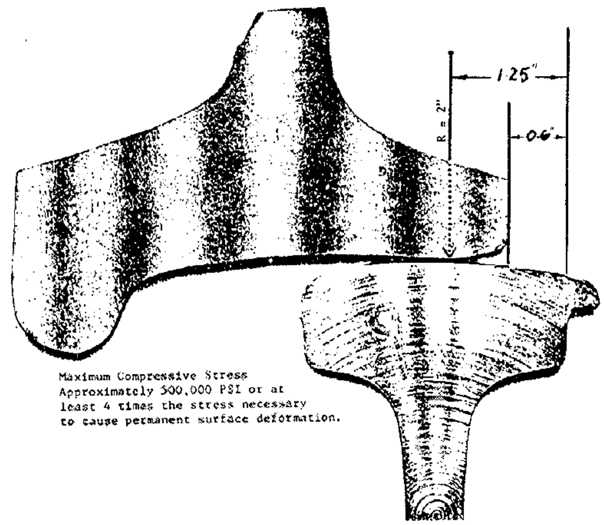


Fig. 10. Contact between wheel and rail in curves.

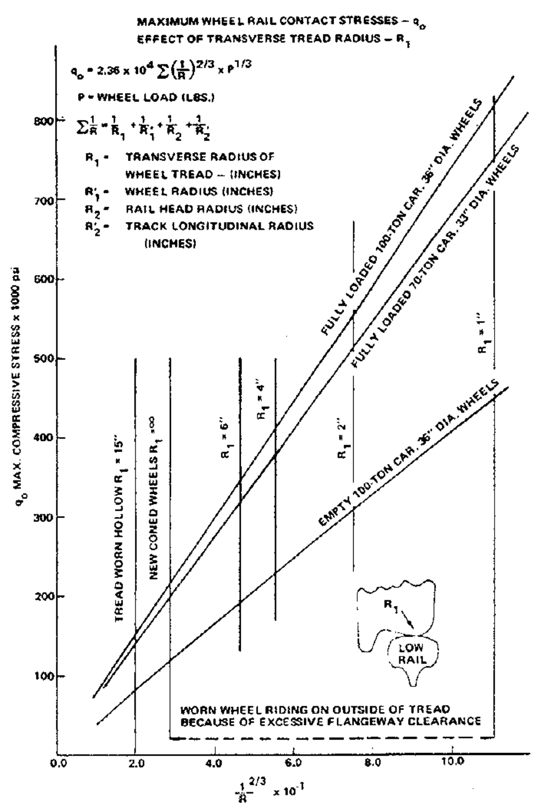


Fig. 11. Effect of transverse tread radius on maximum wheel rail contact stresses.

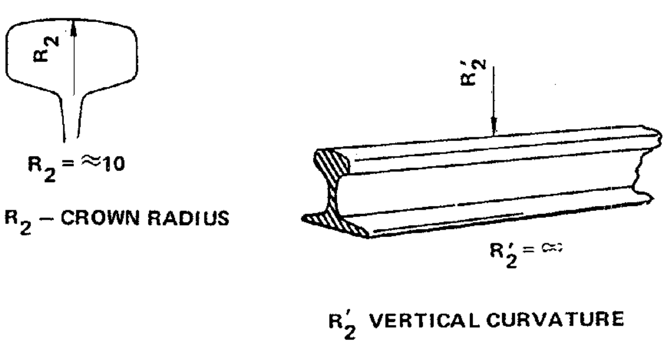


Fig. 9. R_2 and R_2' rail radii.

wear, wide gauge, or wheel flange wear.

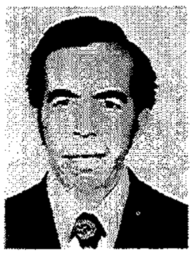
Other remedial measures, although not aimed at the specific mechanisms causing the rail wear problem, have considerable merit and should also be incorporated into the remedial action program. These are:

1. Judicious use of flange oilers to reduce gauge face wear.
2. Use of rail steel with higher yield point to reduce both gauge face wear and head flow.
3. Grinding of the rail to remove existing corrugations before these become so deep that grinding becomes impractical.
4. Avoidance of overspeed or underspeed in curves, as this aggravates all wear conditions.

For convenience, the required remedial action is tabulated below:

REMEDIAL ACTION	PURPOSE OF ACTION	FUNCTION RESPONSIBLE FOR APPLYING ACTION
Buy and maintain all wheels to a special profile with increased conicity	To reduce gauge face wear on high rail in curves. Also increases wheel life and reduces head flow and corru-gation on low rail	Equipment
Eliminate reverse curvature on outside of wheel tread	To reduce or elim-inate head flow and corrugation on the low rail in curves	Equipment
Develop and use a truck with improved curving properties	Eliminate or reduce gauge face wear. Also reduces head flow and corrugation	Equipment (with Technical Research)
Maintain close control of wide gauge in curves	Eliminate or reduce head flow and corrugation	Engineering
Maintain existing flange oilers and extend their use as found necessary	Reduce gauge face wear	Engineering
Use rail steel with higher yield point	Reduce gauge face wear and head flow	Engineering
Grind out rail corruga-tions	Remove existing corrugations	Engineering
Avoid overspeed or underspeed in curves	Aggravates all conditions	Transportation

Conclusion. The mechanisms causing the rail wear conditions have been described and are well understood. Effective remedial action is possible, but it requires a concerted effort by Engineering, Equipment, and Transportation Functions. There is no "quick fix" that can be brought and applied. Moreover, it must be realized that although effective action can be initiated almost immediately, it may be some time before the full benefits can be assessed. However, if no action is taken, the situation will not improve or go away; it can only deteriorate. If we wish to move bulk commodities economically in unit trains, we must attack the problem in an organized manner.



Gonzalo Rivera Diaz
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National Railways of Mexico

Mr. Diaz graduated from the University of Mexico in 1962 with a degree in Civil Engineering. He then received a scholarship and spent the next two years in France studying the construction and design of bridges (SNCF, STUP).

He has been with the National Railways of Mexico since 1961, involved in the design, supervision and construction of bridges. He assumed his present position of Bridge Engineer in 1972.

HEAVY AXLE LOADS IN NATIONAL RAILWAYS OF MEXICO AND NEED FOR STRENGTHENING OF BRIDGES

Introduction. The purpose of this paper is to give to the delegates of the conference a general idea about the actual condition of bridges in the National Railways of Mexico and the manner in which we are proceeding to solve the problems of low capacity, taking account of the increase of heavy traffic during the past few years. We are not going to present spectacular jobs or outstanding bridges, but some of our routine work for the more common types of existing bridges in our system.

Generalities. At the present time the National Railways of Mexico is confronted with a very serious problem caused by the traffic of heavy equipment. Use of this kind of equipment was established to satisfy the need caused by traffic increases of the past few years. It is common for heavy trains to be running over bridge structures that were not designed to support such important live loads.

In the system there are 10,020 bridges with a total length of 64.15 miles. Of these 52% are steel or concrete bridges and 48% wood trestles and mixed bridges with steel beams and wood bents. Most of the steel bridges were built during the last years of the 19th century or in the first ten years of the present century and were designed for a live load according to the traffic importance of each line at the time. Thus we have, in the Mexico-El Paso Railroad, bridges built for Cooper E60 live load, the Mexico-Manzanillo route with E55 bridge capacity; some bridges in the Mexico-Laredo Railroad route have E45 capacity, and in the Coatzacoalcos-Salina Cruz line, through the Tehuantepec Isthmus, the bridges were designed for E40. There were other lines with E35 bridges for narrow track gauge at Mexico-Veracruz (Interocean Line), Mexico-Acambaro, etc., where a widening was made during the period 1950 to 1960.

Before 1970 the design live load in National Railways of Mexico was Cooper E60. In that year the new live load adopted was Cooper E72, which has been kept in use until today. The E72 was established to support the heavy traffic of equipment consisting of 3,000 or 3,600 hp. diesel electric locomotives and freight cars with heavy axle loads, like gondolas handling minerals, hopper cars, or "jumbo" tank cars.

The heaviest four-axle locomotives operating now in the system produce 67.36 kips per axle, corresponding to the B-B AAR designation. The heaviest six-axle locomotives, type C-C, produce 63.33 kips per axle. These two types of locomotives can operate over bridges of any length with original-capacity Cooper E60, taking account of the reduction of impact allowed by AREA.

For diesel electric equipment, nevertheless, in short span bridges or short panels of trusses or through girders with original E50 capacity, the effects of the above locomotives put these bridges in a disadvantageous condition. In our system we can observe the trend of increasing the axle loads of the locomotives. The most serious problem is represented by the trains that handle minerals. Presently these trains go over the main lines of the country carrying iron minerals for the steel industry. The trains are integrated with gondolas 44'10" in length and with a total weight of 264 kips, 66 kips per axle. Because of the great density, this equipment has greater effects than the locomotives on medium and long bridges.

Besides the above-mentioned problems, we find another is caused by heavy freight cars proceeding from the U.S. which enter into our system in regular trains. These cars, like the "jumbo" tank cars, become as harmful to weak bridges as the gondolas for handling minerals.

Actions to Support Traffic. Actions taken by National Railways of Mexico to support the heavier traffic include the following:

1. Actual capacity revision of bridges, considering that the system now operates only diesel electric locomotives, and thus less impact is produced than with the steam locomotives used a few years ago in Mexico.
2. Placing of slow orders for traffic on bridges for trains whose effects are greater than the actual capacity.
3. Track improvement on bridges, eliminating anomalies of alignment and level of track and placing neoprene plates under the rail base to reduce the impact forces.
4. Strengthening of bridges in cases where it is economically possible.
5. Replacement of bridges of very low capacity or bad actual condition.

Strengthening. Temporary strengthening measures generally consist of placing pile wood bents under steel girders to reduce the working span or in trusses to isolate the panels by building the bent under the joints.

Various methods of permanent strengthening have been applied, depending on the type of structure.

Deck girders have been reinforced with the addition of riveted or welded cover plates and stiffeners or by increasing the number of beams under the track.

The capacity of some through girder bridges has been elevated by placing cover plates and additional stiffeners on the master beams. The floor system has been modified with the addition of suplementary floor beams at the middle of the original panel. This is the case with the Huamantla bridge in the Mexico-Veracruz route. (Figs. 1, 2, and 3.) This bridge was supported by pile wood bents under each panel. All the work was done while preserving the normal traffic of the line.

Deck truss reinforcements have been studied in several bridges of the Guadalajara Manzanillo line, by adding a third central truss to the existing pair for each span. In the Mexico-Queretaro route strengthening was done at the Barranca Honda bridge (Figs. 4 and 5). From the calculations and revision of the existing 210-ft. deck truss, we found it necessary to raise the capacity of girders and the bracing system, which were the bridge's weakest parts. The girders were reinforced with cover plates and perfectly fixed to the floor beams. The moment of inertia of all the members of the bracing system was increased with the addition of plates and angles.

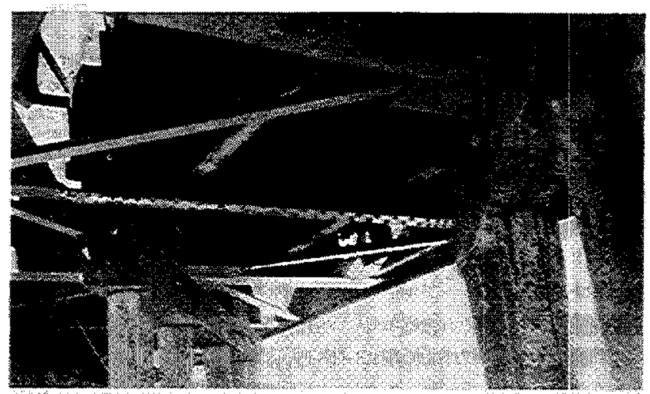

Fig. 1. Strengthening of Huamantla Bridge Mexico-Vera Cruz line.

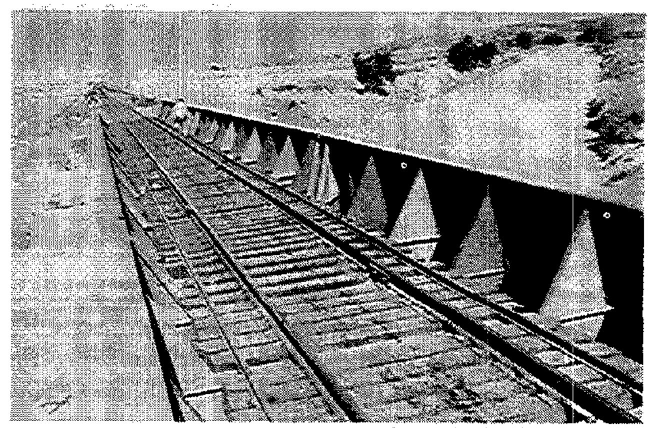

Fig. 2. Strengthening of Huamantla Bridge.

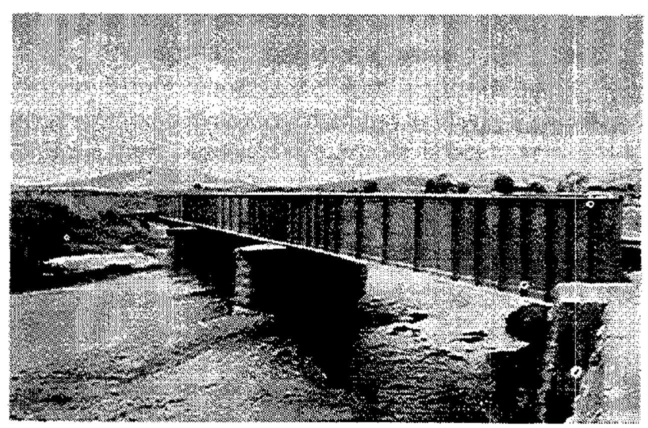

Fig. 3. Strengthening of Huamantla Bridge.

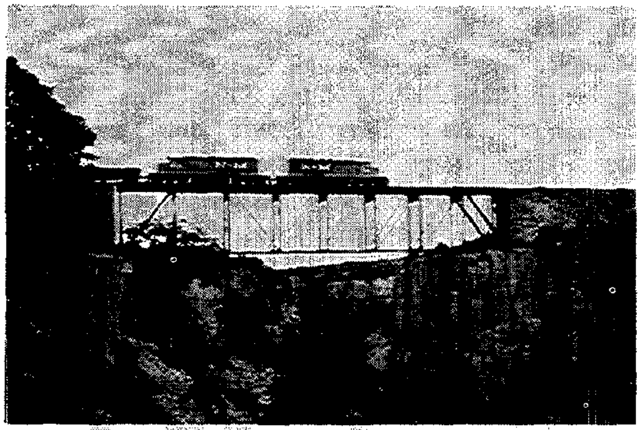

Fig. 4. Barranca Honda Bridge Mexico-Queretaro route.

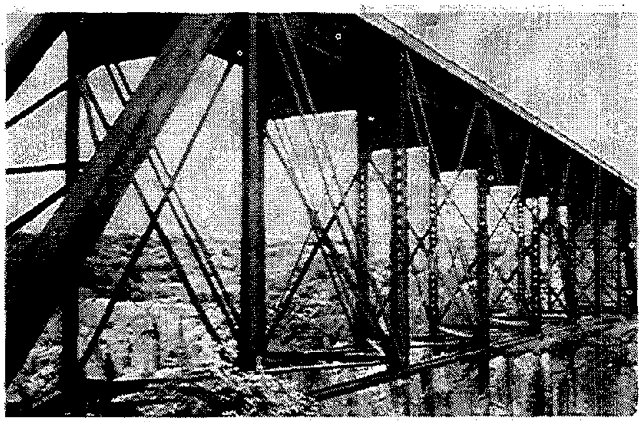

Fig. 5. Barranca Honda Bridge Mexico-Queretaro route.

For riveted truss bridges, strengthening has been done by adding plates and profiles to the weak members. We have found more difficulties in reinforcing pin through trusses with eye-bar tension members. In order to strengthen the structure some tension members and pins have been changed in a few bridges temporarily supported by false work.

A study was done to elevate the capacity of pin trusses. The method consists of prestressing the structure by placing parabolic high-strength strands under the bottom chord. In this manner upward forces are applied against each joint of the truss, voiding the dead load of the bridge. It is possible to raise the truss capacity by about 15%.

Wood trestles have been strengthened by increasing the number of wood stringers or rails when the superstructure is formed with rail girders.

Substitution of Structures. In some bridges with very low capacity or in bad condition reinforcement proves uneconomical. In this case a substitution program has been established. In spans below 60 ft. the use of prestressed concrete girders is very common. Up to 60 ft., steel structures are generally chosen for the substitution.

In the Veracruz-Isthmus line, in which bridges are E40, many replacements of structures are being done in order to raise the capacity of the line to E72, considering the growth of tonnage transported in this route. We can mention a few of the renewed bridges in this line.

The Paso del Toro bridge over the Cotaxtla River is a two 100-ft. span bridge. Abutments and pier were reinforced. Two through trusses of reduced clearance were eliminated in order to install two E72 trusses of standard clearance. Figs. 6 to 13 show some of the stages of the installation of the trusses.

The first stage required four hours to take off the two old trusses without the floor system. This operation was done with two 40-ton cranes. The floor system stayed, like a temporary bridge. In the second stage the new trusses were installed with a traffic interruption of four hours. The new trusses were assembled away from the bridge on a side track and carried with the two cranes from the side track to the correct position in the bridge. A third and a fourth stage without traffic for three hours was necessary to remove the old floor system of

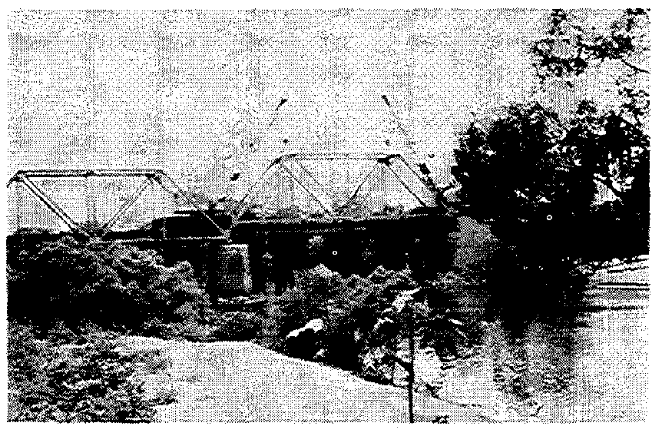

Fig. 6. Substitution of Bridge over Cotaxtla River Vera Cruz-Isthmus line.

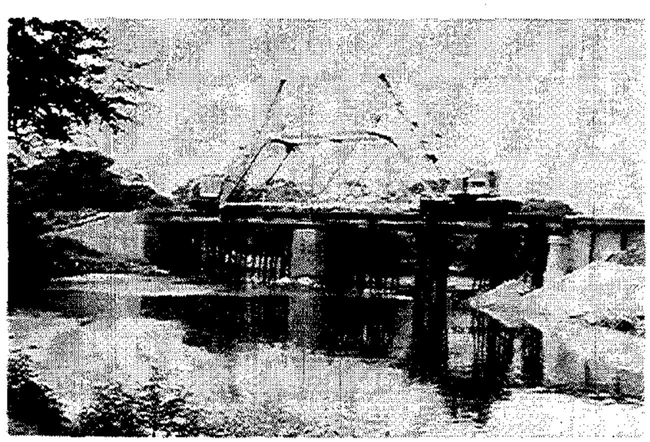

Fig. 7. Bridge over Cotaxtla River.

each truss and to place the new one. Working like this we avoided a longer interruption of traffic, since the assembly was done out of the main track.

Figs. 14 to 17 illustrate the installation of a 120-ft. through truss bridge at Naranjal River. To take off the old truss and install the new one took three hours; we programmed the installation in the period we had between two trains.

Figs. 18 to 24 represent another 120-ft. through bridge installation at Juanita River. The old structure presented failures in the abutments and the superstructure was E40. The new truss was installed in a six-hour period, being totally assembled out of the main track. In order to place the new truss we slid it over the track, pulling with two locomotives, and then moved it to the final position in the bridge, parallel to the old one. Here we did not need false work for the installation.

In the Monterrey-Laredo line over the Salado River there is a four-span bridge. Three spans are 75-ft. deck girders, and the other is a 180-ft. through truss. This bridge represents an obstacle to the passing of big shipments because of the reduced clearance of the truss.

Figs. 25 to 27 show the initial work to install the superstructure. The three 75-ft. deck girders will be installed with two 40-ton crances. To install the new truss a false work is now being built below the existing truss, using one span of the recovered 75-ft. girders in the middle of the river bed and

Fig. 8. New trusses for bridge over Cotaxtla River.

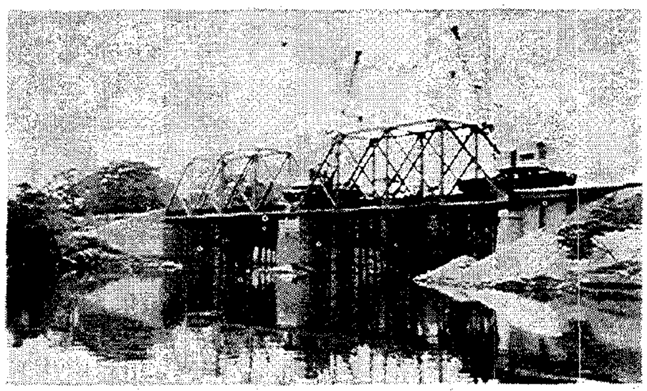

Fig. 11. Installation of new trusses Cotaxla River Bridge.

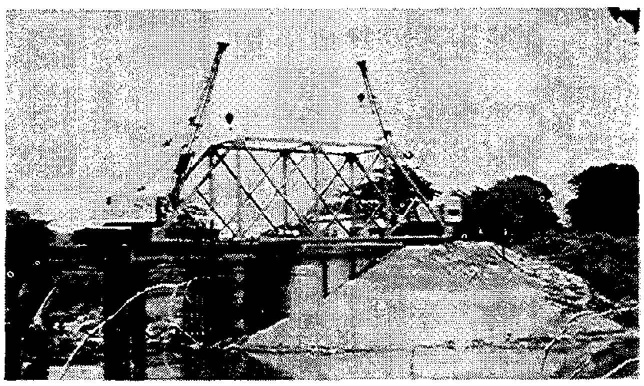

Fig. 9. Installation of new trusses Cotaxtla River Bridge.

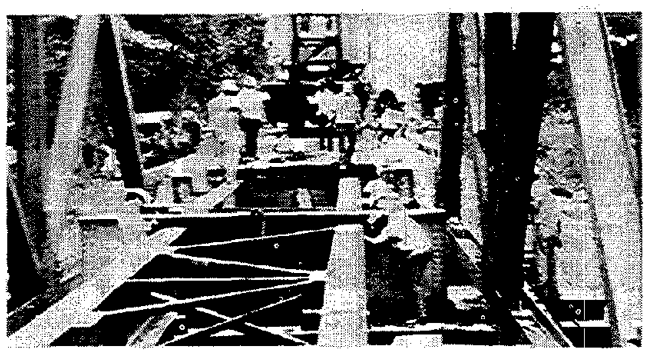

Fig. 12. Removing the old floor system Cotaxtla River Bridge.

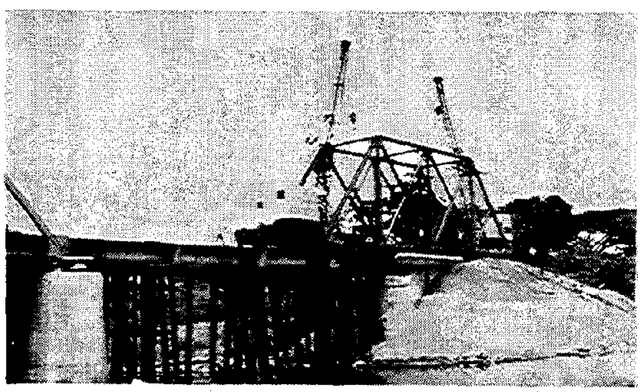

Fig. 10. Installation of new trusses Cotaxla River Bridge.

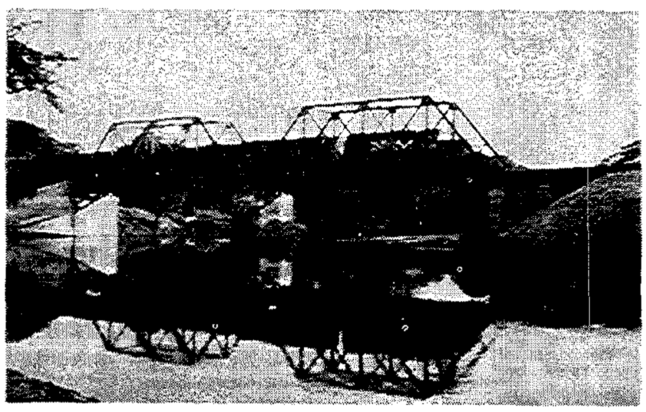

Fig. 13. Renewed bridge over Cotaxtla River.

two 50-ft. stringers besides. The new truss is now assembled out of the bridge. To take off the old truss and put on the new one, two flatcars and hydraulic jacks will be employed instead of cranes. A six-hour period is estimated for this work, without including the substitution of the floor system, which will be programmed to be done afterwards. A very common case of bridge substitution is the wood trestle for concrete trestle.

Figs. 28 and 29 show some work which is being done at Lamadrid Brook in Coahuila to construct a new trestle formed with prestressed concrete bents, to support 20, prestressed concrete spans 27 ft. each. All these tasks have been done without important traffic interruptions. This bridge is located over the Monclova-La Perla Railroad, whose main function is to transport iron ore for the iron works Altos Hornos de Mexico.

Finally, I would like to say that the National Railways of Mexico has developed a large program to strengthen the bridges of the system in order to support the heavy axle loads up to now. But today, when we are in the midst of the project, we ask what will be the limit of heavy axle loads in the future.

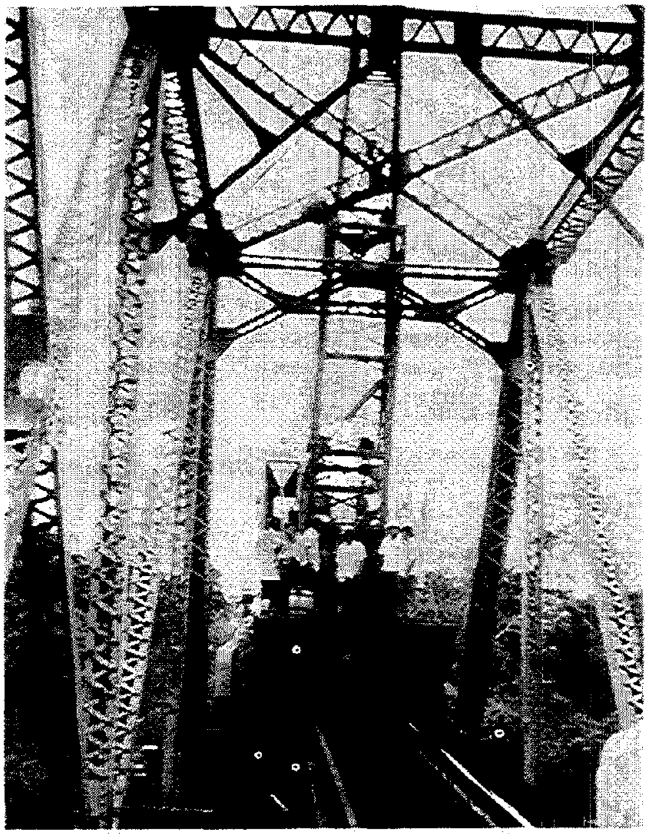

Fig. 14. Old bridge over Naranjal River.

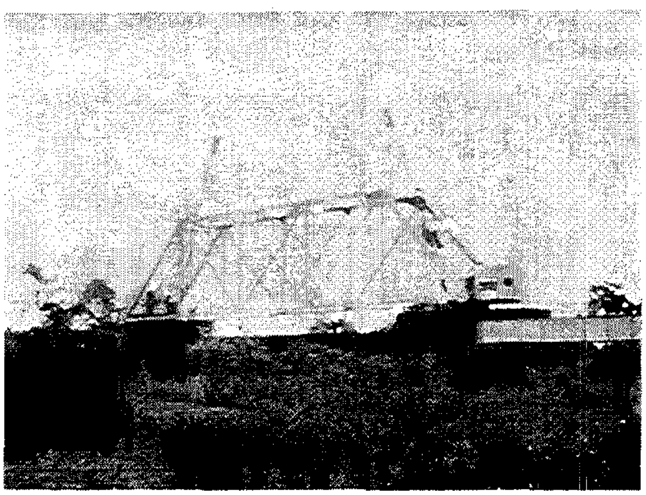

Fig. 15. Removing the old truss Naranjal River.

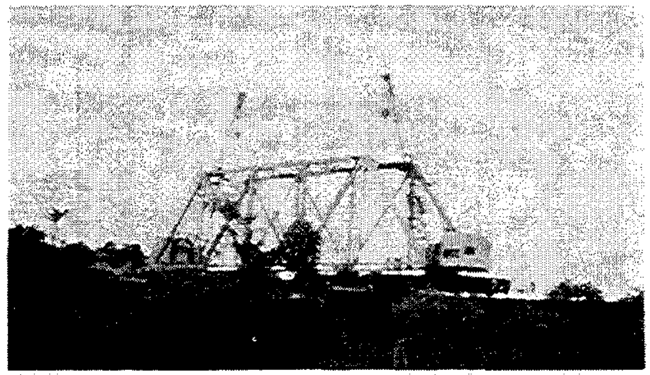

Fig. 16. Naranjal River Bridge installation of the new truss.

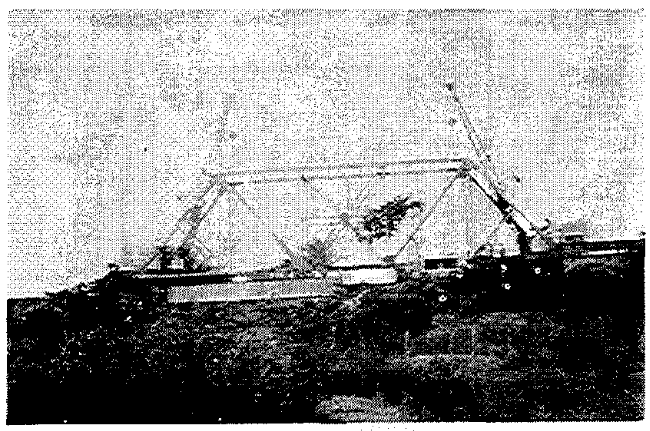

Fig. 17. Naranjal River Bridge.

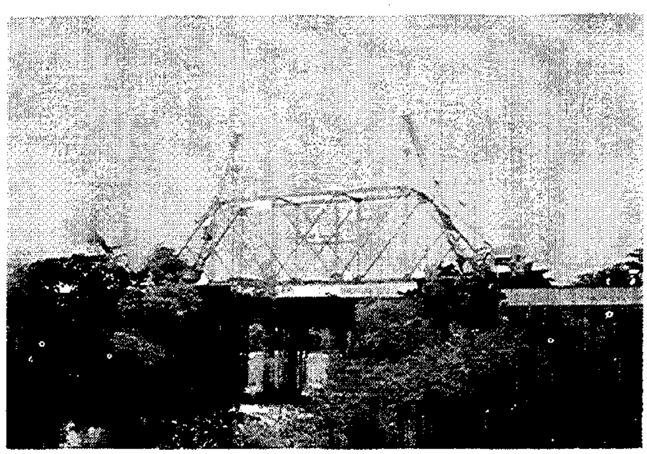

Fig. 18. Juanita River Bridge installation scheme.

Fig. 19. Juanita River Bridge installation of the new truss.

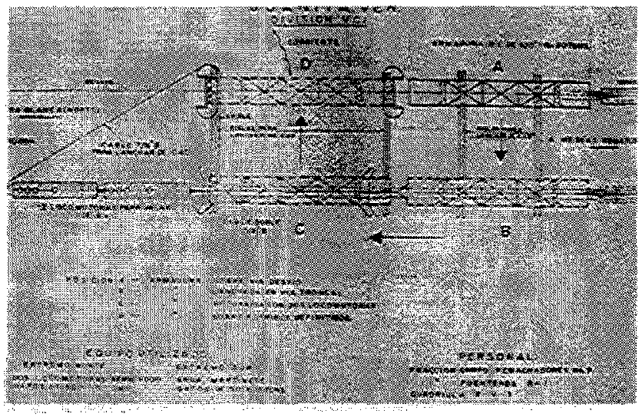

Fig. 20. Juanita River Bridge installation of the new truss.

Fig. 21. Juanita River installation of the new truss.

Fig. 22. Juanita River pulling the new truss over the old bridge.

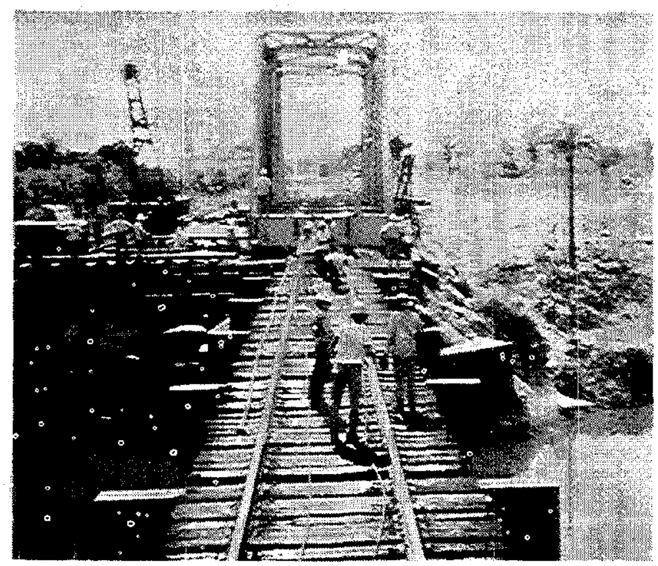

Fig. 23. Juanita River Bridge installation of the new truss.

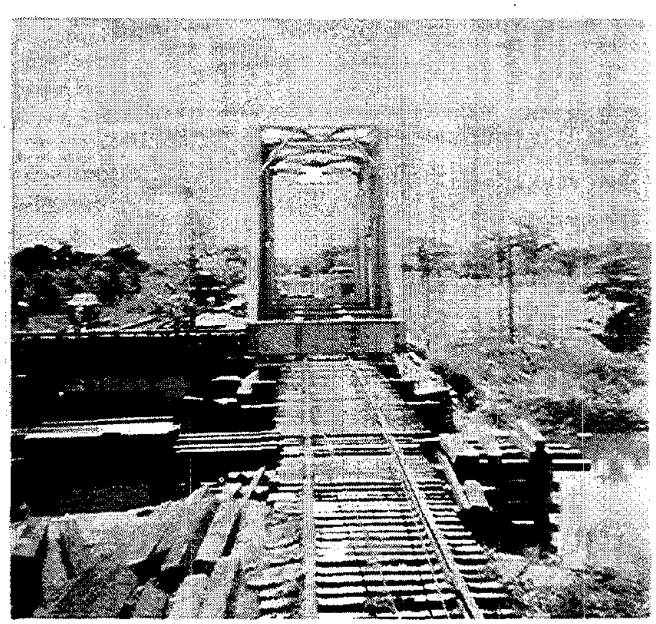

Fig. 24. Juanita River Bridge.

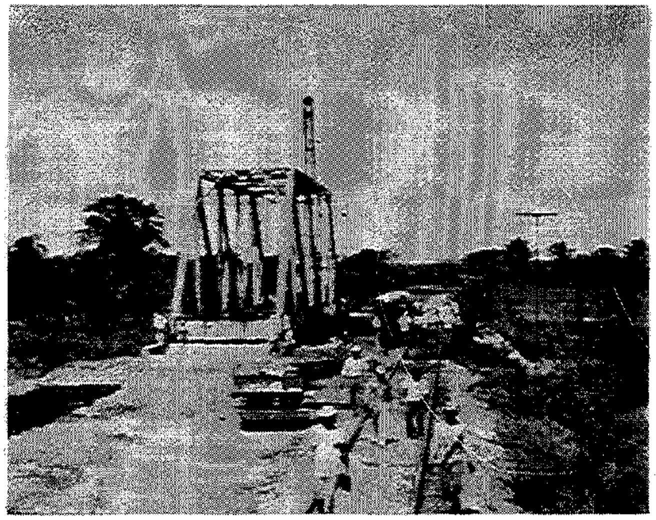

Fig. 25 Bridge over Salado River Monterrey-Laredo line.

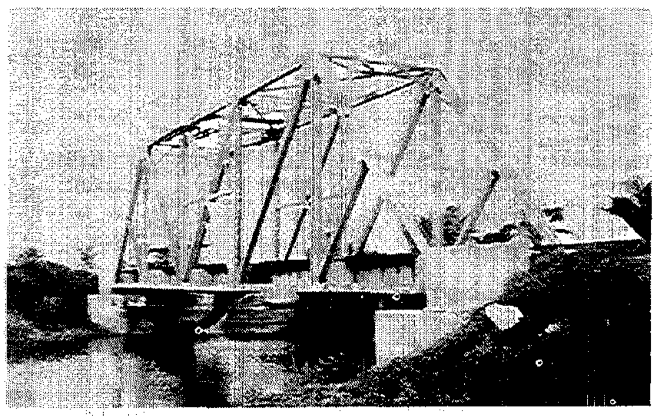

Fig. 26. Installation of a deck girder Salado River.

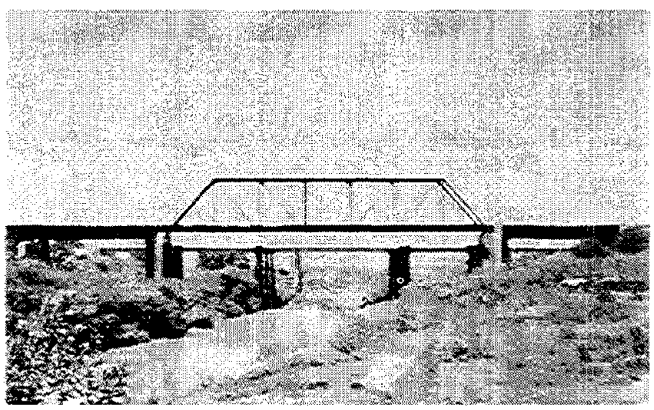

Fig. 27. Bridge over Salado River.

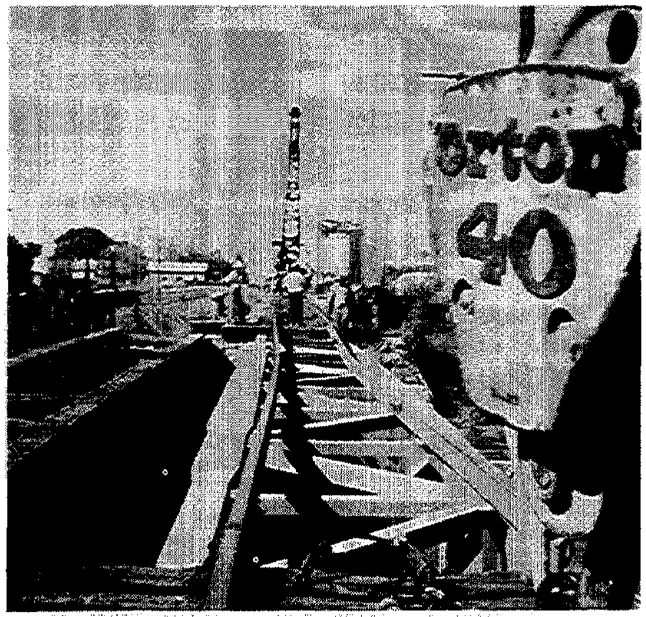

Fig. 28. Lamadrid trestle.

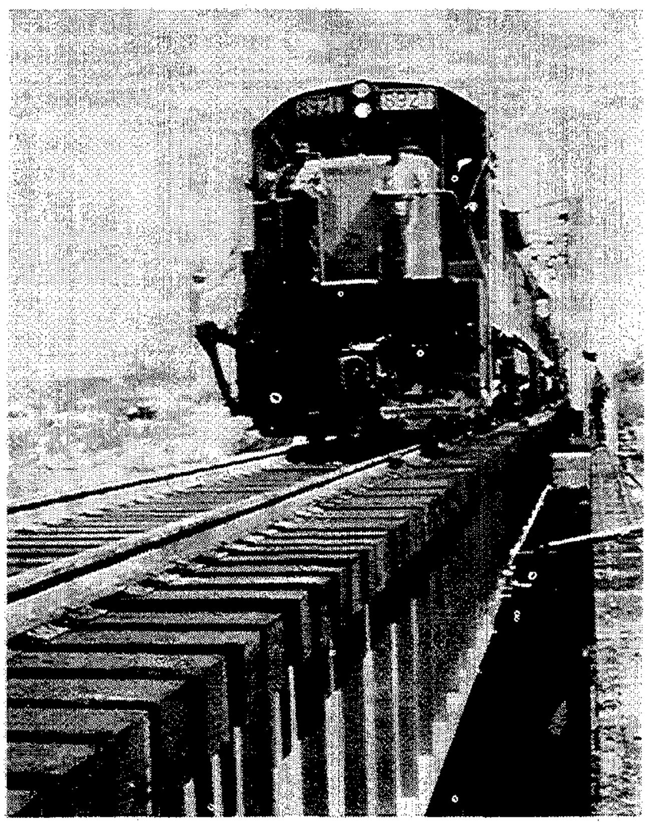

Fig. 29. Lamadrid trestle.

DISCUSSION LEADER
Dr. William J. Harris, Jr.
Vice President--Research and Test Department
Association of American Railroads

Dr. William J. Harris, Jr., is Vice President of the Research and Test Department of the Association of American Railroads, Washington, D.C. He received both B.S. and M.S. degrees in Engineering from Purdue University in 1940 and was awarded the degree of Doctor of Science in Metallurgy by the Massachusetts Institute of Technology in 1948.

Harris filled staff positions in various agencies before being appointed Assistant Executive Secretary, Planning, of the Division of Engineering of the National Academy of Sciences--National Research Council in 1960. Later he became Assistant Director of the Columbus Laboratories and Head of the Washington Office for the Battelle Memorial Institute. He assumed his position with the AAR in 1970.

Harris is the author of about 40 technical papers and co-editor of Perspectives in Materials Research. He has been an active member of the professional community, serving as president or chairman of a number of associations, boards, and advisory committees.

The issue that has been under discussion in this conference has represented three or four critical and important elements of the problem of economic tradeoffs that affect the relationship between heavy cars and rails. In the first place, it is patently clear from the experimental, analytical, and technical work, that we are beginning to understand something about the mechanism of failure of components. But I respectfully submit that in the discussion this morning when we talked about the failure of components, we were talking about the wear and fatigue of metal parts and were alluding indirectly to the fracture of metal parts. We were not talking about the gradual loss of capacity of spike holding by wooden ties. We weren't talking about the loss of capacity of other tie holddown devices on concrete ties. We were not talking about the question of deterioration of ballast. We were not talking about the mechanisms by which stresses move from the railhead into all the elements of track structure. Therefore, relevant as the sessions were this morning, they were not comprehensive in dealing with all of the processes or mechanisms that are critical to an understanding of the cost elements of track deterioration as traffic passes over it.

Now, those issues are matters of concern to many of you, to the railroads in general, and to research programs that FRA and AAR are pursuing. Nonetheless, when we talk about the relationship between equipment and track, we cannot be satisfied with talking only about the metal components in those systems. The progress we are making is nevertheless significant and will lead to much more quantitative insight in the future.

The very useful presentation by Mr. Selzer should lead us to recognize the critical importance of the kind of engineering we were all taught--that is, engineering with economic value associated with it. The inability in this industry to take into account a whole sequence of events causes a great many difficulties that this Conference has addressed in previous years. We do not really understand what the value of improved trucks is, so we do not buy them. We do not really understand what the true advantage of heat-treated rail is over ordinary rail or non-heat-treated rail, so we have a hard time justifying incremental cost increases for these products. If we don't have the cost figures in these somewhat simpler cases, how much more difficult it is to look at the economic factors in two sets of systems, the car-train system and the track system as a whole.

There is one matter that I think Mr. Selzer might wish to take into account in further discussion and consideration of the economic factors in the use of "jumbo" cars. This is the fact that the heavy car per se is important, but we should not forget the kind of dynamic input you get from a light car with "hunting" trucks. In the work that we have done in track/train dynamics and on many of your railroads,the derailment tendencies in light cars under truck hunting conditions are serious; they cause more difficulty in some respects than the loaded heavy car movement. I think that dynamic input has to be recognized as a critical element. Considering the change in average car weight from 46.3 tons to 71.5 tons, it must be recognized that use of the lighter cars would mean that we would have to have trains about 60 or 70% longer. We also know from track/train dynamics and other studies that longer and longer trains also introduce a series of dynamic elements in the equation. You can't just look at a lighter car and a heavier car. You have to look at the whole system.

I have tried to make commitments in our research program to resolve some of these issues. For two years we have had on our docket a commitment to programs on the so-called heavy car problem. It has been difficult to find the right people that we need to get on with that job. We have not made the progress that I hoped we might have made by today. I'm sure all of us will be continuing to look at and analyze in detail the issues and the problems that the heavy car does relate to, but the heavy car can't be looked at out of context.

The floor is now open for discussion of the very excellent papers that were presented this morning.

COMMENTS/DISCUSSION PERIOD

Delegate Comment: Have you considered the effects of different profiles of wheels on the same axle in creating a stability situation within a truck?

Panel Response: Our field data indicates that in most cases effective wheel tapers of two wheels on an axle are very similar; however, wheel tapers differ from axle to axle. To simulate actual operating conditions, we have incorporated in the program variations of axle loads, wheel-diameters, and effective wheel taper for each axle.

Delegate Comment: Was this on the same axle, or was this on an adjacent axle?

Panel Response: On different axles.

Delegate Comment: How about on the same axle?

Panel Response: On the same axles. We took the same profile. We have not looked at different profiles of wheels on the same axle.

Delegate Comment: I am from the Transportation Industry Marketing Department of the IBM Company. Ed Ward asked me to make a few very brief observations about the aplication of the computer system to the analytical work you have been discussing. I would like to do that, and I shall keep the remarks very brief.

Approximately two years ago our group began a functional definition of computer-assisted systems which could be applicable to the railroad maintenance-of-way department and the interface, if any, which would be desirable between the maintenance-of-way department's own system and the data files the railroad's host or central computer system. Believing that the wheel should never be reinvented, our initial work was done with two selected railroads, both of which have made significant accomplishments in applying the information-processing capabilities of commercial computers to the work of their maintenance-of-way departments. Further, through conducting seminars for way officials, both this year and last, which largely featured presentations from the railroads themselves on maintenance of rail-related systems which they had developed, other good work in this area was documented and made available to the industry.

In should be pointed out that while there are quite a few productive computer applications existing today in the engineering and way areas, some of which we have heard about at this Conference, these efforts appear to be fragmented. Perhaps only one road is on the threshold of closing the loops on maintenance-of-way and engineering computer systems. That effort concerns measuring track conditions, putting this data into the computer, analyzing and grading the track defects which are present, and using this information to prioritize schedules and monitor maintenance work. It is interesting to note that the maintenance-of-ways systems currently operational which are considered to be the most productive in analyzing track conditions are those systems that use (that is, interface with) the already existing data in their roads' own central computer systems. These systems tend to verify the results of our own work, which also indicate the need for establishing a data base of maintenance-of-way information which should, for maximum productivity, be augmented by other data, namely transportation data already available in the central system.

With this background, I am at the point where I would like to suggest to this audience the desirability of using data already existing in your railroad central computer system while conducting your own analytical work of track and roadway conditions. The theme of this Conference, I would like to note again, is the effect of heavy axle loads on tracks. Although, as we have heard here, there are many ramifications to this subject, I propose that a complete methodology has yet to be offered which can fully evaluate or answer the questions implied by our Conference theme. This is particularly true of the cost analysis or cost tradeoffs which are so desperately needed by rail management in this area. Yet the sufficient track measuring technology and causal data is probably there, waiting to be used in a valid correlated manner. The challenge, I think, is still before us.

The work currently being done on the Bessemer Lake Erie, the Rio Grande, the Chessie, and, as we have heard, on the Illinois Central Gulf may have contributed some of the best answers so far. Another road has also made substantial progress toward being able to measure track conditions and then to analyze the conditions; that is, to correlate the condition with causal factors such as specific maintenance activities previously applied or omitted, traffic densities, and frequency of the passage of special car types over roadway segments, notably of the 90-ton plus variety. Once these correlations are satisfactorily identified, then, of course, maintenance and transportation policies can be adjusted accordingly to improve the cost effectiveness of maintenance-of-way expenditures on the railroad.

It appears then that much is still left to be done to apply the capabilities of the computer to analyzing track and raodway conditions. I am suggesting that you consider the use of your railroad computer data base as a powerful resource to further your analytical work. I would also solicit any comments, now or after this session, which you may have concerning this utilization of systemwide data as an analytical resource.

Delegate Comment: I noted that in your discussion of the loading factors taken into consideration in bridge design, there did not seem to be any figure given for lateral live load on bridges. In connection with Dr. Harris' comment on the hunting of cars, some recent measurements taken on Canadian National on a roughly 90-foot open deck girder bridge indicated significant lateral loading on the bridge structure, particularly in the crossbridging, on the passage of empty cars moving at roughly 40 mph. It's probably the experience of many railways that most of their bridge structures are suitable for main line speeds of 60 mph. It has become evident recently, and throughout this Conference, that many of the cars we operate today under empty conditions undergo hunting oscillation at speeds 40 mph and up. I wonder if this is now being considered in the fatigue loading on some of the existing bridge structures.

Panel Response: I was referring to design of new bridges when I mentioned other lateral forces. There are three kinds of lateral forces that are acting on a bridge: The wind on the loaded bridge, the wind on the unloaded bridge, and forces from equipment. The AREA specifications, Chapter 15, which cover steel structures specify a force of 20,000 lbs. to be applied at the base of rail as the lateral effect in designing of a new bridge. Section 7 of Chapter 15 deals with the rating of existing structures and currently is under revision. There has been a lot of discussion as to how much of a lateral force you should consider in the rating of an old bridge. Unfortunately, in my opinion, too many railroads are inclined to use the rating allowables and consider only the live load and dead load forces and impact forces in deciding what a bridge can carry. Too many railroads are ignoring the lateral effects and braking, and, as these structures get older, I think they will be experiencing serious problems.

Delegate Comment: With regard to your investigation of rail head flow, has the Canadian National attempted to do any field investigations of the rate of rail head flow? Specifically, have they tried to investigate what types of wheel passage are most responsible for damage to the flow? How often are you actually getting the travel on the outer tread of the wheel that you show in your figures?

Panel Response: We have not made any measurements of rate of rail head wear. However, we have made extensive analysis of rail replacement rates based on specific defects such as head flow, curve wear, etc. and we can relate these replacements to specific curves. Some curves have required rail replacements due to head flow in one and one-half to three years.

The rate of wear depends on the percentage of heavy vehicles riding over the track. In 1967 this percentage was about 10%, it is now around 30%. By that I mean 30% of the gross annual tonnage is in fully loaded 100-ton cars. It is a changing picture, but at present we are obtaining a rail life of about three years or 90 million gross tons. I do not know how often the outer tread of the wheel travels on the top of the rail.

Delegate Comment: Could you discuss the mechanics of using your theory of the transfer of corrugations on curves from one rail to another?

Panel Response: I think it is due to the force being transmitted along the axle to the other wheel. On our own lines, corrugations start on the low rail and in certain cases they do progress to the high rail. However, on some roads the picture is reversed; the corrugations may start on the high rail and then progress to the low rail.

This is why I must admit that I do not know the exact mechanism by which these corrugations start. It is probably a vibration mechanism but other things are certainly involved. Flange oilers, for example, make a significant difference in the amount of corrugation which occurs.

Delegate Comment: Recognizing that the complexities of stability grow very fast, nevertheless, have you examined the case of locomotives in tandem, let's say two or three locomotives coupled together, as they are frequently operated? If not, I would like to solicit your comments. Would you expect much coupling through the locomotive couplers?

Panel Response: I agree, that when two or three locomotives are coupled together into one consist, there will be some coupling effect. Recent work of Dr. Blader at Queen's University in Canada has indicated that the coupling effect is very small and it does not change the critical speed of the vehicle.

Delegate Comment: The public at large has become increasingly concerned in recent years with long-term effects; the whole ecology-environment business is a concern for the future. When we talk about a short-run decision making in the railroad industry, which is really a do-it-now-and-to-hell-with-tomorrow sort of view, is this not potentially a very difficult public relations point to put over, and politically a very dangerous or difficult one?

Panel Response: I think the best way to start to answer that question is to quote John Maynard Keynes, who said in the long run we are all dead, implying that the short run is when decision making has to take place. Now, whether it is politically wise to admit or even mention the fact that people make short-run decisions, I think is irrelevant. I think that people tend to discount the future. Some people tend to discount it more than others, but you cannot say that the future is as important as the present.

Delegate Comment: I'd like to comment briefly on that question because it's a matter of great concern to those of us who are watching national policies emerge in regard to the railroad situation. I personally am convinced that the technology created by the railroads in the latter part of the past century gave them such a technological advantage that they could, without further serious analysis, effectively proceed to be a totally viable economical entity until the interstate system in the United States was completed. The short-term decisions made following that, to go to longer trains and heavier cars and so on and so on, were done in the very best interests of becoming competitive.

Faced with a current 1975 circumstance in which our rates are depressed by $4.5 billion in order to accommodate the subsidies to other modes of transportation, we find ourselves trapped by the consequences of a sequence of decisions that were not thought to have long-term negative consequences. However, we are not able to recover from the results of those decisions because we don't have $4.5 billion per year to take the corrective measures that ordinarily we could in relatively freer economic circumstances. So the problem that we are really addressing in this conference has to be looked at, I think, in terms of a much broader set of economic competitive forces which deny us resources. Under ordinary simplistic competitive circumstances, these forces would not exist.

Delegate Comment: I have a question about corrugations--are the corrugations initiated by flow or wear phenomena? Second, do you ever see corrugations reflected in the wheel treads?

Panel Response: The initiation of corrugation is a very complex phenomenon and in particular circumstances may be due to different causes. My contention is that if head flow of the rail could be eliminated, serious corrugation problems would not occur. This remark does not aply to very short wavelength corrugations which are another problem.

Delegate Comment: What is the wavelength of the corrugations that you are finding?

Panel Response: On the same rail length of about 70 feet, we have found corrugations ranging from 8 to 30 inches. I am unable to reconcile this with a vibration phenomenon although there may be others who can. I believe corrugations such as we experience are related to tie founding although this relationship requires further study. The causes of corrugation may be different in differing circumstances, but they can probably be avoided by eliminating head flow. All the rail which I examined was invariably flowed as well as corrugated.

Delegate Comment: In a fully developed situation of corrugation, the bulk of the unevenness is related to wear rather than flow, although the initiation is flow?

Panel Response: My colleagues from the Canadian Pacific insist that wear is quite important; I don't think it is. Again, their corrugation, to me, is slightly different from ours; their track is certainly different, their loadings are somewhat different (they have heavier loading), they have different speeds, and they have different maintenance practices. Wear could be a factor; I just don't know.

Delegate Comment: Under the assumptions of the linear model and small amplitude oscillation, how do you deal with the free play that you get in roller bearings? I am thinking of the Hyatt bearing design and the free play that exists in the gib clearance. Do you think it has a large importance in the critical speeds, and will it affect your validation tests?

Panel Response: Are we talking about the freight car truck?

Delegate Comment: The locomotive truck. There is a free clearance in the roller bearing itself and there is a clearance also in the gib in the bearing box.

Panel Response: First, the linear model does not allow for an evaluation of free play or clearances, but our field tests have shown a significant influence of clearances between the wheel-axle assembly and the truck frame. It appears that liberal lateral clearances and minimal longitudinal clearances are desirable for the higher critical speed.

Delegate Comment: Could you comment on how close your validation tests were with the locomotive lateral stability models?

Panel Response: In most cases results were within 10 to 15%.

Delegate Comment: Under high tractive effort you essentially lock out the secondary suspension of the locomotive truck. Do you take that into consideration in your model?

Panel Response: Yes.

Delegate Comment: Could you comment further on your own experience with higher strength rails and what plans you have in the future to look at this more extensively?

Panel Response: We have not had a great deal of experience with premium rail, although we have done some field testing. Field tests are difficult to control. The variables could be more closely controlled if a circular test were used to test rail wear. One of the varables I would like to see controlled would be the use of flange oilers. The effect of speed could also be evaluated.

The use of premium rail will obviously provide some relief for head flow and corrugation in rails because of its higher yield strength. However, this is only one of a number of remedial actions which should be taken. In Canada, with the suport of the Transportation Development Agency, we are initiating some tests on alloy rail. We hope to improve the situation considerably by the use of permium rails, whether heat treated or alloyed.

Delegate Comment: Was any consideration given to adding additional degrees of freedom in your apparatus, and if so, what difficulty do you encounter, other than budgetary?

Panel Response: I'll take that in two parts. There have been considerations for additional degrees of freedom, and those are currently planned in the future expansion of that system. What's going to be incorporated is another complement for a rear truck, again with the same operating mode that currently exists, and the introduction of lateral motion at the wheel-rail contact point. Now, economics certainly have been a factor in the evolution of that, but there are no serious technical problems in implementing the hardware. In particular the most difficult device will be a three direction of freedom bearing which must reside on top of the platform that you have seen before to support the wheel and allow the truck to yaw and impart the lateral translation in the wheel set. But, that design at this point is, I would say, well in hand, the budgets are appropriate, and we don't see any major difficulties in implementing that hardware in the next 12 months.

Delegate Comment: Have you also checked the effect of the transverse coupling between the trucks of the locomotive, in regard to the reduction of the lateral forces to the rail?

Panel Response: I presume that you mean, transverse coupling between the trucks is made through the carbody at the center plates.

In the model, two trucks have been coupled through the carbody and the interaction between them has been considered.

I have an additional comment for the delegate who asked about gib clearance, which we call the pedestal clearance. Getting away from a freight car truck, things that pertain to the freight car truck stability are quite different for a rigid locomotive truck with a primary suspension. In a locomotive, longitudinal clearance in the pedestal region has significant effect on critical hunting speed. A large clearance with high wheel taper is desirable for curving, but is very disadvantageous for lateral stability. This allows a wheel-axle set the freedom to go into a yaw mode of oscillation.

As you probably know, under track/train dynamics with FRA funding there will be characterization of a much broader range of trucks than heretofore. So we will be getting some numbers that will be quite valuable and useful in the math model development.

Delegate Comment: You seemed to approach and get right to the threshold of the question about where do we go from here, but you didn't proceed into the subject. It seems to me that there will be a compounding of these problems in the future due to economics and cross tradeoffs and the competitive forces at work. I think that we should close the conference with some resolute determination that these problems are all solveable within the technology available right in this room and that we should lay a format for the future. We see on the horizon the need for a longer car. The 45-ft. trailer is already a reality, in certain states, certainly in Canada where they run trailers and in the West, where they run full trailers and semitrailers pulled by large rigs. Sooner or later the trailer train and tractor will become a reality on American railroads. These are problems that must be solved with sound engineering principles, but at the same time we feel that the future is going to generate an additional set of problems. So we had better be on our way to get ready for these.

Panel Response: I couldn't agree more with your conclusion. It seems to me, however, that there are issues that we must deal with potentially through legislative relief, which will cause some rebalancing of the support for various modes of transportation. I do not believe there is a sufficient degree of freedom either in technology or management for restructuring of this industry within itself to solve all of the problems that we have. But we must make the most aggressive effort that we can, collectively, to do what we can for ourselves, simultaneously pursuing, external to the industry, those changes that must be made elsewhere so as to give us an opportunity for relief.

Bob Parsons and I are dedicated to the principle that in all appropriate ways we will merge our resources and try mutually to expand them and to draw on all of your talents in looking toward the resolution of these matters. I am sure I am only saying what Bob said earlier in his remarks to you.

If there are no more questions, I hope you find this to be an appropriate final commentary for this conference, and I appreciate your making it. I would like to add just a small personal note. I join with the others who have spoken of Jack Loftis; he was my friend and counsellor and a very important consultant to us. We will miss him.

I would like to extend my deep appreciation to the FRA for having organized and made possible this Conference and to Ed Ward in particular for his superb work in planning and executing it. Thank you very much.

SESSION IV

TRANSPORTATION TEST CENTER

This session of the 12th annual Railroad Engineering Conference was held at the Transportation Test Center.

Mr. Jack B. Stauffer, Center Director, provided the conference delegates with an "Overview of Center Activities," and Mr. Ross Gill, Center Staff Engineer, briefed delegates as to railroad test projects being conducted.

Delegates then toured the facilities of the Test Center which included viewing a Test of a Trailer Train car on the vertical shaker in the Rail Dynamics Laboratory, inspection of the new Center Services Building where locomotive and car maintenance and modifications are performed, and visits to the ASF Test Train, Dresser Transportation Equipment's test car and the Standard Light Rail Vehicle.

U.S. DEPARTMENT OF TRANSPORTATION
FEDERAL RAILROAD ADMINISTRATION
TWELFTH ANNUAL RAILROAD ENGINEERING CONFERENCE

LIST OF DELEGATES IN ATTENDANCE

Abbott, R. E.	Office of Safety	FRA
Adler, F. P.	V.P.-Eng.	Pullman Transportation Leasing Co.
Ahmed, N.	Office of R.&D.	FRA
Albenese, D.	Mgr.-Tech. Center	Nat'l. Castings Div., Midland-Ross
Allen, J. W.	V.P.-Eng. R.&D.	Stanray Corp.
Alonzo, J.	Eng.-Track Research	Spanish National Railways
Anderson, B. G.	Asst. V.P.-Eng.	Burlington-Northern
Atkins, P.		International Mill Service
Autrey, W. S.	Chief Eng.-System	Atchison, Topeka & Santa Fe Rwy. Co.
Baier, J. A.	Transp. Eng.	P.U.C. of Colorado
Bailey, R. W.	M-O-W Planning	Chicago & North Western Trans. Co.
Barnes, J. A.	Asst. V.P.-Ch. Eng.	Chicago & North Western Trans. Co.
Beach, G.	Reg'l. Eng. Mgr.	Canadian Transport Commission
Beck, R. F.	Chief Eng.	Elgin, Joliet & Eastern Rwy.
Berg, N. A.	Mgr.-Tech. Serv. Sales	Griffin Wheel Co.
Bexon, H. J.	Mgr.-Prod. Dev. Dept.	DOFASCO
Billingsley, R.	Dir.-Eng.	AMCAR Div., A.C.F. Industries, Inc.
Blakeman, E. C.	V.P.-Sales	American Steel Foundries
Bossong, J. E.	V.P.-Ops.	Griffin Wheel Co.
Bramwell, S.	Mgr.	Quebec, Labrador, North Shore Rwy.
Bray, R. E.	Supt.-Car Dept.	Elgin, Joliet & Eastern Rwy.
Bridge, P. H.	Mech. & Elec. Eng.	British Columbia Railway Co.
Brodeur, R.	Dir.-Eng. & Research	Trailer Train Co.
Brosnan, D. W.	Consultant	Southern Railway Co.
Brown, R. M.	Chief Eng.	Union Pacific R.R.
Brown, T. R.	Dep. Task Mgr. TTD Proj.	AAR Technical Center
Bullock, R. L.	Sr. Proj. Eng.	Standard Car Truck Co.
Byrne, R.	Mgr.-Research	Southern Pacific Transportation Co.
Caldwell, N.	Sr. Res. Eng.	Canadian National Railway Co.
Campbell, D.	President	Cardwell-Westinghouse
Cantey, W. E.	Actg. Assoc. Admin. Policy & Prog. Dev.	FRA
Charles, R. J.	Res. Off. Transit Sys. R&D Br.	Ontario Ministry of Trans. & Com.
Christian, Jr., F.	Mech. Eng.	Chessie System
Cirillo, R. R.	Coord. of Rail Eng.	Texasgulf, Inc.
Claasen, H. R.	Sales Eng.	Tamper
Comiskey, C.	Chief Eng.	Procor, Ltd.
Cope, G. W.	Dir.-Eng.	Dresser, T.E.D., Dresser Ind., Inc.
Cortes, A. D.	Asst. Gen. Mgr. M.P. & R.S.	National Railways of Mexico
Crane, L. S.	Exec. V.P. Ops.	Southern Railway
Cruz, G. F.	Supt.-Mech. Dept.	R.F.F.S.A., Brazil
Cunningham, W. C.	Pres.	Alco Spring Industries, Inc.
Cyr, W. H.	Chief-M.P. & C.E.	Canadian National Rwy. Co.
Danahy, F. A.	Exec. Dir.-Mech. Div.	AAR
Dancu, T. E.	Dir.-Eng.	Thrall Car Manufacturing Co.

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Danielson, V. S.	V.P.-Mech. Ops.	Cardwell-Westinghouse
Davis, L. D.	V.P.-Eng.	American Steel Foundries
Day, J. C.	V.P.-Marketing	American Steel Foundries
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Demuth, H. P.	Deputy Div. Mgr.	ENSCO
dos Santos, G.		R.F.F.S.A., Brazil
Downes, J. R.	Dir.-Eng. Mass Transit	Buckeye Steel Castings
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Ellsworth, K.	Mechanical Editor	Railway Age Magazine
Eshelman, L. L.	Associate	A.T. Kearny, Inc.
Evans, R. A.	Proj. Dir.-R.R. Truck Safety Res.	AAR Technical Center
Everly, J. D.	Pres., Gen. Mgr.	West Virginia Northern R.R. Co.
Fay, G.	Office of R&D	FRA
Ferris, E. W.		Grumman
Fillion, SH. H.	Consultant	Dresser T.E.D., Dresser Ind., Inc.
Fleming, S. E.	V.P.	Brotherhood of Maintenance of Way Employees
Flohr, B.	Dep. Fed. R.R. Administrator	FRA
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Freeman, L. D.	Exec. V.P.	Speno Railroad Ballast Cleaning Co., Inc.
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Garg, V. K.	Sr. Proj. Eng.	Electro-Motive Div., GMC
Garin, P. V.	Asst. V.P.-Res.	Southern Pacific Transportation Co.
Gates, R. E.	Sr. Rail Eg. Eng.	Shell Oil
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Irwin, H. D.	Sr. Mech. Eng.,-Trans. Eq. Div.	Du Pont De Nemours & Co., E.I.

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O'Sullivan, W. B.	Track Eng.	FRA
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Selzer, L.	Industry Analyst	FRA
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Smith, N. E.	Asst. Ch. Eng.-Maint.	Chicago, Milwaukee, St. Paul & Pacific R.R.
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Staten, L.	Test Controller-Rail Systems	Transportation Test Center
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Strutynski, M.	Ch. Eng. R.R.	Pacific Car & Foundry Co.

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Taylor, T. M.	Consultant	
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Tennikait, H. G.	Mgr.-Test Eng.	American Steel Foundries
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Ward, E. J.	Consultant	Seaboard Coast Line R.R. Co.
Ward, J. T.	Sr. Asst. Ch. Eng.	St. Louis-San Francisco Rwy. Co.
Warfel, G. E.	Chief Eng.	Denver & Rio Grande Western R.R. Co.
Waring, E. H.	Chief Eng.	Southern Railway Company
Webb, C. E.	Asst. V.P.-Eng. & Res.	
Weinstock, H.	Mech. Eng.-Office of Eng.	Transportation Systems Center
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Wiebe, D.	Mgr.-Res. & Eng.	A. Stucki Co.
Wilhelm, J.	Materials Scientist	Standord Research Institute
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Worley, B. J.	V.P.-Ch. Eng.	Chicago, Milwaukee, St. Paul & Pacific R.R. Co.
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Yaloutes, P.	Service Eng.	Scullin Steel Company
Yang, T. L.	Sr. Group Eng.	ENSCO
Zakrzewski, A. S.	Adv. Eng. Group-Prod. Div. Eng.	DOFASCO
Zirkle, W. D.	Adv. Ind. Market Rep.-Trans.	I.B.M.


Ed Ward
J. Herbert Lund
Don Wilhite
J.C. Day


W.F. Greenwood
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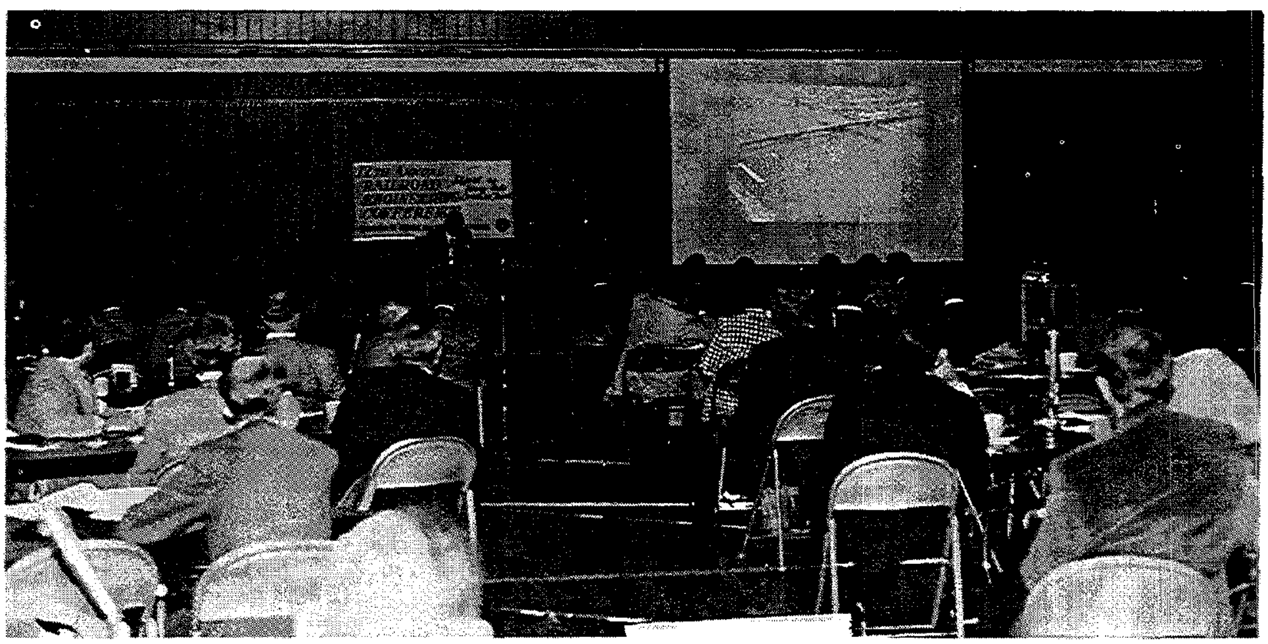

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