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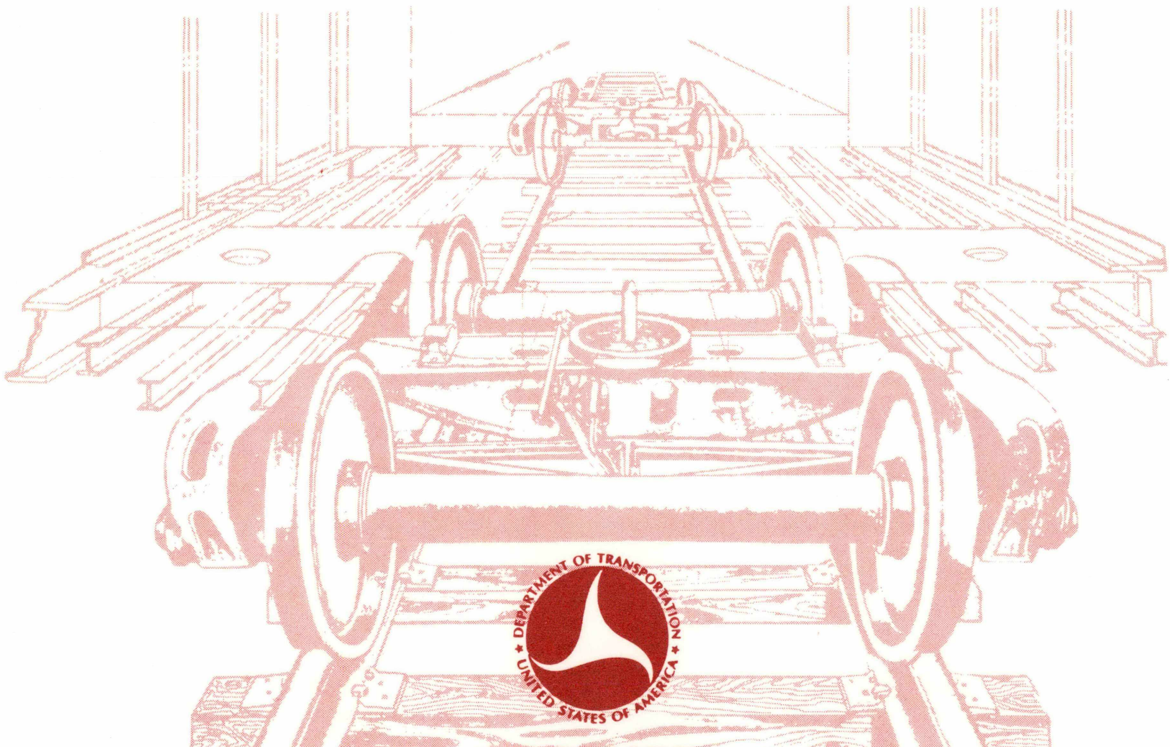
# FREIGHT CAR TRUCK DESIGN OPTIMIZATION

## LITERATURE SEARCH - VOLUME II


Southern Pacific Transportation Company
Technical Research and Development Group

JULY 1975

INTERIM REPORT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This interim report represents the second of a three-volume set. Volumes I and III bear the same report number with A and C suffixes respectively.

16. Abstract


Volume II of the TDOP Literature Search contains the sections entitled: "Truck Components" and "Track-Train Dynamics As Related To Truck Performance." Each of the two sections contains:

- An introduction dealing with literature selected for reprinting
- Reprints of articles judged particularly representative or salient
- A bibliography alphabetized by author

The "Bibliography--Truck Components" is further organized into the following subsections:

- Brakes and Brake Rigging
- Centerplates
- Side Frames and Bolsters
- Snubbers and Dampers
- Springs
- Wheels, Axles, and Roller Bearings
- Miscellaneous Component Systems


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CONTENTS

Volume II

## Section 3

## TRUCK COMPONENTS

### INTRODUCTION

The following subsections are comprised of reprinted articles, which were selected from the assembled literature concerning truck components, and the TDOP truck components bibliography. The reports selected for reprinting are: "Improved Freight Car Running Gear - A Necessity For The 70's," by J. Angold; "Effect Of Design Variation On Service Stresses in Railroad Wheels," by J. P. Bruner, R. D. Jones, S. Levy, and J. M. Wandrisco; "Brake Rigging Efficiency Of Railroad Freight Cars," by R. W. Carman; "Development Of Fatigue Standards For Freight Car Truck Components And Wheels," by M. R. Johnson; "A Field-Service Evaluation Of Various Centerplate Lubricants And Liners," by W. E. Lasky and M. A. Hanson; "Pullman's Approach To Center Plate Problems," by G. L. Rousseau; "Outer Pedestal Legs Of Narrow Pedestal Side Frames," by C. E. Tack; and "Investigation Of The Thermal Capacity Of Railroad Wheels Using COBRA Brake Shoes," by G. R. Weaver, P. A. Archibald, E. B. Brenneman, and G. M. Cabble.

The bibliography, which follows the reprints, is divided into the following subject groupings:

- Brakes and Brake Rigging
- Centerplates
- Side Frames and Bolsters
- Snubbers and Dampers
- Springs
- Wheels, Axles, and Roller Bearings
- Miscellaneous Component Systems

References under each subsection are listed alphabetically by author. Supplemental pages will be added to the "Bibliography--Truck Components," as new articles become available throughout the course of the TDOP.

**JOHN ANGOLD**, Director Technical Research and Development,
The Atchison, Topeka and Santa Fe Railway Company

*John Angold, after graduation from Kansas State University in 1938 joined the Atchison, Topeka and Santa Fe Railway as Assistant Detector Car Operator. He was appointed Assistant Engineer of Tests in 1946, Engineer of Tests in 1959, and in 1965 to his present position of Director, Technical Research and Development.*

*He attended the Santa Fe's Institute of Business Economics at the University of Southern California. He is past chairman of the National Association of Engineers of Tests, has served on the AAR Committee on Specifications for Materials and on the Coupler and Draft Gear Committee, and is a member of the American Society for Testing and Materials.*

# "Improved Freight Car Running Gear - A Necessity For The 70's"

Thank you, Loren. It is good to be back again this year. Last year was the first time I had attended this conference and my colleagues back in Topeka accused me of going to any length in order to return, even if it meant the preparation of a paper.

Seriously though, I was quite pleased when Jack Loftis asked if I would present a paper, since he was quite generous in allowing me to choose the subject, and since it would provide an opportunity to advance my views on one of my favorite subjects—the need for better freight car running gear and suspensions. Several of the points I wish to make have been pretty well covered here in the papers we have already heard, but I feel the need for improved running gear is so acute that the story needs to be told again and again in order to initiate development in this important area.

I suppose there would be as many answers as there were respondents in answer to the question of the area of the greatest need of the railroad industry today, but, I have long felt that improved freight car running gear and suspensions represents one of our most pressing needs. I am afraid I have made quite a nuisance of myself at times by needling supplier representatives in an endeavor to get them to carry the message back home that a serious need exists and that the field is wide open.

Some may ask what is wrong with our present trucks, and I'll ask you what is right with the truck whose lateral and vertical motion approaches its forward motion? That may be a little exaggerated, but if you don't believe there is a lot of substance to it, just go out and take a few rides in freight cars operating in high-speed service and you will have no difficulty in recognizing the conditions to which I refer. I have done this many times and it can be most revealing. You don't have to be too observant to detect the cause for many of our maintenance woes and worse yet, the reason why some shippers refuse to use our services.

My interest in running gear became acute about ten years ago when we were developing a freight car cushioning system, and I was keeping tab on handling of the prototype car provided for shipments of easily damaged lading, through the use of the impact register. The impact register on one shipment was not the usual longitudinal - only type, but had a stylus indicating vertical action as well. In examining this record, I was very surprised to see the almost continuous heavy vertical action which lasted for considerable periods of time. A photograph of such an impact register tape is shown in Figure 1. A study of the territories involved showed this phenomenon to occur in areas where higher speed operation would be encountered. My first deduction was that the truck springs were excited at their resonant frequency above a certain train speed, and that this was the cause of the severe vertical action shown on the impact register tape.

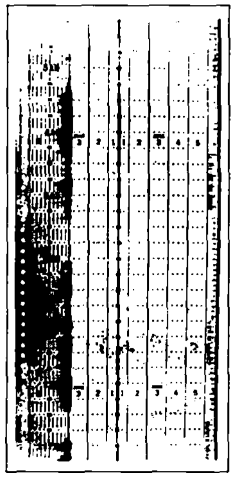

**Fig. 1. Section of impact register tape showing periods of continuous vertical disturbance.**

Reprinted from Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, Dresser Industries, Inc., 2 Main Street, Depew, New York 14043.

Further investigation proved this assumption to be in error. The continuous vertical action was actually a resultant of the severe unstable lateral action of the truck. This action, causing the trucks to strike back and forth from one rail to the other, has a pitching action component in the vertical direction, which causes a large part of the disturbance noted on the vertical impact register tape.

Remembering this disturbance noted on the impact register tapes, my next encounter with this action was in tests on automobile carrying rack cars, in which failure of the vertical rack posts was occurring after only a relatively short amount of service. These tests were performed with a better grade of instrumentation, and it was possible to secure results to which a number value could be related. It was soon determined that the rack post stresses were caused by a racking of the car structure induced by the severe unstable lateral action of the trucks. Figure 2 shows an oscillograph record of these stresses in which the direction of travel is from the left to right. You will note that the stresses are practically nil until the critical speed is reached, at which time the severe lateral action is initiated which induces the stresses in the posts. This action started at approximately 48 miles an hour in this particular car and continued up to the maximum test speed. With the frequency of this disturbance being approximately 2½ cycles a second, it is little wonder that the rack posts were failing. The more serious concern, of course, is the effect this disturbance has upon our customers' lading. In short test trains, where train slack action was not present, I have seen 3,000 pound pallets moved eighteen inches transversely in a car by this action, in less than 100 miles of operation.

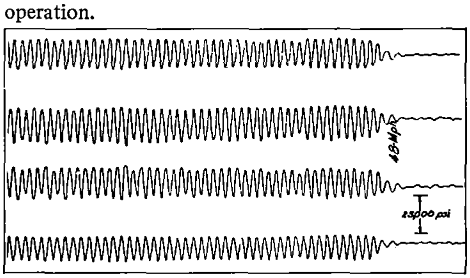

**Fig. 2 Oscillograph record of stresses induced in posts of automobile carrying rack when unstable lateral action of the trucks occurred.**

Previous to our tests, I had supposed this unstable lateral action was the result of the entire truck turning as a unit from side to side about a vertical axis through the center pin, as indicated in Figure 3. Observations made in the autoveyor tests however, indicated this was not the case, but that the action is as shown in Figure 4, wherein the side frames move longitudinally relative to each other, inducing a see-saw action of the truck bolster with the male body center plate acting as the fulcrum. Further instrumented tests in which we participated, employing displacement transducers and TV equipment, have corroborated these earlier observations.

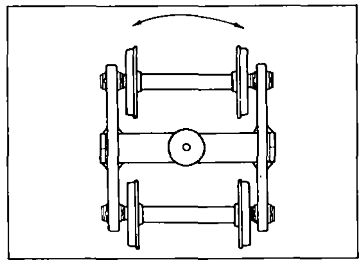

**Fig. 3. Severe truck hunting first thought to be a swiveling action around axis through center pin, but later tests indicated it to be as shown in Fig. 4.**

This action causes the trucks to change tram or "parallelogram", resulting in the axles being alternately realigned in rapid succession so that the wheels strike back and forth between the rails at a rapid rate. The frequency of this action is around 2½ cps. The speed at which it is first noted seems to be affected by several factors, such as the rigidity of the construction of the car, the gross weight of the car, the condition of the wheel treads, and the atmospheric conditions. I have seen it start at speeds as low as 38 mph on a long empty flat car with worn wheels, and if there is an upper speed at which it ceases to exist, it was not found in the speed range of our tests which ranged to well over 90 mph.

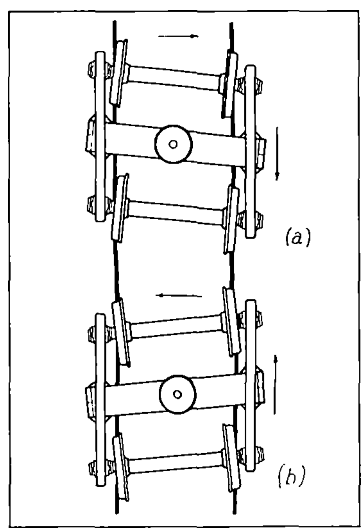


**Fig. 4. Parallelogramming of trucks actually cause of severe lateral instability action.**

When the car wheels are striking back and forth from one rail to the other at this rate, the action is quite violent. An understanding of this action provides an explanation for high wear rates on several of the car components. Have you ever wondered why there was more wear fore and aft on a center plate than there is transversely? I could never rationalize this occurence until I became better acquainted with the motions associated with the lateral instability action of the trucks, since measured transverse accelerations on a freight car are considerably greater in frequency than are longitudinal accelerations. However, the motion as indicated in Figure 4 leaves little doubt as to why the wear areas are fore and aft, and the frequency of this action explains why the wear is considerable.

This truck action also explains why the wear rate is high on roller bearing adapters. With the angularity of the axles with respect to the side frames continually changing when this unstable action occurs, the top of the adapter is subjected to a continual grinding action against the roof of the pedestal area. In addition, with the pounding back and forth from side to side, wear is increased on the inner side of the adapter against the roller bearing outer race.

Flange wear is also accelerated as the wheels strike alternately from rail to rail in this rapid synchronous action. In fact, the observer can generally tell when this unstable action is initiated by the "pinging" noise which is associated with the flange striking the rail sharply.

Another area of wear associated with this action is in the coupler shank and coupler carrier. The car body moves from side to side as indicated in Figure 5, as the wheels strike from one rail to the other, moving the coupler carrier laterally with respect to the coupler shank which is held in the line of draft.

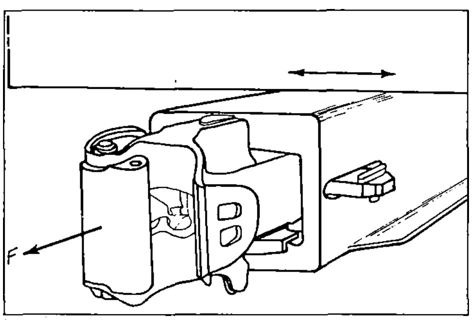


**Fig. 5. Synchronous lateral movement of the car body resulting from truck instability, accelerates coupler shank and coupler carrier wear.**

Along about now, you're probably asking, "What's new about wear on the center plates, roller bearing adapters, wheel flanges, coupler shanks and coupler carriers – we've had that for a long time?" There is nothing new about them, but with our higher train speeds of today we now operate a large percentage of the time in the speed range where the wear in these areas is greatly accelerated because of this rapid synchronous unstable action of the trucks, and these factors take on added significance. I believe we have, for the most part, just accepted these higher maintenance items as a matter of course and have done very little to investigate why they occur and what might be done to minimize their effect. As a result, few people are actually aware of this action in the trucks which I have described.

With the industry's profit picture what it is today, we can no longer afford to sweep this problem under the carpet. Not only is it costing us heavily in maintenance dollars, it also exacts its toll in the loss and damage column. And, worse still, the fragile or perishable type of shipment which is delivered in a homogenized condition, may well lose the industry a shipper. And we can't afford that either. So let's get our design engineers, researchers and field engineers together and come up with a running gear and suspension which will reduce maintenance costs and loss and damage, and make our shippers so pleased we will increase our share of the transportation dollar.

For those who take the position that our present freight car trucks are adequate, I would recommend that they make more than a casual inspection of the cars on the repair track, and while there look over the contents of the scrap pile; ride a few freight trains observing the car action; and obtain the reactions of shippers. I know a desperate need exists in this area, and I am confident that the ingenuity of our suppliers and industry forces working in combination can come up with a workable solution, if given the proper direction and incentive.

Those of you who keep abreast of the developments in the rapid transit sector know the important advances made in running gear in that area. Now, if such advanced thinking can be applied to a segment of the industry which contributes little or nothing to the profit picture, surely we can afford to make at least as great an effort for the very backbone of our industry! Other than a few refinements added along the way, our present basic truck design dates back to the days of the wood burning locomotive, and although I don't advocate making a change just for the sake of changing, there is definitely a great need for improvement in this important part of the freight car which is so closely related to the continuity of our operations.

In all fairness, mention should be made of a couple of recent developments by truck manufacturers which appear to do a very good job in controlling the truck instability problem. One of these accomplishes this by providing an increased amount of lateral freedom in the truck, which freedom is controlled by a swing-hanger type of action. This development is built around the conventional type truck and provision is made to reduce the possibility of relative longitudinal movement between the side frames.

The other development is a considerable departure from the conventional freight truck, in that it employs a semi-rigid truck frame which prevents the parallelogramming action in the truck.

From observations made to date, both of these developments appear to provide a very considerable decrease in the wear rates of the components mentioned earlier in connection with the truck hunting problem. The most readily noticeable feature when inspecting these trucks is in the lack of flange wear. I recently saw a carset of the semi-rigid frame type trucks which had accumulated over 175,000 miles, and the flanges hardly had a mark on them. While both these trucks appeared to provide a considerable improvement over the conventional equipment, very little seems to have been done toward furthering their position in the market, which I understand is a result of surveys not indicating enough interest in them at the present to warrant further expenditures at this time. Now, this is a pretty sad state of affairs when we may be overlooking considerable potential savings, because of a few dollars additional initial investment.

I would like to briefly mention a couple of developments which we are working on in connection with trucks and running gear. One is the conical type center plate which we have been testing for a couple of years. A sketch of this arrangement is shown in Figure 6. First, you will notice that the female portion is on the car body rather than on the truck bolster, thus eliminating the "catch basin" which is basic in the conventional arrangement. Next, note that the heaviest section in the body portion is in the area of the greatest bending moments, quite the opposite from the conventional arrangement in which failures are common in this area. There was initial concern as to whether the car would tend to ride the side

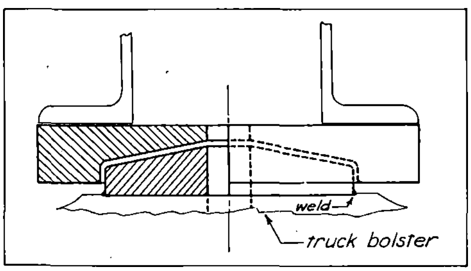


**Fig. 6. Conical center plate in which male portion is located on truck bolster.**

bearings with this arrangement, but tests indicated that this was not the case. With this arrangement, the longitudinal and transverse forces are taken on the tapered surfaces, resulting in little or no wear on the vertical surface, and making the use of the manganese wear ring unnecessary. We have tested this arrangement in 100-ton cars in very high mileage unit train service, and the results have been very encouraging.

Some installations have lubrication in the form of plugs in the male portion, and some of them have been under test without the use of lubrication. The latter appears very promising and we are now installing fifty additional carsets in the same high mileage 100-ton cars without provision for lubrication. This installation could be made readily as original equipment if patterns were available for casting the male portion integral with the truck bolster, although it is really quite easily applied to existing conventional truck bolsters as shown in Figure 7, wherein the vertical ring around the bowl is cut off and the conical plate is welded into place. In the present installation of fifty carsets, we are using a high manganese male portion, and the body plate is being machined from 4130 grade steel. The original installation was of LAHT material and it has now accumulated more than 200,000 miles, whereas conventional body plates have failed with less than 100,000 miles in this same service. It is anticipated the high manganese and 4130 combination will provide well over 500,000 miles of service.

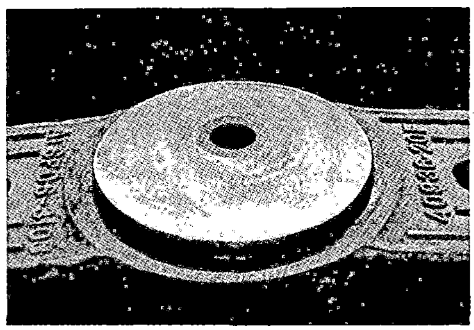

**Fig. 7. Male conical center plate applied to conventional truck bolster.**

The other development to which I referred is the coaxial train concept which is still in the design stage. Figures 8 and 9 are photographs of a small model of this development, and you can see that it employs a radically different type of suspension. This is not a train in the usual sense, in that it is not a train made up of a number of cars, but is a train in itself, made to any desired length for the particular service. If the design proves to have merit, it is intended it will be used in high speed container service.

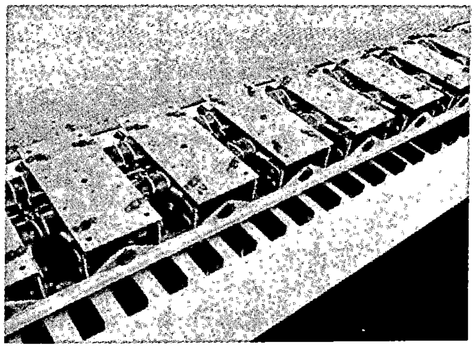

Fig. 8. Top view of model of coaxial train running gear.

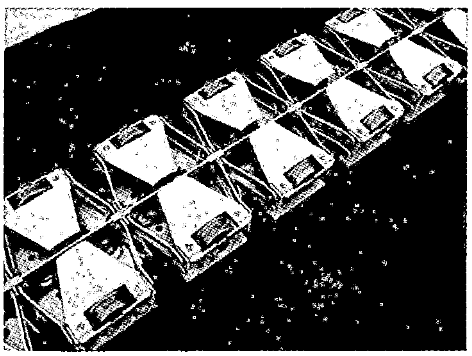

Fig. 9. Underneath view of coaxial train model showing individual "knee-action" wheel suspensions and continuous center sill.

Basically, the coaxial train has a continuous center sill throughout its length, with a "knee-action" suspension from each wheel, connected to it. The wheels are not mounted on through axles, but instead, are mounted on stub shafts and act independently of each other. It is anticipated that each wheel will be driven by its own motor. Various types of springing have been considered, but air springs now appear to be the most feasible. One of the criteria in developing this concept was that it could operate at speeds in excess of 125 mph. Providing for adequate braking from these speeds has proven to be one of our most serious problems.

We expect to make a full scale prototype of a one wheel pair module in the near future for evaluation. If, after evaluation, the concept still appears to have merit, it is intended to make up a series of these modules attached to a continuous center sill and initiate a test program on the unit.

Anticipated advantages of this coaxial train concept are its ability to maintain operating stability even though there is a rapid change of load on a wheel due to a track defect; the ability to operate at higher speeds over track maintained to today's standards; low center of gravity; no train slack action; improved vertical riding qualities; and no necessity for cushioning devices to protect against impacts.

Well, now assuming there is substance to my contention that our freight car trucks are grossly inadequate for present and future operating conditions and requirements, where do we go from here? Merely talking about the problem isn't going to solve it. Positive action is needed involving our best research and design talent, testing techniques, and above all, an effective program to apprise our managements of the urgent need for such a development and the economic benefits which may be expected from it. Such an effort must be industry wide and universally accepted, for there is no other alternative in our system of interchange.

I understand that the AAR Research Consulting Committee on the Design and Performance of Freight Car Trucks has an objective which I believe is to develop adequate performance requirements. Perhaps this is the vehicle which can properly initiate such a program. It is hoped that the scope of their efforts will not be limited to conventional trucks, but will also make investigation into non-conventional types of running gear and suspension. Areas which should be considered in the requirements for freight car running gear are:

1. The allowable change in wheel load for a certain track defect;
2. The elimination of the lateral instability problem (that is, unstable condition of truck hunting) at speeds up to, and in excess of, 100 mph.;
3. Improved riding qualities, with special emphasis on reducing the high frequency "noise" transmitted into the car structure and lading;
4. A design which will reduce jackknifing tendency to a minimum under conditions of heavy buff forces;
5. A design which will reduce related maintenance requirements to a minimum, and;
6. Realistic test conditions under which any selected design would be tested.

This latter point is important, because too often the principals in a truck test specify that all parts be in a new condition. I especially take exception to the use of new wheels in truck tests. Most any truck will ride fairly well when the wheel tread contours are in perfect condition, but when you find the truck that will turn in a good performance when the wheel treads are nearing condemning, you then have something to become excited about. And, as a word of caution when selecting worn wheels for truck tests, do not pick them at random, but select them in truck sets. If this is not done, the wheel pairs may work against each other and nullify any lateral instability which may otherwise be inherent in the system. Another important point is to restrict testing to clean, dry rail, since the lateral instability problem is seldom manifested in the normal operating speed range when the rail is wet or dirty.

The proposed DOT wheel-rail dynamics test facility, which was described here this morning, should provide considerable information when developing criteria which should be included in running gear and suspension requirements. Information which can be generated by such a facility as this is difficult and often impossible to develop by other methods.

Replacing our antiquated trucks with running gear and suspension designed for present and future operating requirements may be considered by some as a luxury which our industry cannot afford. The same thinking was advanced when the transition was made from link and pin to the automatic coupler. But, where would the industry be today if this had not taken place? With the number of unexplained derailments increasing, with the heavy economic drain from the maintenance of equipment as well as from loss and damage, and with the tendency for the railroad to be relegated more and more to the handling of lower revenue lading, it is no longer a matter of whether we can afford improved running gear. The point has been reached where we can no longer afford to not have it. Thank you very much.

DELEGATE COMMENT: Mr. Angold, could you elaborate on the rack post failures? Did they occur near the attachment of the post to the car?

MR. ANGOLD: Yes. Near the place of attachment.

DELEGATE COMMENT: How was this problem circumvented?

MR. ANGOLD: It was a basic design weakness in this car. It was helped by stiffening up the center sill by boxing it in so that the car would not rack as readily.

DELEGATE COMMENT: Was this the car where the rack was an independent design or was the rack an inherent part of the total car design?

MR. ANGOLD: It was an integral part of the car structure.

CHAIRMAN SMITH: Before we go to lunch, I have a comment to make on John's paper. He sort of embarrassed the truck manufacturers, of which we are one, about not coming up with new design trucks and pursuing the goal of a suitable truck for our customers. I wish to point out we designed and built a high speed truck—the XL-50—back in 1946. It was well tested with favorable results. 134 carsets were placed in service on six railroads. We exhibited one of these trucks here at our first Conference in 1964. It had gone over 800,000 miles without replacement of parts. Ten years ago we decided on the basis of the performance of the XL principal to proceed with a new 70-ton truck. We went through with the truck and had it AAR tested, proved it out, made road tests, offered it to our customers, you people. I wonder how many of you have made a real study of it. Now, John speaks of overlooking trucks. John, when did you put the XL truck under your cars?

MR. ANGOLD: Well, we have tested it. In fairness to ourselves, we did put our "money where our mouth is" by making a fairly substantial investment in one of the trucks which I mentioned earlier, which our tests did show to eliminate the lateral instability problem.

CHAIRMAN SMITH: I can say, with the exception of CN, and the PFE, the railroads have not made a real test—road test, service test—of the XL-70. As a representative of the truck manufacturers, I can somewhat indict the railroad people, too. I concur with you that it is both our problem. But, it is very difficult to bring out a truck, most difficult, as you people know. There is more to it than just talk. There is a lot of money invested, and a lot of time, and much cooperation needed. I think it was pointed out last year that there is a need for a complete system in regard to trucks and also the cooperation of the railroaids to furnish the truck designer adequate time and information to arrive at what the customer wants. Truck manufacturers are limited in their facilities. You people alone have the actual test site. With that, let's go to lunch.

CHAIRMAN SMITH: The first paper this afternoon will be given by Mr. George R. Reed, Director Engineering and Research, American Car and Foundry Division, ACF Industries.

# Effect of Design Variation on Service Stresses in Railroad Wheels

J. P. BRUNER R. D. JONES SAMUEL LEVY J. M. WANDRISCO

## INTRODUCTION

Present wheel configurations have evolved from design changes dictated by service performance experience. The satisfactory performance of a railroad wheel depends on its ability to withstand not only the repeated stresses imposed on it by normal loads and braking conditions, but also the occasional high stresses that develop under abnormal operating conditions.

Wheel failures are infrequent, but when they occur, investigations often disclose them to result either from unusual service conditions which cause thermal damage or impose high stresses on the wheel, or from normal service conditions in which stress concentrations occur at discontinuities in the wheel. A small but increasing incidence of plate failures of wheels in which no discontinuities can be found, suggests that the continual increase in wheel loads, operating speeds, and braking severity associated with present railroad operating practices and new types of equipment may impose stresses on the wheels which approach the limit of safe performance.

The manufacturers of wrought steel wheels, through the Technical Committee on Railroad Materials of the American Iron & Steel Institute have therefore been considering whether the present designs are the best attainable, and if not, what can be done to improve them. This is not a simple problem because the optimum designs, from a service standpoint, are those which can best withstand the combined effects of stresses resulting from the following conditions:

1 Thermal gradient effects resulting from the conversion of the kinetic energy of the train into heat when the brake shoes are applied to the wheel tread. The stresses caused by braking are considered to be of a steady nature as they do not fluctuate, i.e., vary in magnitude, during the course of each wheel revolution.

2 Vertical loads due to equipment and lading. These loads can apply at any point across the wheel tread fluctuating once in each revolution of the wheel, or about one million times in the course of 1800 miles travel, and may be accentuated by dynamic effects due to track deviations and operating conditions.

3 Lateral loads applied against the front of the flange as a result of curve negotiation, hunting and nosing; against the rim faces from the action of car retarders; and against the back of the flange by guardrail and special track-work. These loads occur less frequently than vertical loads, but also fluctuate during each revolution of the wheel.

4 Tractive loads, also of a fluctuating nature, which act in the rail contact area of driving wheels.

## COMPUTER PROGRAM FOR SIMULATED BRAKING AND TRACK LOADING

In 1964, the manufacturers of wrought steel wheels[1] commissioned General Electric Company's Advanced Technology Laboratory (now Research and Development Center) to develop a method for determining the combined effects of these stresses so that different wheel configurations could be evaluated. Using basic mechanical and thermal concepts, a mathematical approach was adopted. Two computer programs were developed; one to deal with thermally induced stresses resulting from braking, and one to deal with track load stresses. A detailed account of this portion of the work was presented at the December 1965 meeting of the Amercian Society of Mechanical Engineers (1).[2]

For that study, the computer input information consisted of a mathematical description of the geometry of the wheel under consideration, together with details of the physical properties of the wheel steel. Loading data were described in the case of the thermal stress program, as the time history of the heat input to various locations on the tread and in some cases on the flange. From these data, the increase in temperature at

[1] Armco Steel Corporation, Bethlehem Steel Corporation, Canadian Steel Wheel Limited, Edgewater Steel Company, Standard Steel Division of Baldwin-Lima-Hamilton Corporation, and the United States Steel Corporation.

[2] Numbers in parentheses designate References at the end of the paper.

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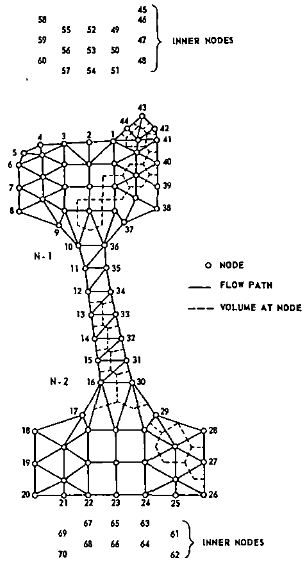



Fig.1 Wheel schematic cross section showing thermal nodes and heat flow paths used in computer study. Volumes and surfaces associated with nodes are shown dotted at typical locations

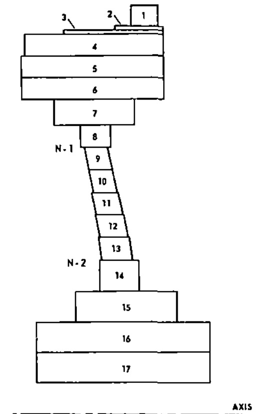



Fig.2 Wheel schematic cross section showing 17 structural rings used in computer thermal stress analysis

each of seventy different positions in the wheel cross section were measured, Fig.1. Then the resultant stress changes on the surfaces of the wheel were calculated by considering the wheel to be an assembly of seventeen concentric rings. This is shown schematically in Fig.2. The tendency of each ring to expand as its temperature increased, was balanced by interfacial forces and moments, just sufficient to overcome the effects of the different expansions of the rings. The stresses in the wheel, required to counteract these forces, were then calculated.

For the track load program, the computer input information was similar with the exception that thermal data were excluded. The same structural ring concept as shown in Fig.2 was used. Circumferential distribution of stresses was taken into account, since these loads are applied at only one point on the rim. Hertzian stresses in the wheel-rail contact area were not included.

Details of these computer programs have been presented to the Association of American Railroads by the wrought steel wheel manufacturers through the American Iron and Steel Institute.

*HEATING AND LOADING CONDITIONS*

Using these programs for simulated braking and track loading, a computer analysis of several different wheel designs was made using a variety of braking conditions and track loads. The programs were used separately and 225 computer runs were made to determine the specific effects of different types of loading. The effects of rapid-stop versus drag braking were examined as well as the effects of brake shoe position on the tread, application points of track loads, and traction loads. Basic designs examined were A40 multiple-wear, F36 multiple-wear, and B33 one-wear wheels. Design variations studied included plate slope, plate shape, plate thickness, fillet radii, and rim thickness. The specific configurations are shown in Figs.3-5 inclusive.

For these analyses, two types of braking conditions were used. In the first case, a constant rate of heat input was used to simulate prolonged drag braking. Heat inputs of 50,000 Btu/hr and 20,000 Btu/hr were used representing two different degrees of drag braking. The highest heat input was chosen to produce stresses exceeding the yield strength and arbitrarily used on the A40 design. The lower heat input representing a moderate braking condition was used on the F36 and B33 designs. The second case simulated rapid-stop braking with a high initial rate of heat input. To simulate a rapid-stop, an initial rate of 175,000 Btu/hr was used for the A40 wheels. Much lower initial rates were used for the other wheels. The heat input curves for both cases are shown in Fig.6 for the A40 wheels and Fig.7 for the F36 and B33 wheels. Heating conditions are described in Table 1 and the location of the brake shoe on the tread is shown in Fig.8.

Several track loading conditions were simulated. Vertical loads of 60,000 lb were used,

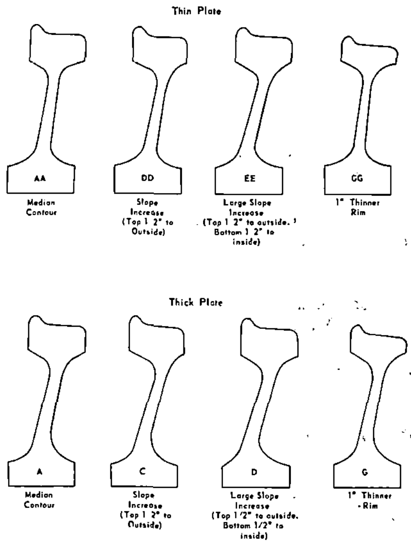



Fig.3 A40 wheel designs

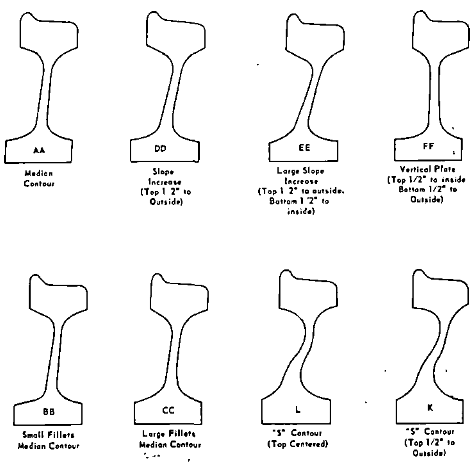



Fig.4 F36 wheel designs

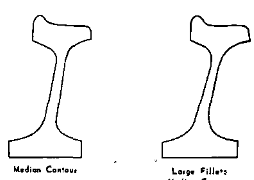



Fig.5 B33 wheel designs

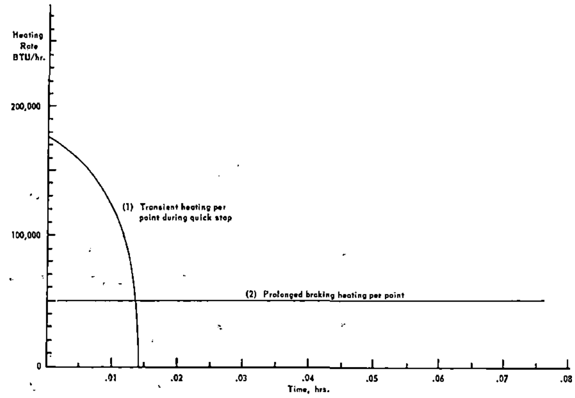



Fig.6 Braking conditions for A40 wheels

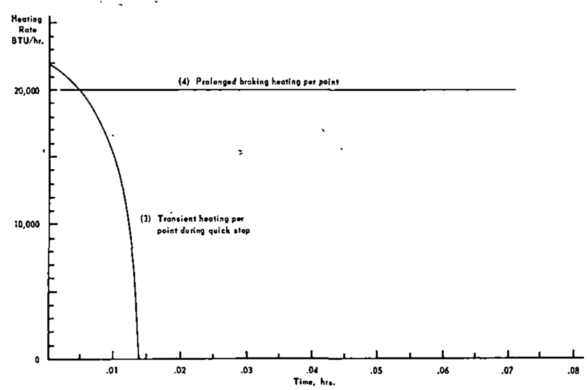



Fig.7 Braking conditions for F36 wheels

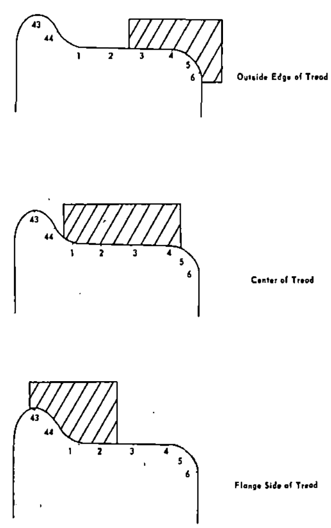



Fig.8 Brake shoe locations

TABLE 1

SIMULATED BRAKING CYCLES

A40 Wheels

1. Variable braking according to Curve (1) of Fig. 6 for quick stop. Brake applied centrally on tread (points 1, 2, 3, 4). Stresses after .0265 hrs. (1.59 min.).

2. Variable braking according to Curve (1) of Fig. 6 for quick stop. Brake applied towards outside of tread (points 3, 4, 5, 6). Stresses after .0065 hrs. (0.39 min.) except where otherwise noted. Temperatures after .0065 hrs. were excessive at point 5.

3. Variable braking according to Curve (1) of Fig. 6 for quick stop. Brake applied towards flange side of tread (points 43, 44, 1, 2). Stresses after .0265 hrs. (1.59 min.).

4. Prolonged braking according to Curve (2) of Fig. 6. Brake applied centrally on tread (points 1, 2, 3, 4). Stresses after .0705 hrs. (4.22 min.).

5. Prolonged braking according to Curve (2) of Fig. 6. Brake applied towards outside of tread (points 3, 4, 5, 6). Stresses after .0705 hrs. (4.22 min.).

6. Prolonged braking according to Curve (2) of Fig. 6. Brake applied towards flange side of tread (points 43, 44, 1, 2). Stresses after .0705 hrs. (4.22 min.).

B33 and F36 Wheels

1. Variable braking according to Curve (3) of Fig. 7 for quick stop. Brake applied centrally on tread (points 1, 2, 3, 4). Stresses after .0265 hrs. (1.59 min.) unless otherwise noted.

2. Variable braking according to Curve (3) of Fig. 7 for quick stop. Brake applied towards flange side of tread (points 43, 44, 1, 2). Stresses after .0265 hrs. (1.59 min.) unless otherwise noted.

3. Prolonged braking according to Curve (4) of Fig. 7. Brake applied centrally on tread (points 1, 2, 3, 4). Stresses after .0705 hrs. (4.22 min.).

4. Prolonged braking according to Curve (4) of Fig. 7. Brake applied towards flange side of tread (points 43, 44, 1, 2). Stresses after .0705 hrs. (4.22 min.).

TABLE 2

SIMULATED LOADING CYCLES

A40 Wheels

1. Vertical load of 60,000 lbs. applied centrally on tread at 2.85 inches from flange apex.

2. Vertical load of 60,000 lbs. applied towards outside of tread at 4.275 inches from flange apex.

3. Vertical load of 60,000 lbs. applied towards flange side of tread at 1.4 inches from flange apex.

4. Vertical load of 60,000 lbs. applied centrally on tread at 2.85 inches from apex and lateral load of 30,000 lbs. towards flange.

5. Vertical load of 60,000 lbs. applied centrally on tread at 2.85 inches from flange apex and lateral load of 30,000 lbs. away from flange.

6. Vertical load of 60,000 lbs. applied centrally on tread at 2.85 inches from flange apex and traction force of 30,000 lbs.

B33 and F36 Wheels

1. Vertical load of 60,000 lbs. applied centrally on tread at 2.85 inches from flange apex.

2. Vertical load of 60,000 lbs. applied towards outside of tread at 4.275 inches from flange apex.

3. Lateral load of 20,000 lbs. on the flange at 0.47 inches above the flange apex.

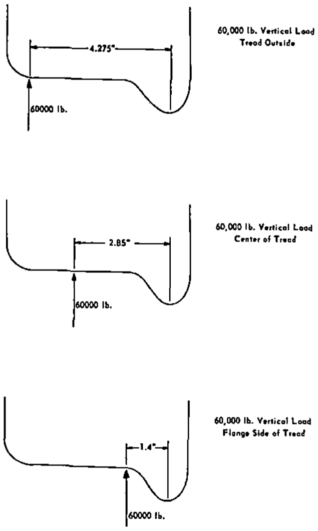



Fig.9 Track loads

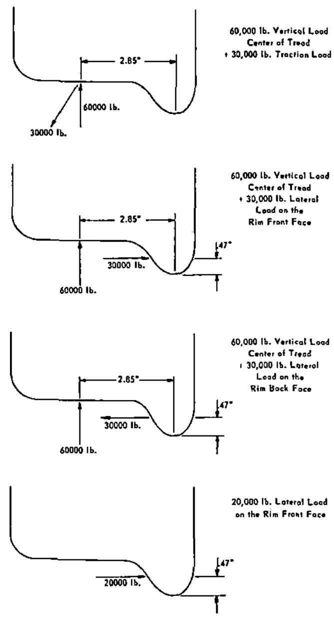



Fig.9 (continued) Track loads

about twice the normal loading. This allowed for increased loading due to dynamic effects. On the A40 wheels, lateral loads of 30,000 lb, much higher than normal, were combined with the vertical loads. These were applied against both front and back rim faces. A 30,000-lb tractive load was also investigated on the A40 wheels. For calculations on the F36 and B33 wheels, a 20,000-lb lateral load on the front of the flange was used without vertical loads. Specific loading conditions are listed in Table 2 and illustrated in Fig.9.

STRESS CONCEPTS

Previous investigations have shown that the maximum principal stresses developed in the wheel plates act radially and occur at the back face N1 position (BFN1) and front face N2 position (FFN2). These are the critical plate locations from the standpoint of fatigue, and most plate failures in service have originated at one of these two locations.

It is known that heating of the rim from braking friction develops high radial tensile stresses at the critical plate locations BFN1 and FFN2, whereas compressive radial stresses are developed on the opposite sides of the plate. Previous investigations have also shown that loads on the front of the flange develop stresses that are opposed to the vertical loading stresses, so the combined effects of vertical and lateral loads are less at the critical plate locations. Lateral loads on the back of the flange add to the vertical loading stresses so the combined effects are greater. Lateral load investigations reported in the literature have invariably dealt with loads against the front of the flange. The loads against the back of the flange encountered with guard-rail or special track-work and loads against the back rim with some retarders have not been reported.

In the discussion, heating stresses are considered separately from vertical and lateral loading stresses. Heating stresses are steady stresses, since they remain essentially constant in one wheel revolution, while track loading stresses fluctuate with each revolution. For purposes of evaluation, fluctuating stresses are considered as a range of stress through which the stress varies equally in magnitude from a mean stress. A sample calculation is shown below for radial stresses at one of the critical locations, BFN1 or FFN2.

| | Point of Track Load 0 deg | Opposite Track Load 180 deg |
|---|---|---|
| Steady stress (heating) | +40,000 psi | +40,000 psi |
| Fluctuating stress(loading) 60,000 lb vertical load 30,000 lb front face lateral load | +20,000 psi | -10,000 psi |
| Combined stresses | +60,000 psi Max | +30,000 psi Min |

TABLE 3

EFFECT OF BRAKING HEAT INPUT
ON RADIAL STRESSES

| CONDITION | LOCATION | A40 STRESS (ksi) MEDIAN CONTOUR (AA) | A40 STRESS (ksi) SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) | F36 STRESS (ksi) MEDIAN CONTOUR (AA) | F36 STRESS (ksi) SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) | F36 STRESS (ksi) VERTICAL PLATE (FF) |
|---|---|---|---|---|---|---|
| Rapid Stop (Tread Center) | BFN1<br>FFN2 | + 21<br>+ 32 | + 25<br>+ 32 | + 4<br>+ 5 | + 4<br>+ 5 | + 1<br>+ 3 |
| Drag Braking (Tread Center) | BFN1<br>FFN2 | + 35<br>+ 52 | + 38<br>+ 50 | + 22<br>+ 26 | + 21<br>+ 25 | + 8<br>+ 13 |

TABLE 4

EFFECT OF BRAKE SHOE LOCATION ON RADIAL
STRESSES IN A40 WHEELS

| CONDITION | LOCATION | STRESS (ksi) MEDIAN CONTOUR (AA) | STRESS (ksi) SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) | STRESS (ksi) RIM 1" THINNER (GG) |
|---|---|---|---|---|
| Prolonged Heating (Tread Outside) | BFN1<br>FFN2 | + 28<br>+ 71 | + 33<br>+ 66 | + 15<br>+ 82 |
| Prolonged Heating (Tread Center) | BFN1<br>FFN2 | + 35<br>+ 52 | + 38<br>+ 49 | + 37<br>+ 63 |
| Prolonged Heating (Flange Side of Tread) | BFN1<br>FFN2 | + 56<br>+ 45 | + 60<br>+ 44 | + 65<br>+ 51 |

$$\text{Mean stress} = \frac{\text{max} + \text{min}}{2} = +45{,}000 \text{ psi}$$

Stress range = max-min = +30,000 psi or ±15,000 psi

Note that heating stresses remain constant when the wheel is rotated from 0 deg to 180 deg, but the loading stresses vary. In one wheel revolution the combined stresses at any one point fluctuate from +60,000 psi maximum at 0 deg to +30,000 psi minimum at 180 deg. These data would be reported in the tables as +40,000 psi heating stress and ±15,000 psi fluctuating stress. In the graphs, however, the ±15,000 psi would be plotted about the +45,000 psi mean stress. There is a +5,000 psi component of mean stress because the track load fluctuates between +20,000 psi and -10,000 psi.

COMPUTER RESULTS

Rapid-Stop Versus Drag Braking

In all designs examined in the programs, simulated drag braking produced much higher tensile stresses at BFN1 and FFN2 positions than did rapid-stop braking. A few A40 and F36 wheel designs were selected for illustration, Table 3. Note that A40 wheels had higher stresses from both types of braking. This is due to the higher heat inputs. For the remaining comparisons, only the lower steady heat input from simulated drag braking will be considered.

Effect of Brake Shoe Location

Heat was applied toward the outside of the tread, at the tread center, and toward the flange side of the tread as shown in Fig.8. This simu-

TABLE 5

EFFECT OF CHANGING VERTICAL LOAD APPLICATION POINTS ON THE WHEEL TREAD ON RADIAL STRESSES IN A40 WHEELS

| 60,000 LB. VERTICAL LOAD APPLICATION POINT | LOCATION | STRESS (ksi) | | |
|---|---|---|---|---|
| | | MEDIAN CONTOUR (AA) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (DD) | LARGE SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) |
| Tread Outside | BFN1<br>FFN2 | ± 3<br>± 9 | ± 4<br>± 8 | ± 4<br>± 6 |
| Tread Center | BFN1<br>FFN2 | ± 5<br>± 6 | ± 6<br>± 6 | ± 6<br>± 6 |
| Tread Inside | BFN1<br>FFN2 | ± 6<br>± 4 | ± 8<br>± 4 | ± 8<br>± 4 |

TABLE 6

EFFECT OF TRACTION LOADING ON RADIAL STRESSES IN A40 AND F36 WHEELS

| CONDITION | LOCATION | A40 STRESS (ksi) | | | F36 STRESS (ksi) | | |
|---|---|---|---|---|---|---|---|
| | | MEDIAN CONTOUR (AA) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (DD) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) | MEDIAN CONTOUR (AA) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (DD) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) |
| 60,000 lb. Vertical (Tread Center) | BFN1<br>FFN2 | ± 5<br>± 6 | + 6<br>± 6 | ± 6<br>± 6 | ± 7<br>± 8 | ± 8<br>± 8 | ± 8<br>± 8 |
| 60,000 lb. Vertical (Tread Center) +30,000 lb. Traction | BFN1<br>FFN2 | ± 5<br>± 6 | ± 6<br>± 6 | ± 6<br>± 6 | ± 7<br>± 8 | ± 8<br>± 8 | ± 8<br>± 8 |

lates the effect of the brake shoe riding on different points of the tread. Results are shown on A40 wheels in Table 4.

Steady tensile stresses at BFN1 were lowest with heating at the tread outside corner, increasing as the flange was approached. For example, on a thin rim A40 with 50,000 Btu/hr for about 4 min, the stress increased from a low of 15,000 psi to a high of 65,000 psi. Conversely, FFN2 stresses were highest with heating at the outside corner decreasing as the flange was approached. In the same example, these decreased from 82,000 psi to 51,000 psi. All wheel designs showed the same trend with the shift in heat input location but the magnitudes varied. These data demonstrate how the brake shoe location affects the stress pattern and the magnitude of tensile stresses that can be developed in braking.

For the remaining comparisons, only heating at the tread center is considered.

Effect of Vertical Load Positions on the Tread

Lateral displacement of the car wheel relative to the rail causes the contact point to move across the tread. Simulated loading tests were made representing the shift in contact points of the wheel and rail. Computer runs were made simulating a 60,000 lb vertical load at three locations across the tread surface, shown in Fig.9. Some results are shown in Table 5 on some A40 design variations.

The stress change trends were the same for

TABLE 7

EFFECT OF PLATE SLOPE ON RADIAL STRESSES
IN A40 AND F36 WHEELS

| CONDITION | LOCATION | A40 STRESS (ksi) | | | F36 STRESS (ksi) | | | |
|---|---|---|---|---|---|---|---|---|
| | | MEDIAN CONTOUR (AA) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (DD) | LARGE SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) | MEDIAN CONTOUR (AA) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (DD) | LARGE SLOPE INCREASE (TOP 1/2" TO OUTSIDE, BOTTOM 1/2" TO INSIDE) (EE) | VERTICAL PLATE (TOP 1/2" TO INSIDE, BOTTOM 1/2" TO OUTSIDE) (FF) |
| Prolonged Braking (Tread Center) | BFN1<br>FFN2 | + 35<br>+ 52 | + 42<br>+ 52 | + 38<br>+ 49 | + 22<br>+ 26 | + 23<br>+ 26 | + 21<br>+ 25 | + 8<br>+ 13 |
| 60,000 lb. Vertical (Tread Center) | BFN1<br>FFN2 | ± 5<br>± 6 | ± 6<br>± 6 | ± 6<br>± 6 | ± 7<br>± 8 | ± 8<br>± 8 | ± 8<br>± 8 | ± 4<br>± 6 |
| 60,000 lb. Vertical 30,000 lb. Lateral Front Face | BFN1<br>FFN2 | ± 1<br>± 13 | ± 2<br>± 11 | ± 3<br>± 8 | | | | |
| 60,000 lb. Vertical 30,000 lb. Lateral Back Face | BFN1<br>FFN2 | ± 11<br>± 26 | ± 10<br>± 22 | ± 8<br>± 20 | | | | |
| 20,000 lb. Lateral Front Face | BFN1<br>FFN2 | | | | ± 7<br>± 14 | ± 6<br>± 11 | ± 4<br>± 9 | ± 9<br>± 18 |

each design variation. Also the magnitude of change was about the same for each design variation shown in Table 5. At BFN1, fluctuating stresses of about ±4000 psi were measured when the load was applied to the outside of the tread. These increased to about ±8000 psi when the load shifted toward the flange. At FFN2, the stresses were of similar magnitude but they decreased as the point of load application was shifted from rim side of the tread to the flange side.

For the remaining comparisons, only loading on the tread center is considered.

Effect of Traction Loading

A 30,000-lb tractive load was applied to all wheel designs tested, in conjunction with a 60,000-lb vertical load on the tread center. In Table 6, results are compared with a 60,000-lb vertical load only for several A40 and F36 wheel design variations.

There was no effect from traction loading on the maximum and minimum stresses (usually at 0 deg and 180 deg) though there was a small effect at other angles.

Effect of Plate Slope

One of the first design changes studied was the effect of plate slope. The slope was varied from the median in A40 and F36 wheels by changing the plate attachment points at the hub and rim fillets. For the purpose of this discussion, the median slope is the deviation produced by dishing when the plate is attached at the mid-width of the rim and hub. Increasing plate slope denotes greater deviation from vertical. The stresses from heating and loading are summarized in Table 7.

Heating Stresses. Substantially increasing the plate slope from the median had little effect on the stresses developed at either BFN1 or FFN2 from heating. Only the F36 design was checked for a decreasing slope from the median, approaching a vertical plate. Here, a 50 to 65 percent reduction occurred in the tensile stresses which dropped from a range of 22,000-26,000 psi to a range of 8000-13,000 psi.

Loading Stresses. Increasing the plate slope from the median had little effect on the vertical loading stresses. However, when lateral loads were combined with vertical loads on either the front or back rim faces, the larger fluctuating stresses were reduced about 20 to 40 percent as illustrated in Table 7 for the A40 wheels. This can be seen more clearly on the F36 wheels for the case of a 20,000-lb lateral load alone on the front of the flange. In this case also, stress reductions up to 40 percent were attained.

A vertical plate raised the fluctuating lateral load stress of the F36 wheel about 25 percent, but lowered the vertical load stresses slightly.

TABLE 8

EFFECT OF PLATE SHAPE ON RADIAL STRESSES IN F36 WHEELS

| CONDITION | LOCATION | STRAIGHT PLATE STRESS (ksi) MEDIAN CONTOUR (AA) | STRAIGHT PLATE STRESS (ksi) SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (DD) | "S" PLATE STRESS (ksi) MEDIAN CONTOUR (C) | "S" PLATE STRESS (ksi) SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (K) |
|---|---|---|---|---|---|
| Prolonged Braking (Tread Center) | BFN1<br>FFN2 | + 22<br>+ 26 | + 23<br>+ 26 | + 18<br>+ 22 | + 19<br>+ 18 |
| 60,000 lb. Vertical (Tread Center) | BFN1<br>FFN2 | ± 7<br>± 8 | ± 8<br>± 8 | ± 5<br>± 6 | ± 8<br>± 5 |
| 20,000 lb. Lateral Front Face | BFN1<br>FFN2 | ± 8<br>± 14 | ± 6<br>± 11 | ± 1<br>± 5 | ± 0<br>± 4 |

TABLE 9

EFFECT OF PLATE THICKNESS ON RADIAL STRESSES IN A40 WHEELS

| CONDITION | LOCATION | MEDIAN CONTOUR STRESS (ksi) THIN (AA) | MEDIAN CONTOUR STRESS (ksi) THICK (A) | SLOPE INCREASE STRESS (ksi) THIN (DD) | SLOPE INCREASE STRESS (ksi) THICK (C) |
|---|---|---|---|---|---|
| Prolonged Braking (Tread Center) | BFN1<br>FFN2 | + 35<br>+ 52 | + 32<br>+ 42 | + 42<br>+ 52 | + 37<br>+ 40 |
| 60,000 lb. Vertical (Tread Center) | BFN1<br>FFN2 | ± 5<br>± 6 | ± 4<br>± 5 | ± 6<br>± 6 | ± 6<br>± 5 |
| 60,000 lb. Vertical +30,000 lb. Lateral Front Face | BFN1<br>FFN2 | ± 1<br>± 13 | ± 3<br>± 9 | ± 5<br>± 11 | ± 5<br>± 6 |
| 60,000 lb. Vertical +30,000 lb. Lateral Back Face | BFN1<br>FFN2 | ± 11<br>± 25 | ± 6<br>± 18 | ± 10<br>± 22 | ± 6<br>± 15 |

Combined Heating and Loading Stresses. For the designs investigated, a plate with a greater slope resulted in the lowest fluctuating stresses, while a vertical plate produced the lowest heating stresses. Since heating stresses were not affected by increasing the slope from the median and fluctuating stresses were lower, a plate with a greater slope would appear to be desirable.

Effect of Plate Shape

Two F36 "S" shaped plate designs were compared with straight plates. One of the "S" plates had the upper portion of the plate shifted 1/2 in. toward the outside. This straightened the upper plate so only a semi-"S" shape was achieved. These results are tabulated in Table 8.

Heating Stresses. These were reduced about 20 percent in the "S" shaped plates at both BFN1 and FFN2 positions, from about 22,000-26,000 psi to 18,000-22,000 psi.

Loading Stresses. The fluctuating stresses developed from vertical loading were reduced by the "S" contour. The most significant change occurred in lateral loading, where the stresses were lowered at least 60 percent in the "S" shaped plates. A stress range of ±14,000 psi was lowered to ±5000 psi, a very significant reduction.

Combined Heating and Loading Stresses. Both

TABLE 10

EFFECT OF FILLET RADII ON RADIAL STRESSES IN F36 AND B33 WHEELS

| CONDITION | LOCATION | F36 STRESS (ksi) MEDIAN CONTOUR (AA) | F36 STRESS (ksi) SMALL FILLETS (BB) | F36 STRESS (ksi) LARGE FILLETS (CC) | B33 STRESS (ksi) MEDIAN CONTOUR | B33 STRESS (ksi) LARGE HUB FILLET |
|---|---|---|---|---|---|---|
| Prolonged Braking (Tread Center) | BFN1<br>FFN2 | + 22<br>+ 26 | + 22<br>+ 26 | + 18<br>+ 26 | + 23<br>+ 29 | + 19<br>+ 22 |
| 60,000 lb. Vertical (Tread Center) | BFN1<br>FFN2 | ± 7<br>± 8 | ± 8<br>± 9 | ± 6<br>± 8 | ± 8<br>± 9 | ± 8<br>± 7 |
| 20,000 lb. Lateral Front Face | BFN1<br>FFN2 | ± 8<br>± 14 | ± 8<br>± 14 | ± 5<br>± 13 | ± 5<br>± 14 | ± 4<br>± 12 |

TABLE 11

EFFECT OF RIM THICKNESS ON RADIAL STRESSES IN A40 WHEELS

| CONDITION | LOCATION | THIN PLATE STRESS (ksi) MEDIAN CONTOUR (AA) | THIN PLATE STRESS (ksi) 1" THINNER RIM (GG) | THICK PLATE STRESS (ksi) MEDIAN CONTOUR (A) | THICK PLATE STRESS (ksi) 1" THINNER RIM (G) |
|---|---|---|---|---|---|
| Prolonged Braking (Tread Center) | BFN1<br>FFN2 | + 35<br>+ 52 | + 37<br>+ 63 | + 32<br>+ 42 | + 34<br>+ 51 |
| 60,000 lb. Vertical (Tread Center) | BFN1<br>FFN2 | ± 5<br>± 6 | ± 5<br>± 6 | ± 4<br>± 5 | ± 4<br>± 5 |
| 60,000 lb. Vertical (Tread Center) +30,000 lb. Lateral Front Face | BFN1<br>FFN2 | ± 1<br>± 13 | ± 0<br>± 13 | ± 5<br>± 9 | ± 5<br>± 8 |
| 60,000 lb. Vertical (Tread Center) +30,000 lb. Lateral Back Face | BFN1<br>FFN2 | ± 11<br>± 25 | ± 10<br>± 26 | ± 6<br>± 18 | ± 6<br>± 18 |

steady heating and fluctuating loading stresses were reduced significant amounts in the "S" shaped plates.

Effect of Plate Thickness

Plate thickness effects were studied in A40 wheels by comparing minimum thickness plates (N1 = 1 in., N2 = $1^3/16$ in.) with those about 1/16 in. heavier. Results are shown in Table 9.

Heating Stresses. The steady heating stresses in the thicker plates decreased about 10 percent at BFN1, from about 40,000 to 35,000 psi, and 25 percent at FFN2, from about 50,000 to 40,000 psi. Both median slope and greater slope contours showed the same effect.

Loading Stresses. Fluctuating stresses in the thicker plate wheels were lower for all loading conditions studied. Stresses at FFN2 decreased from ±13,000 to ±9000 psi for combined vertical and front of the flange lateral loading, and from ±25,000 to ±18,000 psi for combined vertical and back of the flange lateral loading. These decreases ranged from 30 to 60 percent.

Combined Heating and Loading Stresses. It should be noted that both braking and loading stresses can be lowered substantially by increasing the plate thickness regardless of slope.

Effect of Fillet Radii

Another design variation studied was the ef-

TABLE 12

EFFECT OF WHEEL DIAMETER ON RADIAL STRESSES IN F36 AND A40 WHEELS

| CONDITION | WHEEL DESIGN | THIN PLATE (ksi) | | THICK PLATE (ksi) | |
|---|---|---|---|---|---|
| | | MEDIAN CONTOUR (AA) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (DD) | MEDIAN CONTOUR (A) | SLOPE INCREASE (TOP 1/2" TO OUTSIDE) (C) |
| Prolonged Braking | F36 (BFN1) | + 55.7 | + 57.1 | + 42.0 | + 43.5 |
| | A40 (BFN1) | + 35.0 | + 41.9 | + 31.6 | + 37.4 |
| | Ratio (BFN1) | .63 | .73 | .75 | .86 |
| | F36 (FFN2) | + 66.2 | + 64.5 | + 59.0 | + 55.7 |
| | A40 (FFN2) | + 52.1 | + 52.4 | + 42.0 | + 40.0 |
| | Ratio (FFN2) | .79 | .81 | .72 | .72 |
| 60,000 lb. Vertical (Tread Center) | F36 (BFN1) | ± 7.7 | ± 8.0 | ± 5.5 | ± 6.5 |
| | A40 (BFN1) | ± 4.9 | ± 5.9 | ± 4.3 | ± 5.6 |
| | Ratio (BFN1) | .64 | .74 | .78 | .86 |
| | F36 (FFN2) | ± 8.5 | ± 8.1 | ± 6.2 | ± 5.5 |
| | A40 (FFN2) | ± 6.3 | ± 5.9 | ± 4.9 | ± 4.7 |
| | Ratio (FFN2) | .74 | .73 | .79 | .85 |

fect of different fillet radii. Data for F36 wheels with both rim and hub fillet radii changed are shown in Table 10, along with a B33 wheel with only different hub fillet radii. Increasing the fillet radii had the effect of thickening the plate with a consequent reduction in stresses.

Effect of Rim Thickness

A40 wheels representing full rim conditions and rims after substantial service wear were compared, Table 11.

Heating Stresses. Decreasing the rim thickness 1 in. raised the FFN2 stresses about 20 percent in the 40,000-60,000 psi range. BFN1 stresses were affected only slightly.

Loading Stresses. There were only very small changes in any of the fluctuating stress ranges due to rim thickness.

Effect of Wheel Diameter

The effect of wheel diameter may be observed in Table 12, where stresses for 36-in. and 40-in-dia wheels are shown. Heating stresses for the F36 wheels were multiplied by 2.5 to make them comparable with those used for the A40 wheels. This straight line adjustment was necessary because the heat input was 20,000 Btu/hr for the F36 wheels and 50,000 Btu/hr for the A40 wheels, and is compatible with the theoretical aspects of the computer program.

It can be seen that the 36-in-dia wheels developed larger stresses than the 40-in. wheels. The ratios are in the range of 0.63 to 0.86, roughly inversely proportional to the square of the wheel diameter. These data show that increasing the wheel diameter for a given service is effective in reducing plate stress resulting from both the heating and track loading.

EFFECT OF COMBINED HEATING AND LOADING STRESSES ON WHEEL FATIGUE

In the initial part of this study, the effects of steady braking stresses and fluctuating loading stresses were determined separately. This study gives information which would be useful in selecting a wheel configuration for a particular service where perhaps one type of loading might occur with above average frequency. But, how can we look at the overall picture? How can these data be used to determine the combined effect of heating and loading stresses?

A method of combining steady and fluctuating stresses and comparing with the fatigue strength of a material is the modified Goodman diagram (2). For this method residual stress induced in the manufacturing process can also be considered a steady stress. Residual stress remains constant during a wheel revolution, and is added directly to the heating stress. The fatigue limit is the maximum fluctuating stress which a material can withstand without failure. For as-rolled Class U wheel steel the fatigue limit is about ±22 500 psi for 10,000,000 cycles if the steady stress is zero. As the steady stress (residual, heating, and mean stress for the load) is increased in ten-

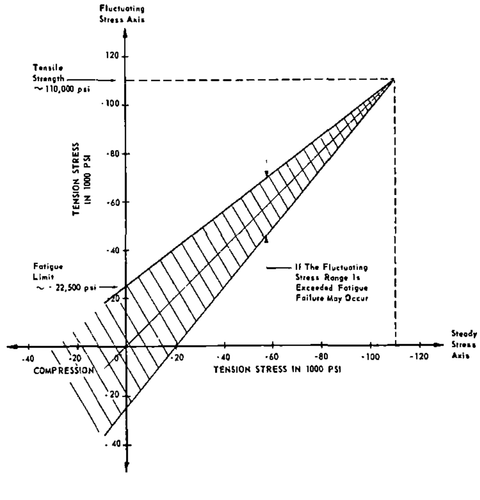



Figure 10 - Modified Goodman Diagram Relating Steady And Fluctuating Stresses To Fatigue Strength

Fig.10 Modified Goodman diagram relating steady and fluctuating stresses to fatigue strength

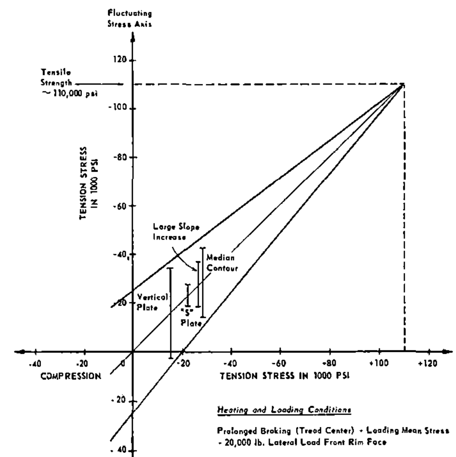



Fig.11 Modified Goodman diagram combining heating and loading stresses at FFN2 for several A40 wheel designs

sion, the magnitude of fluctuating stress (vertical and lateral loads) required to exceed the fatigue limit is reduced. This is shown by the modified Goodman diagram in Fig.10. The fatigue limit is reduced to about ±15,000 psi at 40,000 psi tension steady stress and to ±11,000 psi at 60,000 psi steady stress for wheels. Note that steady compressive stress is beneficial. Combined stresses which remain in this "fatigue envelope" (cross-hatched in Fig.10) may be considered safe.

To illustrate this fatigue concept, combined heating and loading stresses for several of the wheel designs investigated are plotted on modified Goodman diagrams in Fig.11. These examples were selected to emphasize differences and demonstrate how the diagrams might be used. For simplicity only the FFN2 stresses are shown, although BFN1 could just as well have been used. Vertical loading stresses alone are not plotted because the fluctuating stress ranges were relatively low. Also residual stresses are not included, although they may add several thousand psi to the steady heating tensile stress on rim-treated wheels.

Fig.11 shows stress ranges for three A40 designs with lateral loading both on front and back flanges. Note that the tensile heating stress plus the mean stress for loading are about 40,000 psi for a median contour plate, 50,000 psi for a median contour plate, and 60,000 psi for a median contour thin plate. Combining these steady stresses with high fluctuating stresses from loading caused the fatigue limit to be exceeded in some cases. Stresses due to the back lateral loads were consistently higher than those due to

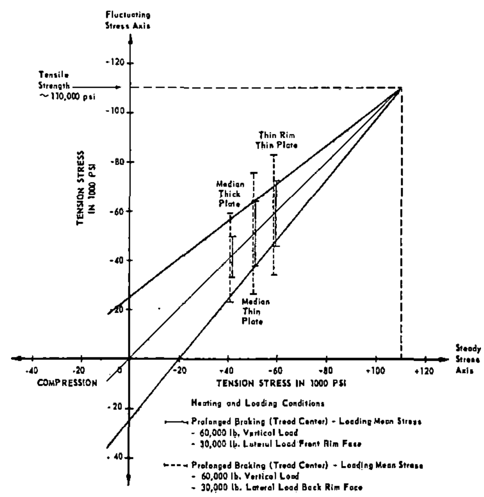


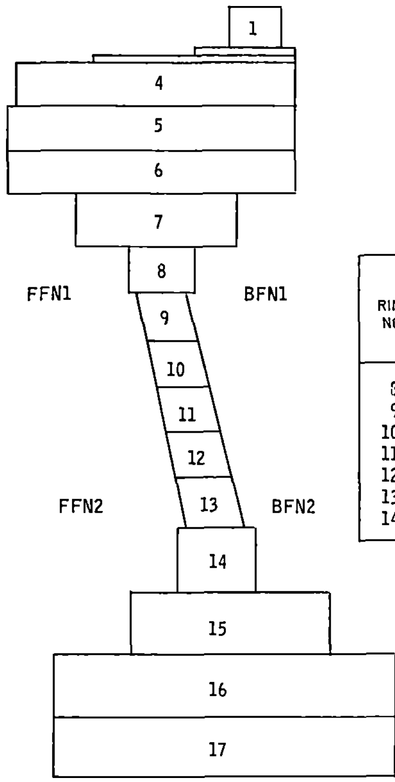

Fig.12 Modified Goodman diagram combining heating and loading stresses at FFN2 for several F36 wheel designs

front lateral loads because they add to, rather than subtract from, the vertical loads. Extreme cases are illustrated since heat input for these A40 wheels was very high and loads were at an expected maximum. These data show the advantages of a thicker plate and the disadvantage of a thin rim.

A more moderate case is illustrated by selected F36 data in Fig.12. Heat inputs were

TABLE 1

RATIOS OF FLUCTUATING STRESS TO THE FATIGUE LIMIT
AT DIFFERENT RING LOCATIONS ON THE PLATE OF AN F36 WHEEL
WITH AN "S" SHAPE




| RING NO. | FLUCTUATING STRESS/STRESS AT FATIGUE LIMIT | | | |
|---|---|---|---|---|
| | INITIAL CONTOUR | | FINAL CONTOUR | |
| | OUTSIDE | FLANGE SIDE | OUTSIDE | FLANGE SIDE |
| 8 | .04 | .47 | .15 | .19 |
| 9 | .13 | 2.00 | .18 | .19 |
| 10 | .08 | .26 | .24 | .19 |
| 11 | .29 | .07 | .37 | .12 |
| 12 | 1.43 | .08 | .70 | .11 |
| 13 | 1.75 | .08 | .69 | .09 |
| 14 | .56 | .04 | .59 | .06 |

only 40 percent of those on the A40 wheels (20,000 Btu/hr versus 50,000 Btu/hr), and lateral loads were only 20,000 lb on the front of the flange. The effect of lower heat input can be immediately seen by the much lower steady stress, in the 10,000 to 25,000 psi range. This allows a much higher fluctuating stress before the fatigue envelope is exceeded.

Even so, the disadvantage of a thin vertical plate and advantage of an "S" shaped plate can be seen. Steady stress in the vertical plate is low, but the very high fluctuating lateral loading stress approaches the fatigue limit. On the other hand, the moderate steady stress plus the very low lateral loading stress of the "S" plate recommends it for severe lateral loading service.

OPTIMIZATION OF WHEEL PLATE CONTOUR

Comparing the combined stresses with the fatigue strength pointed out certain trends, but the study was taken a step further. A qualitative method of determining the optimum wheel design for specific heating and loading conditions was developed.

This method used more realistic heating and loading conditions and included residual stress data. In an example, heat input was a steady 10,800 Btu/hr, simulating prolonged braking on a long grade (3). This was half of the lowest heat input previously used on the F36 wheels, but was applied for a much longer time. Several track loading conditions were used, including 25,000 to 60,000-lb vertical loads applied at three locations across the tread. These were combined with lateral loads of zero, 5000 lb, and 20,000 lb on the front rim face, and zero and 20,000 lb on the back rim face. Residual stress values were also included in the computer programs for the first time. These averaged about 20,000 psi tension at BFN1 and FFN2.

First, the ratios of the fluctuating stresses to the fatigue limit were determined for the initial contour. This was done not only at BFN1 and FFN2 positions, but also at seven locations on each side of the plate from the rim fillet to the hub fillet. These locations were at rings 8, 9, 10, 11, 12, 13, and 14 shown in Fig.2 and in Table 13. Ratios in excess of 1.0 indicated the fatigue limit was exceeded and fatigue damage could occur. Examples of computed stress ratios at each plate location are shown in Table 13. These were determined from the highest combined stresses for each ring location, regardless of specific cycle. Note that the ratios greatly exceeded 1.0 at ring 9 on the flange side (BFN1) and rings 12 and 13 on the outside (FFN2). These are the critical plate positions observed throughout this study.

The aim of this optimization approach was to lower all fatigue damage ratios to below 1.0, thus eliminating critical stress locations. The next step was to change the contour by adding metal at the plate locations where the highest stress ratios occurred, and removing metal where the lowest ratios occurred. Several successive computations were made until the final contour produced fatigue damage ratios which were below 1.0 in all positions, Table 13. The optimum plate contour suggested for these particular braking and loading conditions had a relatively thick plate. It is strongly emphasized that an entirely different plate configuration could have been obtained under other combined loading conditions.

Prior test data as well as data from this computer study indicate both plate slope and plate thickness have an effect on the rim relaxation and stress buildup involved in propagating thermal cracks through the rim and into the plate. A greater plate slope seems beneficial for both thermal crack and fatigue crack resistance. However, increasing the plate thickness can be detrimental as demonstrated by University of Illinois tests (4). Thicker plates had poorer thermal crack failure resistance, but better fatigue failure resistance. Any overall design optimization should take this into consideration and evaluate the relative hazards.

While some data exists (5-10), additional data relating to the magnitudes of various service loads and their relative frequencies of application are urgently needed to make the fullest use of design optimization for fatigue failure resistance: (a) the maximum lateral loads imposed by car retarders and special track work should be measured together with the vertical track loads at the moment of application, (b) thermal loads developed in prolonged drag (speed control) braking and in short duration (emergency) braking should be determined together with typical vertical and lateral loads applied to the wheel in the heated condition, and (c) lateral and vertical loads under a variety of equipment and operating conditions should be measured. It is believed this information would be best obtained through the Association of American Railroads.

The analytical stress data developed by this computer program have been validated for certain conditions. However, no design changes should be made until the superiority of a specific wheel configuration has been demonstrated by laboratory and service tests.

## SUMMARY


1 Computer analyses were made of several different A40, F36, and B33 wheels with different plate contours using a variety of simulated braking conditions and track loads. It was apparent there was no "one best" plate contour for all possible service conditions, but several design variations showed promise for improved resistance to wheel plate fatigue. (a) Increasing the plate slope from the median reduced loading stresses 20 to 40 percent, with no significant rise in heating stresses. (b) An "S" shaped plate contour reduced both heating and loading stresses by at least 20 percent. (c) Increasing the plate thickness 1/16 in. over the minimum reduced the loading stresses 30 to 60 percent and the heating stresses 10 to 25 percent. Such a change could have a detrimental effect on thermal crack propagation, and should not be considered where thermal cracking is prevalent.

2 A method was proposed for optimizing wheel plate contours for specific service conditions to safeguard against fatigue damage of wheel plates.

3 It was evident that different optimum plate configurations could have resulted from other combined loading conditions. Therefore, actual service braking and loading data must be obtained to utilize the techniques developed by this study.


## ACKNOWLEDGMENTS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# Brake Rigging Efficiency of Railway Freight Cars

**R. W. CARMAN**

THE OPERATING PROBLEM

Southern Railway has a unique operating problem, in that it runs through trains over a route with one of the steepest mainline railroad grades in the United States. Located 24 miles southeast of Asheville, North Carolina (Fig. 1) near the small town of Saluda, the line descends 600 ft in a distance of less than four miles, and with a grade in excess of 4 percent over most of this distance. This route carries heavy traffic, as it is a main connecting link for traffic originating in the midwest and destined to the eastern sea coast in addition to coal, pulpwood, paper, and other local traffic between eastern Tennessee and the Carolinas.

This grade has always been an operating problem and, for that reason, is equipped with special safety devices, such as a runaway track at the bottom of the 4 percent grade with a switch controlled by an electrical timing mechanism such that if a train arrives at the bottom of the grade earlier than a predetermined time period after leaving the top, it will be diverted into the runaway track and not continue down the mainline with possible hazard to the communities which are in that area.

This operating problem, however, has always been somewhat alleviated by the fact that eastbound general merchandise trains usually have a considerable number of empty box cars, pulpwood cars, or other equipment which contribute a large amount of braking in relation to their gross weight such that the whole train can be handled satisfactorily with a combination of dynamic breaking and normal air braking.

In 1960, Southern Railway purchased a group of 750 aluminum gondolas for rotary dump coal service which were to form one of the first unit trains in the United States (Fig. 2). Because of the light weight of these cars, it was necessary to equip them with load-empty brakes (Table 1), as the AAR braking specification requirements could not be met using single capacity brakes.

This was fortunate because in later years as unit trains were made up which operated between

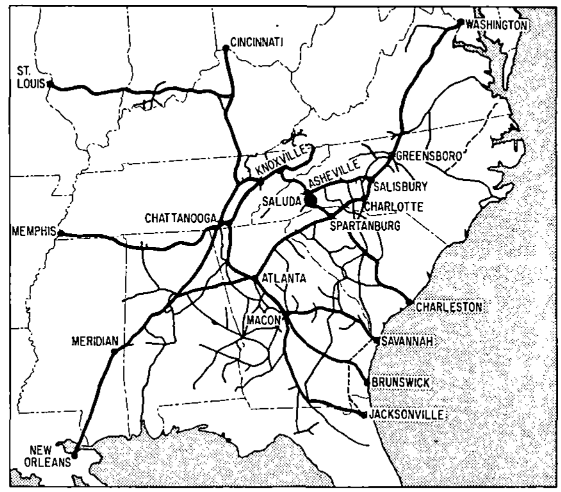


Fig. 1 Line map of Southern Railway system

the coal region in western Virginia and eastern Kentucky into the Carolinas, it was necessary to operate solid trains of these cars on this route, and the load empty brakes provided superior braking capability.

Between 1964 and 1966, Southern Railway purchased 1600 open-top, 100-ton coal hoppers to replace an aging fleet of 50-ton cars. However, because of their higher light weight, it was deemed unnecessary to equip these cars with load-empty brakes as the standard single capacity brakes were adequate, so these cars were equipped with truck-mounted single capacity brakes with composition brake shoes (Table 1). Then in 1969, Southern went into the 100-ton hopper market again and purchased 600 open-top cars equipped with composition brake shoes and foundation brake rigging (Table 1). The cars built in 1964 and 1966, because of the relatively large cubic capacity for a coal car, 4000 cu ft, became known on the railroad as "Big Reds" (Fig. 3) while the 1969 built cars, which had 10 percent less cubic capacity, have become known as "Little Big Reds" (Fig. 4).

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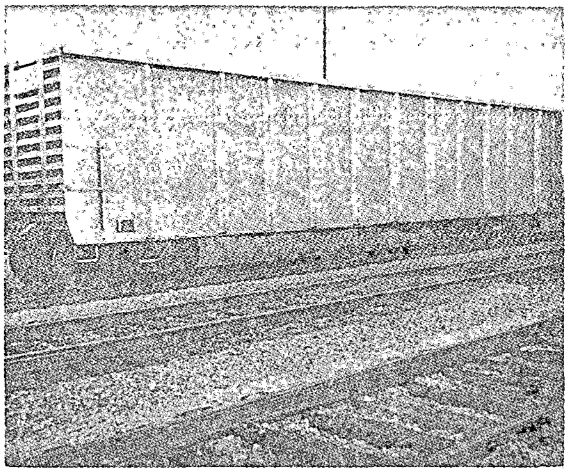

Fig. 2 100-ton "Silversides" aluminum gondola with load-empty brake, conventional rigging, composition brake shoes

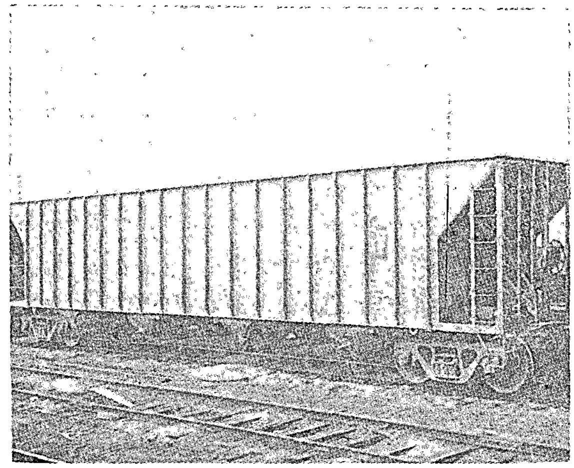

Fig. 4 100-ton "Little Big Red" hopper with conventional rigging, composition brake shoes

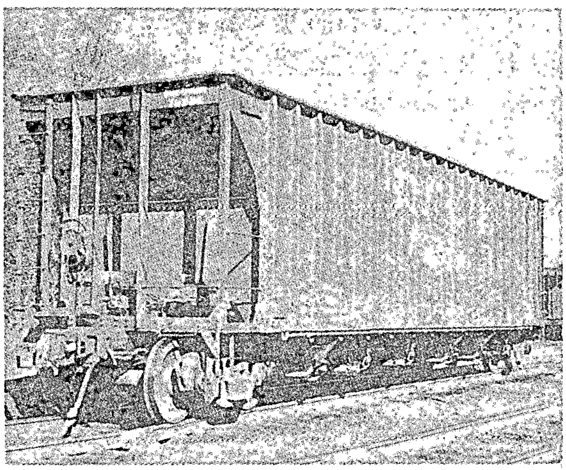

Fig. 3 100-ton "Big Red" hopper with WABCOPAC brake, composition brake shoes

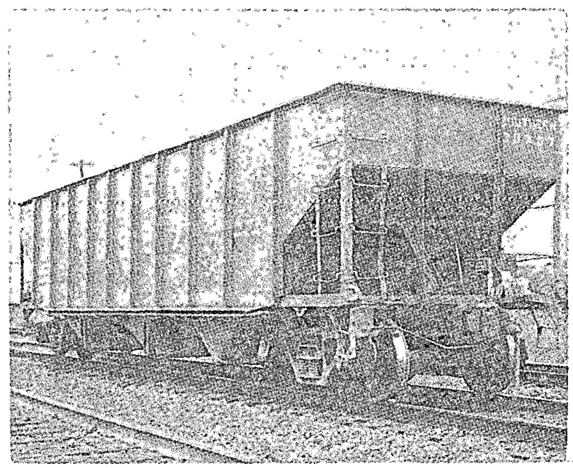

Fig. 5 70-ton hopper with conventional rigging, cast-iron brake shoes

As the unit train concept continued to grow, it became necessary that these later built cars be used in unit train service, including those trains operating over the grade at Saluda Mountain. Operating experience showed that there was some lack of control in handling a train which had large number of these cars in the train, however, so a restriction was temporarily placed on the number of these cars which could be handled together in a train. A dynamometer test was made by Westinghouse Air Brake Company which simulated operation of "Big Red" cars down this grade and showed that the cars could be safely handled. However, the management remained skeptical about the result of this work, and it was finally decided to run an instrumented single car test on the grade to determine whether these cars could be stopped using air brakes alone.

DEVELOPMENT OF TEST PROGRAM

The test program, which was formulated to

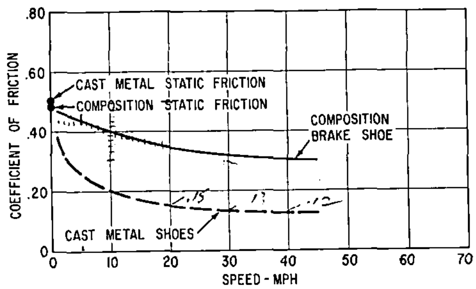


Fig. 6 Coefficient of friction versus speed[1]

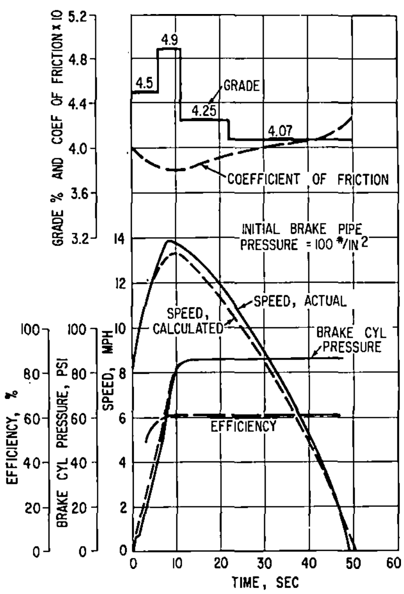


Fig. 7 100-ton gondola: actual versus simulated braking performance (actual speed[2])

determine the braking capacity of coal-handling freight cars, included the following cars:

1 The 100-ton "Big Red" hopper cars built in 1964 and 1966, equipped with truck-mounted brakes and composition brake shoes: Testing of this car was a primary objective of the test at the inception of the test program.

2 100-ton aluminum "Silversides" gondolas: These cars were tested because they were known to operate very satisfactorily, and this car was to be used as a standard of measurement for the other test cars. Equipped with conventional brake rigging, load-empty brake, and composition brake shoes, Table 1.

3 100-ton "Little Big Red" hopper cars built in 1969: These cars were just recently built at the time of the test program and had

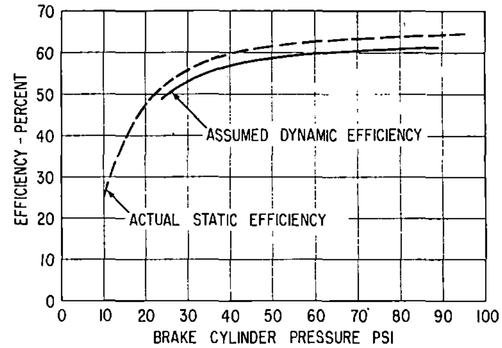


Fig. 8 100-ton gondola: static versus dynamic efficiency (actual static efficiency[2])

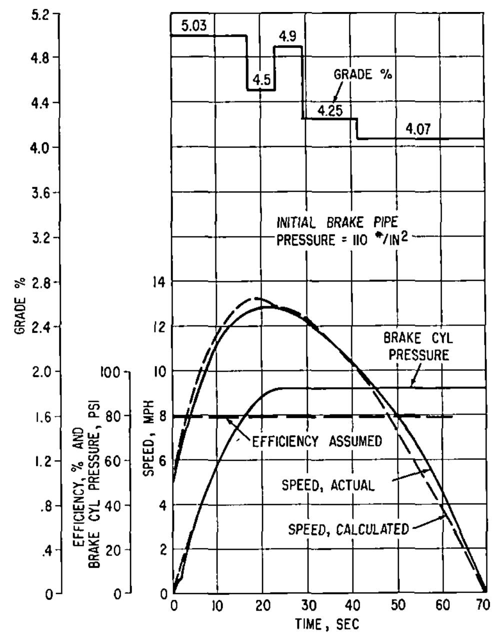


Fig. 9 100-ton "Big Red" hopper: actual and simulated braking performance (actual speed[2])

very little service experience at the time; however, it was felt that they probably braked satisfactorily. It was decided to test these cars to have a measure of foundation truck rigging performance versus that of the truck-mounted brake. Cars equipped with conventional brake rigging and composition brake shoes, Table 1.

4 70-ton hopper cars built in 1957 (Fig. 5): One purpose in including this car in the test was to determine the performance of cars equipped with cast iron brake shoes Table 1, to determine if moderate-sized freight equipment built prior to 1960 with cast iron shoes and foundation rigging

Table 1 Freight Car Mechanical Data

| | 100-ton gondola AB | 100-ton "Big Red" hopper ABD | 100-ton "Little Big Red" hopper ABD | 70-ton hopper AB |
|---|---|---|---|---|
| Type brake Equipment | Load-empty | Single capacity | Single capacity | Single capacity |
| Brake rigging | Conventional | WABCOPAC | Conventional | Conventional |
| Brake shoes | Composition | Composition | Composition | Cast metal |
| Brake cylinder, in. | 7 5/8 - 12 | 8 1/2 (4) | 10 | 10 |
| Lever ratio | 5.82:1 | 2:1 | 6.80:1 | 9.85:1 |
| Cubic capacity, cu ft | 3620 | 4000 | 3600 | 2678 |
| Nominal capacity, lb | 215,000 | 195,000 | 198,500 | 167,000 |
| Average light weight, lb | 48,000 | 68,000 | 64,500 | 53,000 |
| Test gross weight, lb | 268,000 | 263,000 | 259,700 | 219,600 |

performed as good or better than modern day heavy weight equipment with complex brake rigging.

The test series was to include the following separate tests:

1 A static dynamometer brake show test to determine the static brake rigging efficiency.

2 Level track stopping test: This test consisted of pushing the car to predetermined speed with a switch engine and uncoupling and then making an emergency brake application. The rate of brake cylinder pressure buildup and speed as a function of time were recorded. These tests made it possible to calculate some preliminary dynamic brake rigging efficiencies prior to going ahead with the actual grade tests and helped

Table 2 Comparison of Static and Dynamic Efficiency

| Type Car | Brake cylinder pressure, psi | Static Efficiency, percent | Dynamic Efficiency, percent | Percent Difference |
|---|---|---|---|---|
| 100-ton gondola | 86 | 64 | 61 | - 4.7 |
| 100-ton "Big Red" hopper | 92 | 84 | 79 | - 6.0 |
| 100-ton "Little Big Red" hopper | 95 | 41 | 53.5 | +30.5 |
| 70-ton hopper | 93 | 66.5 | 54.5 | -18.0 |

formulate decisions as to the method of testing the cars on the 4 percent grade.

3 Four percent grade test: These tests were performed by pulling the cars up a predetermined distance on the 4 percent grade at Saluda, North Carolina, releasing them from a switch engine, and making an emergency brake application. The brake cylinder pressure buildup rate and speed versus time were recorded.

METHOD OF DETERMINING DYNAMIC BRAKE RIGGING EFFICIENCY

It was anticipated at the outset of this program that one of the primary results which could be obtained from such a test would be a determination of the dynamic brake rigging efficiency for the various cars involved, as this was the only real unknown which prevented making an engineering estimate of the performance of the cars prior to the test. The relation of coefficient of friction to speed[1] has been well documented and is known to vary as shown in Fig. 6. Although this can vary with temperature and is somewhat dependent upon brake shoe force, these figures can be used for temperatures up to 150 deg and for brake shoe forces in the normal operating range. Knowing the grade, the rolling resistance of the car on straight track which is estimated to be 6 lb/ton of gross weight, the brake cylinder pressure and the lever ratio, it is then possible to calculate the net retarding force or accelerating force acting on the car

[1] Brenneman, E. B., "Static Friction," Westinghouse Air Brake Division, Westinghouse Air Brake Co., Fig. 1.

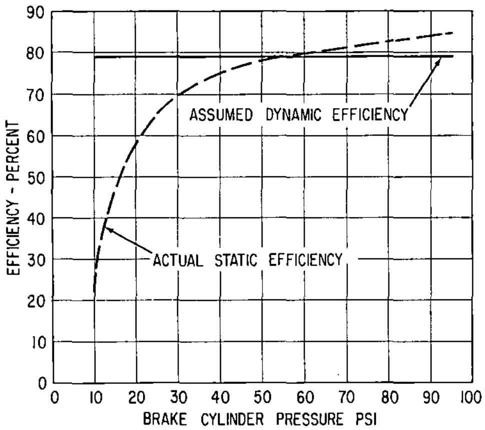


Fig. 10 100-ton "Big Red" hopper: static and dynamic efficiency (actual static efficiency[2])

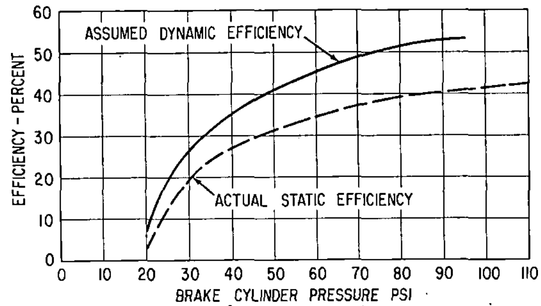


Fig. 12 100-ton "Little Big Red" hopper: static and dynamic efficiency (actual static efficiency[2])

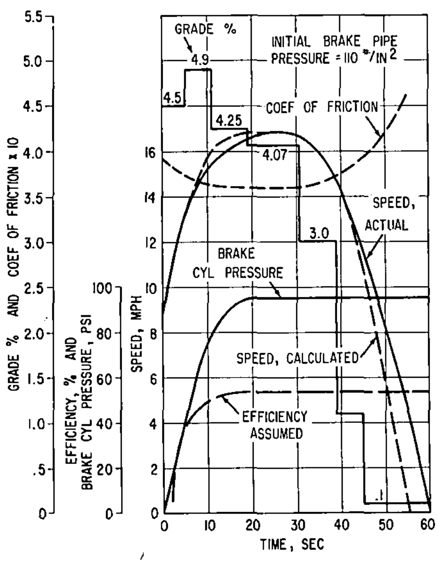


Fig. 11 100-ton "Little Big Red" hopper: actual and simulated braking performance (actual speed[2])

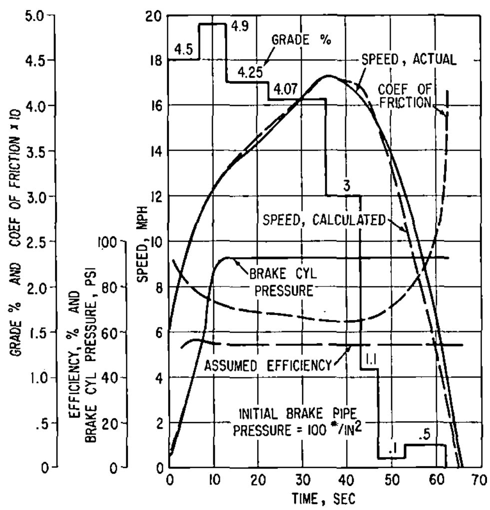


Fig. 13 70-ton hopper: actual and simulated braking performance (actual speed[2])

and determine whether the car will stop or accelerate if the dynamic efficiency is known. To date, no one has been successful in measuring the dynamic brake rigging efficiency. In order to do this, it would be necessary to have a dynamometer brake shoe which could be used in a moving car, and this is not presently available. It was, therefore, one of the prime purposes of this test to calculate the dynamic brake rigging efficiency and, if possible, correlate this with the measured static brake efficiencies.

The equations for motion for single freight car under braking conditions are as follows:

$$F_a = F_g - F_r - F_b \tag{1}$$

where

$F_a$ = net accelerating (retarding) force

$F_g$ = force due to gravity

$F_r$ = rolling resistance

$F_b$ = braking force

$$F_g = W \times G \tag{2}$$

$$F_r = 6 \text{ lb} \times W/2000 = 0.003 \times W \tag{3}$$

$$F_b = P \times A \times E \times C \times R \tag{4}$$

where

G = grade, percent

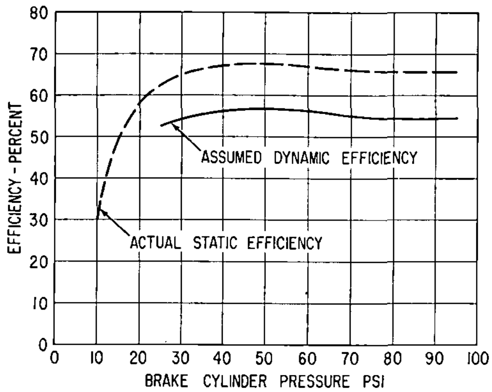


Fig. 14 70-ton hopper: static and dynamic brake rigging efficiency (actual static efficiency[2])

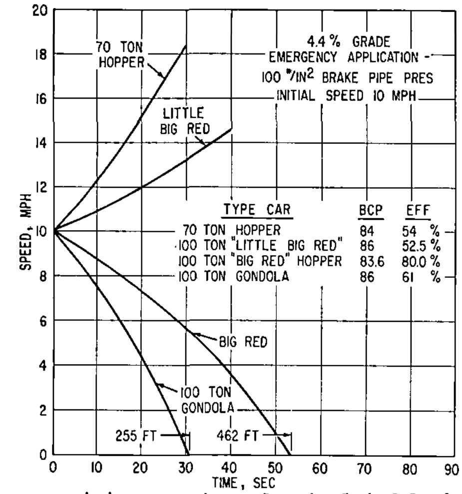


Fig. 15 4.4 percent grade simulated braking performance

W = gross weight, lb
P = brake cylinder pressure, psi
E = brake rigging efficiency
C = coefficient of friction
R = brake lever ratio
A = brake cylinder area, psi

Therefore,

$$F_a = W \times G - 0.003 \times W - P \times A \times E \times C \times R \qquad (5)$$

The acceleration of the car is then:

$$A = \frac{Fa}{W} \times g \qquad (6)$$

$$g = 32.2 \text{ ft/sec}^2 \qquad (7)$$

For a small time increment, t, in which retarding force, and thus the acceleration, remain constant, the change in velocity

$$V_1 - V_0 = A \times t \qquad (8)$$

and the distance traveled given by:

$$D = \frac{(V_1 + V_0)}{2} \times t \qquad (9)$$

To analyze this data, a computer program was used which makes a calculation of the net retarding force every 1/10 of a second, and then determines the velocity and distance traveled per the foregoing equations. The following relationships are taken into account:

1 Grade: An actual railroad grade is not constant, but varies. This is taken into account by including in a computer program a table which gives a percent grade and distance for each location where the grade changes.

2 Brake cylinder pressure: If that part of the test is to be included which includes the application of the brakes, it is necessary to include sufficient data in the program to calculate the brake cylinder pressure of the function of time. This is a polynomial equation of the form

$$P = K_o + K_1 \times T + K_2 \times T^2 + K_3 \times T^3 \; .... + K_n \times T^n \qquad (10)$$

The coefficients ($K_o$, $K_1$, $K_2$, etc.) and the number of terms used in this equation are determined by a separate computer program for polynomial curve fitting which rapidly gives the coefficients and number of terms which provide the best fit. For functions which do not have abrupt changes, it is usually possible to fit a curve in this manner and duplicate the original results within an accuracy of 1 percent. Likewise, the coefficient of friction as a function of speed is expressed as a polynomial equation.

Since the efficiency remains the only variable which is not known, the project consists of finding the efficiency value which would most closely provide speed and distance versus time results similar to the actual test data. This was done by trying two different assumptions as to the relationship of efficiency to brake cylinder pressure:

1 That the efficiency was uniform, i.e., independent of brake cylinder pressure
2 That the efficiency was dependent on brake cylinder pressure with this relationship

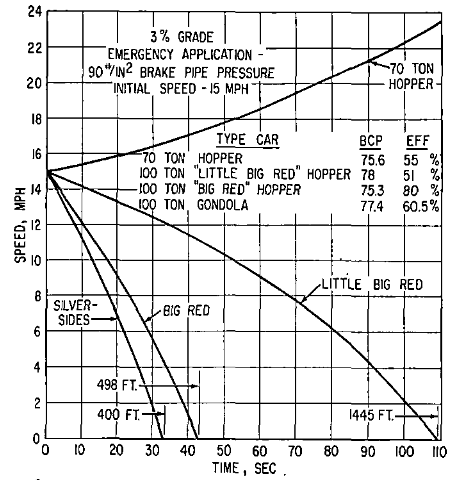


Fig. 16 3 percent grade simulated braking performance

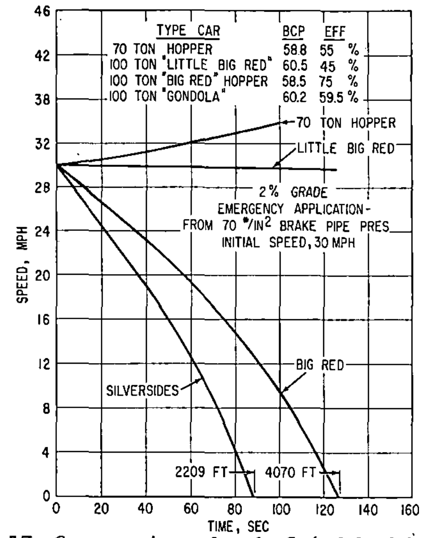


Fig. 17 2 percent grade simulated braking performance

being established by the static dynamometer brake shoe test.

RESULTS OF DYNAMIC BRAKE RIGGING EFFICIENCY CALCULATIONS

The results of this simulation technique are as follows:

1 100-Ton aluminum gondola: Fig. 7 illustrates the computer simulation, speed versus time results shown in dashed lines and the actual test results shown in solid line (actual speed[2]). The dynamic brake rigging efficiency which resulted in most closely simulating the actual test results was 61 percent at 86 lb brake cylinder pressure. This compares with a measured static brake rigging efficiency of 64 percent at 86 lb brake cylinder pressure. It was assumed that at lower pressures, the dynamic efficiency was likewise reduced amount proportional to the difference at 86 lb. The comparison of the measured static and assumed dynamic brake rigging efficiencies is shown in Fig. 8.

2 100-Ton "Big Red" hopper car: The dynamic brake cylinder efficiency, which was found to most closely simulate the actual braking performance, was 79 percent compared to 84 percent measured at under static test. In this case, it was found that better correlation was found in the computer simulation if the efficiency was assumed to be independent of brake cylinder pressure [Fig. 9 (actual speed[2])]. The static brake rigging efficiency versus brake cylinder pressure is shown in Fig. 10 (actual static efficiency[2]).

3 100-Ton "Little Big Red" hopper car: The dynamic brake efficiency of 53.5 percent at 95 psi brake cylinder pressure resulted in closely simulating the actual test, Fig. 11 (actual speed[2]). This compares with a static rigging efficiency of 42 percent. It was assumed that the variation of brake rigging efficiency with brake cylinder pressure was proportional to that variation under the static conditions [Fig. 12 (actual static efficiency[2])].

4 70-Ton hopper cars: A dynamic efficiency of 54 percent at psi brake cylinder pressure most closely approximated the actual test conditions [Fig. 13 (actual speed[2])]. This compares with a measured static brake rigging efficiency of 66 percent, as shown in Fig. 14 (actual static efficiency[2]).

SIMULATING CONSTANT GRADE OPERATION

As the final part of the analytical work in this project, a simulation of the operation of these cars on constant grades with constant brake cylinder pressures was made.

The reasons for this are twofold:

1 Although these tests were made on the actual grade which was of concern, the test conclusions were vague as to whether some of the test cars could be stopped under actual operating conditions. This was because 110-lb brake pipe

[2] "Braking Performance Tests," Westinghouse Air Brake Division Report No. E-B3098-1, pp. T-6473. 4-1A; 4-2C, -3C, -4C; 14-3A; 14-1B, -4B.

pressure was used to test some of the cars and in some cases the car speed exceeded the speed limit on the grade before full brake cylinder pressure was reached. Therefore, using the dynamic brake rigging efficiency calculated in simulating the actual performance on the grade, a simulation was made of a 4.4 percent grade, with brake cylinder pressure resulting from an emergency application with a brake pipe pressure of 100 psi and from an initial speed of 10 mph.

2 The tested performance of the cars raised a question of the adequacy of the car braking for less severe grades. Although the 4.4 percent grade is unique, the practice of operating at 10 mph and using 100-psi brake pipe pressure tends to compensate for the severity of the grade. It was, therefore, decided to simulate braking on a 3 percent grade with a brake pipe pressure of 90 psi and an initial speed of 15 mph, and a 2 percent grade with a brake pipe pressure of 70 psi and a speed of 30 mph.

The results of these simulations are as follows:

1 4.4 Percent Grade — 100-lb brake pipe pressure — initial speed 10 mph (Fig. 15): The 100-ton gondola and the 100-ton "Big Red" hopper can be safely stopped on this grade from the initial speed of 10 mph. The 100-ton "Little Big Red" hopper car and the 70-ton hopper would fail to stop and would continue to accelerate.

2 3 Percent Grade — 90 lb brake pipe pressure — initial speed, 15 mph (Fig. 16): The 100-ton gondola and the 100-ton "Big Red" hopper can be safely stopped under these conditions. The "Little Big Red" car is marginal, requiring over 1/4 mile to stop. The 70-ton hopper will not stop and continues to accelerate.

3 2 Percent Grade — 70-lb brake pipe pressure — initial speed, 30 mph (Fig. 17): The 100-ton gondola and the 100-ton "Big Red" hopper car can be stopped under these conditions. The "Little Big Red" hopper just balances the grade, losing an insignificant amount of speed. The 70-ton hopper cannot be stopped and continues to accelerate.

## CONCLUSIONS

### Correlation between Static and Dynamic Brake Rigging Efficiency

Table 2 is a comparison of static and dynamic efficiencies at the brake cylinder pressures used in the various grade tests. Correlation between the static efficiency and dynamic efficiencies is good for the 100-ton gondola and the 100-ton "Big Red" hopper with truck-mounted brake, but poor for the 70-ton hopper with cast iron shoes and the 100-ton "Little Big Red" hopper with foundation brake rigging and composition shoes.

### Need for A Dynamic Dynamometer Brake Shoe

There is a definite need for a device which can measure brake shoe forces under moving conditions so that dynamic efficiencies can be directly measured. While the conclusions of this study show that for most cars the dynamic efficiency is probably in the same range as the static efficiency, it also raises the suspicion that there can be substantial variation between the static and dynamic efficiency such that problems may occasionally arise as a result of using static brake rigging efficiencies for the design of brake ratios.

### Variation of Static Brake Rigging Efficiency between Similar Cars

The static dynamometer brake shoe test, made as a part of this project and made on other occasions, has shown substantial variation in car to car variability in the measurement of static forces. Some of this may be due to limitations in the accuracy of the potentiometers, strain gages, and other instrumentation, while variations in tolerances between brake shoes and brake beams and truck sides, pin tolerances, jaw dimensions, and variations in fit-up which result in different lever angularities and clearances, may contribute to the variation also. There is a need for additional testing to isolate the variation caused by each of these sources of variation so that more consistent performance can be obtained from the braking system.

### Capability of Car Stopping on Grade

Freight cars being built today, as well as cars built over the last 10 to 20 years, may be incapable of being stopped by their own braking system when loaded to full capacity under current speed limit and brake pipe pressure operating conditions. This is true not only on the 4.4 percent grade conditions discussed in this paper, but on more common grades of 2 to 3 percent as well. Actual testing on grades in the 2 to 3 percent range is suggested to confirm this.

### AAR Braking Ratio Specifications

There is a need to rewrite the AAR braking ratio specification so as to:

1 Take into account the brake rigging efficiency. Until such time as a means is develop-

ed to measure the dynamic brake rigging efficiency, the static brake rigging efficiency should be used.

2 Increase the braking force on loaded cars. Present regulations allow single capacity brakes to be applied on 100-ton cars as light as 53,500 lb. One result would be to raise the car light weight on which single capacity brakes can be used.

Recommendations which will put into effect the foregoing specification changes have been proposed as a letter ballot item and will go into effect in 1972 if approved.

Correlation with Dynamometer Test

While no direct correlation was made between the dynamometer test of the 100-ton "Big Red" hopper car with Wabcopac and composition shoes mentioned earlier in this paper, it appears that the grade test performed at Saluda, North Carolina, confirmed the conclusions of the dynamometer test that these cars could be safely handled.

Other Operating Practices

The foregoing conclusions are based on brake applications made without the use of retainer valves, and does not preclude the possibility that satisfactory braking can be obtained under any of the foregoing conditions by the use of retainer valves in conjunction with repeated application and release cycles to obtain higher brake cylinder pressures than would be obtained from applications made with zero initial brake cylinder pressure.

ACKNOWLEDGMENTS

The author wishes to acknowledge the assistance and cooperation of the Westinghouse Air Brake Company and the Railroad Friction Products Corporation, who provided instrumentation and personnel to perform the static and grade tests and reduced the data obtained during these tests to a usable form.

# Development of Fatigue Standards for Freight Car Truck Components and Wheels

M. R. JOHNSON

INTRODUCTION

The fatigue failure of freight car truck components causes a significant number of train derailments. The number of these failures could be reduced by the utilization of fatigue performance standards, which were designed to achieve a desired level of operational reliability.

At the present time the only freight car truck components which must be designed to satisfy fatigue requirements are side frames and roller bearings, and these criteria need to be updated to reflect specific reliability goals and the actual loads encountered in service. Truck bolsters, wheels, and axles are not required to pass fatigue test specifications although fatigue tests have influenced present standard axle designs.

In this paper the factors which must be considered in the development of fatigue performance standards which will provide a desired reliability against fatigue failure are discussed. Also presented are typical load spectra describing the freight car truck fluctuating load environment on which such standards would be based.

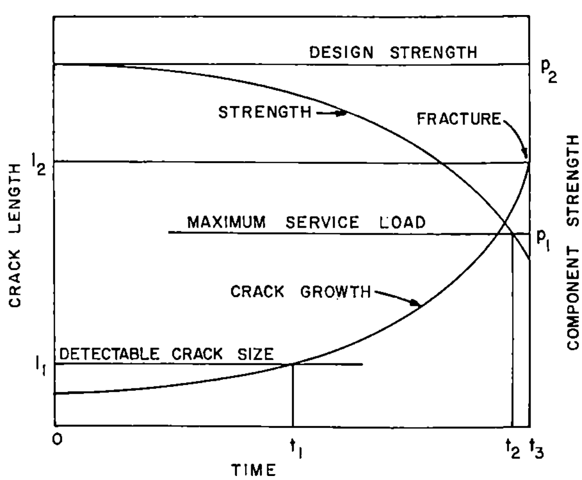


Fig. 1 Fatigue crack growth

ESTABLISHMENT OF RELIABILITY GOALS

System Reliability

The basis for the establishment of fatigue standards for freight car truck components and wheels would be the desired level of safe operations. Standards for individual components must be based on the reliability of the entire truck. The present reliability for freight car truck operations is indicated by Federal Railroad Administration accident statistics, which show that in 1971, 942 of 5131 train derailments were attributed to the failure of truck components and wheels. In this period 29.2 billion freight car miles were operated. Considering each truck of the car as an independent system, this is a failure rate of $1.6 \times 10^{-8}$ per truck mile. Fatigue failure, which is the cause of only a portion of these derailments, would have a lower failure rate.

Reliability goals for individual components would have to be greater than for the system as a whole. For example, if the truck were idealized as consisting of two side frames and a bolster, and if the reliability of each of these components was 0.9999 over their lifetime, then the truck has an overall reliability of 0.9997.

Inspection Interval

The fatigue requirements implied by a reliability goal will be influenced by the frequency and quality of service inspections. This is illustrated by Fig. 1, where the crack length and strength of a representative structural element are plotted as a function of time. The structural element is assumed subjected to a fluctuating load which is stationary in time. The figure indicates the growth of a typical flaw with time. The structure initially would contain microscopic flaws, which are an inherent material property or the result of manufacturing processes. As the component is subjected to a time varying load, a typical flaw would increase in size. At time $t_1$ the crack length reaches the threshold of detectability, and as the crack length increases, its growth is accelerated. Finally, it

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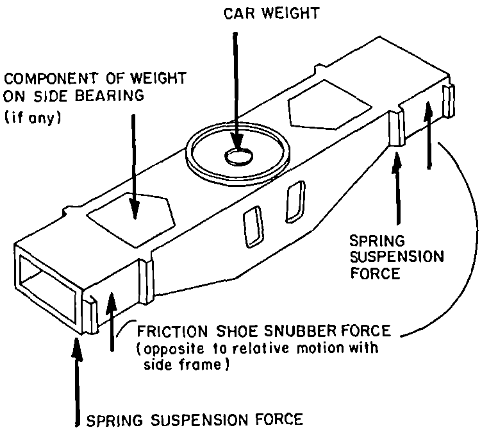

Fig. 2 Bolster vertical loads

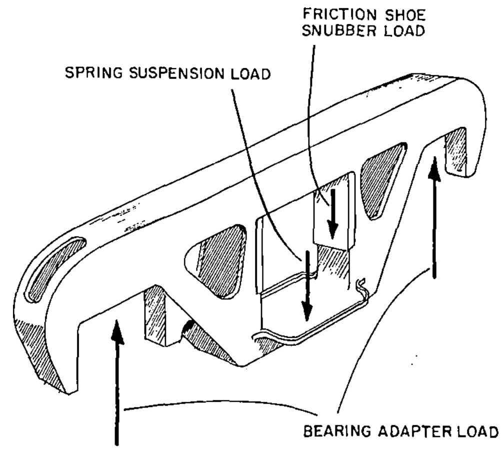

Fig. 3 Side frame vertical loads

is unable to sustain the load and fracture occurs at $t_3$.

The decrease in strength of the element which accompanies crack growth is also plotted in Fig. 1. The ratio of the initial design strength, $p_2$, to the maximum expected service load, $p_1$, represents the static design safety factor. A decrease in strength accompanies the crack growth as illustrated in the figure. At time $t_2$, the part is no longer able to sustain the maximum anticipated service load. Failure of the part would follow shortly thereafter.

If a component has to operate in a region of time greater than $t_1$, it is obvious that the minimum inspection interval should be less than $t_2$-$t_1$. Also note that improved inspection techniques, which lower the threshold size of detectable flaws, would allow an increased interval in the period between inspections.

### Fail-Safe or Safe-Life Design

Fatigue standards which are based on a specific reliability goal will be influenced by whether or not the structural design philosophy is fail-safe or safe-life. Fail-safe design relies on utilization of redundant structural members. The probability of a catastrophic failure is small because failure requires simultaneous defects in each member. Low probabilities of failure can be developed even though the separate members possess only moderate fatigue properties. Fail-safe structures are commonly allowed to operate in the region of detectable cracks, as illustrated in Fig. 1. There are only a few examples of fail-safe design principles in the freight car truck. One of these is the truck-bolster/body-bolster connection where the center plate rim and the center pin provide redundant structural elements in the connection between the two bolsters.

Safe-life design requires maintaining the integrity of each structural element. Structural components must be designed for an extremely low probability for failure during intervals between inspections. Conventional freight car truck construction is based primarily on safe-life principles. Failure of one of the major structural elements, such as the side frames, bolster, wheels, or axles, inevitably leads to derailment. Safe-life structures are generally restricted to operate in the region where no detectable cracks will occur.

## SERVICE LOAD ENVIRONMENT

The establishment of fatigue standards requires an adequate understanding of the freight car truck operational load environment and the influence of truck design and operational parameters on this environment.

### Vertical Load

Vertical load is a result of the transfer of the weight of the car and lading to the rails. It is characterized by an average value representing the car weight and fluctuations about this level.

Vertical loads acting on the bolster are illustrated in Fig. 2. The line-of-action of the weight of the car is normally at the center of the bolster, but this can be shifted because of load transfer to one of the side bearings. The vertical load on a side frame is applied

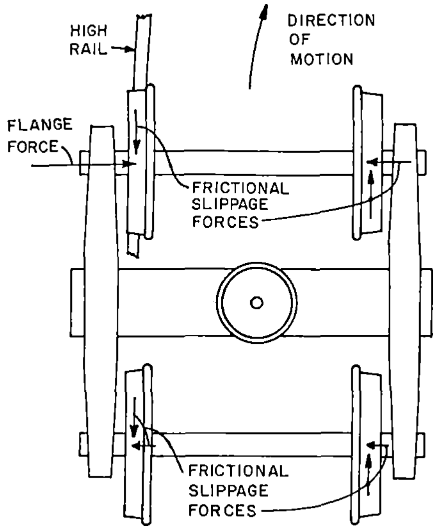


Fig. 4 Curved track freight car truck forces at wheel/rail interface

between the base of the spring suspension and the roller bearing adapters as shown in Fig. 3. The side frame is subjected to higher shock load forces than the bolster, such as those due to crossings or wheel flat spots, because it does not have the benefit or the spring suspension system for the isolation of these loads.

Vertical loads on a wheel/axle set are applied between the journals and the wheel/rail contact points. The offset positions of these loads cause a bending moment which is carried through the axle. Because of the changing orientation of the axle as the wheel rotates, a cycle of stress is produced in the axle once per revolution, and this becomes the primary consideration in the evaluation of fatigue strength.

The vertical load on the wheel produces high stresses in the vicinity of the wheel/rail contact point and this stress pattern is repeated once per wheel revolution. The vertical load also produces fluctuating stresses in the wheel plate as the wheel rotates, but these stress levels are usually low compared to stresses developed from lateral loads at the wheel/rail interface.

Lateral Loads

Lateral loads on truck components are the result of transfer of force from the car body to the rail, or the result of internal truck forces accompanying the traversal of curved track. Lateral force transfer from the car to the rail is due to coupler angularity on curves, speed not commensurate with superelevation on curves, etc. Internal forces are the result of the yawing slippage of wheel/axle sets when operating over curved track.

The forces which act on a railroad car truck on curved track and their relationship to its unique design characteristics (e.g., the rigid interconnection of wheels on the axle, wheel tread profile, flange guidance, etc.) are discussed in works by Porter (1),[1] Minchin (2), Cain (3), and others. These analyses are concerned with quasi-static, steady-state forces acting on the wheels of the truck due to the combined rolling and slippage of the wheels. If a tapered wheel is used, the lateral displacement of the wheel/axle set from the centerline of the track can accommodate some curvature, but a condition of creep or slip must develop to physically maintain the wheel/axle set in the proper orientation if the curvature is above approximately 2 deg.

A typical set of forces acting on a four-wheel truck while traversing curved track is shown in Fig. 4. The forces are those acting on the wheels at the rail in the absence of lateral forces from the car. A "rigid" truck is assumed, which implies that the positions of the wheels are maintained in a fixed rectangular array regardless of the forces acting on the truck. Normally the only point where a wheel flange makes contact with the rail is at the outer wheel of the lead axle. Other conditions are possible, such as the flange of the inner wheel of the trailing axle contacting the rail. This can occur when the degree of curvature is high and the track gage clearance is low. Another possibility is the flange of the outer wheel of the trailing axle contacting the rail. This can take place if the rigidity of the truck is not maintained and it undergoes a skewing deformation.

The analysis of the truck forces indicates that the individual wheel/axle sets undergo both a rolling and a slipping motion. The slipping component of the motion can be described by a yawing rotation of the axle about a fixed point. The location of this point is different with each axle, but in each case it lies on a radial line with respect to the curve which is perpendicular to the axis of the truck. For the rear axle this point lies at the intersection of the truck axis with the radial line, whereas for the front axle the point is located near the inner rail. The forces acting on the basic truck

[1] Numbers in parentheses designate References at end of paper.

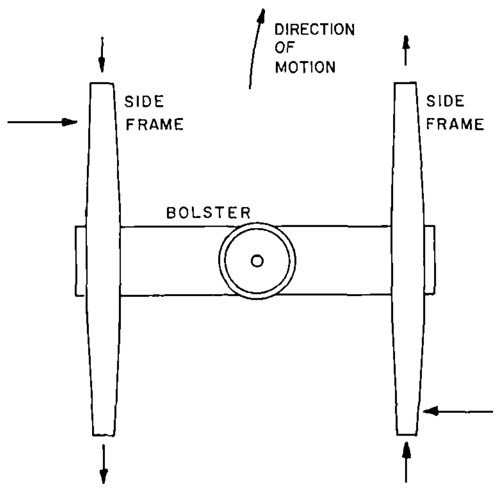


Fig. 5 Curved track forces on truck frame

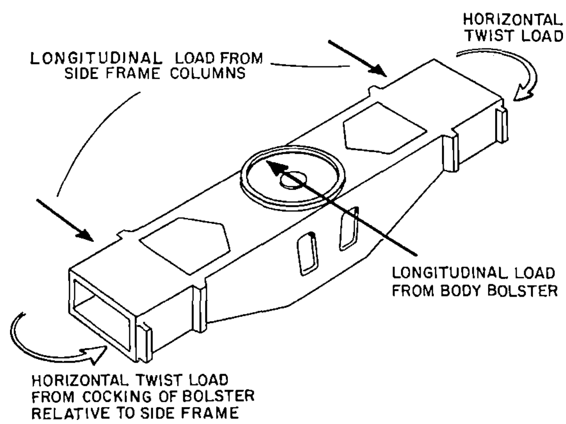


Fig. 6 Bolster longitudinal and horizontal twist loads

frame (here taken to include the side frames and bolster) as a result of the wheel/rail interaction forces illustrated in Fig. 4, is illustrated in Fig. 5.

The effects of internal truck forces due to the traversal of curved track vary from one component to another. The primary effect on the bolster is to induce a horizontal twist load as shown in Fig. 6. The side frame is subjected to lateral load, which is applied between the side frame column and the roller bearing adapters, as illustrated in Fig. 7. Fig. 7 also shows the bolster horizontal twist load applied to the side frame columns.

Lateral loads on the wheel/axle resulting

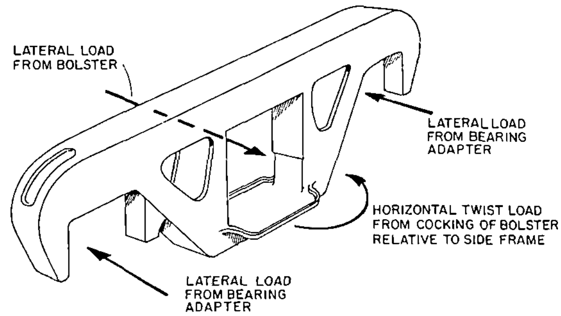


Fig. 7 Side frame lateral and horizontal twist loads

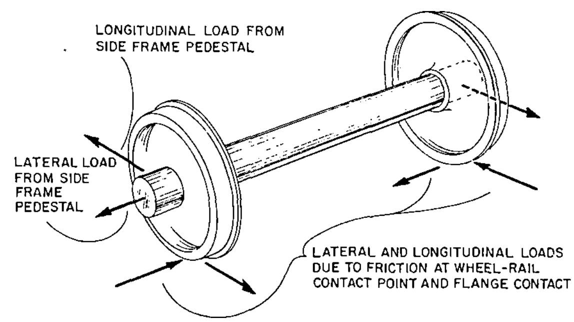


Fig. 8 Wheel-axle lateral and longitudinal loads

from the traversal of curved track are illustrated in Fig. 8. These forces cause plate stresses which alternate once per wheel revolution. This stress pattern is opposite to that caused by the vertical load. In many cases, however, the stresses due to the lateral load are larger than those caused by the vertical load so that they become the controlling stresses in the initiation of fatigue cracks. The front face hub fillet is the most critical area for these cracks.

Longitudinal Loads

As illustrated in Fig. 6, a longitudinal load on the bolster may be applied between the truck-bolster/body-bolster center plate interface and the sides of the bolster mating with the side frame columns. This load is due to braking and inertial forces accompanying acceleration of the truck. The most severe forces occur under car impact conditions when an unloaded car is coupled at high speed.

Longitudinal loads on the side frame are applied between the side frame column at the bolster and the pedestal jaw faces. As illus-

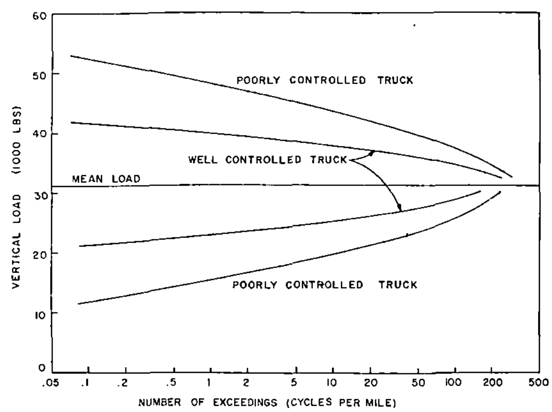


Fig. 9 Load spectra for side-frame/roller-bearing-adapter

trated in Fig. 8, longitudinal loads on the wheel/axle set result in axle torque. This torque is applied to the axle from opposing couples due to longitudinal loads at the wheel/rail interface and at the bearing.

Wheel Thermal Load

Wheels are also subjected to thermal load due to the absorption of energy on the tread of the wheel from tread brakes. The deposition of this energy in the rim leads to high thermal gradients within the wheel. This causes an expansion and twisting of the rim relative to the plate which induces steady plate tension stresses and bending moments in the plate in regions which are highly stressed by lateral loads. The plate becomes more sensitive to fatigue damage at these high stress locations.

Format for Presentation of Fluctuating Load Data

Load data must be presented in a form that is useful for an assessment of the fatigue problem. This requires information on both the fluctuating load intensity and the frequency of application. A study of test data has revealed that the load environment can be characterized by a mean value and a fluctuating component. The mean load may be zero, a value established by the weight of the car, or an operating condition (e.g., the steady component of lateral wheel load on curves). The optimum description of the load environment includes definition of the force levels on each major component. This requires load data at various interfaces between components such as the wheel/rail interface, the side-frame/roller-bearing-adapter interface, the side-frame/spring-group/bolster interface, and the center-plate/side-bearing/truck-bolster interface. Truck design parameters which influence the environment include the spring travel of the suspension system, the type of damping mechanism, the degree of wheel wear, etc. Operational parameters, such as the weight of the car, train speed, track conditions, etc., also influence the load environment.

The fluctuating load data are conveniently summarized by counting the peak loads that are within certain force ranges between crossings of the mean level. This format provides an accurate summary of the load environment and it can be readily used for the specification of fatigue performance tests. The mean-level-peak-crossing data are displayed on a "load spectrum," which is a plot of the peak levels (both positive and negative) of the alternating component of the load versus the number of times the load level is exceeded in a given counting interval.

Examples of Service Load Data

Recent load spectra data have become available which describe the freight car truck load environment as a function of operating conditions and design parameters (4). These data have been obtained by the Bessemer and Lake Erie Railroad using their road test facilities (5). The load data are presented for a 100-ton capacity car (263,000 lb rail load).

Spectra for side-frame-pedestal/roller-bearing-adapter interface load data. The spectra obtained from the measurement of the fluctuating loads at the interface between the side frame pedestals and the roller bearing adapter (SF/BA) are representative of the vertical truck load environment. Data from a number of tests are summarized by the load spectra shown in Fig. 9. The data generally fall into one of two groups as represented by the two spectra shown in this figure. The lower curve shows the most favorable case. It is associated with long travel truck suspension springs (D3 or D5) and a properly-adjusted snubbing system. The worst case is representative of two conditions: an unsnubbed short-travel (D2) spring group or a long-travel spring group with defective springs or a defective snubber.

The data are expressed in terms of cyclic load counts per mile. They represent a typical test track consisting of 0.60 mile of tangent track, 0.25 mile of curved track up to 6 deg curvature, and 0.15 mile of curved track 6 to 10 deg curvature per mile. It is also assumed that there is one turnout every 3 miles.

The SF/BA spectrum represents the full vertical load on a bearing, one-half of the total

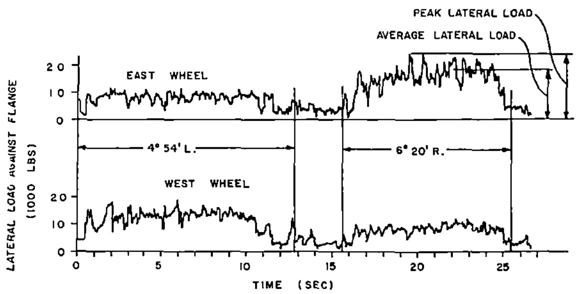


Fig. 10 Lateral wheel loads on lead axle (Northbound Run 35 mph)

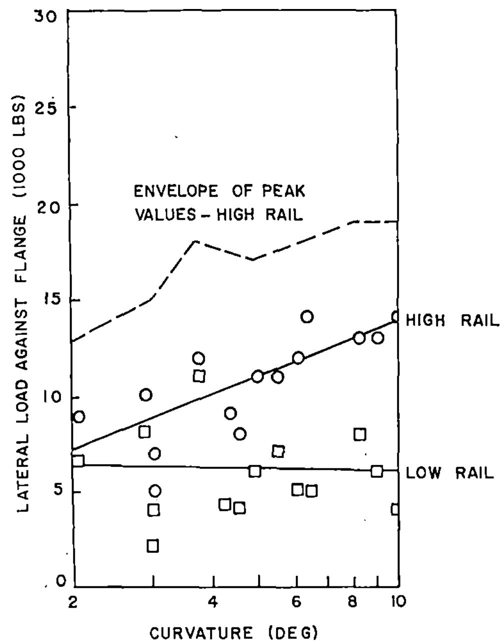


Fig. 11 Lateral wheel load as a function of curvature, 20 ft-6 in. truck centers car

vertical load on the side frame, and one-fourth the total vertical bolster load. Complete definition of the vertical bolster load must also consider that the line of action of the load would be modified by load transfer to the side bearings. The spectrum also represents the vertical wheel load at the wheel/rail interface. Although the instantaneous values of each of the two vertical journal loads are not necessarily identical, this difference is insignificant in the definition of the spectrum for vertical wheel load.

Lateral loads at the wheel/rail interface. The primary effect of lateral load at the wheel/

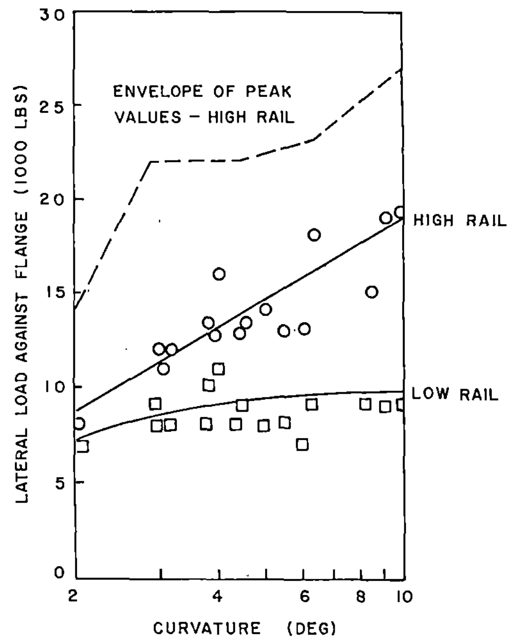


Fig. 12 Lateral wheel load as a function of curvature, 40 ft-9 in. truck centers car

rail interface is to produce alternating stresses in the wheel plate. If there is a net load on a wheel/axle set from the two lateral wheel loads, this load will tend to distort the rectangular configuration of the truck introducing lateral loads on the bearing and side frame and a twist load on the bolster and side frame.

When operating on tangent track, the lateral load at the wheel/rail interface is characterized by an alternating load with the major component directed against the flange and a minor component directed away from the flange. This is due to the normal side-to-side hunting motion of the wheel/axle set.

When traversing curved track, the alternating load component is superimposed on a steady component of load which varies with the wheel position in the truck, the wheel on the high rail of the lead axle usually being subjected to the highest loads. This load is directed against the flange. The wheel on the opposite side of the axle is subjected to similar forces directed against the flange, but of lesser magnitude. The component of steady lateral load on the trailing axle of the truck is usually small in comparison to the lead axle.

Fig. 10 shows typical lateral wheel load test data. The data are for the lead axle of a

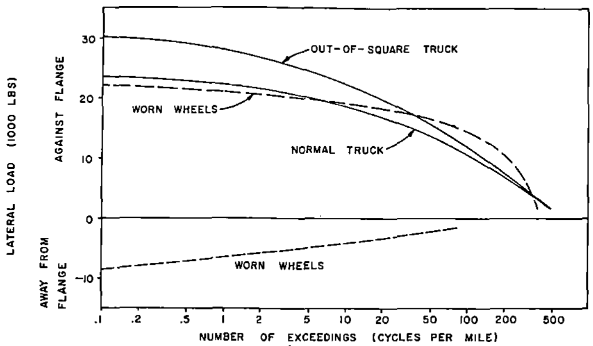


Fig. 13 Lateral wheel load spectra

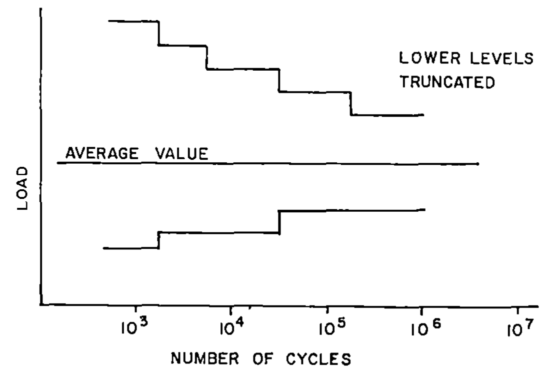


Fig. 14 Step load representation of load spectra

truck moving through a 4°54' left curve and into a 6 deg-20 ft right curve. The figure illustrates the steady and fluctuating components of the load and the larger loads associated with the high rail.

Test results show that the steady component of lateral load is a function of the degree of curvature. This is illustrated in Figs. 11 and 12 where points are plotted indicating the peak average running loads on both the high rail and low rail as a function of degree of curvature. Although there is some scatter in the data, there is an obvious trend showing an increase in force with increasing degree of curvature. Figs. 11 and 12 also show an envelope of the peak lateral load (representing the sum of the steady and fluctuating components) as a function of curvature.

A major parameter influencing lateral load at the wheel/rail interface is the car length, the lateral loads increasing with the truck center distance. This is evident from a comparison of Fig. 11, which is for a 20-ft, 6-in. truck center distance car, and Fig. 12, which is for a 40-ft, 9-in. truck center distance car.

Load spectra can be developed for lateral wheel load data which take into account the rotation of the wheel. That is, a steady load acting against the flange will produce one cycle of load for every revolution of the wheel. Typical spectra are shown in Fig. 13 representing three conditions. These spectra are plotted in terms of cycles per mile and assume 0.60-mile tangent track, 0.25-mile curved track up to 6 deg, and 0.15-mile curved track 6 to 10 deg curvature per mile. The spectra show that allowing a truck to be distorted into an out-of-square configuration can lead to higher lateral loads. They also show that a worn wheel contour can lead to forces away from the flange.

FATIGUE PERFORMANCE CRITERIA

Fatigue performance criteria would consist of two principal elements: (a) a description of the test load including the maximum and minimum values of the alternating load, the relative number of cycles at various load ranges, and the manner in which the load is to be applied; (b) a fatigue-life factor-of-safety which states the minimum acceptable ratio of the lifetime number of cycles of the part under test to the number of cycles representing the operating environment. The factor of safety with respect to fatigue life would be based on these considerations:

- The desired reliability of operation
- The statistical distribution of the load spectrum reflecting different conditions of truck operation
- The statistical distribution of the experimentally determined fatigue strength of individual truck components
- The number of specimens used to evaluate fatigue strength.

Cyclic Loading Procedure

The preferred method for conducting fatigue tests would be to make use of randomly sequenced variable-amplitude cyclic load tests since this procedure would be most representative of the manner in which loads are applied to service and would also eliminate block-loading effects. The amplitude of the cyclic loads would be selected to follow the load spectrum for the component being tested. For test purposes, the load spectrum would be represented by a stepwise distribution of constant amplitude loads as shown in Fig. 14. The load would be defined by the peak

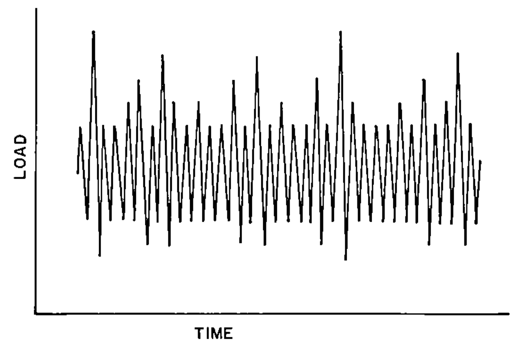


Fig. 15 Load history of randomly sequenced step load

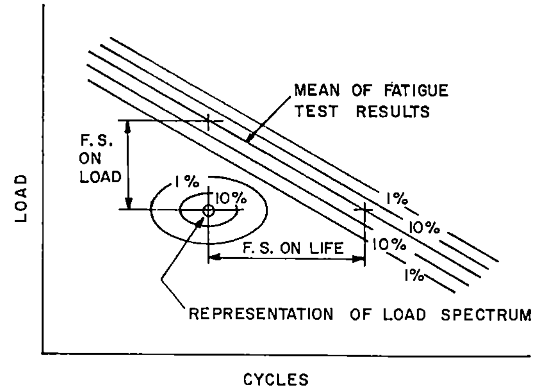


Fig. 16 Comparison of fatigue test results with load spectrum

alternating load level, the mean load, relationships of the lower alternating load levels to the peak alternating load, and the relative number of cycles at each load level. Low amplitude cycles which are below the level causing significant fatigue damage would be truncated. A fatigue test loading sequence illustrating these characteristics is shown in Fig. 15.

A finite number of load cycles distributed over the various load levels would be included in the specification to represent the desired lifetime of the component. Fatigue tests would normally be conducted by repeating the sequence of loads until failure of the part occurs.

Statistical Considerations

It is important to review some of the statistical characteristics of the load environmental and fatigue test data and how they relate to the establishment of a factor of safety. Fig. 16 shows a plot comparing a load environment with fatigue test data. In this plot, the variable-amplitude load sequence is characterized by the peak load. The lifetime of the environment is represented by a specific number of cycles. Thus, the load environment can be characterized by a single point on the figure. A position at some other ordinate than the one indicated would imply that all loads in the spectrum had been increased or decreased proportionately.

Fig. 16 also shows a plot of fatigue test data. These data would be obtained from tests utilizing the environmental load spectra at various levels of amplification. As indicated in the figure, both the fatigue test data and the load environmental data would be most accurately described by a series of curves, indicating the statistical distribution of the data. It would be desirable to have these data down to low values of probability of occurrence, because then one could make direct calculations of the probability of failure.

It is impractical, however, to get sufficient test data, both with respect to load environmental and fatigue strength, to be able to define the curves representing the extremes in the statistical distribution (e.g., 0.1 or 0.01 percent). The definition of these curves would require a large number of tests, which would be impractical to perform. Therefore, it is necessary to base the fatigue design criteria on a more limited understanding of the distribution of test points and an arbitrary specification for the degree of separation of the load environment and fatigue curves.

The idealized distribution curves for environmental load and fatigue data shown in Fig. 16 illustrate the fact that not only is the degree of separation of the two sets of curves important, but that the degree of scatter in the data is significant as well. If, for example, there is a high degree of scatter in the fatigue life data, the probability of failure increases because there will be a larger overlap in the two distributions.

The log-normal distribution is commonly used as an overall fit to fatigue test data. Substantial deviations from this distribution have been noted at the extreme values in some larger samples of data (6). It nevertheless represents the best assumed distribution for the purpose of making statistical calculations involving the confidence in test data and probability of survival.

The scatter of the test data is measured by the standard deviation, $\sigma$. Since it is impractical to test large numbers of any given test specimen to accurately determine the standard deviation, this parameter must be estimated. The concept of using "pooled data" has been found useful in this regard (6). This involves combining data from a number of similar tests to gain a more reliable estimate of $\sigma$ for a given class of structures and materials. This procedure could be followed once the fatigue testing of freight car truck components is carried out in a standardized manner.

Factor of Safety

Since freight car trucks must be considered as a safe-life structure, a large factor of safety in design against fatigue failure is required. As shown in Fig. 16, a factor of safety can be established either with respect to the fatigue life or the load intensity. The usual practice is to specify a factor of safety on life, and this is generally referred to as a "scatter factor." The scatter factor is selected to account for variations in both the load environment and fatigue strength. Various scatter factors are specified by various regulating agencies. For example, the U. S. Air Force requires demonstration of a scatter factor of 4 when testing a full-scale complete airframe (7).

An important consideration in the establishment of a scatter factor is the number of specimens to be used in the fatigue test group. The group of test specimens is regarded as a sample from the total population of the specific type and design of component. The larger the group of test specimens, the more nearly the average fatigue properties of this group can be expected to represent the fatigue properties of the population at large. Therefore, the scatter factor can be inversely related to the size of the test group in specifying the fatigue life that must be demonstrated by the tests; see Albrecht (8).

Tests at Amplified Load Levels

It would be desirable in the formulation of fatigue performance standards to provide for tests utilizing an amplified version of the load spectrum which would allow for demonstration of the required safety factor with a reduced number of load cycles. The amplified load spectrum would be formulated by multiplying the average value of the load and each of the alternating load levels by a constant factor. In this way, the relative relationship between the peak alternating load, the mean load, and the intermediate levels of alternating load would be maintained. The amplified load spectrum would be particularly useful for evaluating the fatigue strength when the load spectrum is in the vicinity of the fatigue limit of the component. Under these conditions, demonstration of a safety margin with respect to lifetime (number of cycles) would require long periods of testing.

Summary

Based on the factors discussed in this paper, an acceptable fatigue performance would include the following elements:

1 The standard would be based on specific reliability goals.

2 The standard would be expressed as performance test requirements.

3 The operational load environment for each truck component would be represented by a load spectrum, based on test data depicting severe operating conditions.

4 For test purposes, the test load spectrum would be represented by a stepwise distribution of constant amplitude loads.

5 Tests would be conducted by randomly sequencing the variable-amplitude loads in the test spectrum.

6 Specimens subjected to the test load spectra would be required to survive a number of loading cycles exceeding the number of cycles defined in the lifetime environment. The scatter factor by which the mean life of the group of test specimens must exceed the lifetime cycles depends on the number of specimens in a test group. Scatter factors based on 99 percent confidence in 0.9999 probability of survival with $\sigma$ of 0.10 are listed in the following for various size groups of test specimens:

| Number of Test Specimens | Scatter Factor |
|---|---|
| 1 | 4.0 |
| 2 | 3.5 |
| 3 | 3.2 |
| 4 | 3.0 |

7 Tests may be conducted with a reduced number of cycles by utilizing an amplified load level subject to the following restrictions:

- Each load in the test spectrum must be proportionately increased (including the average component, if any).
- Two groups of specimens would be tested at two different amplified load levels.

The expected lifetime of the component at the normal level of loading would be determined by linear extrapolation on a plot of load level versus log-cycles, as illustrated in Fig. 17, from points representing average lifetimes of specimens at the two elevated load levels. The expected lifetime, at the nominal load level would then be used to determine whether or not the scatter factor requirement has been met.

Note that the scatter factors presented in the foregoing summary are based on statistical assumptions including the shape of the distribution curve representing fatigue test results and its standard deviation. These statistical assumptions should be continually reviewed as additional fatigue test data are obtained.

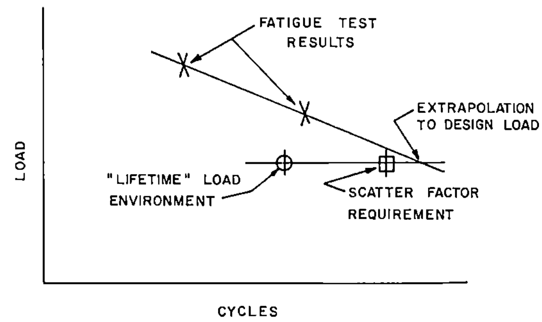


Fig. 17 Extrapolation of amplified load level tests to design level for reference to scatter factor requirement

CONCLUSIONS

Fatigue standards for freight car truck components and wheels are required to ensure that operational reliability goals are fulfilled. These standards would be based on the load environment to which the truck components are subjected in service and the fatigue properties. Recent studies have provided useful data which clarify the nature of the fluctuating load environment. Test data are now required to establish the fatigue properties of the various truck components. The greatest need is for data establishing the spread of the fatigue strength of components which are of the same design and which are subjected to the same levels of fluctuating load. These data would allow a more accurate determination of the margin between the load environment and the fatigue strength which is required to attain given reliability goals.

Conservative fatigue standards are required for freight car truck components because of the safe-life design philosophy that is generally followed in their design. The introduction of fail-safe design features would be a desirable means of increasing the safety and reliability of truck operation. Freight car truck designs need to be reviewed to determine if fail-safe principles can be introduced. Both technical and economic aspects need to be considered in a component-by-component review of the function and design of each part to determine the areas where changes are feasible.

ACKNOWLEDGMENTS


The work described in this paper was carried out by IIT Research Institute under contract to the Federal Railroad Administration as part of their program for the development of suitable safety standards for railroad equipment. The work included tests conducted by the Bessemer and Lake Erie Railroad under the direction of Mr. L. A. Peterson for the measurement of the load environmental data presented in the Section on Service Load Environment.


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# A Field-Service Evaluation of Various Center-Plate Lubricants and Liners

**WAYNE E. LASKY** **M. A. HANSON**

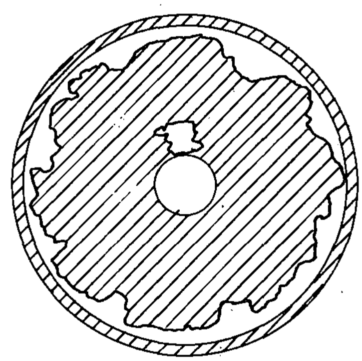

Fig. 1 Worn center plates

For many years AAR Interchange Rule 61 has required the lubrication of center plates when a car is on a repair track and the body is jacked off the trucks. No specifications have been published as to the lubricant except to require that grease shall be used. Interchange Rule 20 provides for the use of a one-piece metal liner between the upper and lower center plate surfaces under certain conditions. The exact composition of the liner is not specified.

Recent changes in car designs and railroad operating practices have demonstrated the need for lubricants of greater serviceability than those commonly used in the past. The benefits to be gained from reasonably free swiveling of freight car trucks are numerous. Some of the benefits are reduced wheel flange wear, reduction of lateral thrust on journal bearings, reduced wear on center plate surfaces, reduction in hot boxes, and probably most important -- more nearly uniform train resistance.

With the increasing use of hump yards for classification of cars, the humping speed is frequently dictated by the cars with the least favorable rolling characteristics. The high torque required to move a partially bound center plate increases rolling resistance and tends to increase humping speeds. Excessive humping speed is damaging to the equipment, impairs bearing performance and most probably is a major cause for damage claims.

There are two diametrically opposed aspects of the center-plate problem. The center plates on heavily loaded cars, operated at low speed, are frequently too rigid. The center plates on lightly loaded cars, operated at high speed, give too little truck restraint.

EARLY OBSERVATIONS

Late in 1953, the Gulf, Mobile & Ohio Railroad purchased 300 ore cars equipped with $6\frac{1}{2}$ x 12-in. roller bearings. The cars were intended to be used for the transport of iron ore. These cars were regularly loaded to their full maximum load limit on rail. The cars were built with conventional pedestal-type freight car trucks -- a portion of them with spring planks and the remainder were of the spring plankless type. They were all equipped with 14-in-dia center plates. The cars had a nominal load capacity of 95 tons, but owing to the high specific gravity of the lading were of such size as a conventional 50-ton hopper car. Shortly after these cars were placed in service, excessive wheel-flange wear was noted.

TABLE 1

FLANGE WEAR AFTER 11 MONTHS SERVICE

| Railroad | GM&O | | GM&O | | A | | B | | C | |
|---|---|---|---|---|---|---|---|---|---|---|
| Truck | A3 | | Barber Plank | | Barber | | A3 | | | |
| No. of Cars | 19 | | 43 | | 51 | | 42 | | 50 | |
| No. of Wheels | 152 | | 344 | | 408 | | 336 | | 400 | |
| Flange Wear | No. | % | No. | % | No. | % | No. | % | No. | % |
| 0/16" | 6 | 3.9 | 27 | 7.8 | 0 | 0.0 | 51 | 15.2 | 129 | 32.25 |
| 1/16" | 3 | 2.0 | 23 | 6.7 | 1 | 0.25 | 54 | 16.1 | 51 | 12.75 |
| 2/16" | 44 | 28.9 | 119 | 34.6 | 60 | 14.7 | 172 | 51.1 | 130 | 32.5 |
| 3/16" | 64 | 42.1 | 117 | 34.0 | 179 | 43.9 | 54 | 16.1 | 81 | 20.25 |
| 4/16" | 31 | 20.4 | 44 | 12.8 | 124 | 30.4 | 5 | 1.5 | 7 | 1.75 |
| 5/16" | 3 | 2.0 | 10 | 2.9 | 33 | 8.1 | 0 | 0.0 | 1 | 0.25 |
| 6/16" | 1 | 0.7 | 3 | 0.9 | 7 | 1.7 | 0 | 0.0 | 1 | 0.25 |
| 7/16" | 0 | 0.0 | 1 | 0.3 | 3 | 0.7 | 0 | 0.0 | 0 | 0.0 |
| 8/16" | 0 | 0.0 | 0 | 0.0 | 1 | 0.25 | 0 | 0.0 | 0 | 0.0 |
| Ave. Wear | 0.18" | | 0.16" | | 0.21" | | 0.11" | | 0.09" | |

Careful observations were made on some of these cars moving away from the ore loading docks. It was noted when the cars were loaded with the trucks out of square, the trucks would not square themselves until enough speed was attained partially to unload the center plates due to carbody bounce.

Observations made of the track over which the locked trucks were operating disclosed accumulated loose scuffed metal as deep as 2 in. alongside the rail. The source of the metal was scuff-

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TABLE 2

FLANGE WEAR AFTER 33 MONTHS SERVICE

| Railroad | GM&O | GM&O | GM&O |
|---|---|---|---|
| Truck | A3 | Barber Plank | Both |
| No. of Cars | 19 | 48 | 67 |
| No. of Wheels | 152 | 384 | 526 |
| Flange Wear | % | % | % |
| 0/16" | 0.0 | 8.0 | 5.5 |
| 1/16" | 2.0 | 5.0 | 4.1 |
| 2/16" | 15.8 | 28.0 | 24.7 |
| 3/16" | 46.7 | 27.2 | 32.8 |
| 4/16" | 21.0 | 23.0 | 22.4 |
| 5/16" | 9.9 | 5.7 | 6.9 |
| 6/16" | 4.6 | 2.6 | 3.2 |
| 7/16" | 0.0 | 0.5 | 0.4 |
| 8/16" | 0.0 | 0.0 | 0.0 |
| Avg. Wear | 0.21" | 0.18" | 0.19" |
| No. of Wheels Turned | 1 pair | 9 pair | 10 pair |

ing of the face of the wheel flanges and scuffing of the mating rail surfaces. The wheel flanges had a series of half-moon-shaped scuff marks around the inner faces.

At the same time as the Gulf, Mobile & Ohio purchase, three other railroads bought substantially identical cars for the same type of service. Careful examinations of these cars disclosed that the severity of the wheel conditions varied inversely with the quality of roadway over which they operated. Two of the railroads had relatively good main line tracks, with moderate curvatures. The cars operated on these lines evidenced wheel conditions much better than the conditions noted on the G. M. & O. cars.

The fourth railroad had track conditions inferior to that on the G. M. & O. and the wheel conditions were even worse than on the G. M. & O. cars. Table 1 shows wheel measurements made after approximately 11 months of service.

Numerous inspections were made in order to determine the cause for the trucks not swiveling. The inspections disclosed severe galling of the center plates. Early observations clearly showed the inadequacy of the residuum-type grease which had been used to lubricate the center plates when new. The center plates were substantially dry even after only a few days of service. It was, therefore, decided to lubricate the center plates, using approximately 2 oz of 50 percent molybdenum disulfide and 50 percent Stoddard solvent by volume. An inspection one year later showed a marked improvement in center-plate conditions with galling apparent only in rare instances. Most of the prior galling had healed in service. It was then decided to lubricate all the center plates regularly with the molybdenum-disulfide mixture on a periodic basis to coincide with the relubrication period of the roller bearings. Table 2 shows wheel measurements made after 22 months of additional service or a total of 33 months.

From 1954 until 1958 the relubrication period was on a one-year basis; since that time relubrication has been on an 18-month basis. These cars accumulate relatively low mileage. The exact mileage of individual cars is not known but as a group, they accumulate between 8000 and 10,000 miles a year. In spite of periodic lubrication of the center plates, the vertical wear on the truck-bolster center plates of these cars now is approximately 1/4 in. and the rim of the truck center plate is elongated approximately 7/32 in. longitudinally. Table 3 gives detailed measurements of a number of the center plates of these cars.

TABLE 3

95 TON GM&O ORE CARS
AVERAGED WEAR MEASUREMENTS
16 CENTER PLATES-1962

| | |
|---|---|
| Truck Center Plate Wear | |
| Vertical | 0.23" |
| Rim Diametrical - Transverse to Car | 0.12" |
| Longitudinal to Car | 0.24" |
| Body Center Plate Wear | |
| Diametrical - Transverse to Car | 0.06" |
| Longitudinal to Car | 0.15" |

Currently almost all of the center plates are worn sufficiently for the truck center-plate rim to foul the body bolster. The clearance new at this location was not measured but was estimated to be 5/16 in. Thus, this is the wear which has occurred in 100,000 miles or less. This condition is believed to be somewhat typical of what can be expected from heavily loaded cars operated over secondary rail lines.

A considerable number of railroads for many years have operated solid-bearing-equipped box cars in head-end service, usually with a load maximum of 60,000 lb on 50-ton cars. Such cars have not exhibited any unusual center-plate wear.

About three years ago one railroad placed into service a group of 70-ton 50-ft box cars in head-end service equipped with roller bearings with only a few thousandths of an inch bearing lateral. These cars operated approximately 6000 miles per month in high-speed passenger service. After 8 months of service, the truck center plate was worn 1/4 in. vertically and the bowl was elongated up to 2 in. The center-plate bowl-rim wear was not longitudinal but off at an angle from the longitudinal position, giving rise to the supposition that braking action on the car may have been a factor in location of the wear. A possible explanation for the excessive wear on

TABLE 4

**AAR CENTER PLATE TEST RESULTS**

| LUBRICANT | Vertical Load (lb) | Torque, (ft. lb) | | Final Reading | |
|---|---|---|---|---|---|
| | | Start | 5,000 Cycles | No. Cycles | Torque (ft. lb.) |
| #2 Grease In Plastic Bag | 40,000 | 1,865 | 4,810 | | |
| #2 Grease with Coarse Babbitt | 40,000 | 3,850 | 3,060 | | |
| | 60,000 | 4,450 | 4,560 | | |
| #2 Grease with Fine Babbitt | 40,000 | 2,040 | 3,140 | | |
| Powdered Copper and Lead | 40,000 | 2,260 | ---- | 1,000 | 3,740* |
| Babbitt Metal Chips | 40,000 | 5,190 | 4,400 | | |
| Graphite Grease | 40,000 | 1,980 | 2,190 | | |
| Graphite Grease | 40,000 | 1,530 | 4,420 | | |
| Powdered Copper and Lead | 40,000 | 3,200 | 3,020 | | |
| Plastic Liner with Graphite | 40,000 | 2,810 | 9,380 | | |
| Composition Liner with Graphite | 40,000 | 3,100 | 5,880 | | |
| Grease and Molybdenum Disulfide | 40,000 | 2,520 | 4,730 | | |
| Steel Liner | 40,000 | 1,790 | ----- | 33,000 | 9,400** |
| Steel Liner | 40,000 | 2,060 | ----- | 3,000 | 9,400 |
| Graphite Grease(#2)with Coarse Babbitt | 40,000 | 1,870 | 1,780 | | |
| | 80,000 | 4,200 | 3,780 | | |
| Graphite Grease(#2)with Fine Babbitt | 80,000 | 3,070 | 3,070 | 24,375 | 8,580 |
| Graphite Grease(#2)with Lead Shot | 80,000 | 5,280 | 3,980 | 22,000 | 9,090 |
| Molybdenum Disulfide and Naphtha | 80,000 | 3,740 | ----- | 1,000 | 17,600 |
| Molybdenum Disulfide and Stoddard Solvent | 80,000 | 1,235 | 4,315 | 11,155 | 9,865 |
| Bronze and Babbitt Chips | 80,000 | 1,850 | 2,160 | 25,000 | 3,700 |
| White Lead | 80,000 | 1,850 | 7,400 | 7,000 | 10,790 |
| Plastic Liner (dry) | 40,000 | 1,460 | 4,315 | 25,000 | 5,550 |
| | 80,000 | 10,425 | ----- | ----- | ----- |
| Plastic Liner and #2 Grease | 80,000 | 2,150 | 1,460 | 25,000 | 1,850 |
| PRR Grease and Oil Mixture | 80,000 | 3,700 | 3,700 | 25,000 | 4,010 |
| Grease w/lead Powder | 80,000 | 4,940 | ----- | 58 | 6,780 |
| Load Reduced to | 60,000 | 5,625 | at 60 cycles | 350 | 7,400 |
| Molybdenum Disulfide in Grease | 80,000 | 4,470 | ----- | 1,000 | 6,245 |
| Load Reduced to | 60,000 | 4,470 | at 1025 cycles | 2,000 | 4,470 |
| Graphite Grease (2) with Coarse Babbitt | 80,000 | 4,470 | 3,085 | 10,000 | 3,375 |
| Molybdenum Disulfide and Grease | 80,000 | 3,700 | 7,400 | 12,250 | 9,250 |
| Brass Chips with Oil on Plastic Disc. | 80,000 | 3,235 | 8,785 | 5,200 | 9,095 |
| Load Reduced to | 60,000 | 8,180 | at 5,201 cycles | 6,000 | 9,560 |
| Load Reduced to | 40,000 | 8,180 | at 6,020 cycles | 10,000 | 6,785 |

*Test discontinued. Excessive vibration.
**Liner pulled apart. Excessive galling.

the center plate of these cars is the restriction of the lateral freedom of the bearings.

AAR Road Tests[1] have shown an increase in bearing lateral freedom to be beneficial in reducing the Lateral Damage Index at high speeds with light loads.

Based on the judgment that additional truck-bolster restraint was needed to correct the excessive center-plate wear, the trucks were overhauled and modified. The vertical surface of the rim of the center plate was reinforced with an abrasion-resistant steel ring and the car was equipped with constant-friction side bearings, imposing approximately 4000 lb force on the truck bolster. New body center plates were fabricated from 1045 steel and the bottom surface of the body center plate was flame-hardened. These modified trucks have now been in service 18 months and center-plate wear has been reduced to a normal level. At high-speed operation such as 60 mph and above, it is hypothesized that increasing the frictional forces in the center plate would be beneficial. No certain proof as yet exists as to whether wear can be reduced by a satisfactory lubricant when excessive truck swiveling is occurring.

REVIEW OF LABORATORY TESTS

In 1961, W. H. Cyr reported on the results of center-plate lubrication tests, utilizing a test stand simulating freight car operations.[2] His test data indicated machining of center plates to be beneficial and lubrication of center plates with molybdenum disulfide mixed with a solvent, temporarily to be quite beneficial. He determined the reversals of rotation on a freight-car truck in service to be 2 cpm and estimated a freight car to average 72,000 cycles in approximately 6 months of service. During the tests, other methods of lubrication were tried, including grease, thin liners, and bonded graphite, but it was concluded that all of them were unsatisfactory due to short service life.

The AAR Research Department also ran a series of laboratory tests covering 28 different types of lubricants.[3] The tests were made with three

[1] A.A.R. Summary Report of Road Tests, Cooperative Freight Truck and Snubber Research Program, Report No. F-3800, June 1951, Appendix B, pp. 17-24.

[2] W. H. Cyr, ASME Paper No. 61-WA-239, 1961, "Measurements of Center Plate Friction on Freight Cars."

[3] AAR Research Department, Report MR-438, December 1963.

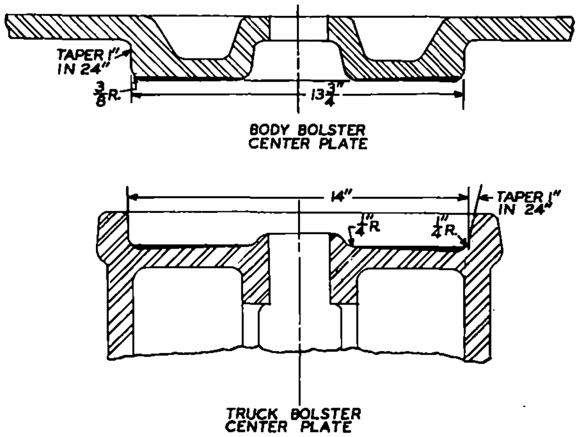

Fig. 2 Machined center plates (no contact in lined areas)

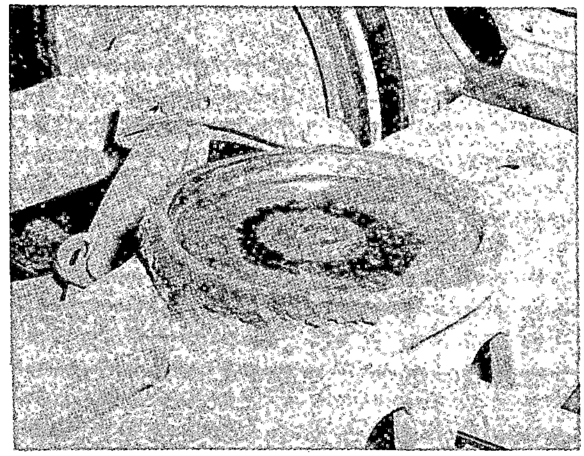
Fig. 4

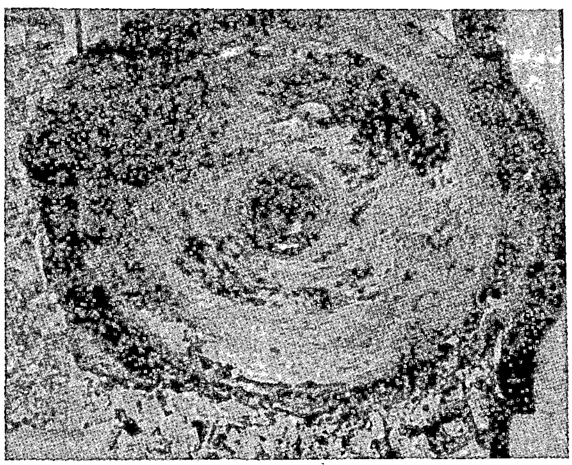
Fig. 3

TABLE 5

WEIGHT PER SQUARE INCH ON CENTER PLATES
AT MAXIMUM GROSS RAIL LOAD

| NOMINAL CAR CAPACITY | MAX. GROSS LOAD ON RAIL | CENTER PLATE DIAMETER | P.S.I. |
|---|---|---|---|
| 40 TON | 142,000 | 12" | 783 |
| 50 TON | 177,000 | 12" | 989 |
| 70 TON | 220,000 | 14" | 845 |
| 70 TON | 220,000 | 15" | 718 |
| 70 TON | 220,000 | 16" | 618 |
| 100 TON | 263,000 | 14" | 1010 |
| 100 TON | 263,000 | 15" | 859 |
| 100 TON | 263,000 | 16" | 739 |
| 125 TON | 315,000 | 16" | 890 |

center-plate loads of 50,000 - 60,000 and 80,000 lb. The torque was measured at the start of the test, after 5000 cycles and at 25,000 cycles. To establish a baseline, a standard unmachined center plate was tested without lubrication or shims with a 40,000-lb vertical load. The initial torque was 10,400 ft-lb which increased to 13,700 ft-lb after only 500 cycles. When the machine was stopped because of the excessively high torque, the center-plate surfaces were examined and it was found that severe galling had resulted. Table 4 is a reproduction of the AAR test results.

IMPOSED LOADS ON CENTER PLATES

The unit loads on center plates are quite modest if it is assumed the loads are uniformly distributed. Table 5 gives the calculated unit load for the various size center plates in service.

It is commonly recognized that momentary unit loading may be much higher than indicated in Table 5 owing to roll of the truck bolster during impact and severe sway of the car body under adverse track conditions.

Some evidence also indicates the body center plate may be deflecting rather severely under heavy loading. Frequently on new cars with machined center plates, the load-bearing area is confined to an irregular area near the edge of the center plate as shown in Fig. 1.

Also the wear on old center plates seems to be most pronounced near the outer edges as shown in Fig. 2. In addition, a distinct wear step is

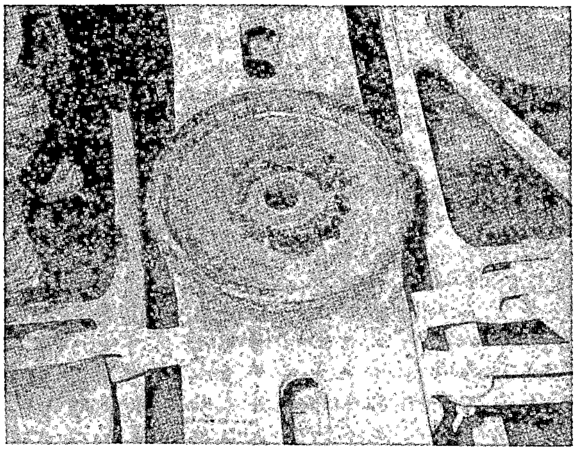

Fig. 5

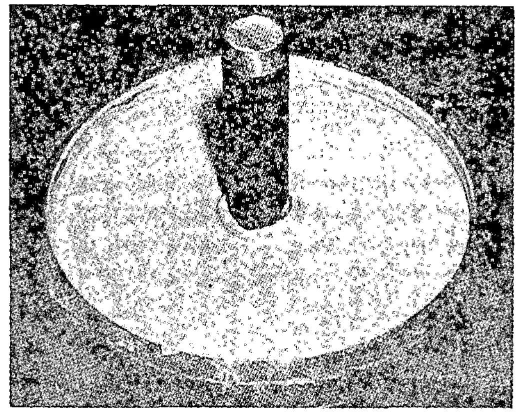

Fig. 7

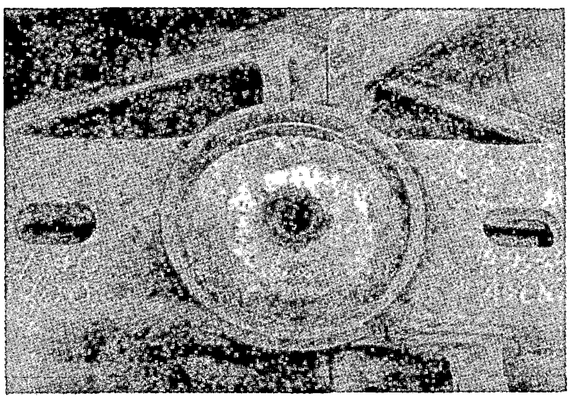

Fig. 6

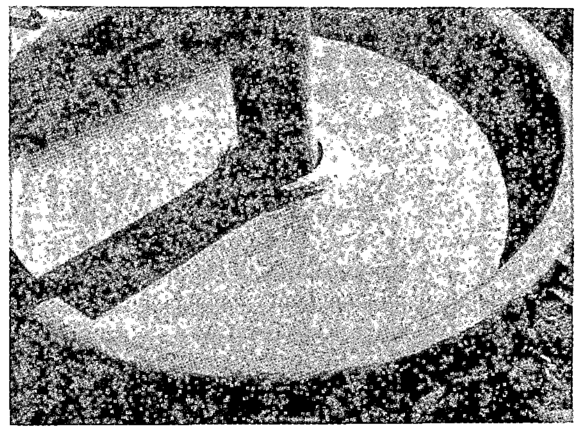

Fig. 8

formed due to the 3/8-in. radius at the edge of the body center plate.

FIELD SERVICE TESTS

During 1960 a number of hot boxes and instances of excessive journal-bearing end wear were experienced on new cars shortly after leaving the car builders' shops. Subsequent investigations disclosed inadequate center-plate lubrication was the major contributing factor. The AAR Lubrication Committee, therefore, appointed a special Subcommittee to investigate the proper lubricant for use in freight car center plates.

Numerous types of lubricants were applied to center plates of cars for field testing. A considerable number of the test cars were heavily loaded hopper cars in iron-ore service. Later, after the center plate testing machine was placed in service at the AAR Research Center, field service testing was expanded to include additional types of lubricants which showed promise on the AAR test machine.

Since many of the cars in the field had center plates worn to a degree to require shimming, it was decided also to include various types of liners in the field testing.

To date in the field-test program, 171 center plates have been inspected and photographed after 12 to 18 months of service. The test program has included 41 various lubricants and liners. Seven proprietary plain greases have been tested, including ball and roller-bearing grease, various gear compounds, antirust compounds and hot-box coolants. Sixteen items incorporating solid lubricants have been tested, including the 50-50

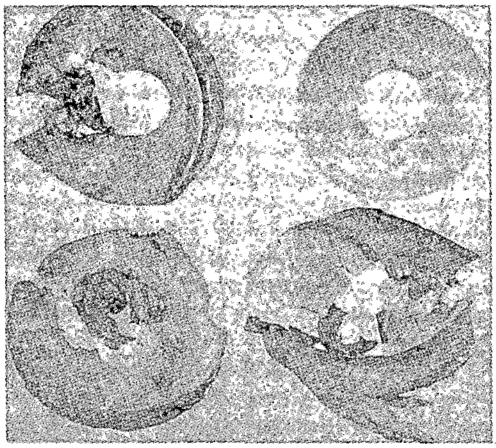

Fig. 9

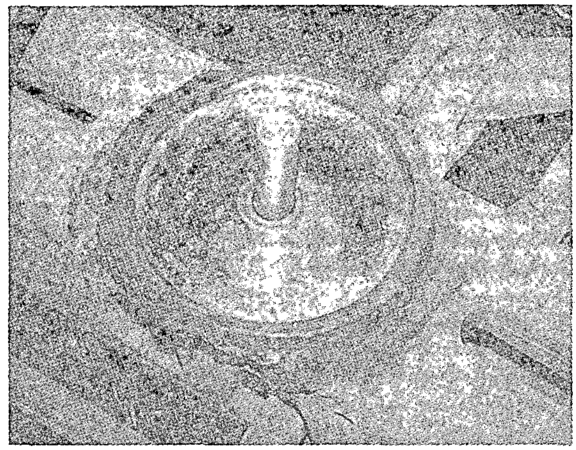

Fig. 10

mixture of molybdenum disulfide-volatile solvent; greases, lacquers, or volatile solvents compounded with various solid lubricants, including molybdenum disulfide, graphite, powdered lead, lead shot, babbitt chips, and brass turnings. Six designs of thin plastic disks impregnated with either molybdenum disulfide or graphite also were tested. Ten types of liners varying from 1/8 to 1/4 in. in thickness have been tested, including some manufactured from laminated fabric impregnated with a solid lubricant, two compositions of high tensile bronze liners, soft-steel and hardened-steel liners, both solid and perforated as well as some steel liners with a solid lubricant bonded to the surfaces.

All of the proprietary plain greases failed to provide adequate lubrication even for short periods. Fig. 3 shows a typical center plate of a high-tonnage car having galled surfaces after service with a gear grease. The plain greases completely disappeared from the center-plate contact area after a short period of time, extruding over the rim of the truck center plate. The center-plate contact surfaces were dry and freshly galled areas were present.

Most of the sixteen items compounded with solid lubricants when used in sufficient quantities were found to be reasonably effective for the one-year test period since they adequately prevented seizure and galling. Satisfactory lubrication was experienced even after the carrier for the solid lubricant had disappeared. The plating of the contact surfaces with the solid lubricant apparently prevented seizure or galling of the metal. It is believed that enough of the solid lubricant remained in the contact area for sufficient time to plate the contact surfaces thoroughly. Additional service testing will be required, however, to establish the full service life expectancy.

The brass turnings were found unsatisfactory even though used in sizable quantities. Some of the other compounded solid lubricants were used in relatively small quantities, as recommended by the supplier, and, in general, these were unsatisfactory for the year's service test irrespective of the type of solid lubricant used.

Reasonably effective results for the one-year test were obtained with molybdenum disulfide, graphite, powdered lead, lead shot or babbitt chips when mixed with a No. 1 or No. 2 grease and applied to a depth of 1/16 in. on the contact area of the truck center plate. During the test period they adequately prevented metal seizure and galling. Fig. 4 is a center plate after 14 months of service with a No. 2 grease containing 4 percent molybdenum disulfide and applied in quantities of 4 oz per center plate.

Fig. 5 is a center plate after one year of service with a 50-50 mixture by volume of No. 2 grease and babbitt chips applied in quantities of 3 oz per center plate. The surface appeared to be covered with a thin coating of babbitt although the surface appeared dry and the grease had extruded over the rim of the plate.

Fig. 6 is a center plate after one year of service with graphite applied in a lacquer carrier to depth of 1/16 in. No fresh galling or metal seizure was noted although surfaces appeared dry and free of lacquer.

Fig. 7 is a center plate after application of 50-50 mixture of molybdenum disulfide and vola-

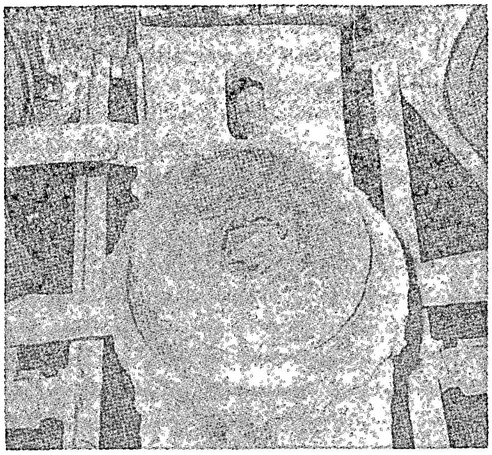

Fig. 11

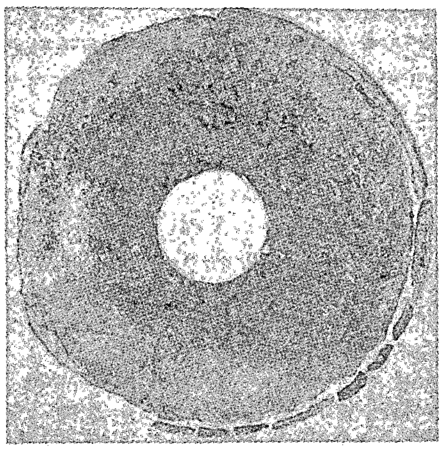

Fig. 12

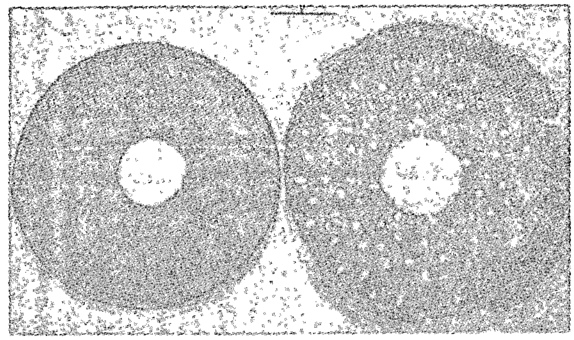

Fig. 13

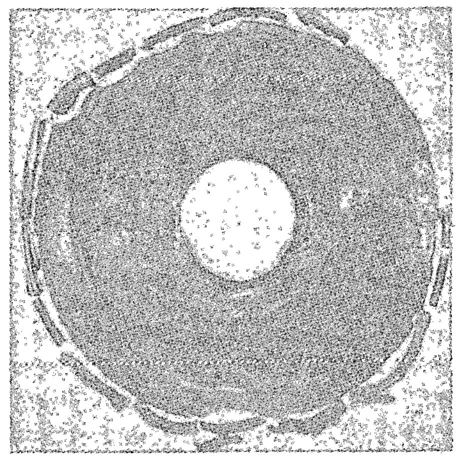

Fig. 14

Fig. 15

tile solvent to a depth of 1/16 in. Note both horizontal and vertical contact areas are coated.

Fig. 8 is a center plate after one year of service with 50-50 mixture of molybdenum disulfide and volatile solvent. Surfaces are bright and shiny but perfectly dry. No fresh metal seizure or galling is present and old galled areas have partially healed.

All of the six types of thin plastic disks impregnated with either molybdenum disulfide or graphite appeared to lubricate the center plates adequately and prevent metal seizure although the disks themselves eventually disintegrated and portions were extruded over the rim of the center plate or wrapped about the center-plate pin. Fig. 9 is a photograph of one new disk and three disks that have seen various lengths of service. Fig. 10 is a center plate after one year of service with a thin plastic disk impregnated with molybdenum disulfide. Note the shiny polished plated surface with no evidence of fresh galling.

It appears that on the high-tonnage cars tested with solid lubricants, it was most important that the solid lubricant be retained in the center plate for a sufficient length of service for the contact areas to become uniformly coated or plated with the solid lubricant. In some instances it appeared as if the solid lubricant was carried out of the center plate by the vehicle prior to the solid lubricant having become bonded to the metal surfaces.

Of the ten types of liners tested, those manufactured from laminated fabric impregnated with a solid lubricant and those made from bronze appeared to offer satisfactory lubrication as well as function as a liner in high-tonnage freight cars. Fig. 11 is a center plate with laminated fabric after a short period of service. Fig. 12 is a bronze liner after one year of service. No evidence of metal seizure or galling was noted but when used in old center plates, both types of liners tended to disintegrate at the outer periphery as well as at the periphery of the center hole.

Soft steel liners applied to heavily loaded cars thinned rapidly and metal eventually extruded over the rim of the center plate. Fig. 13 shows two soft-steel liners after short service periods. The solid liner has extruded to the vertical rim of the center plate and has continued upward along the wall for 1/2 to 1 in. The perforated liner has extruded over the center-plate rim, forming sharp scalloped configurations outside the rim.

Hardened steel liners, when applied to old center plates on heavily loaded cars, cracked and chipped at the outer periphery. This may be due to the liner becoming lodged on a wear step. Metal chips eventually lodged under the liner, causing severe gouges and indentation of female center plate surface. Fig. 14 is a hardened-steel, center-plate liner removed after one year of service. Fig. 15 is a close-up view of a center plate surface showing deep gouges caused by the broken pieces of hardened liner lodging under the liner.

## CONCLUSIONS

1 Center plate lubrication with molybdenum disulfide was effective in reducing wheel flange wear of heavily loaded cars operated in slow to moderate speed service.

2 Solid Lubricants - molybdenum disulfide, graphite, babbitt chips, lead shot and lead powder are all more effective for center-plate lubrication than any oils or plain greases previously used.

3 The total effective service life of any of these lubricants is not known as yet but they are known to be effective in preventing galling for one year under the test conditions.

4 High-tensile bronze liners and phenolic laminated fabric liners were satisfactory for the test period of one year.

5 Soft-steel liners were unsatisfactory in heavily loaded cars.

6 325 Brinell liners were unsatisfactory in worn center plates on account of liner breakage and resultant center plate damage from the broken pieces.

Mr. Rousseau was born in Massachusetts and received an ASME degree at Worcester Junior College. He also received Certificates of Recognition at the School of Industrial Management, Worcester Polytechnic Institute and the Indiana Executive Program, Indiana University.

Mr. Rousseau entered the carbuilding industry at Pullman-Standard Car Manufacturing Company, Worcester, Massachusetts in 1940. Since that time he has held positions in Manufacturing and Engineering including Assistant Superintendent of Production; Project Engineer; Project Manager of Product Development; Assistant General Manager of Production Engineering; and General Manager of Freight Car Engineering, his present position.

**GEORGE L. ROUSSEAU**

**Mr. Rousseau:** Thank you, Loren, I would like to discuss Pullman's approach to center plate problems and design. For the past several years the inspection of center plates has revealed an increasing frequency of deterioration, wear and breakdown. This problem gained prominence as the percentage cost per car of center plates increased. Center plates on rigid underframe cars, acceptable by past experience and performance — although not trouble-free, had been used for basic design features on cushioned underframe cars. Initial reactions to early center plate problems varied, resulting in a variety of sizes and shapes — some of which worked; however, many did not.

Beginning in 1967 when Sales and Service reports indicated an increase in the normal center plate wear, Pullman-Standard initiated full-scale testing and field surveys to define what the reasons and causes were for increased wear and breakdown. Trailer Train had recognized center plate problems on high-mileage cars earlier and initiated preventive maintenance programs with center plate replacement running at 70% due to breakage or a 5/8" condemning wear. While field data was being compiled, testing began on fabricated cushioned underframe center plates using a Hy-Cube car, since this car was the first to indicate deficiency in service. Rock and roll was highly suspect from preliminary data, and so two car rocking devices were designed and built for exciting a fully-loaded Hy-Cube car at its natural roll frequency.

The testing began with center plates similar to those originally specified by the Hy-Cube Committee. These were used to establish a basis of evaluating improvement in subsequent tests. Strain gage data quickly indicated very high strains in the 3/8" radius transition between the bowl and base plate, which is a standard AAR requirement. In a fatigue environment the stress range from positive to negative exceeded the endurance limit as indicated on modified Goodman diagrams. Initial test results reproduced the semi-circular fractures on the transverse centerline observed in the field. Therefore, two parameters for center plate breakdown had been defined: roll input and a stress concentration in the 3/8" bowl transition radius.

Field Service reports, along with static testing conducted by Pittsburgh Forgings, Symington-Wayne and Buckeye Castings, correlated the test results. Field observation of both body center plate and truck bowl also indicated heavy wear, which would of course accelerate body center plate breakdown. The incentive rates in August 1967 for 100,000 pounds of paper per car required double tiers of paper rolls, resulting in lower critical speeds that fall into the operating range. Track conditions, with the decline of passenger service, began to worsen noticeably — adding to the accelerating rock and roll problem. Financing for track maintenance became increasingly difficult, while rail loads increased steadily.

Testing continued on a variety of cushioned underframe center plates, including both fabricated and cast designs, recognizing the fact that the industry needed substantially improved designs for reliable performance in an increasingly demanding service environment.

*The film that you now see indicates the nature of developmental testing conducted at Pullman-Standard. The first car, generously provided by the DT&I, is the*

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*86-foot Hy-Cube car now common to the industry. The close-up view that you now see shows the placement of strain gages in the 3/8" transition radius and on the top surface. The loaded car in the next scene has been excited by the rocking device, and its roll mode is being monitored by the oscilloscope and strain gages. The developmental testing included the wing type center plates shown in this close-up view of the body bolster-truck undergoing cyclic fatigue. During some of the testing a dynamic vertical drop or bounce was observed. The next scene shifts to a later stage of the testing where a flat center plate with a modified bolster was used. The bolster had gussets applied to the webs at the offsets in the bottom cover plate. The advantage of the flat application was that attachment stresses were considerably reduced due to reduced surface mating. This contributed to longer fatigue life when compared to earlier wing type center plates that extend up the slope of the bolster. The flat application has but one surface to match versus two surfaces and the angles on a wing type.*

*The next car selected for testing was a 100-ton covered hopper car that was provided by the Chicago, Milwaukee, St. Paul & Pacific Railroad because of their interest in center plate performance on rigid underframe cars, because of the high center of gravity, and because of our follow-up on data obtained from our Field Service personnel The testing, now in progress, is expected to show that similar problems exist on both rigid and cushioned underframe cars. At this time we have not developed enough information to state conclusively the contributing factors as they relate to this type of car; however, rock and roll does create high cyclic stresses in the center plate. This phase of testing will eventually include the new design center plate developed by the A.A.R. and A.R.C.I. and other features depending upon test results. These pictures were taken at the start of the program, and we cannot correlate test results with life expectancy yet. This concludes the film on our testing to date.*

In returning to the area of cushioned underframe testing, additional developmental work continued on a 50-foot cushioned underframe box car, one that is used in a greater variety of service, included paper. During this phase of testing cause and effect relationships were studied to determine proper stiffness ratios such as reinforcement to base plate thickness, optimum reinforcement spacing, and minimum bowl radius to bring the stress range within the fatigue allowable.

The severity of the dynamic tests was to the degree that several truck springs had to be replaced — an observation not prevalent in the field, although heavy side bearing wear and even loss of side bearing have been observed. The conclusion of the two-year test program on cushioned underframe cars has indicated these results:

1. Rock and roll is instrumental in center plate wear and breakdown.
2. Breakdown originates in the transition radius and propagates into the transverse and longitudinal reinforcement.
3. The 3/8" radius is not satisfactory for today's fatigue environment. A 3/4" radius provides maximum or near optimum reduction of stress concentration in the interface area between the bowl and base plate at a minimum increase in bowl depth.
4. Longitudinal reinforcement on the center plate should be located over the outer circumference of the bowl.
5. A flat center plate application significantly reduces the attachment pre-stress in the transition radius because of simplified surface matching.
6. Both fabricated and cast center plates can develop cracking unless fatigue design is considered.
7. The center plate application is related to car configuration. Bolster modification for center plate application cannot be arbitrary.
8. Supplemental snubbing devices that reduce car body roll would increase the fatigue life of center plates.

Naturally, cost versus utlization is a vital factor, especially on cushioned underframe cars. Materials such as Grade "E" castings obviously cannot be purchased at the same cost as Grade "C" or a fabrication. The objective of the test was to develop an acceptable product in both fabricated and cast designs. We feel that this has been accomplished on cushion underframe cars. To parenthetically state that continued track deterioration in the future will not cause additional problems is perhaps being optimistic, if not facetious.

For this reason Pullman has initiated a second phase of testing on a rigid underframe covered hopper car. Many of the preliminary results of the cushioned car tests and Trailer Train inspections indicated that these cars, subjected to the same rock and roll, might develop similar problems. Concerned men in the industry indicated that there was indeed a problem. Working through the ARCI, a new standard center plate was developed, incorporating the collective thoughts and best features test-proven to date. This center plate will be dynamically tested in the near future in the same manner as you observed in the film.

The rigid underframe center plate improvement does present a coupler height problem, particularly on lighter cars. Presently, alternate bolster designs are being developed to implement the improved center plate.

What is required in this particular area is a near continuous test program, routine inspection of all cars with side bearing adjustment and center plate lubrication performed as required — particularly on those cars more prone to rock and roll — . . . and a continual awareness and response to the carbuilders of new loading and handling conditions as they develop or are conceived. Trailer Train has established and use a condemning or out-of-round gage for center plate wear. This is an example for the industry.

On behalf of Pullman-Standard, may I extend our appreciation for this opportunity to discuss with you the nature of our work in this area — past, present and future. We sincerely believe that there is no substitute for full-scale dynamic tests; and, though they are expensive, we will continue our programs, knowing they are necessary. This is our contribution to an industry that we believe can flexibly attack and cooperatively eliminate its problems.

(APPLAUSE)

**Chairman Smith:** The next speaker is Mr. John F. Krause, Chief Engineer, Fleet Maintenance, General American Transportation Corp.

# Outer Pedestal Legs of Narrow Pedestal Side Frames

C. E. TACK

HISTORY

With the advent of the heavy-duty, light-weight roller bearing, the suppliers of freight-car, truck side-frame castings were able to omit the use of a journal box. The bearing being self-contained and sealed could operate without protection from weather and railway operating conditions. Therefore, the pedestal end to accept the roller bearing and adaptor casting was accepted as standard. When the compact bearing design evolved an alternate standard "narrow" pedestal design was adopted by the AAR. This latter design had the advantage of lighter frame weight.

Early in 1960, several fractures of the outer pedestal leg were reported on frames having the AAR alternate standard narrow pedestal. In all, a total of eight pedestals were reported broken. The reported fractures were confined to castings produced by only two of the manufacturers. Available data indicated that occurrence of the fractures was during cold weather and associated with severe end impact, in most cases during switching. In other instances, fractures were traceable to recent humping operations during winter months. Derailments as the result of broken pedestal legs were quite expensive to the railroads and became a matter of concern, since epidemic occurrence could not be tolerated.

Data gathered on service conditions indicated the fractures started in the radius at the junction of the journal-bearing seat and outer pedestal leg and extended at an angle of 65 to 75 deg above the horizontal. In one case, the angle was almost 90 deg to the horizontal. Fracture sections showed no evidence of the progressive type indications commonly associated with fatigue but had the chevron pattern typical of brittle-type fractures. Metal thicknesses were close to nominal and the metal was solid with satisfactory heat-treatment. Chemical analyses were within the AAR Specification M-201 for side-frame and bolster materials. Sulfur and phosphorous contents were on the high side of the specification. It is interesting to note that the frames involved were produced by a process which normally results in sulfur and phosphorous on the high side of the range allowed in present specification.

The loads causing the fractures are inertial, applied to the outer pedestal leg by roller bearing, wheel and axle assembly. Therefore, increased impact speeds impose heavier loads on the pedestal legs. After consideration of fracture histories and results of these investigations, a combination of low-temperature embrittlement and overspeed impact was suspected.

APPROACH TO THE PROBLEM

With the available facilities, investigations were set up along the following paths to resolve the pedestal-failure problem.

1 Analytical - to reevaluate standard procedures of design calculation.

2 Static test investigations - to compare various pedestal designs, isolate stress concentrations, determine stress values and check analytical procedures.

3 Fatigue investigations - to compare the effect of repeated loadings on various designs and to determine if fatigue was a factor in failures.

4 Laboratory impact tests - to compare impact resistance of the standard and proposed designs.

5 Concurrently car-impact test results were made available to the investigators by National Castings Company.

During the foregoing program of laboratory investigation of outer pedestals, the following four designs were studied. The designs as listed below are shown in Fig.1.

Design A - Original AAR alternate standard design (3/8 in. radius, 5/8 in. metal thickness).

Design B - Design proposed when initial fractures were reported (3/8 in. radius, 5/8 in. metal thickness, $1\frac{1}{4}$ in. extension on rib over journal).

Design C - Design evolved as a result of studies and later proposed by Truck Manufacturers' Engineers' Committee to Association of American Railroad (5/8 in. radius and 3/4 in. metal thickness).

Design D - Design evolved as a result of studies on above three (11/16 in. radius and 13/16 in. metal thickness).

ANALYTICAL STUDIES

In the theoretical stress analysis of the various pedestal contours, bending in a curved section was encountered. It is generally conceded that the stress distribution in curved sections is not linear, but follows a hyperbolic curve. Also,

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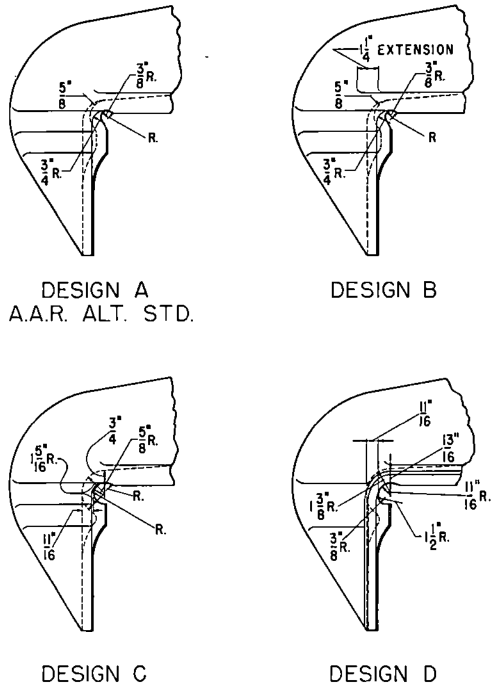


Fig. 1 Details of radii contours and metal thickness of 70-ton-capacity outer pedestal legs of narrow pedestal side frames

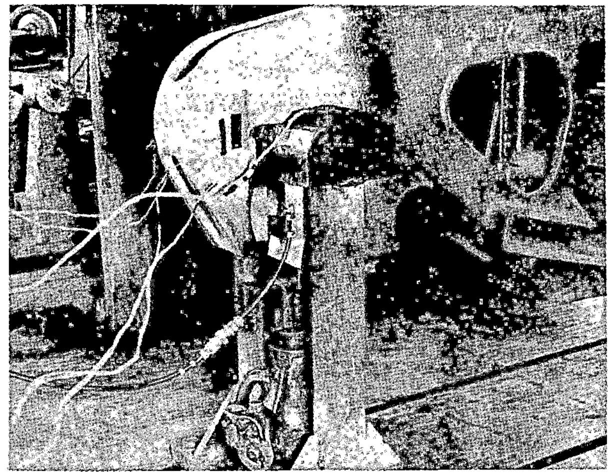

**Fig. 2 Static test setup. Loading on outer pedestal at roller-bearing centerline through a load cell by means of a wedge and jack arrangment**

the neutral axis does not coincide with the gravity axis, being shifted toward the concave surface. The curved-bar bending theory was applied to the outer pedestal-leg calculations and by this method, stresses and stress concentrations could be calculated at various angles throughout the junction. Detailed application of this theory can be found in the Appendix.

From the analysis of each design under study, the following stress-concentration factors were determined:

| Design No. | Stress-concentration factor at 45° | 60° | 75° |
|---|---|---|---|
| A | 3.042 | 2.951 | 2.929 |
| B | 3.249 | 3.067 | 3.007 |
| C | 2.306 | 2.321 | 2.337 |
| D | 2.245 | 2.254 | 2.278 |

Theoretical calculations on the four pedestal end designs indicated a 2 to 5 percent increase in strength with the addition of the $1\frac{1}{4}$-in. rib extension to the original design. With the changing of the radius from 3/8 to 5/8 in. and increasing the metal section to 3/4 in. a theoretical increase in strength of from 20 to 24 percent in the critical radius was indicated. This latter is the TMEC accepted design for standard pedestal production. Design D represents an increase of $2\frac{1}{2}$ to 3 percent over the accepted new standard design.

STATIC STRESS INVESTIGATIONS

Static tests were set up and conducted as shown in Fig.2. Loads were applied at the roller-bearing centerline through a load cell by a jack-and-wedge arrangement. Each pedestal was first stress-coated to determine areas of stress concentration. By application of SR-4 resistance-type strain gages at points of high stress, the quantitative stress values were determined.

Suspicions of the high stress being in the radius at the junction of the journal bearing and outer pedestal leg were confirmed as illustrated in Fig.3, showing the pattern developed in the stress coat during a static load application. The high point of stress was in the inner radius (3/8 in.), which was the apparent point from which service failure originated and propagated to complete failure of the pedestal end. The highest stresses on the 3/8-in. radius were located 12 deg above the horizontal. A similar stress-coat test on the original design with the $1\frac{1}{4}$-in. rib extension indicated the high stress to be also 12 deg above the horizontal. On the design with the 5/8-in. radius and 3/4-in. metal section, the highest stress-coat indication was 28 deg above the horizontal. On the design with the 11/16-in. radius and 13/16-in. metal thickness the highest stress indication was 25 deg above horizontal.

SR-4 resistance-type strain gages were applied to the inner critical radii on the three de

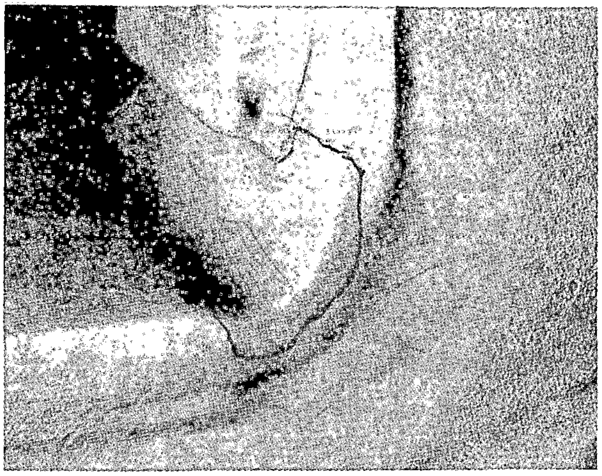

Fig. 3 Static stresscoat pattern; pattern developed by static load applied to outer pedestal at roller-bearing centerline

TABLE 1

STATIC STRESS RESULTS
DESIGN COMPARISON DESIGN "A"
3/8" RADIUS - 5/8" SECTION
NO RIB EXTENSION

| LOAD | AVERAGE STRESS AT 12° | 70° |
|---|---|---|
| 0# | 0 PSI | 0 PSI |
| 2500 | 6180 | 4290 |
| 5000 | 11895 | 8325 |
| 7500 | 17550 | 12390 |
| 10000 | 23235 | 16470 |
| 12500 | 28830 | 20505 |
| 15000 | 34275 | 24300 |
| 17500 | 40110 | 28380 |

TABLE 2

STATIC STRESS RESULTS
DESIGN COMPARISON DESIGN "B"
3/8" RADIUS - 5/8" SECTION
1¼" RIB EXTENSION

| LOAD | AVERAGE STRESS AT 12° | 70° |
|---|---|---|
| 0# | 0 PSI | 0 PSI |
| 2500 | 5535 | 3915 |
| 5000 | 11130 | 7875 |
| 7500 | 16740 | 11835 |
| 10000 | 22110 | 15750 |
| 12500 | 27450 | 19575 |
| 15000 | 32955 | 23400 |
| 17500 | 38550 | 27180 |

signs at high points of stress indicated by the stress coat and as shown in Figs.4, 5 and 6. Quantitative values of stress recorded during static tests are shown in Tables 1, 2, 3 and 4. These values are comparatively plotted in the graph, Fig.7. Comparison of the stress values on the critical radius show that the addition of the $1\frac{1}{4}$-in. rib extension to the original design increased the overall strength by 2 to 5 percent under

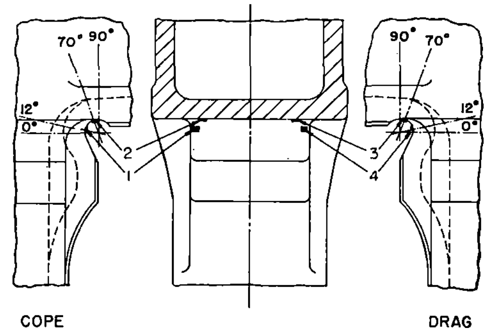


Fig. 4 Location of SR-4 resistance-type strain gages on radii of outer pedestal leg of Designs A and B during static test

TABLE 3

STATIC STRESS RESULTS
DESIGN COMPARISON DESIGN "C"
5/8" RADIUS - 3/4" SECTION
NO RIB EXTENSION

| LOAD | AVERAGE STRESS AT 75° | 28° |
|---|---|---|
| 0# | 0 PSI | 0 PSI |
| 2500 | 3450 | 4215 |
| 5000 | 7335 | 9000 |
| 7500 | 11145 | 13605 |
| 10000 | 14655 | 18150 |
| 12500 | 18195 | 22830 |
| 15000 | 21765 | 27480 |
| 17500 | 25620 | 32250 |

TABLE 4

STATIC STRESS RESULTS
DESIGN COMPARISON DESIGN "D"
11/16" RADIUS - 13/16" SECTION
NO RIB EXTENSION

| LOAD | AVERAGE STRESS AT 25° | 52° |
|---|---|---|
| 3000 | 3075 PSI | 3885 PSI |
| 5000 | 6765 | 8700 |
| 7000 | 9870 | 12630 |
| 9000 | 13380 | 17340 |
| 11000 | 16350 | 21300 |
| 13000 | 19740 | 25680 |
| 15000 | 23310 | 30225 |
| 17000 | 26325 | 34650 |

static-load conditions. By a change in contour from 3/8 to the 5/8-in. radius, and increasing the metal section to 3/4 in., a reduction in stress on the inner critical radius of 20 percent was effected. Increasing the radius to 11/16 in. and the section to 13/16 in. did not significantly increase the static strength over the 5/8-in. radius. The theoretical calculations were based on a high stress at 75 deg above the horizontal; however, the magnitude of the stress reduction due to changes in radius contour and increased section was the same as calculated.

An ultimate load was developed on the outer

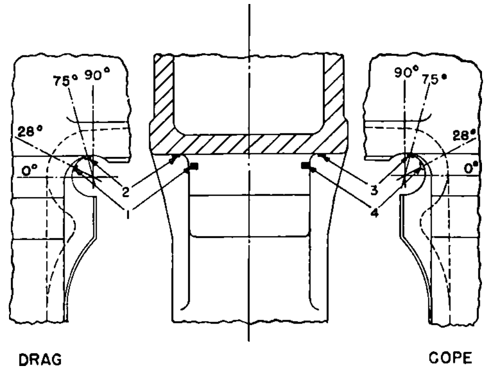


Fig. 5 Location of SR-4 resistance-type strain gages on radii of outer pedestal leg of Design C during static test

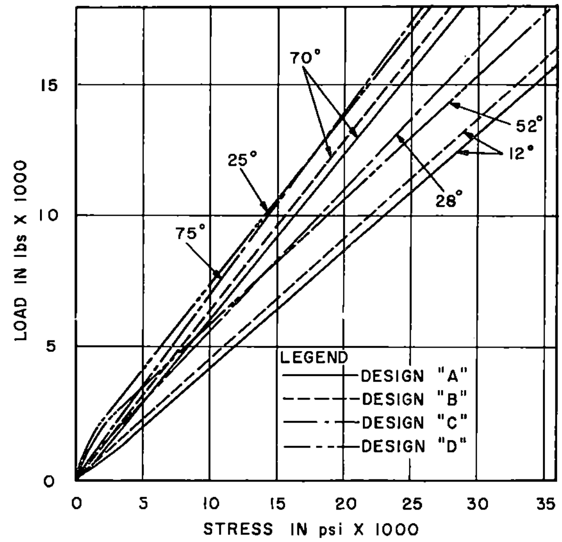


Fig. 7 Comparison of static stresses in four designs on outer pedestal; average of cope and drag stresses

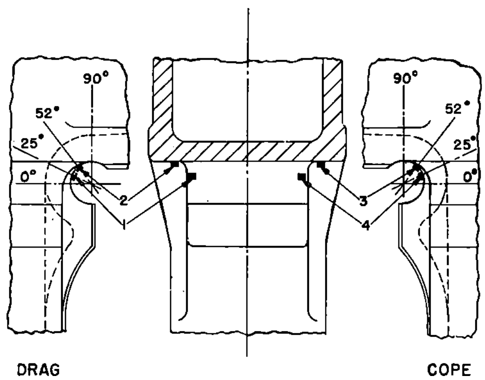


Fig. 6 Location of SR-4 resistance-type strain gages on radii of outer pedestal leg of Design D during static test

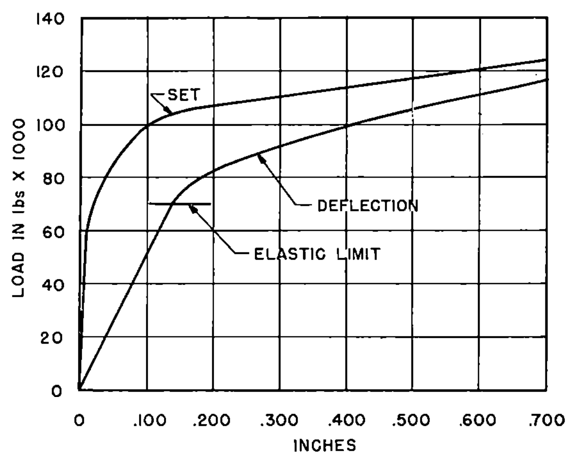


Fig. 8 Static deflection and set curve showing pedestal jaw spread on Design D

leg with 11/16 in. radius and 13/16 in. section. Local yielding was found at a load of 27,500 lb and a stress of 56,500 psi with the SR-4 gages. The graph, Fig.8, shows a load deflection-set curve of the static results of Table 5. Using the Johnson 50 percent method, the structure elastic limit was established at a load of 70,000 lb. The ultimate load occurred at 136,700 lb. Several fractures opened transversely as shown in Fig.9. Permanent jaw spread was greater than 1 in.

FATIGUE TESTS

Fatigue tests on outer pedestal legs were conducted in an automatic press of 100,000 lb capacity using a setup as shown in Fig.10. The load was applied at the roller-bearing centerline of the outer pedestal leg through a calibrated spring for adjusting load. The rocker assembly was used to change the load direction 180 deg to the pedestal leg.

Preliminary fatigue tests were made using a 28,000-lb load application to the outer leg of the pedestal. This load was inadequate to develop fatigue cracks in the critical radius. To determine an adequate load for fatigue, one pedestal was stress coated to locate SR-4 gages for calibration. Previous experience indicated a stress of 60,000 psi is needed to develop fatigue cracks in grade B steel. This stress level was found at

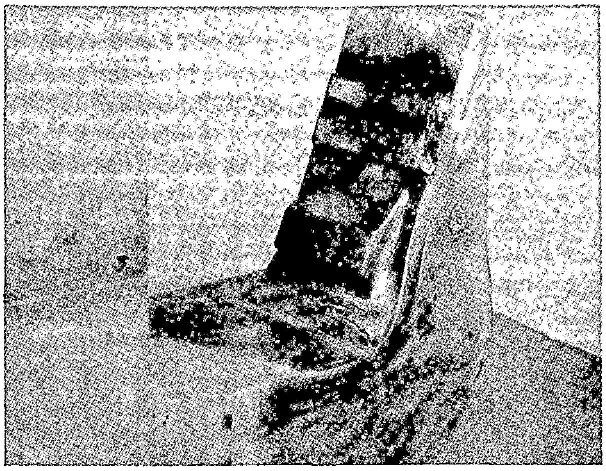

Fig. 9 Outer pedestal leg after static ultimate-load test to load of 136, 700 lb

TABLE 5

STATIC DEFLECTION & SET RESULTS

| LOAD | DEFLECTION | SET |
|---|---|---|
| 3000 | .006" | .0000" |
| 5000 | .010" | .0000" |
| 7000 | .014" | .0002" |
| 9000 | .018" | .0002" |
| 11000 | .022" | .0010" |
| 13000 | .026" | .0010" |
| 15000 | .031" | .0017" |
| 17000 | .034" | .0015" |
| 19000 | .039" | .0010" |
| 21000 | .042" | .0010" |
| 23000 | .046" | .0010" |
| 25000 | .050" | .0025" |
| 27000 | .056" | .0040" |
| 29000 | .058" | .0042" |
| 31000 | .061" | .0045" |
| 33000 | .066" | .0050" |
| 35000 | .070" | .0062" |
| 40000 | .080" | .0067" |
| 45000 | .091" | .0075" |
| 50000 | .098" | .0080" |
| 55000 | .109" | .0090" |
| 60000 | .119" | .0120" |
| 65000 | .129" | .0185" |
| 70000 | .141" | .0225" |
| 80000 | .181" | .0360" |
| 90000 | .284" | .0620" |
| 100000 | .407" | .1020" |
| 110000 | .576" | .2840" |
| 120000 | .773" | .5850" |

a load of 42,000 lb. All fatigue tests were conducted at this level.

Fatigue tests were made on the four designs illustrated in Fig.1, one group ground to nominal dimension for design tests and a second group as taken from production. The results of the fatigue tests are shown in Fig.11 with the number of cycles to the development of the first crack being the basis for comparison. The pedestal leg with the 5/8-in. radius and 3/4-in. metal section proved superior from both the design and production standpoints. No pedestal ends from production were tested with 11/16 in. radius and 13/16 in. section; however, the design tests proved this contour to be superior to other designs studied by the fatigue-test procedure. No pedestal legs were broken during fatigue test, although up to 600,000 loadings were accumulated in some instances. From the results of fatigue tests and the conditions of service fracture, it is indicated that fatigue was not a factor to be considered in resolving the problem.

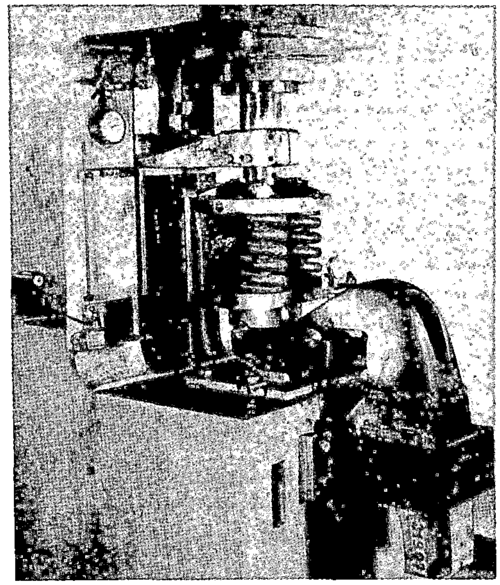

Fig.10 Fatigue test setup. Load applied at roller-bearing centerline through a calibrated load group and transfer rocker arrangement

TABLE 6

IMPACT TEST RESULTS
DESIGN COMPARISON DESIGN "A" - 3/8" RADIUS - 5/8" SECTION
NO RIB EXTENSION

DESIGN "A" (ORIGINAL) - 3/8" RADIUS - 5/8" SECTION

| DROP HEIGHT | TEMPERATURE | JAW SPREAD (SET) COPE | DRAG |
|---|---|---|---|
| 1' | 0°F | | |
| 1' | 0°F | | |
| 2' | 0°F | 3/" | 4/" |
| 2' | 0°F | 6/" | 7/" |
| 3' | 0°F | 10/" | 13/" |
| 3' | 0°F | 15/" | 17/" |
| 4' | 0°F | 25/" | 27/" |
| 4' | 0°F | 32/" | 33/" |
| 5' | 0°F | 42/" | 45/" |
| 5' | 5°F | End Snapped Off | |

Set In 64th Inch

## IMPACT TESTS

Impact tests were conducted by dropping a tup of 915 lb on a wedge arrangement of the same principle as used in static tests, except that a

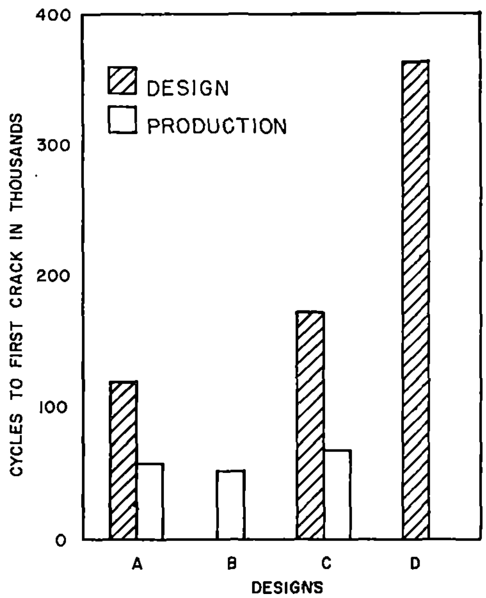


Fig. 11 Comparison of fatigue-test results; test stress level - 60, 000 psi

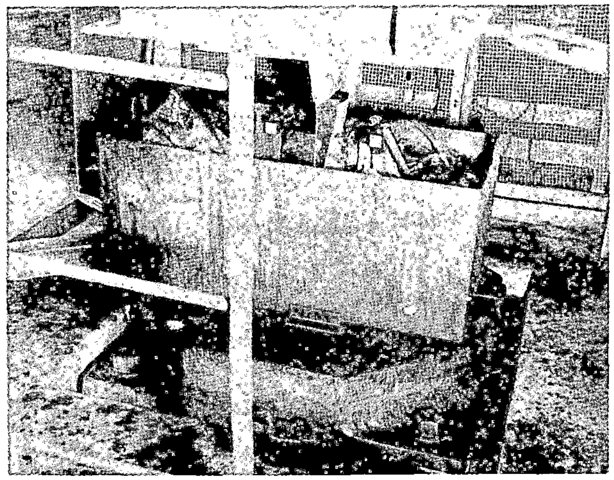

Fig. 12 Impact test setup; 915-lb tup dropped on wedge arrangement to apply load at roller-bearing centerline. Specimen immersed in ice-brine solution for testing at reduced temperatures

TABLE 7

IMPACT TEST RESULTS
DESIGN COMPARISON DESIGN "B" - 3/8" RADIUS - 5/8" SECTION
1¼" RIB EXTENSION

| DROP HEIGHT | TEMPERATURE | JAW SPREAD (SET) COPE | JAW SPREAD (SET) DRAG |
|---|---|---|---|
| Datum | 1 to 3°F | | |
| 1' | 1 to 3°F | | |
| 1' | 1 to 3°F | 1/" | 1/" |
| 2' | 1 to 3°F | 3/" | 3/" |
| 2' | 1 to 3°F | 8/" | 8/" |
| 3' | 1 to 3°F | 16/" | 17/" |
| 3' | 1 to 3°F | 22/" | 24/" |
| 4' | 1 to 3°F | 34/" | 35/" |
| 4' | 1 to 3°F | Fracture - End Snapped Off | |

Set In 64th Inch

TABLE 8

IMPACT TEST RESULTS
DESIGN COMPARISON DESIGN "C" - 5/8" RADIUS - 3/4" SECTION
NO RIB EXTENSION

| DROP HEIGHT | TEMPERATURE | JAW SPREAD (SET) COPE | JAW SPREAD (SET) DRAG |
|---|---|---|---|
| Datum | 1 to 3° F | | |
| 1' | 1 to 3° F | | |
| 1' | 1 to 3° F | | |
| 2' | 1 to 3° F | 4/" | 3/" |
| 2' | 1 to 3° F | 8/" | 7/" |
| 3' | 1 to 3° F | 15/" | 15/" |
| 3' | 1 to 3° F | 24/" | 20/" |
| 4' | 1 to 3° F | 33/" | 31/" |
| 4' | 1 to 3° F | 44/" | 41/" |
| 5' | 1 to 3° F | 56/" | 53/" |
| 5' | 1 to 3° F | 1-5/" | 1-2/" |
| 6' | 1 to 3° F | Fracture - End Snapped Off | |

Set In 64th Inch

heavier block to withstand the impact forces was substituted for the load cell. Loading was again on the roller-bearing centerline. After several attempts to obtain comparative data on the impact resistance of the designs illustrated in Fig.1, it was decided to reduce the test temperature. To reduce the temperature, the entire pedestal was immersed in a tank of ice and water with salt added proportionately to give the desired temperature. The pedestal was held in a specially designed fixture, so the test could be made while the pedestal was immersed, thereby maintaining a constant temperature during impact. The impact setup is shown in Fig. 12.

Tests for comparison of design strength were conducted in the 0 to 5 F temperature range. Impacts developed at speeds to be abusive to the point of fracture at room temperature were beyond the range of our laboratory equipment. This paralleled the car-impact tests which were conducted with ambient temperature well above freezing. Therefore, it was necessary to reduce the plasticity of the material to do comparative testing in the laboratory.

Tables 6, 7, 8 and 9 show the average results of the impact tests on several pedestals of each design. These are also graphically illustrated in Fig.13. The actual range of chemistry of the pedestals used in this series of impact test was as follows:

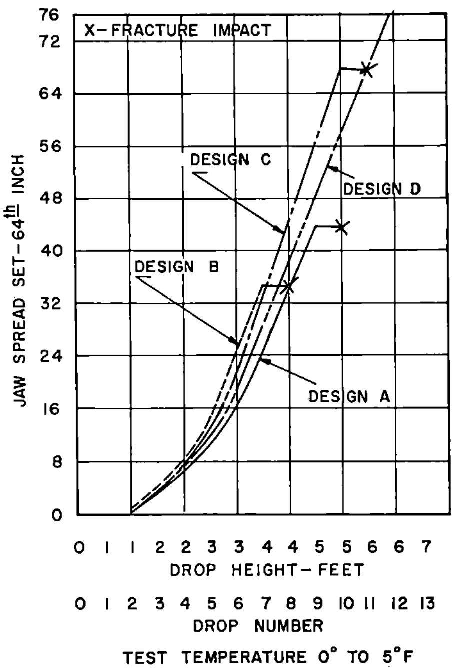


Fig. 13 Comparison of impact resistance of outer pedestal designs; loading at roller-bearing centerline

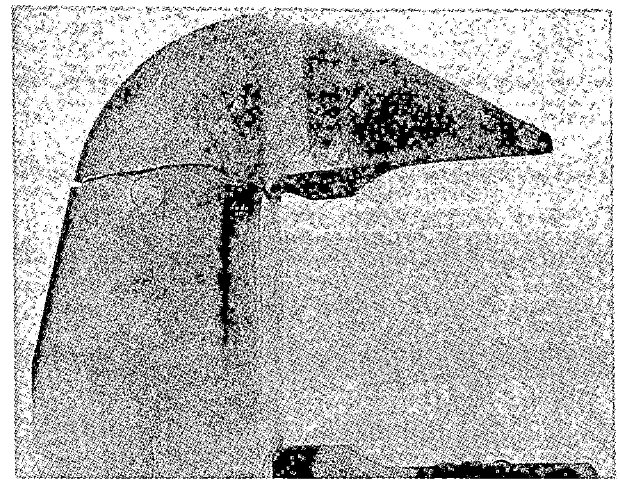

Fig. 14 Fracture due to low-temperature impact test on outer pedestal leg. The fracture was a duplication of those occurring in service

TABLE 9

IMPACT TEST RESULTS
DESIGN COMPARISON DESIGN "D" - 11/16" RADIUS - 13/16" SECTION
NO RIB EXTENSION

| DROP HEIGHT | TEMPERATURE | JAW SPREAD (SET) COPE | JAW SPREAD (SET) DRAG |
|---|---|---|---|
| 1' | 2 to 3° F | | 1/" |
| 1' | 2 to 3° F | | 1/" |
| 2' | 2 to 3° F | 3/" | 3/" |
| 2' | 2 to 3° F | 6/" | 6/" |
| 3' | 2 to 3° F | 11/" | 10/" |
| 3' | 2 to 3° F | 19/" | 19/" |
| 4' | 2 to 3° F | 28/" | 28/" |
| 4' | 2 to 3° F | 38/" | 38/" |
| 5' | 2 to 3° F | 48/" | 48/" |
| 5' | 2 to 3° F | 57/" | 58/" |
| 6' | 2 to 3° F | 67/" | 68/" |
| 6' | 2 to 3° F | 76/" | 77/" |
| 7' | 2 to 3° F | 87/'' | 88/" |

Set In 64th Inch

| C | MN | Si |
|---|---|---|
| 0.28 - 0.31 | 0.65 - 0.67 | 0.38 - 0.41 |

| P | S |
|---|---|
| 0.005 - 0.009 | 0.017 - 0.021 |

Results show that the addition of the $1\frac{1}{4}$-in. extension to the rib over the journal of the original design did not appreciably improve the impact resistance of the outer pedestal leg. Improvement was found; however, when the 5/8-in. radius and 3/4-in. metal section were incorporated in the design. The greatest improvement was in Design D (11/16 in. radius). In this case fracture did not occur. The use of the larger radius reduced the offset between the critical inner radius and the outer radius to allow the stresses caused by impact to be uniformly distributed across the entire width of the pedestal.

Examination of the fractures showed that in the original design and the design with the rib extension with the 3/8-in. radius, the fracture started approximately 45 deg above the horizontal and extended in an arcuate pattern to the vertical at an angle of 65 to 75 deg from the 3/8-in. radius. In the case where the 5/8-in. radius was incorporated in the design, the fracture started at a point on the inner radius approximately 30 deg above the horizontal and extended into the metal section for a distance of 1 in. at an angle of 50 to 55 deg above the horizontal. From this point, the fracture curved in an arcuate pattern to the vertical. The fracture on the casting with the extended rib passed directly through the heaviest part of the rib extension. Fig.14 shows a typical fracture of Design C with 5/8 in. radius. The chevron pattern of the brittle fracture is shown in Fig.15. The effect of the test on Design D (11/16 in. radius) is shown in Fig.16.

The foregoing impact tests were based entirely on drop height. To complete the impact story, it was necessary to determine force applied

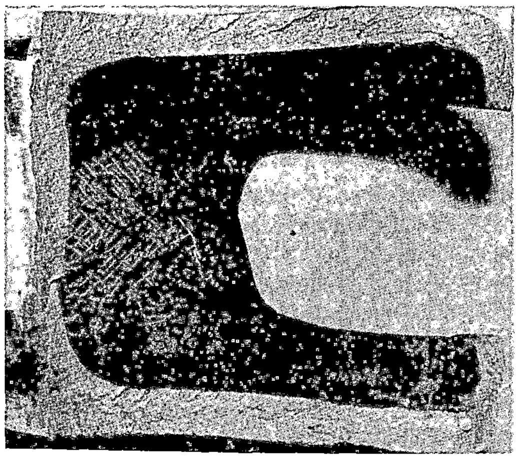

Fig. 15 Metal section after fracture in low-temperature impact test. Note chevron pattern in metal surface indicating a brittle-type fracture

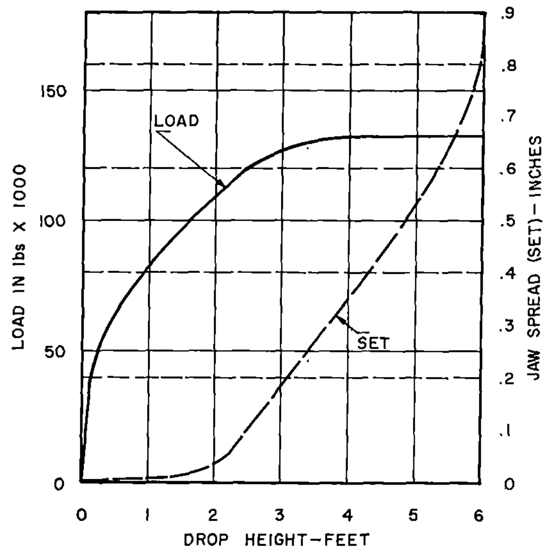


Fig. 17 Load induced by impact on outer pedestals

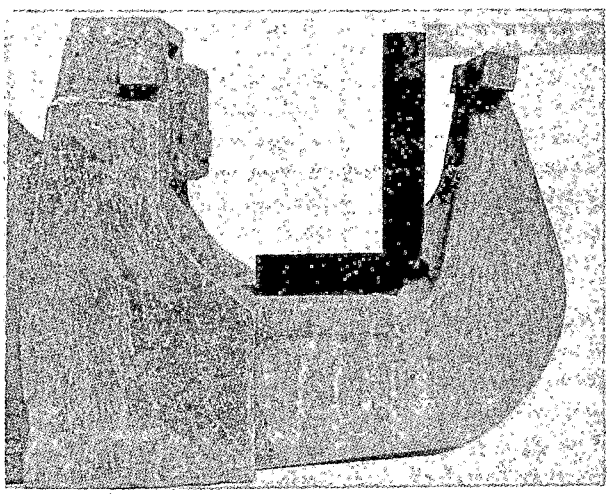

Fig. 16 Outer pedestal of Design D (11/16-in. radius - 13/16-in. section) of impact tests under 915-lb drop hammer from a maximum height of 7 ft

TABLE 10

COMPARISON OF CHEMISTRY ON IMPACT RESISTANCE
DESIGN "A" - 3/8" RADIUS - 5/8" SECTION
NO RIB EXTENSION

| CHEMICAL ANALYSES | C. | MN. | SI. | P. | S. |
|---|---|---|---|---|---|
| Specimen No. 1 | .264 | .70 | .52 | .018 | .039 |
| Specimen No. 2 | .29 | .65 | .38 | .010 | .019 |

| DROP HEIGHT | TEMPERATURE | JAW SPREAD SPECIMEN NO. 1 COPE | JAW SPREAD SPECIMEN NO. 1 DRAG | JAW SPREAD SPECIMEN NO. 2 COPE | JAW SPREAD SPECIMEN NO. 2 DRAG |
|---|---|---|---|---|---|
| 1' | +25° F | 0" | 0" | 0" | 0" |
| 1' | 25° F | 0" | 0" | 0" | 0" |
| 2' | 25° F | 4" | 4" | 4" | 4" |
| 2' | 25° F | 7" | 7" | 7" | 7" |
| 3' | 25° F | 12" | 13" | 16" | 16" |
| 3' | 25° F | 19" | 20" | 25" | 25" |
| 3' | 25° F | 22" | 22" | 32" | 33" |
| TEMPERATURE LOWERED | | | | | |
| 3' | 10° F | | | 40 | 41 |
| 4' | 10° F | End Snapped Off | | 47 | 48 |
| 4' | 10° F | | | 55 | 56 |
| 5' | 10° F | | | 1-1/ | 1-3/ |
| 5' | 10° F | | | 1-6/ | 1-11/ |
| 6' | 10° F | | | 1-20/ | 1-24/ |

Set in 64th Inch.

to the outer pedestal. A heavy-duty load cell was substituted for the solid load block and an impact test conducted on a pedestal end. An oscilloscope and Polaroid camera were used for recording equipment. Fig.17 shows the impact load on the pedestal and the resulting set. The maximum force of 132,600 lb was applied at 4-ft drop height. This value remained constant through the 6-ft drop. It was found that as the plastic deformation of the pedestal leg increases, the rate of force application decreases.

## METALLURGICAL STUDIES

Indications from service and the foregoing laboratory tests that the fractures were of a brittle nature led to a study of the variations in the chemistry range allowable in AAR Specification M-201. Research laboratory studies show that as the phosphorous and sulfur content of steel increases, the transition temperature of the steel is increased. The transition temperature is generally understood to be that temperature at which

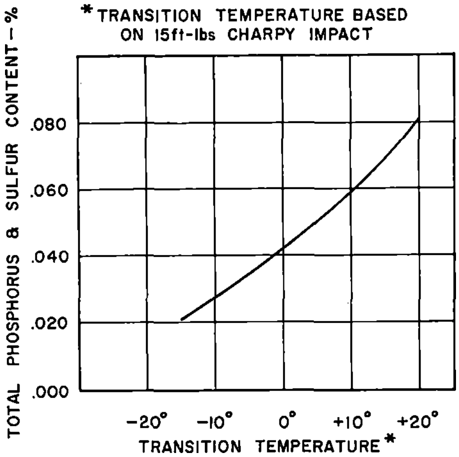


Fig. 18 Effect of phosphorous and surfur. Content of transition temperature of Grade B steel

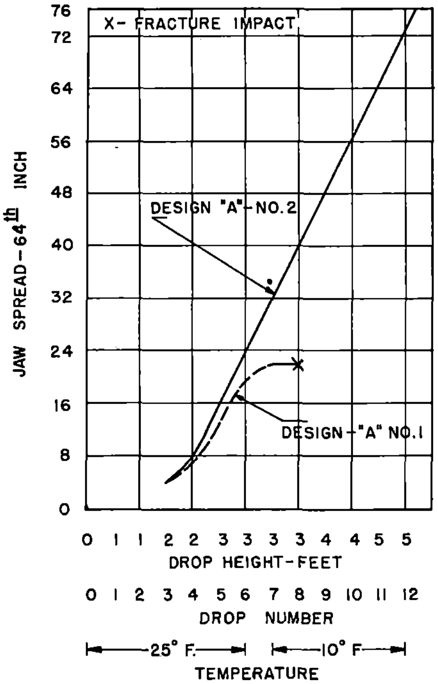


Fig. 19 Comparison of variation in Metal Chemistry and effect on impact resistance

the elasticity of the material is reduced and a condition of brittleness is inherent in the material. Some investigators have chosen an arbitrary value of 15 ft-lb as determined in Charpy V notch tests as the point where brittleness can be expected. Results of charpy V-notch tests conducted on various steel compositions are shown in Fig.18.

Results of impact tests on two pedestal ends of Design A (original), but of different chemistries, are shown in Table 10 and are graphically compared in Fig.19. The results of these tests show that with a total sulfur and phosphorous content of 0.029 percent, it did not reach the transition temperature at 10 F and continued to take set by spreading, indicating little, if any, loss in ductility. However, when the phosphorous-sulfur content was increased to 0.057 percent ductility was drastically reduced and it is indicated 10 F is below the transition temperature of the steel.

SUMMARY

Static tests were conducted on outer pedestals to the point of disclosing the location, orientation and value of stress concentration. These tests confirmed the analytical studies of the strengths of the various designs. Static tests to ultimate, disclosed the structure to have a yield point at a load of 70,000 lb and an ultimate load of 136,700 lb. Yield stress of the material was 56,500 psi. These values are beyond the range of any static load that could be applied to the pedestal in normal service, thereby eliminating this loading as a cause of failure.

Fatigue tests at a stress level of 60,000 psi which is above the 56,500 psi localized yield point found in static test, did not cause failure even though loadings on the order of 600,000 were accumulated on each pedestal.

Impact tests at ambient laboratory temperatures did not produce fractures as evidenced in service. However, considerable bending outward of the pedestal was caused. Had this deformation occurred in service it would be readily noticeable and the conditions corrected before any serious complications could develop. This deformed condition has not been reported in service to any extent.

To reproduce service failure, it was necessary to reduce the temperature of the specimen. We were then able to duplicate the condition insofar as fracture location and appearance is concerned. The results of the impact tests on the various designs are interesting, in that by modification to the radius of curvature at the junction of the pedestal leg and increasing metal sections, the impact resistance can be improved. It was found that the chemistry of the material can have an equal effect in improving impact resistance. Tests on two pedestals of comparable contour and metal thickness indicated a greater impact resistance at lower temperatures when the phosphorous and sulfur content of the material was reduced, even though both specimens were produced within the chemical composition tolerances.

CONCLUSIONS

Laboratory investigations eliminated the possibility of fracture from static or fatigue loading. The possibility of fracture from impact at normal operating temperatures is also remote. It was fairly well established that fracture was due to severe impact at low ambient temperature.

The design with increased radius and metal section can have improved impact resistance even under conditions of low temperature.

The original design has excellent impact resistance and has given satisfactory performance. It is felt important that phosphorous and sulfur content be kept below the maximum permitted by the present specification, or that other steps be taken to keep the low-temperature transition point as low as practical.

ACKNOWLEDGMENT

The data used to determine service conditions were supplied by the several members of the Truck Manufacturers Engineers' Committee. National Castings Company conducted a series of car impact tests and supplied data to the TMEC which assisted greatly in evaluating the results of the laboratory studies which were conducted by American Steel Foundries Incorporated.

An independent study was subsidized by Birdsboro Corporation and the conclusions were made available to the TMEC. Credit must be given to these companies, their engineering staffs, and to the Engineering Staff of American Steel Foundries for the gathering of widely scattered information which led to the successful conclusion of this study.

APPENDIX

CURVED-BAR BENDING THEORY

In many machine-design problems involving the stress concentration caused by curvature of the main member, it has been found that the use of curved-bar bending theory will provide an accurate solution. Electric strain-gage measurements have indicated good agreement with this theory. The agreement for the pedestal end of a side frame is well within the accuracy desired, and so a brief description of the method of computation is given.

It is generally known that the stress distribution in curved bars is not linear but follows a hyperbolic curve. Also, the neutral axis for a curved-bar section does not coincide with the gravity axis, as in the case of straight-bar bending.

One solution for this problem is developed by S. Timoshenko in "Strength Of Materials," Part II (p. 65, second edition). It is stated that this solution is approximate, but it is in good agreement with the exact solution. A brief explanation of fundamental principles involved in curved-bar bending may be summarized.

During bending, the fibers of a straight bar are elongated on the convex side and shortened on the concave side. The strain of compression and tension fibers equidistant from the gravity axis are equal in magnitude, as the elemental lengths between two plane sections before bending are equal. Strain is determined as the quotient of the length of deformation divided by the original length. However, the strain due to bending in curved bars is not equal for compression and tension fibers equidistant from the neutral axis. The deformations at equidistant points above and below the neutral axis are equal, but the elemental lengths between two radial sections vary. It may be seen that the elemental arc at the concave surface is minimum and the deformation is maximum, therefore, the strain on this surface is maximum.

In order to meet the condition of equilibrium on a cross section, the neutral axis will not coincide with the gravity axis. The neutral axis

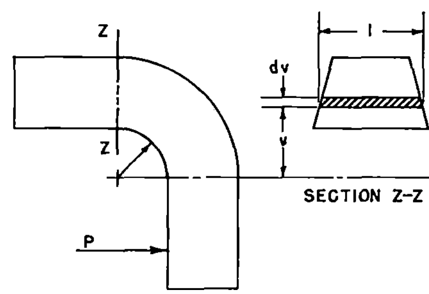


Fig. 20 Typical application section

is shifted toward the concave surface, so that the summation of the distributed normal forces on the cross section is equal to zero.

The equation denoting the location of the neutral surface with respect to the center of curvature for any cross section is as shown in Fig. 20:

$$r = \frac{A}{\int_A (dA/v)}$$

where

$$dA = \ell \, d\,v$$

$v$ = distance from center of curvature to any point in section

$A$ = cross section area of section being considered

$\ell$ = width of section (written as a function of $v$)

The procedure for determining the bending stress with stress concentration is shown in Fig.

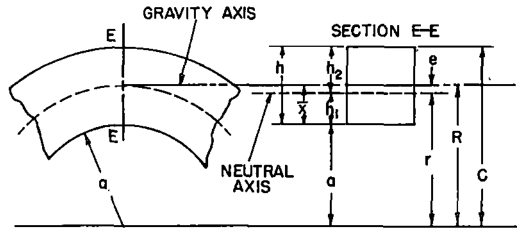


Fig. 21 Theoretical nomenclature of problem

21. In this figure the nomenclature is as follows:

$e = R - r$

$h_1 = \bar{x} - e$

$h_2 = \bar{x} + e$

$\sigma_1 = \dfrac{Mh_1}{Aea}$

$\sigma_2 = \dfrac{Mh_2}{Aec}$

$S = \dfrac{M}{SM}$

$a$ = inner radius of curvature

$c$ = outer radius of curvature

$\bar{x}$ = distance from base of section to gravity axis

$R$ = distance from center of curvature to gravity axis

$r$ = distance from center of curvature to neutral axis

$M$ = bending moment with respect to gravity axis

$\sigma_1$ = bending stress with stress concentration at concave surface

$\sigma_2$ = bending stress with stress concentration at convex surface

$e$ = distance of neutral axis from gravity axis

$S$ = nominal bending stress

$SM$ = section modulus

For irregular sections, the location of the neutral axis with respect to the center of curvature may be determined by dividing the irregular section into a number of elementary section, i.e., rectangles, triangles, and so on, and using the summation in the general equation. This means that the numerator is equal to the total area of the section, and the denominator is the summation of the values for the particular integrals of each elementary section.

A numerical example using the principle of summing elementary sections to obtain the shift of the neutral axis of a irregular section may be illustrated in the example shown in Fig.22.

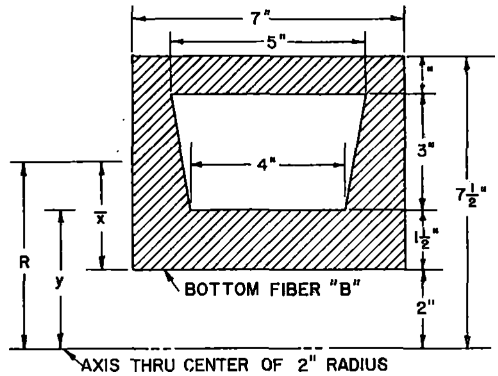


Fig. 22 Example problem for application of theory

$A = 25.00$ sq in.

$\bar{x} = 2.585$ in.

$R = 4.585$ in.

$I = 85.03$ in$^4$

$S.M_B = 32.89$ in$^3$

$r = 3.847$ in.

$e = 0.738$ in.

$h_1 = 1.847$ in.

$$\sigma_1 = \frac{1.847\ M}{25.00 \times 0.738 \times 2.00} = 0.0501\ M$$

$$S = \frac{M}{32.89} = 0.0304\ M \text{ (tension)}$$

The stress-concentration factor in this example is

$$\frac{0.0501}{0.0304} = 1.65$$

This method has been found to be quite practical in problems similar to this example, and good accuracy has been indicated by laboratory stress analysis.

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# Investigation of the Thermal Capacity of Railroad Wheels Using COBRA Brake Shoes

*An objective evaluation of the thermal capacity of 36 in. CR wheels braked with COBRA brake shoes is presented and related to that of wheels braked with cast metal brake shoes. The wheel tread conditions, hardness and macrostructure of the rim, and residual stress patterns, which developed from high speed dynamometer braking-were investigated through three progressive test series. The results indicate that the thermal capacity of wheels braked with COBRA brake shoes far exceeds limits previously established for cast metal shoes.*

## Introduction

A GROWING concern has developed within the railroad industry that on-tread braking may be approaching its limit, due to the inability of the wheel to adequately act as a brake drum with the increasing demands imposed by new heavier cars and higher operating speeds. One cause of this concern was two explosive type wheel failures, which occurred on two commuter type passenger cars during the winter of 1968. These unfortunate wheel failures, and the events which followed, focused the attention of the industry on the subject and thereby established the basis for the present investigation.

The particular cars involved were the Penn-Central "Silverliners," which were equipped with CR wheels and on-tread brakes with composition brake shoes. As a result of the wheel failures, and a lack of documented data on wheel performance with composition shoes, the decision was made to change from CR to AR wheels on these cars. In accordance with this decision, the CR wheels on the Northeast Corridor Project cars, commonly known as the Metroliners, were also replaced with AR wheels. This change from Class C to Class A wheel was predicated on published data covering wheel performance experienced with cast metal brake shoes.

While several research projects have been conducted in the past with the purpose of understanding and solving wheel problems related to braking heat, all of the tests reported in such research have been conducted with cast metal type brake shoes. Consequently, the results of these tests have established limitations with respect to the braking duty which wheels can safely endure with cast metal brake shoes. It should not be taken for granted, however, that the thermal limitations established through these tests automatically apply to wheels braked with composition brake shoes; in fact, experience in service has indicated the contrary to be true. Specifically, three prototype Pioneer Cars which operated in the same commuter type service as the "Silverliners" for over seven years were equipped with CR wheels and COBRA brake shoes, and during that time no thermal problems were experienced with the CR wheels.

In view of this situation, it was deemed advisable to develop data on the thermal limitations of CR wheels braked with COBRA brakes shoes, and to relate these thermal limitations to specific braking duty.

Throughout this paper COBRA composition brake shoes are referred to because they were used in the tests. It is not known if other brands would produce identical results.

The wheel tests described in this paper were conducted through the combined efforts of a railroad, a composition brake shoe manufacturer, and a wrought steel wheel manufacturer. The railroad provided the necessary test wheels along with unbraked control wheels from the same heat, which were taken from the wheels removed from Metroliner cars prior to being placed in service. The brake shoe manufacturer provided the full scale brake dynamometer facilities and the necessary test shoes. The wheel manufacturer, who manufactured the test wheels originally, provided laboratory test facilities for metallurgical analysis of all test wheels and control wheels.

## Highlights of Previously Published Data

The most familiar studies of wheel and brake shoe limits in the past were carried on at the University of Illinois. The first is reported in the University of Illinois Engineering Experiment Station Bulletin No. 301, "The Friction of Railway Brake Shoes at High Speeds and High Pressure," by Herman J. Schrader, May

Contributed by the Railroad Division for presentation at the Winter Annual Meeting, Los Angeles, Calif., November 16–20, 1969, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Manuscript received at ASME Headquarters, July 22, 1969. Paper No. 69-WA/RR-2.

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1938. The second is the University of Illinois Engineering Experiment Station Bulletin No. 387, "The Effect of Brake Shoe Action on Thermal Cracking and on Failure of Wrought Steel Railway Car Wheels," by Wetnkamp, Sidebottom and Schrader in cooperation with the Technical Board of the Wrought Steel Industry, June 1950. Both of these test series were conducted on a full scale dynamometer, using cast metal brake shoes and a clasp brake arrangement.

The 1938 tests were significant in establishing a limitation for on-tread braking duty. Although wheels at that time were not classified by the Association of American Railroads with respect to heat treatment or carbon content, the particular wheels used in the tests were 0.73 percent carbon, homologous to a Class U wheel in the present specification.

One of the conclusions drawn from the results of these tests was: *"Shoe pressures of 20,000 pounds, combined with high speeds, cracked the wheel tread at a very rapid rate, and the rate of performing work on the wheel should be kept below 125,000 ft.-lb. per second in order to avoid this type of failure."*

However, it was clearly stated that the conclusions were applicable only to the types of brake shoes and wheel tested.

The 1950 Illinois tests were more extensive, consisting of both high energy stop-type braking and long duration drag-type braking. The drag test phase of this test series primarily related the occurrence of brittle wheel fractures to wheel design and heat treatment, and the stop test phase related the development of thermal cracks to the carbon content of the wheels.

The stop tests were conducted with a simulated wheel load of 20,000 lb, stopping from 115 mph, with a shoe force of 20,000 lb on each of two cast iron brake shoes in a clasp arrangement. This braking duty represented a total kinetic energy of 9,000,000, ft-lb per stop.

Of the five Class CR wheels which were tested in this manner, three developed thermal cracks after only one stop. The remaining two wheels developed cracks after five stops and seven stops, respectively.

A third series of tests, very similar to those reported in Illinois Bulletin No. 387, was presented in ASME Paper 60-RR-1, 1960, by J. M. Wandrisco and F. J. Dewez, Jr., titled "Study of the Defects That Originate and Develop in the Treads of Railroad Wheels During Service."

These tests simulated stops with a 31,000 pound wheel load, from the same speed of 115 mph, and with the same shoe force of 20,000 pounds on each of two cast metal brake shoes.

This study was extremely valuable in that it resulted in a much better understanding of the mechanism theories for the formation of defects that develop in wheels as a result of braking heat. It is noteworthy to extract one particular result of these tests which supports the results obtained in 1950 at the University of Illinois: the average number of stops required to produce a thermal crack in CR wheels was *one* stop.

## Test Facilities

The full scale, dual wheel brake dynamometer on which the braking tests were conducted is located in Wilmerding, Pa. A general view of this facility is shown in Fig. 1. The machine is one of the newest brake dynamometers in the world, designed specifically to simulate actual railroad braking conditions. The dual wheel arrangement permits a direct performance comparison of different types of brake shoes or wheels, while maintaining identical test conditions.

Operation of the dynamometer is controlled from an instrument console, located in a control room overlooking the actual test stand. The instrument control panel and the adjacent recording instruments provide the means of keeping continuous, permanent records of speed, brake cylinder pressure, wheel tread temperature, torque, stop distance, and stop time, during all modes of testing.

The metallurgical laboratories are located at the wheel manufacturing facility at Burnham, Pa. For this program, test wheels were subjected to metallurgical tests for macrostructure, hardness, and stress analysis, the latter by a strain gage technique. Results of these tests are included in the paper.

## General Test Data

The subject wheel tests were conducted in three phases, each of which was devised to investigate particular questions concerning the braking limitations of Class CR wheels. The nature of each of these phases, as they are designated in this paper, was:

Test Phase 1 involved a series of dynamometer stop tests investigating the relative effects of COBRA brake shoes and cast metal brake shoes, under identical braking conditions.

Test Phase 2 involved a higher level of braking than Phase 1, investigating the relative effects of plain and flanged style COBRA brake shoes.

Test phase 3 involved an even higher braking level, again investigating both plain and flanged style COBRA brake shoes, with the intent of establishing an upper braking limit which Class CR wheels can endure with COBRA brake shoes.

The three test phases basically consisted of conducting series of dynamometer stop braking tests on two test wheels, under systematic, carefully controlled test conditions, and forwarding each wheel to the laboratory for metallurgical analysis upon completion of the prescribed braking tests.

The wheels utilized for these tests were 36 in. CR, fully machined, shotpeened wheels of modified Standard AAR design having a conventionally sloped plate and multiwear rim. The particular wheels used for any single test phase were from the same heat, and a third wheel from each heat was used as an unbraked, control wheel to provide metallurgical data for such wheels as they were manufactured.

## Test Phase 1

The purpose of this test series was to obtain a direct comparison of wheel performance of wheels braked with COBRA brake shoes to that of wheels braked with cast metal brake shoes, and also to obtain a relative verification of past test results with cast metal shoes. In line with these objectives, the braking conditions for the dynamometer stops were established as a combination of the conditions used in past wheel tests and Metroliner braking conditions.

The equivalent wheel load and COBRA brake shoe braking force directly simulated the Metroliner cars. The kinetic energy level per stop was set at slightly over 9,000,000 ft-lb to match the work level per stop of the 1950 University of Illinois tests. With the Metroliner wheel load, this required an initial speed of 110 mph. Table 1 presents the dynamometer test conditions for this series of emergency stop tests.

The dynamometer stop test series consisted of making alternate, 110 mph, simulated Metroliner emergency stops on wheel No. 10793, braked with a single flanged COBRA brake shoe, and wheel No. 8925, braked with cast metal brake shoes in a clasp brake arrangement. To insure that the braking work done on wheel No. 8925 with cast metal brake shoes would be directly comparable to the braking work done on wheel No. 10793 with the COBRA brake shoe, it was necessary that the work rate for each stop, as well as the total work for each stop, be equal. Since the wheel load and initial speed were the same for both wheels, the total work per stop ($1/2\ mv^2$) was necessarily equal for both wheels. To obtain equal work rates, it was therefore only necessary to obtain equal stop times on the two test wheels.

The shoe force on the COBRA flanged brake shoe was set to match the Metroliner shoe force for an emergency application. The cast metal brake shoe force on clasp brakes was fixed to obtain the same stop time as that obtained with the COBRA brake shoe, from 110 mph.

**Table 1**

| | Wheel No. 8925 cast metal brake shoes | Wheel No. 10793 COBRA flanged brake shoes |
|---|---|---|
| Wheel location | Outside | Inside |
| Equivalent wheel load | 23,128 lb | 23,083 lb |
| Total simulated car weight | 185,024 lb | 184,644 lb |
| Brake arrangement | clasp shoes | single shoes |
| Net shoe force per wheel | 19,150 lb | 7,419 lb |
| Initial speed | 110 mph | 110 mph |
| Kinetic energy per stop ft-lb | 9,359,413 | 9,341,203 |
| Average stop distance | 6,845 ft | 5,225 ft |
| Average stop time | 68.02 sec | 66.31 sec |
| Average work rate per stop ft-lb/sec | 137,598 | 140,872 |
| Average maximum wheel tread temperature | 1033°F | 664°F |
| Average retardation rate | 1.62 mphps | 1.66 mphps |
| Total braking work done on wheels* ft-lb | 470,379,039 | 697,053,524 |

*Includes wear-in stops

Wheel No. 10793, with the single flanged COBRA brake shoe, was mounted on the inside location on the dynamometer, and wheel No. 8925, with the clasp cast metal brake shoes, was mounted on the outside location. The inside wheel is positioned closest to the dynamometer inertia disks; the actual positions of the two wheel locations are shown in Fig. 1.

Prior to beginning the emergency stop series from 110 mph, light stops from 60 mph were made to wear the brake shoes into the exact contour of the wheel. The 110 mph emergency stops were then made, alternately, on the two test wheels. After each stop, both wheels were visually examined for any signs of thermal cracking. Since wheel No. 8925, braked with the metal brake shoes, developed a very rough surface of shoe metal deposits with each stop, the wheel tread was cleaned with a grinding disk after each stop. Both wheels were allowed to air cool to 300 deg F after each pair of stops. They were then water cooled to 100 deg F prior to the next stop.

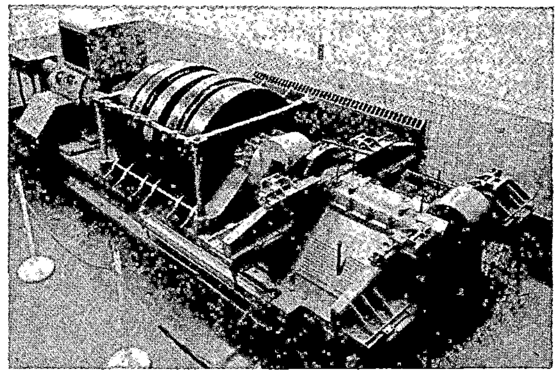


Fig. 1 View of test dynamometer showing dual wheel arrangement

It was the original intent to make fifty emergency stops on each wheel, provided defects did not develop in either wheel which would create an unsafe testing condition.

## Dynamometer Test Results

After approximately twenty emergency stops were completed on each wheel, wheel No. 8925 (metal brake shoes) began to develop hairline, skin-deep thermal checks. The abundance of these thermal checks increased with each successive stop. When thirty-seven stops were completed on wheel No. 10793 with the flanged COBRA brake shoe, and the dynamometer was being accelerated for the thirty-seventh stop on wheel No. 8925, a sharp "ping" was heard. Instead of proceeding with the emergency stop, the machine was stopped without brakes on the wheel in question. An examination revealed a severe thermal crack extending almost across the complete width of the tread of wheel No. 8925. The tests were discontinued at this point.

This braking duty, dissipating over 9,000,000 foot-pounds of kinetic energy at rates of over 137,000 foot-pounds per second with cast iron brake shoes and over 140,000 foot-pounds per second with the COBRA brake shoe, is severe by any practical service standards, and exceeds the previously recommended limit of 125,000 foot-pounds per second.

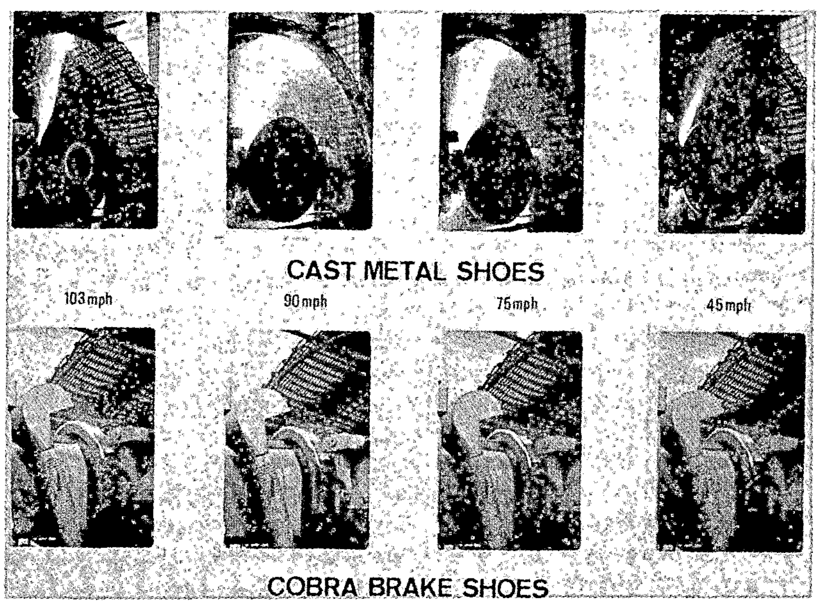



Fig. 2 Sequential photographs taken during typical 110 mph emergency stops

**Table 2**

| | C | Mn | P | S | Si |
|---|---|---|---|---|---|
| Wheel 8927 (unbraked) | 0.75 | 0.68 | 0.015 | 0.034 | 0.38 |
| Wheel 8925 (metal) | 0.73 | 0.74 | 0.012 | 0.040 | 0.35 |
| Wheel 0793 (COBRA) | 0.73 | 0.74 | 0.013 | 0.040 | 0.33 |
| M-107 Cl. CR. | 0.67/0.77 | 0.60/0.85 | 0.05 max | 0.05 max | 0.15 min |

The severity of the braking conditions for these 110 mph stops was portrayed vividly by the spectacular flaming which occurred with the clasp cast metal brake shoes. Fig. 2 is a series of photographs taken in sequence during a typical 110 mph emergency stop with two cast metal brake shoes and with a single COBRA brake shoe. These photographs show a direct comparison of the severe flaming encountered with the metal brake shoes with the only slight smoking which occurred with the COBRA brake shoe.

## Wheel Metallurgy Test Results

### Chemical Analysis

A check analysis of the test wheels showed that the chemistry of the three wheels were within the requirements of AAR Specification M-107 Class CR (Table 2).

### Magnaflux Examination

The test wheels were wet magnetic particle inspected and no evidence of tread or flange defects was found on wheel serial 10793 (flanged COBRA brake shoes).

Wheel No. 8925 revealed only the visual thermal crack progressed 3 in. across the tread from the front face to the flange, as shown in Fig. 3. Fig. 4 shows the fracture face of the 3-in.-long thermal crack on wheel serial 8925 exposed by sectioning the rim

**Fig. 3 Wheel No. 8925 showing thermal crack developed during braking with cast metal brake shoes**

### Macro Examination

One radial section was removed from each wheel for macro inspection. Fig. 5 shows a Brinnell hardness survey of an unbraked, as-manufactured, wheel. Wheel No. 10793, which had been braked with a flanged COBRA brake shoe, showed a freedom from nonuniform thermal penetration effect in the rim as shown in Fig. 6. This wheel did not display any evidence of localized overheating at the tread.

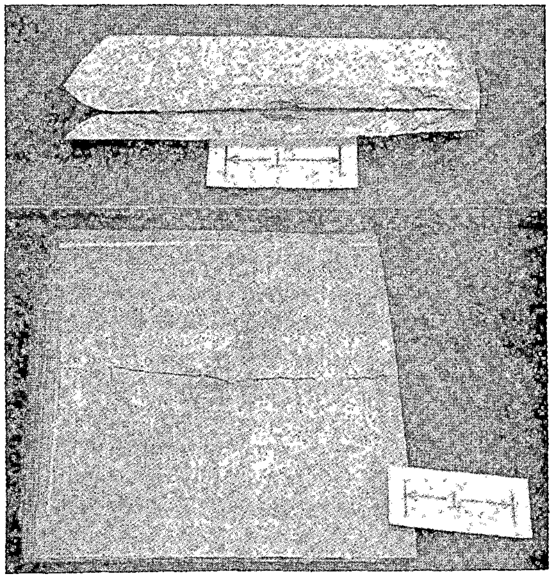


**Fig. 4 Thermal crack exposed by sectioning from wheel No. 8925**

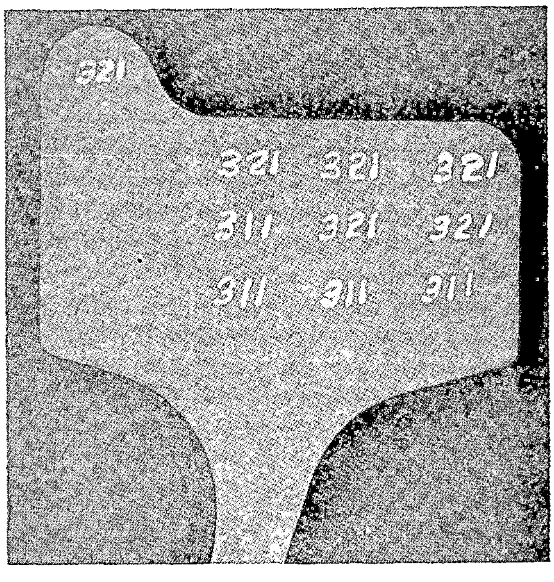



***Fig. 5 Hardness survey of as-manufactured wheel No. 8927***

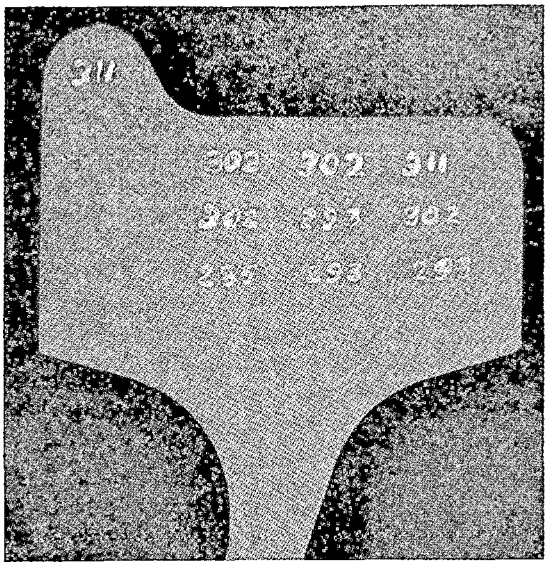



Fig. 6 Hardness survey of wheel No. 10793 braked with flanged COBRA brake shoe

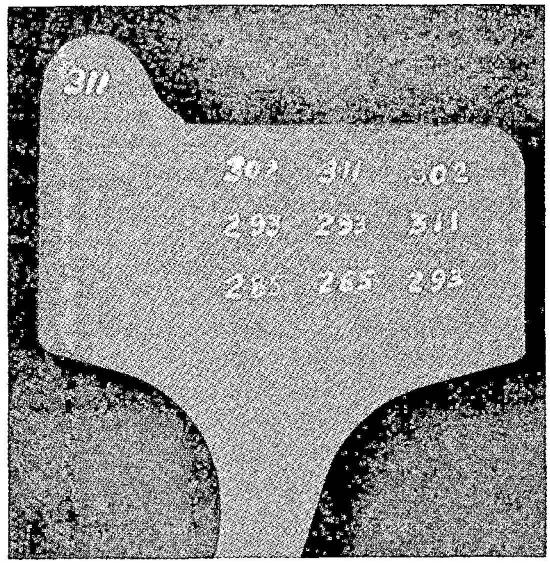



Fig. 7 Hardness survey and heat affected zone of wheel No. 8925 braked with clasp cast metal brake shoes

Wheel No. 8925, braked with clasp cast metal brake shoes, shows indications of severe thermal penetration locally on the tread, as shown in Fig. 7. It was apparent that cast metal brake shoes produced an entirely different type of heat penetration than that of the COBRA brake shoe.

### Stress Results

BLH type AX-5 strain gages were applied to the wheels as shown in Fig. 8. Gages were applied radially at 0 deg and 120 deg on both sides of the wheels. Residual stress measurements are shown in Table 3. Fig. 9 shows the location of these measurements on the wheel cross section.

The results of stress analysis indicate that greater changes in plate and rim compressive stresses occurred in the wheel braked with cast metal shoes. Metallurgical examination also revealed a uniform thermal penetration of the rim of the wheel braked with the COBRA brake shoe compared with a severe localized heating of the tread of the wheel braked with cast metal brake shoes.

## Test Phases 2 and 3

The second series of wheel tests were conducted to investigate the braking limitations of CR wheels braked with COBRA brake shoes, and also to determine the relative thermal effects of plain and flanged style brake shoes under identical braking conditions. The test conditions were very similar to those outlined for Test Phase I, with the exception that the initial speed was 150 mph.

Test Phase 2 consisted of conducting thirty 150 mph simulated Metroliner emergency stops on wheel No. 618, braked with a single COBRA flanged brake shoe, and thirty 150 mph simulated Metroliner emergency stops on wheel No. 566, braked with a single plain-style COBRA brake shoe. The test procedure was identical to that described for Test Phase 1, as alternate stops were made on the inside and outside wheel until thirty stops had been completed on each. Wheel No. 613 was used as an unbraked control wheel for metallurgical examination.

Test Phase 3, the third wheel test series, was conducted to further explore the limitations of CR wheels braked with COBRA

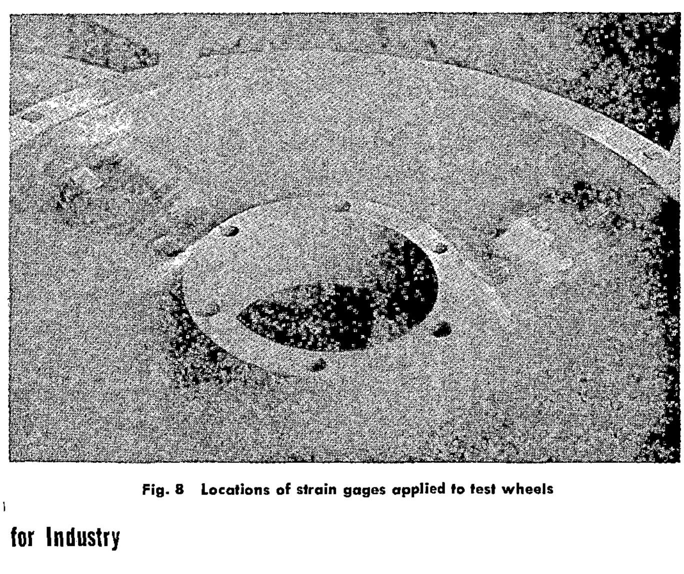


Fig. 8 Locations of strain gages applied to test wheels

**Table 3**

RESIDUAL STRESS (psi x 1000)

| Wheel No. | (Typical Values (as-manufactured) Not Shot Peened | Shot Peened | 10793 110 MPH Tests 37 Stops COBRA Shoe | 8925 110 MPH Tests 36 Stops Metal Shoe |
|---|---|---|---|---|
| BACK FACE | | | | |
| Rim-R | -45 | -25/-45 | -8 | -1 |
| (1) Rim-T | -40 | -25/-45 | -23 | -9 |
| (1) Plate - RF-R | +1 | -35/-45 | -46 | -64 |
| Plate - RF-T | -3 | -20/-40 | -29 | -39 |
| Plate - HF-R | -15 | -30/-40 | -19 | -24 |
| Plate - HF-T | -10 | -25/-45 | -22 | -26 |
| FRONT FACE | | | | |
| Rim-R | -20 | -20/-40 | -22 | -21 |
| (1) Rim-T | -20 | -15/-40 | -20 | -26 |
| Plate - RF-R | -10 | -40/-50 | -19 | -15 |
| Plate - RF-T | +2 | -25/-50 | -14 | -18 |
| (1) Plate - HF-R | +35 | -40/-50 | -18 | -18 |
| Plate - HF-T | +15 | -30/-45 | -15 | -14 |

-R = Radial
-T = Tangential
-RF = Rim Fillet (Point of tangency with plate)
-HF = Hub Fillet (Point of tangency with plate)
(1) Considered to be the critical stress locations

shoes and to verify the results of Test Phase 2 with respect to the relative thermal effects of plain and flanged brake shoes. This series of tests was identical in every way to Phase II, except that fifty stops were made on each of two wheels rather than thirty stops. In this test series, wheel No. 413 was braked with a COBRA flanged brake shoe, and wheel No. 439 was braked with a COBRA plain-style brake shoe. Wheel No. 433 was used as an unbraked control wheel.

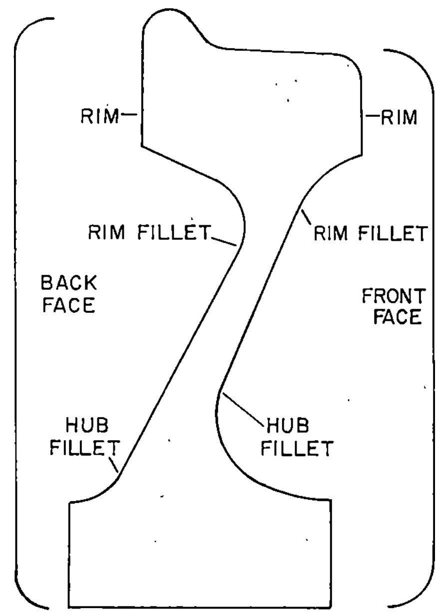



**Fig. 9 Wheel cross section showing stress measurement locations and designations**

## Dynamometer Test Results

Since Phase 2 and Phase 3 of the wheel tests were similar, and for the purpose of avoiding repetition, the dynamometer test results for these Test Phases have been combined in Table 4. All four of the test wheels involved completed the prescribed dynamometer stop braking tests with no visible defects.

The mechanical work done by the brake shoes during each of the 150 mph stops was over 17,000,000 ft-lb which is almost twice that of the 110 mph stops previously discussed in Test Phase 1. In this respect, the stops involved in Phase 2 and Phase 3 were more severe than those of the previously mentioned wheel tests.

Although the total work per stop was much greater for these

**Table 4**

| | Thirty Emergency Stops Wheel No. 566 Plain COBRA Brake Shoe | Thirty Emergency Stops Wheel No. 618 Flanged COBRA Brake Shoe | Fifty Emergency Stops Wheel No. 439 Plain COBRA Brake Shoe | Fifty Emergency Stops Wheel No. 413 Flanged COBRA Brake Shoe |
|---|---|---|---|---|
| Wheel Location | Inside | Outside | Inside | Outside |
| Equivalent Wheel Load | 23,084 lb. | 23,084 lb. | 22,919 lb. | 22,919 lb. |
| Total Simulated Car Wt. | 184,672 lb. | 184,672 lb. | 183,352 lb. | 183,352 lb. |
| Brake Arrangement | Single Shoe | Single Shoe | Single Shoe | Single Shoe |
| Net Shoe Force Per Wheel | 7,419 lb. | 7,419 lb. | 7,419 lb. | 7,419 lb. |
| Initial Speed | 150 mph | 150 mph | 150 mph | 150 mph |
| Kinetic Energy Per Stop, ft. lb. | 17,348,845 | 17,348,845 | 17,232,669 | 17,232,669 |
| Ave. Stop Distance | 20,378 ft. | 16,554 ft. | 20,545 ft. | 15,562 ft. |
| Ave. Stop Time | 171.8 sec. | 150.7 sec. | 174.4 sec. | 142.4 sec. |
| Ave. Work Rate Per Stop, ft. lb./sec. | 100,983 | 115,122 | 98,811 | 121,016 |
| Ave. Max. Wheel Tread Temp. | 806°F | 705°F | 769°F | 772°F |
| Ave. Max. Wheel Flange Temp. | NR | 604°F | NR | 679°F |
| Ave. Retardation Rate | .873 mphps | .995 mphps | .860 mphps | 1.054 mphps |
| Total Braking Work* Done on Wheel, ft. lb. | 955,078,076 | 955,078,076 | 1,193,189,975 | 1,193,189,975 |

*Includes wear-in stops

**Table 5**

| | | C | Mn | P | S | Si |
|---|---|---|---|---|---|---|
| 30 | (Wheel 613 unbraked | 0.74 | 0.70 | 0.033 | 0.041 | 0.33 |
| stops | (Wheel 566 plain | 0.75 | 0.76 | 0.031 | 0.038 | 0.30 |
| | (Wheel 618 flanged | 0.75 | 0.76 | 0.032 | 0.041 | 0.33 |
| 50 | (Wheel 433 unbraked | 0.73 | 0.72 | 0.026 | 0.045 | 0.30 |
| stops | (Wheel 439 plain | 0.74 | 0.72 | 0.024 | 0.039 | 0.29 |
| | (Wheel 413 flanged | 0.74 | 0.72 | 0.024 | 0.041 | 0.30 |
| | M-107, Cl-CR | 0.67/0.77 | 0.60/0.85 | 0.05 max | 0.05 max | 0.15 min |

stops, the average work rate per stop with the flanged COBRA brake shoe was not as high as the work rate per stop achieved for the 110 mph stops. It is significant, however, to compare the rate of work which took place for the first 9,000,000 ft-lb of energy dissipation during the 150 mph stops with the work rate for the complete 110 mph stops. This rate of work with the flanged COBRA brake shoe, accomplished in reducing the speed from 150 mph to 104 mph, was approximately 250,000 ft-lb per second, which is double the previously recommended limit of 125,000 ft-lb per second.

## Wheel Metallurgy Test Results

### Chemical Analysis

The chemistry of the test wheels is shown in Table 5 with the analyses of the two heats involved. The wheels meet the requirements of AAR Specification M-107, Class CR.

**Table 6**

Wheel No. 613, Heat K-9150, Unbraked.

Wheel No. 566, Heat K-9150, Thirty Stops, Plain Shoe.

Wheel No. 618, Heat K-9150, Thirty Stops, Flanged Shoe.

Wheel No. 433, Heat W-7881, Unbraked.

Wheel No. 439, Heat W-7881, Fifty Stops, Plain Shoe.

Wheel No. 413, Heat W-7881, Fifty Stops, Flanged Shoe.

| Wheel No. | Location at Which Brinnell Hardness Was Taken 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|
| 613 | 331 | 341 | 341 | 341 | 341 | 341 | 341 | 321 | 311 | 321 |
| 566 | 341 | 341 | 341 | 341 | 341 | 341 | 341 | 321 | 321 | 321 |
| 618 | 341 | 331 | 331 | 341 | 331 | 331 | 331 | 293 | 311 | 321 |
| 433 | 321 | 321 | 321 | 341 | 321 | 321 | 321 | 302 | 302 | 321 |
| 439 | 311 | 321 | 321 | 321 | 321 | 321 | 321 | 311 | 302 | 311 |
| 413 | 331 | 321 | 321 | 321 | 311 | 311 | 321 | 302 | 311 | 321 |

### Macro Examination

Macro examination of full radial sections removed from the test wheels indicated no noticeable thermal penetration into the rims. There was no visible difference between the wheels braked with plain shoes and flanged shoes at thirty and fifty emergency stops and without braking. None of the wheels showed any evidence of localized overheating at the tread surface. Fig. 10 shows the cross section of wheel No. 413, braked for fifty emergency stops with a flanged COBRA brake shoe, and gives the locations for the hardness values in Table 6.

Photographs of the tread surfaces after the 50 stop test series are shown in Figs. 11 and 12.

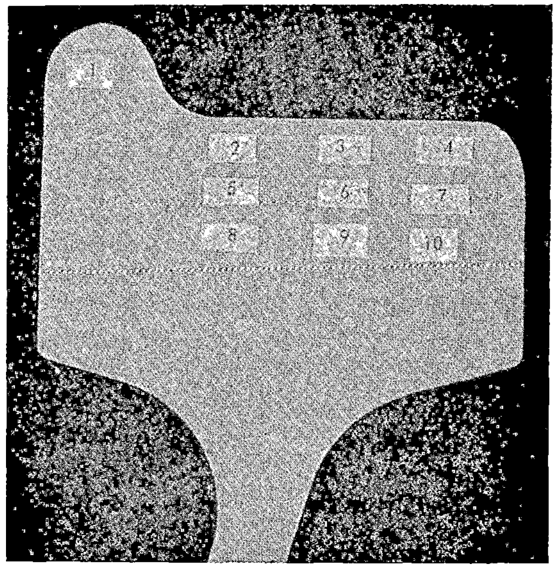



**Fig. 10 Section of wheel No. 413 after fifty 150 mph stops with a flanged COBRA brake shoe. Numerals show locations of hardness values listed in Table 6.**

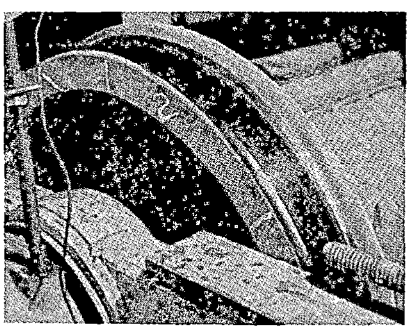


**Fig. 11 Polished tread condition of wheel No. 439 after fifty 150 mph stops with a plain COBRA brake shoe**

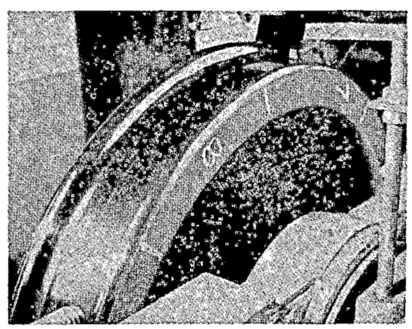


**Fig. 12 Polished tread and flange condition of wheel No. 413 after fifty 150 mph stops with a flanged COBRA brake shoe**

**Table 7**

RESIDUAL STRESS (psi x 1000)

| Wheel No. | Typical Values (as-manufactured) | | 10793 | 8925 | 566 | 618 | 439 | 413 |
|---|---|---|---|---|---|---|---|---|
| | | | 110 MPH Tests | | 150 MPH Tests | | | |
| | | | 37 Stops | 36 Stops | 30 Stops | | 50 Stops | |
| | Not Shot Peened | Shot Peened | COBRA Shoe | Metal Shoe | Plain COBRA Shoe | Flanged COBRA Shoe | Plain COBRA Shoe | Flanged COBRA Shoe |
| BACK FACE | | | | | | | | |
| Rim-R | -45 | -25/-45 | -8 | -1 | -45 | -42 | -20 | -11 |
| (1) Rim-T | -40 | -25/-45 | -23 | -9 | -41 | -50 | -13 | -15 |
| (1) Plate - RF-R | +1 | -35/-45 | -46 | -64 | -50 | -76 | -23 | -41 |
| Plate - RF-T | -3 | -20/-40 | -29 | -39 | -46 | -58 | -26 | -33 |
| Plate - HF-R | -15 | -30/-40 | -19 | -24 | -48 | -44 | -20 | -6 |
| Plate - HF-T | -10 | -25/-45 | -22 | -26 | -51 | -48 | -30 | -8 |
| FRONT FACE | | | | | | | | |
| Rim-R | -20 | -20/-40 | -22 | -21 | -39 | -42 | -10 | -15 |
| (1) Rim-T | -20 | -15/-40 | -20 | -26 | -51 | -51 | -24 | -21 |
| Plate - RF-R | -10 | -40/-50 | -19 | -15 | -54 | -42 | -15 | -4 |
| Plate - RF-T | +2 | -25/-50 | -14 | -18 | -46 | -45 | -11 | -18 |
| (1) Plate - HF-R | +35 | -40/-50 | -18 | -18 | -54 | -61 | -26 | - |
| Plate - HF-T | +15 | -30/-45 | -15 | -14 | -54 | -53 | -18 | - |

-R = Radial
-T = Tangential
-RF = Rim Fillet (Point of tangency with plate)
-HF = Hub Fillet (Point of tangency with plate)

(1) Considered to be the critical stress locations

**Stress Results**

Residual stress measurements were taken in the same manner as described in Phase 1 and are shown in Table 7. Included in this table are typical residual stresses for shot peened and nonshot peened wheels, as-manufactured, as well as the residual stresses for the braked wheels.

The as-manufactured, nonshotpeened stress pattern was as expected with respect to magnitude and type of stress for rim-treated wheels having a conventional configuration. The as-manufactured, shotpeened wheel stress pattern, which to our knowledge has not been previously observed, reflects the effect of inducing surface residual compressive stresses throughout.

The thirty emergency stop test wheels, No. 566 and No. 618, had compressive residual stresses of the magnitude expected for an unbraked wheel at all locations, which indicates that the wheels were unaffected by braking heat with either the plain or flanged COBRA brake shoe. The fifty stop test wheels show some degradation of compression, but even this level of braking was not sufficient to induce tension at any location. This may be due to the beneficial influence of the residual compressive stresses induced by the shot peening process as well as the beneficial effects of the heat input profile generated by the use of the COBRA brake shoe.

## Evaluation

Restraint has been exercised in drawing conclusions from these data because of the inherent variability of results from this kind of testing and the limited data available at this time. However, they do reflect, in a general way, what has been experienced in service and also what theoretical considerations would anticipate. It is deemed reasonable to draw the following tentative conclusions from this phase of the experimental work:

1 The work limits established by the University of Illinois test results are still valid for today's wheels braked with cast metal brake shoes.

2 The CR shotpeened wheel used in conjunction with the COBRA brake shoe, flanged or plain, exceeded the limits established for cast metal brake shoes.

3 Although the braking work rates and the energy dissipation per stop in these tests exceed any service condition in this country, the maximum limits for CR shotpeened wheels used in conjunction with the COBRA brake shoe have not been reached in this phase of this investigation.

4 The results obtained so far in this investigation justify further testing which is now under way.

## Brakes and Brake Rigging

Broadbent, H. R., "Confines Of Braking--5," Railway Gazette, Vol. 97, December 5, 1952, Temple Press Limited, 161-166 Fleet Street, London EC4, England, p 627.

This article discusses the economics of brake systems, especially the fact that extra costs in changes should bring about greater safety, train handling improvements, and easier maintenance. Factors of higher speeds, increased loading as reflected in higher air pressure requirements, problems of brake block and wheel temperatures, as well as increased stresses in the components, are considered. Increased wear in brake blocks and wheels can result either in higher costs or spatial limitations for the hardware necessary to perform the tasks.

RRIS: 037787

TDOP: 03-001

Carman, R. W., "Brake Rigging Efficiency of Railway Freight Cars," ASME, 71-WA/RT-2, 345 East 47th Street, New York, N.Y. 10017.

A test program for evaluating brake rigging efficiency is described, and correlation between static and dynamic efficiency is reported. Computer simulations of constant grade braking are performed.

TDOP: 03-002

Dike, G., "On Optimum Design Of Disk Brakes," ASME, 73-WA/DE-5, 345 East 47th Street, New York, N.Y. 10017.

Means of optimizing disc brake design are described based on temperature as the limiting factor. Consideration is given heat conduction problems, and comparisions with some experimental results obtained in Sweden are drawn.

TDOP: 03-003

Kondo, K., "Air-Brake Developments On Japanese Railways," Railway Gazette, Vol. 124, July 19, 1968, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 347-549.

Application of self-lapping brake for the locomotive as well as for the train and use of brake diaphragms in place of cylinders is discussed in this report. The D.E. 10 General-Purpose C-B diesel-hydraulic locomotive brake system is described in some detail. The use of diaphragms instead of cylinders to operate the brake obviates air leaks, reduces linkage to a minimum, and hence lubrication, and also reduces the need to adjust the linkage to take up brake wear.

RRIS: 037469
TDOP: 03-004

La Plaiche, M., "The Disc Brake--A Contribution Towards The Study Of Its Rational Use On Rail Vehicles," Revue Generale Des Chemins De Fer, July 1972, Societe Nacionale Des Chemins De Fer Francis, 92 Rue Bonaparte, 75 Paris 6E, France.

The author presents comments regarding the comparison of disc brakes, cast iron brakes, and composition brakes based on experience on French Railroads. The report concludes that more study is required regarding the applicability of disc brakes to rail vehicles.

RRIS: 047800
TDOP: 03-005

Rihosek, J., "Freight Car Brake Riggings," Railway Gazette, Vol. 71, September 1, 1939, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 319-320.

The design of the rigging influences the maximum brake effect, and the stretching and deformation of the brake rods, levers, and associated parts unfavorably influences the piston travel. The U. S. type of freight car truck, due to its two independent side frames, has a great disadvantage regarding braking technique. On European cars, clasp brakes are used whereby the axle is not influenced by the brake shoe thrust. The problem of play in joints in the single shoe type of brake is illustrated. The clasp brake system and a diagram of the eccentric force by rods from the body in the U.S. type truck are also shown.

RRIS: 037866

TDOP: 03-006

Temple Press Limited, "Problems In Running Braked and Unbraked Trains," <u>Railway Gazette</u>, Vol. 80, June 9, 1944, 161-166 Fleet Street, London EC4, England, pp. 588-589.

Acknowledgment is made of a paper presented at a technical meeting in 1944 on brake equipment and tests. This paper is an account of the highly specialized equipment that was developed by the Westinghouse Brake and Signal Co., Ltd., to provide a complete braking installation capable of dealing with every kind of service demand.

RRIS: 037970

TDOP: 03-007

Centerplates

Chicago, Milwaukee, St. Paul and Pacific Railroad, Road Test of "Center Plate Extension Pad" Truck Swivel Prevention Device on 100 Ton "Airslide" Hopper MILW 97207, Service Test Report S-681, 1971, Test Department, Milwaukee Shops, 516 W. Jackson Blvd., Chicago, Illinois 60606.

At the request of Standard Car and Truck Company, a series of six (6) road tests were conducted on Trains 99 and 98 to investigate the performance of the centerplate extension pad referred to as "CPEP." The CPEP was designed as a frictional dampening device functioning to reduce truck shimmy and/or hunting.

The study recommends that the CPEP devices operating in MILW 97207 be checked for effectiveness after approximately one year of service. It also advises the examination of car centerplates periodically to determine if use of the CPEP device alleviates centerplate damage and wear on these 100-ton hopper cars.

TDOP: 03-008

Dresser Transportation Equipment Division, Friction Torque Test Procedure And Analysis, 1972, Research and Engineering Department, 2 Main Street, Depew, New York 14043.

This is a brief description of a friction torque test procedure for centerplates and an analytical method to determine the coefficient of friction for a centerplate configuration. Coefficient of friction so obtained is used to predict friction torques that might be developed for 24 in. and 32 in. centerplate configurations.

TDOP: 03-009

Fillion, S. H., "Body Structural Performance (Center Plates)," Technical Proceedings - 1967 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, N.Y. 14043, pp. 28-30.

This article presents an analysis of the carbody bolster and centerplate structures on cushion underframe cars. The harmonic roll problem is described, and design calculations are derived to meet this problem with new designs of car and truck centerplates and carbody bolsters which permit forces to be distributed resulting in reduced stress levels.

TDOP: 03-010

Johnson, M. R., R. E. Welch, and G. Ojdrovich, "Analysis of Truck Bolster Center Plate Rim Response To Impact Loads," ASME, 74-RT-5, December 1973, 345 East 47th Street, New York, N.Y. 10017.

The impact of a moving freight car into a string of standing cars results in a large longitudinal load acting on the centerplate rim of the truck bolster. The load is the result of the rapid deceleration of the truck. The dynamic characteristics of this load and its effects within the rim are discussed. Analytical procedures are described for processing strain gage data to determine the location of load application on the rim.

RRIS: 054008

TDOP: 03-011

Lasky, W. E. and M. A. Hanson, "A Field-Service Evaluation Of Various Center-Plate Lubricants and Liners," ASME, 64-WA/RR-2, 345 East 47th Street, New York, N.Y. 10017.

The report states that service performance has demonstrated the inadequacy of the commonly used centerplate lubricants. Recent service tests have demonstrated reduced galling through the use of solid lubricants. Certain types of plastic disks also markedly improved centerplate conditions. Service tests were also performed on several types of liners where their use was required because of centerplate wear.

TDOP: 03-012

Rousseau, G. L., "Pullman's Approach To Center Plate Problems," Technical Proceedings--1969 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York, 14043, pp. 39-41.

This paper reports on tests conducted on fabricated, cushioned underframe car centerplates using a high-cube car in conjunction with a car-rocking device designed to excite a fully loaded car at its natural roll frequency. Results were correlated with field service reports. Tests were in progress with the rocking device using a 100-ton covered hopper car. The conclusion of the two-year test program on cushioned underframe cars has indicated these results:

- Rock and roll is instrumental in centerplate wear and breakdown.
- Breakdown originates in the transition radius and propagates into the transverse and longitudinal reinforcements.
- The 3/8 in. radius is not satisfactory for today's fatigue environment.
- Longitudinal reinforcement on the centerplate should be located over the outer circumference of the bowl.
- A flat centerplate application significantly reduces the attachment prestress in the transition radius.
- Both fabricated and cast centerplates can develop cracking unless fatigue design is considered.
- The centerplate application is related to car configuration.
- Supplemental snubbing devices that reduce carbody roll would increase the fatigue life of centerplates.

RRIS: 039505

TDOP: 03-013

Seel, M., W. J. Ruprecht, G. L. Rousseau, and J. F. Krause, "Carbuilders' Approach To The Solutions Of The Truck-Car Body Relationships," (panel discussion given at 1969 Railroad Engineering Conference), Technical Proceedings - 1969 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 30-44.

This section of the Technical Proceedings consists of a panel discussion on carbuilders' approaches to body centerplate designs for freight cars. The history of centerplate design is reviewed, including abnormal wear rates and increased rates of breakage of centerplates on long, 70-ton cars and on all 100-ton cars, which became a problem in the mid 1960's. Fore and aft flexing of trailer and container cars and rock and roll of heavy cars are cited as possible contributing factors. Various road and laboratory tests are discussed. Use of wear rings, increased depth in body centerplate, finish of wear surfaces (including truck centerplates), and fit-up of body centerplate to center sills and fillers are cited as solutions in service and are under observation.

RRIS: 039504
039506

TDOP: 03-014

Symington Wayne Corporation, "Discussion of Truck Center Plate Liners," Technical Proceedings - 1966 Railroad Engineering Conference, 2 Main Street, Depew, N.Y. 14043, pp. 48-50.

This discussion is contained in the record of the 1966 Railroad Engineering Conference where one of the topics of conversation was the problem of bolster centerplate wear and the experience of the conferees with various liners on various freight cars.

TDOP: 03-015

Walmsley, R. G., "Non-Metallic Liners For Rolling Stock Bearing Surfaces," Railway Gazette, Vol. 126, May 15, 1970, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 392-394.

Development of asbestos-based bearings and surfaces to obtain specified constant friction characteristics under working conditions is discussed. Asbestos and steel working together will quickly bed down into smooth, highly polished working surfaces. This is probably due not only to the resistance to welding with the asperities of the steel, but also to the mildly abrasive properties of the asbestos, which produce a lapping action and smooth off the sharper edges of the metal asperities.

RRIS: 037758
TDOP: 03-016

Side Frames and Bolsters

American Steel Foundries, Freight Car Truck Progress, by R. B. Cottrell, Chicago, Illinois.

This paper presents a technical and historical discussion of freight car truck design from 1910 to 1947. Side frame and bolster designs and the development of specifications having acceptable side frames and bolsters are reviewed in detail. These specifications relate to 40-, 50-, and 70-ton-capacity side frames and bolsters.

TDOP: 03-017

Association of American Railroads, Effect of Periodic Normalizing Of Truck Side Frames, Research Report No. AAR-MR-219, May 1954, 3140 South Federal Street, Chicago, Illinois 60616.

The two series of tests on eight side frames covered by this report are the basis for the following conclusions:

- Heat treatment of second-hand truck side frames did not increase their fatigue life expectancy.
- The reconditioning of worn column areas by welding when subsequently heat treated does not impair fatigue life expectancy.
- The normalizing heat treatment process (heating to 1550 degrees F and cooling in still air) did not result in improved fatigue test performance in comparison with the stress relieving heat treatment as given the first four frames.

RRIS: 040362

TDOP: 03-018

Association of American Railroads, Investigation Of Coatings To Protect Truck Side Frames From Corrosion, AAR-MR-299, September 1957, 3140 South Federal Street, Chicago, Illinois 60616.

In this report, the relative performance of coatings for the corrosion protection of truck side frames on railroad equipment was investigated. Copper steel was evaluated relative to low carbon steel in terms of reducing the effects of corrosion. Laboratory tests were designed to simulate various corrosive conditions found in service. The effects of brine corrosion and of atmospheric corrosion on the coating systems were studied extensively. The test data indicated the superiority of zinc coatings for corrosion protection of truck side frames. Certain vinyl coatings also demonstrated excellent corrosion resistance.

RRIS: 040367
TDOP: 03-019

Association of American Railroads, Survey Of Cast Steel Truck Side Frame And Truck Bolster Removals By Member Railroads And Private Car Owners, Technical Report No. MR-446, December 1967, Association of American Railroads Research Center, 3140 South Federal Street, Chicago, Illinois 60616.

The critical areas for defects on truck side frames removed from service were the compression and tension members close to the journal boxes. The critical area for defects on truck bolsters removed from service was in the center, especially in and adjacent to the center bearing. The percent of removal for bolsters is 2.4 times greater than for side frames. The report recommends that dynamic test requirements for side frames should be made more severe and bolster dynamic tests should be formulated. At the time of this report, the equipment to handle dynamic investigations of side frames and bolsters at the AAR Research Center had been purchased and was being installed.

RRIS: 039432
TDOP: 03-020

Association of American Railroads, Survey of Cast Steel Truck Side Frame And Truck Bolster Removals By Member Railroads And Private Car Owners, Technical Report No. MR-446A, April 1968, 1920 L Street, Washington, D. C. 20036.

A survey was made of cast steel truck side frames and truck bolsters removed from service for reasons other than normal wear. The incidence of removal of these units increases with service life. Bolsters are removed 2.4 times more often than side frames. Proposed changes in specifications should equalize this replacement rate.

RRIS: 039536
TDOP: 03-021

Bethune, A. E., D. D. Taylor, and R. F. Gonsalves, Testing of Side Frame Stabilizers on 100-Ton Barber S-2 Trucks, Report No. S-445-72, May 1972, Canadian Pacific, Department of Research, Windsor Station, Montreal 101, Quebec.

This document reports on the testing of truck side frame stabilizers to determine the effectiveness of stabilizers in improving truck stiffness, permitting easier alignment to curves, and reducing wheel flange wear and rail wear. Static tests and mainline service tests were conducted. Stabilizer units and instrumentation are described, and stress levels reported.

TDOP: 03-022

Kraichik, M., V. A. Ladugin, V. S. Ignat'Eva, NYa Mikalev, YuS Tarasevich, "Experimental Investigation Of Residual Welding Stresses In Elements Of The Frame of Bogies Of Rolling Stock," Welding Production, Vol. 19, No. 8, August 1972, Welding Institute, Abington Hall, Abington, Cambridge CB1 6 AL, England, pp. 46-48.

Unfavorable residual stresses, reaching the yield point, may develop in welded elements of bogie frames of rolling stock. Considerable reduction of these stresses, as a result of tempering, to the value 350 kg /sq cm should promote an increase of service reliability of the bogie frames.

RRIS: 047895
TDOP: 03-023

Moehling, C. and D. P. Mate, "Computer Analysis Of A Railroad Freight Car Bolster Utilizing The Finite Element Method," Physical Sciences Newsletter, February 1971, Control Data Corporation, 4550 West 77th Street, Minneapolis, Minnesota 55435, pp. 4-6, (abbreviated version of paper presented at ASME Winter Annual Meeting 1970).

This paper discusses a structural analysis of a truck bolster and verification of the analysis through a laboratory stress investigation. The finite element method is claimed to be a means of accurately predicting the action of a complex structure under varying loads. However, the authors indicate that sound engineering judgement is the key to effective computer analysis rather than extensive knowledge of the finite element method.

RRIS: 046378
TDOP: 03-024

Monselle, D., "Truck-Bolster Dynamic Loadings Measured Under Harmonic Roll Conditions," ASME, 71-WA/RT-6, 345 East 47th Street, New York, N.Y. 10017.

In service, two general types of fatigue loading conditions are imposed on railroad freight-car truck bolsters. Both types of loadings involve a relative rolling or rocking motion between the truck and carbody bolsters but differ in the amplitude and frequency of the motion. One condition, the lateral harmonic roll motion action, possible with large and high center of gravity cars, is associated with low train operating speeds and operation on track with periodic low joint conditions. This paper illustrates the truck-bolster dynamic loadings measured under the extreme harmonic roll condition on a specially prepared test track and describes the methods of instrumentation developed to measure such loadings.

TDOP: 03-025

Shadur, L. A., "Problems of Calculation of Bogie Frames For Freight and Passenger Cars," Rail International, No. 7, July 1973, International Railway Congress Association, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 762-786.

Higher train speeds, the need for greater safety, and the need for more effective utilization of materials are making railway specialists calculation conscious. More attention is paid today to methods of calculation for rolling stock. The bogie, the vital and most dynamic element of the rolling stock, is attracting close attention of designers and specialists. This article discusses the complex phenomena accompanying the design of a modern bogie. Special attention is paid to establishing the forces acting on a two-axle bogie, the vertical loads acting on a bogie side frame, and assessment of fatigue strength of bogie parts. The calculations for passenger bogies are also considered. The theoretical discussions are illustrated by practical examples and data.

RRIS: 050607
TDOP: 03-026

Tack, C. E., "Outer Pedestal Legs of Narrow Pedestal Side Frames," ASME, 62-WA-243, 345 East 47th Street, New York, N.Y. 10017.

The author reports on fracture experiences up to 1962 with "narrow" pedestal designs. The static and fatigue tests performed, and the metallurgical tests performed, are also noted. Conclusions associate failure with low temperature and suggest design changes to improve impact resistance.

TDOP: 03-027

Temple Press Limited, "Cast Steel Bogie Frames," Railway Gazette, Vol. 101, November 16, 1954, 161-166 Fleet Street, London EC4, England, p 584.

Several Belgian stainless-steel, multiple-unit electric trains are fitted with bogies with one-piece cast steel frames housing one 250- to 265-hp, nose-suspended traction motor. In view of the number that will eventually be required over a period of several years, a decision was made to subject one of the first castings to comprehensive dynamic tests so that the accuracy of the designers' calculations could be checked, and so that any weak points might be brought to light before serious production began. Loads were applied at a rate of 250 cycles a minute for 133 hours. Horizontal and vertical loads were applied at the same frequency so that diagonal forces would be at a maximum at the same time. Stresses in the frame under static and dynamic loads were checked by 14 strain gauges. Under variable dynamic loading simultaneously applied vertically and horizontally, maximum stress did not exceed 3-1/2 tons per sq. in., and maximum amplitude of stress variation at any one point was not above 2-1/4 tons per sq. in.

RRIS: 039967
TDOP: 03-028

Tennikait, H. Garth, "American Steel Foundries Dynamic Test Facility For Railway Truck Components," Closed Loop Magazine, Vol. 4, No. 2, Spring 1974, MTS Systems Corporation, P. O. Box 24012, Minneapolis, Minnesota 55424, pp. 10-16.

This article describes results of testing of the loads and stresses in 100-ton freight car truck bolsters. Tests were initially conducted in the field to develop loads and frequencies of loads applied to truck bolsters during harmonic roll action of freight cars. A dynamic test machine was installed in the laboratory to evaluate truck bolster fatigue under the same conditions.

TDOP: 03-029

U.S., Federal Coordinator of Transportation, Report of Mechanical Advisory Committee to the Federal Coordinator of Transportation on Freight Cars, 1935, pp. 177-186.

This section of the Report of the Mechanical Advisory Committee covers truck side frames and bolsters and presents a comprehensive review of the design and developments of freight car trucks and truck components. Material and dynamics specifications are presented, and methods for testing are given.

TDOP: 03-030

Snubbers and Dampers

AAR Draft Gear Testing Laboratory, Report On Service-Endurance Test Of Symington Snubbers, by W. E. Gray, May 29, 1929, Purdue University, Lafayette, Indiana.

Results are presented from tests conducted by the Association of American Railroads on two types of friction snubbers for use under refrigerator cars. One of these snubbers had been used in service, and its capacity for energy absorption is measured relative to its capacity in the new, unused condition. Results reported include: installation data and laboratory measurements, general summary for each test car used, absorption curves for each snubber, closure curves for each snubber, closure curves first oscillation, tabular results from bounce test, and sample oscillation diagrams.

TDOP: 03-031

AAR Draft Gear Testing Laboratory, Test of Railway Truck Corporation "Snub-Up" Snubber, by W. E. Gray, April 8, 1943, Purdue University, Lafayette, Indiana.

A proprietary friction snubbing device to be used in the springs of freight cars for controlling vertical oscillation was tested by the Association of American Railroads. Static and bounce tests were performed on these devices. Tabular results, oscillation diagrams, absorption curves, and closure curves (first oscillation) are given for the bounce tests, and load height diagrams are shown for the static tests.

TDOP: 03-032

Batchelor, G. H. and R. C. Stride, "Hydraulic Dampers and Damping," Institution of Locomotive Engineers Journal, Paper No. 712, Vol. 58, Part 6, 1968-69, Institution of Locomotive Engineers, Locomotive House, 30 Buckingham Gate, London SW1, England, pp. 563-629.

After a brief review of the influence of viscous damping on free and forced oscillations of mass-spring systems and its effects on response to impulses, suitable damping factors are quoted for control of body and bogie oscillations on the suspension. This is followed by a discussion of the most suitable forms of force-velocity characteristics for railway applications and a section dealing with damping calculations. The influence of damper flexible mountings is then examined, with particular reference to impulsive suspension response and response to sinusoidal excitation. The paper goes on to deal with experimental work relating to vehicle damping requirements and then discusses the recently issued British Railways specification that stipulates the requirements for dampers for use on British Railways' vehicles. Difficulties in the manufacture of dampers with linear symmetrical force-velocity characteristics are then pointed out, with special reference to the tendency for S-shaped characteristics. A section is devoted to the more common faults and difficulties that have been experienced with hydraulic dampers, and this is followed by a discussion of the testing of dampers to ascertain force-velocity characteristics and endurance. Recent damper developments are mentioned, followed by conclusions covering the principal aspects of hydraulic dampers and damping.

TDOP: 03-033

Batchelor, G. H., "Some Aspects Of The Theory And Practice Of Damping--II," Railway Gazette, Vol. 115, December 1, 1961, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 628-630.

When mounting dampers across the secondary suspension of a vehicle body, pitching and swaying oscillations must be accounted for. The lateral motion on the swinglinks can be damped without difficulty; however, rolling on the bolster springs is difficult to damp with hydraulic units since the frequency of the oscillations is often low (about 0.5 to 0.8 c/s), and the moment arm short. Resonance conditions in the swaying mode must, if at all possible, be avoided at the operationally important speeds, since control by hydraulic damping is unlikely to prove an acceptable solution.

RRIS: 040085

TDOP: 03-034

Canadian National Railways, An Experimental Evaluation of Friction Snubber Springs, by K. A. Henderson, April 1966, Research and Development Department, Technical Research Centre, Montreal, Quebec.

At the writing of this report, approximately 27,000 cars in use on Canadian National Railways were not equipped with trucks incorporating friction snubbing devices. In order to evaluate the effectiveness of commercially available add-on snubber units for possible application to the 27,000 cars, Canadian National conducted a series of field tests applying snubber units to a box car (450182) loaded with pyrite.

Due to the small number of samples tested and the wide variety of results obtained, the test series must be considered only as exploratory. The conclusions reached are as follows:

- There is a large difference in the capacity of different snubbers and also between snubbers of the same type.
- The capacity of the snubber depends greatly on the weight of the car and lading.
- Friction snubbers are potentially capable of providing an amount of damping equal to that offered by a "Barber Stabilized" truck.
- The effect of friction snubber springs on the critical speeds of a vehicle can be measured.

TDOP: 03-035

Koffman, J. L., "The Friction Damper," Rail Engineering International, Vol. 3, No. 9, November 1973, Broadfields Limited (Technical Publishers), Little Leighs, Chelmsford, Essex CM3 1PF, England, pp. 414-420.

Constant-force dampers are attractive for rolling stock with small differences between loaded and empty conditions such as passenger coaches but are not desirable for freight wagons where the tare/loaded weight ratio is high, which calls for displacement-responsive damping. Friction damping and viscous-damping are mathematically compared and assessed. The author warns of pitfalls when "scaling down."

RRIS: 051395
TDOP: 03-036

Koffman, J. L., "Friction Damping," Railway Gazette, Vol. 111, November 13, 1959, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 422-425.

The Alsthom truck for Co-Co electric locomotives, the S.I.G. truck for electric railcars, the Werkspoor truck for electric railcars, and the Allan truck for diesel-electric railcars are photographed showing the spring systems. The action of friction (Coulomb) and hydraulic (viscous) damping is illustrated with the aid of force-displacement diagrams. The effect of friction and hydraulic damping on force transmission and oscillation amplitude ratio as a function of frequency ratio, and the effect of oscillation amplitude on dynamic stiffness of leaf springs as affected by friction is also illustrated. The advantages of friction damping are mentioned.

RRIS: 037702
TDOP: 03-037

Koffman, J. L., "Hydraulic Dampers," Railway Gazette, Vol. 111, October 30, 1959, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 362-366.

As far as vehicle suspension incorporating steel springs is concerned, the number of damping characteristics can be limited to two: friction damping and viscous damping. Friction damping occurs in every system and is maintained by friction forces acting in opposition to the motion. With true (theoretical) viscous damping, the damping force opposing the oscillation is proportional to the velocity of the latter. Many hydraulic dampers meet this requirement over the lower range of their characteristics. The effect of damping on the pattern of vibration decay and the effect of damping on the force and displacement transmissibility and resultant acceleration are shown. The Askania hand vibrograph is illustrated, along with the vibrograph records of diesel-electric locomotives. Typical characteristics of various damping methods encountered with railway vehicles are graphed.

RRIS: 037703

TDOP: 03-038

Temple Press Limited, "Friction-Controlled Suspension for Wagon Bogies," Railway Gazette, Vol. 112, May 13, 1960, 161-166 Fleet Street, London EC4, England, pp. 573-574.

As of this writing, British Railways had 200 cars equipped with friction-controlled trucks with unit brake beams. The integral friction-controlled suspension system for the car trucks is designed to provide maximum shock cushioning at high speeds. The ends of the bolsters are supported on nests of coil springs. Sandwiched between the guide faces of the bolster and the guide surfaces of the frames are spring-loaded friction shoes. The controlled friction damping permits use of long-travel, low-rate coil springs which give maximum cushioning for all loading conditions. The side frames can be supplied with either the hanger-type suspension brake beams, or pockets can be cast in the frames to accommodate the unit-type brake beam. The truck frames can be arranged to accommodate roller bearing axleboxes and either integral or separate plain bearing boxes.

RRIS: 037701

TDOP: 03-039

Springs

AAR Draft Gear Testing Laboratory, Report On Test Of Frost Friction Truck Spring Type 420, by W. E. Gray, December 1942, Purdue University, Lafayette, Indiana.

A bolster friction-snubbing device was tested by the Association of American Railroads at the request of the manufacturer. A report is presented on the results obtained. Bounce test results reported include: tabular results, energy absorption curves, and simple oscillation diagrams. Tabular results are also reported for reaction tests.

TDOP: 03-040

AAR Draft Gear Testing Laboratory, Report On Test Of Frost Friction Truck Springs Nos. 969-973 - Type 360, by W. E. Gray, July 10, 1939, Purdue University, Lafayette, Indiana.

A manufacturer of proprietary friction springs submitted components to the Association of American Railroads for testing to determine the amount and direction of movement of these springs. The report on this testing includes the following data: tabular results for bounce test, energy absorption curves for bounce tests, sample oscillation diagrams for selected bounce-tested springs, and movement test results.

TDOP: 03-041

AAR Draft Gear Testing Laboratory, Report On Test Of G-B Friction Bolster Spring Type A-2, by W. E. Gray, April 1940, Purdue University, Lafayette, Indiana.

A friction bolster spring of proprietary design was submitted to the Association of American Railroads for testing. The following results are included for bounce tests: tabular results, oscillation diagrams, energy absorption curves, and closure curves for first oscillation. Static test results include: closure and release curves and oscillation diagrams.

TDOP: 03-042

AAR Draft Gear Testing Laboratory, Report On Test Of Holland Truck Springs Style B-188, by W. E. Gray, June 1943, Purdue University, Lafayette, Indiana.

Data are presented from tests conducted by the Association of American Railroads on the friction volute bolster spring submitted by a manufacturer for evaluation. Bounce and static tests were performed on ten groups of five springs each. Tabular results and energy absorption curves are given for the bounce tests. Closure and release curves and oscillation diagrams are given for the static tests.

TDOP: 03-043

AAR Draft Gear Testing Laboratory, Report On Test Of Holland Truck Springs Type A-6, by W. E. Gray, June 12, 1940, Purdue University, Lafayette, Indiana.

A report of the Association of American Railroads presents data from tests on freight car truck volute springs submitted by a manufacturer. Bounce and static tests were performed on the springs. Tabular results and energy absorption curves are presented for the bounce tests, and diagrams are given for the static tests.

TDOP: 03-044

American Steel Foundries, Inc., Extended Life Springs, Hammond Division, Hammond, Indiana.

Manufacturer's brochure describes an extended-life spring citing test results, areas of application, and manufacturing technique.

TDOP: 03-045

American Steel Foundries, Inc., How The Hammond Division "Extended Life" Coil Spring Compares With The AAR Specification M114, Bulletin No. JHL-1, 1962, Hammond Division, Hammond, Indiana.

This manufacturer's bulletin describes extended-life coil springs for freight car trucks and compares them with AAR requirements.

TDOP: 03-046

American Steel Foundries, Inc., Performance Characteristics of Long-Travel Truck Springs, Hammond Division, Hammond, Indiana.

This article states that while longer-travel springs are indispensable for easy riding, proper control of this action is of equal importance in the modern, easy-riding truck. As a constant amount of load is required to compress a spring group for each increment of travel, and a corresponding amount is returned in recoil for each increment of travel, a constant amount of frictional control is required in cushioning each such increment of travel. Specifications are presented for various coil spring types and couplers for use in 40-, 50-, and 70-ton trucks.

TDOP: 03-047

Batchelor, J., "Frequency-Dependent Elastomer Suspension Components For Fast Trains," 1972 Proceedings of The Institute of Rubber Industries, Institute of Rubber Industries, London, England.

Requirements for a frequency-dependent spring, with little hysteresis, and for a damping material with little change of modulus with frequency were investigated together, as frequency-dependence of modulus and hysteresis are closely related. Dynamic test data on a variety of polyester urethanes are presented in this report and are shown to be consistent with an amorphous domain structure. The hysteresis correlates well with the rate of change of modulus with frequency.

RRIS: 051464

TDOP: 03-048

Botham, G. J. M., "Practical Aspects of Primary Suspension Design," Institution of Locomotive Engineers Journal, Vol. 56, No. 313, Part 5, Institution of Locomotive Engineers, Locomotive House, 30 Buckingham Gate, London SW1, England, pp. 495-535.

The most common primary suspension systems on car trucks are: leaf spring and horn guide; coil spring, equalizer beam, and horn guide; coil spring and cylindrical guides; and chevron rubber. The design characteristics of these systems are discussed. Damping characteristics, stress loads, and spring rates are given for suspension systems. To fit coil springs to 4-wheeled trucks, the following factors must be considered: satisfactory ride performance can be obtained with laterally rigid suspension on 15-ft wheelbase cars; particular attention has to be given to providing sufficient damping force to control vertical, and, in particular, pitching movements; and, although it is advantageous to do so, such damping need not necessarily be variable between tare and load for the majority of applications.

RRIS: 040498
TDOP: 03-049

Federation des Industriels Ferroviaires, "Research on Pneumatic Suspension Systems - A Study of Diverse Prototype Equipment Effected by The S.N.C.F. Vitry Tests Station," French Railway Techniques, No. 3, 1966, 92 Rue Bonaparte, 75 Paris 6E, France, pp. 149-153.

This paper discusses the testing of pneumatic suspension systems by the S.N.C.F. Two types of pneumatic springs (by Sumitomo) were examined in a program involving both static and dynamic tests. The aim of the static tests was to study vertical and lateral flexibility and the useful surface, as related to load and capacity, of the auxiliary reservoir. The dynamic tests studied spring action under free and sustained oscillations.

Three types of coaches with three different types of trucks were used for these tests: one fitted with the Sumitomo bellows springs with lateral control by two pneumatic springs plus longitudinal links, another fitted with Sumitomo bellows springs with lateral control by pendular-type suspension, and the third system was a "Bruhat" truck fitted with two "Sumiride" diaphragm springs and pneumatic lateral control.

The theory of the operation of the pneumatic springs was analyzed. Both the static and dynamic tests confirmed the theoretical considerations. These results indicated that the two types of pneumatic springs offer about the same qualities of suspension relating to considerable flexibility, simplicity of the damping system in the vertical direction, and the damping of excitations set up by the rolling gear. However, the diaphragm spring offers definite advantages over the bellows springs. Its greater vertical flexibility, which is due to the level-correcting feature of this system, ensures better dynamic flexibility, and while its rigidity in the lateral direction is greater, this is really no drawback.

However, the full advantages of these pneumatic spring systems could not be demonstrated in these tests. The report suggests that these studies should be pursued further to define the structural arrangements of the trucks that would take full advantage of the pneumatic spring suspension.

TDOP: 03-050

Frost, W. E., "Design of Springs," Railway Gazette, Vol. 97, December 26, 1952, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 705-706.

Due to high stresses and abrasive action between laminated spring plates causing a reduction in thickness, it is recommended that a service life be established for the springs and that the springs then be replaced. Several other design weaknesses are described, including the center fastening.

RRIS: 039469
TDOP: 03-051

Hendrick, J. K., "Some Optimal Control Techniques Applicable To Suspension System Design," ASME, 73-ICT-55, 345 East 47th Street, New York, N.Y. 10017.

The need to develop new suspension systems for vehicles of the future has encouraged the application of modern optimization techniques to such proposed suspension designs as magnetic levitation and air cushion suspension. This paper presents an analytical approach to three suspension optimization problems:

- The choice of the optimal parameters for a predetermined suspension configuration given a particular application
- The choice of the optimal passive suspension configuration for a given application
- The choice of the optimal active suspension configuration for a given application

RRIS: 051400
TDOP: 03-052

IPC Transport Press, "The Heart Of The Matter - Truck Suspension," Railway Gazette International, Vol. 126, October 1970, Dorset House, Stamford Street, London SE1, England, p 23.

This document is an advertisement for the Gloucester cast steel freight truck which has a "Metalastik" rubber suspension. The Metalastik springs consist of bonded rubber springs of chevron shape which give angular and vertical flexibility. Derailment and wheel wear is minimized by the transverse angling of the side frames. The use of this simple suspension eliminates the need for bolster springs, guides, and lubricating surfaces.

RRIS: 039486
TDOP: 03-053

Koffman, J. L., "BR Class 40 Locomotive Suspensions Modified," Railway Gazette, Vol. 126, February 20, 1970, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 142-144.

The article discusses the suspension design of the British Railways class 40 locomotive, which was modified to minimize the maintenance and repair of the laminated springs. The previous suspension system subjected the truck frame to high vertical and lateral stresses. The modification replaced the laminated springs by helical springs. The result was an improvement in the vertical riding qualities, especially over 50 mph. A reduction of dynamic forces that will result in longer suspension and truck-frame life was also achieved.

RRIS: 037800
TDOP: 03-054

Koffman, J. L., "Laminated Spring Friction," Railway Gazette, Vol. 120, April 3, 1964, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 279-284.

The advantages and disadvantages of using laminated springs for rolling stock are explored in this paper. Attention is focused on the static aspects of spring friction, the effects of twisted track on vehicle performance, the results of derailment tests, and performance characteristics of laminated and helical springs.

RRIS: 040101

TDOP: 03-055

Koffman, J. L., "Non-Linear Springs," Railway Gazette, Vol. 113, August 5, 1960, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 164-166.

The problem of ensuring satisfactory ride characteristics with vehicles operating in suburban services is particularly difficult because, as far as the bolster springs are concerned, the rush-hour load can exceed the weight of the vehicle body. Consequently, a static deflection of 4 inches under tare load will mean a static deflection of 8 inches and more under overload. General design considerations often make it necessary to deviate from the constant frequency relation, particularly when dealing with heavy overloads. To ensure a nonlinear characteristic, rubber cones are used inside helical bolster springs, and use is also made of suitable rubber springs sometimes incorporating helical springs vulcanized in them to prevent undue barreling out of the hollow rubber cylinders. The use of non-linear centering devices is beneficial in reducing the degree of coupling between bogie hunting and body nosing and swaying in the range of low amplitude and frequency oscillations.

RRIS: 037698

TDOP: 03-056

Koffman, J. L. and G. H. Batchelor, "Performance of Locomotive Bogie Bolster Springs--I," Railway Gazette, Vol. 116, January 5, 1962, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 16-18.

The lateral rigidity of helical springs in bolster suspensions of truck vehicles is evaluated in terms of displacements between the top and bottom bolster planks. The effects of lateral flexibility on working stresses and lateral displacement and oscillation characteristics, particularly nosing, swaying, and lateral oscillations, are calculated.

RRIS: 040088
TDOP: 03-057

Lipsius, J. M., "Separate and Joint Components for the Vertical and Transverse Damping Between Bogie and Vehicle Body," Rail International, August 1967, International Railway Congress Association, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 597-613.

A general discussion of the influence of separate and joint suspension components on lightweight construction and smooth riding of bogie vehicles is followed by a discussion of the technical requirements which the vehicle body suspension must meet. Subsequently, the various types of separate and joint suspension components already in use are examined in order to ascertain to what extent they meet the various requirements. Finally, a number of possible arrangements of joint suspension components is mentioned.

RRIS: 033444
TDOP: 03-058

Matsumiya, S., K. Nishioka, S. Nishimura, and M. Suzuki, "On The Diaphragm Air Spring Sumiride," The Sumitomo Search, No. 2, November 1969, Sumitomo Metal Industries, Ltd., Osaka, Japan, pp. 86-92.

The diaphragm air spring (Sumiride) was developed as a bolster spring for a railroad bogie truck without a swing bolster hanger. The lateral spring action of Sumiride depends mainly on the change of the effective area in this direction. This paper summarizes the analytical results obtained from tests on the spring characteristics of Sumiride. The design of Sumiride and its practical applications to bogie trucks are explained.

TDOP: 03-059

Meacham, H. D., Supplement to Summary Report On Experimental Evaluation of Elastomeric LIM Truck Suspension Elements, DOT-FR-00026, January 1972, National Technical Information Service, Springfield, Virginia 22151.

Seven different elastomeric suspension components were evaluated using Battelle's servo-controlled hydraulic load actuator to stroke the elements at vibration displacements and frequencies corresponding to those encountered in service on the various rail vehicles. Static and dynamic spring rates and damping were obtained. A figure is included illustrating how these values were determined from the Lissajous diagrams (plots of force versus displacement) recorded in the laboratory runs.

TDOP: 03-060

Temple Press Limited, "Rubber in Railway Engineering," Railway Gazette, Vol. 106, January 18, 1957, 161-166 Fleet Street, London EC4, England, pp. 71-72.

This article discusses the design of bogies employing rubber to avoid wear on metal surfaces, thus increasing service life of rolling stock. The Silentbloc suspension system is illustrated. The rubber suspension system designs attempt to eliminate bogie hunting by eliminating all wearing parts, providing spring action in three degrees of freedom, and giving a suspension system with constant periodicity.

RRIS: 037621
TDOP: 03-061

Temple Press Limited, "Rubber Suspension For Bogies," Railway Gazette, Vol. 99, December 4, 1953, 161-166 Fleet Street, London EC4, England, pp. 626-627.

Experiments were undertaken to determine the practicability of suspending whole bogies on rubber, thereby bringing about a reduction in maintenance costs by eliminating all wearing parts such as rubber plates, hanger pins, and bushes, as well as the steel springs. Such suspension would also go a long way toward reducing, if not eliminating, the running noises resulting from conventional type bogie suspension, particularly in tunnels. A preliminary investigation indicated that the cost per bogie was approximately 50 Pounds more than for the standard arrangement. It is anticipated that the life of the rubber units will be approximately ten years, if not longer. The saving in maintenance would more than compensate for the extra initial cost.

RRIS: 037773
TDOP: 03-062

Temple Press Limited, "Rubber Suspension in British Railways Coaching Stock," Railway Gazette, Vol. 101, October 15, 1954, 161-166 Fleet Street, London EC4, England, p 434.

British Railways has been conducting experiments with a new development in rubber, known as the Andre-Neidhart System, adapted to the suspension of railway vehicles. This suspension was adopted to replace the conventional method of axlebox and bolster suspension. The trials have indicated that improved vertical riding would be achieved by the fitting of a much stiffer suspension than was usual and that the transverse cushioning effect was likely to give improved riding qualities.

RRIS: 037770

TDOP: 03-063

Temple Press Limited, "Synthetic Rubber Springs," Railway Gazette, Vol. 117, August 10, 1964, 161-166 Fleet Street, London EC4, England, pp. 167-168.

Neoprene disc springs, developed in Austria, are replacing both the steel and natural rubber springs formerly used because of their superior performance. In service operating tests, neoprene springs have been used on railcar bogies with excellent results. They have all the qualities of steel springs as regards suspension, and also incorporate self-dampening properties which do away with the need for hydraulic shock absorbers. They resist the effects of oil, heat, weather, ozone, and abrasion.

RRIS: 037270

TDOP: 03-064

U.S., Federal Coordinator of Transportation, Report of Mechanical Advisory Committee to Federal Coordinator of Transportation On Freight Cars, 1935, pp. 186-199.

This report includes an historical introduction and a description of different types of springs for freight car trucks. Laboratory and service tests conducted up to the time of the report and important results are included, and the use of nonharmonic truck springs is recommended. Also, there is a discussion of stresses in lading, cars, and track.

TDOP: 03-065

Association of American Railroads, "Analysis Of Hot Box Data Submitted By Fourteen Member Roads," Research Report, AAR MR-210, December 1953, 3140 South Federal Street, Chicago, Illinois 60616.

The results of this study of 2,161 cases of hot boxes indicated that hot boxes occur less frequently on spring plankless trucks. However, the cars with spring plankless trucks were newer cars. It was concluded that, at the time of this writing, the major cause of hot boxes (71.16%) was the conventional waste pack. It was suggested that elimination of this waste pack by substituting an improved method or material to provide the lubrication would eliminate much of the problem. Periodic repacking, and the associated attention to the box assembly at that time, did not reduce the frequency of hot boxes during the months immediately following, as might be expected. The higher capacity cars ran hot less frequently than the lower capacity cars, but the higher capacity cars were generally more modern. The number of hot boxes caused by bearing defects indicated a need for improvement of bearing design and box assembly. Tank and refrigerator car performance was less satisfactory than the average of all other types.

RRIS: 040357
TDOP: 03-066

Bruner, J. P., G. N. Benjamin, and D. M. Bench, "Analysis Of Residual, Thermal, And Loading Stresses In A B33 Wheel And Their Relationship To Fatigue Damage," ASME, 66-WA/RR-3, 345 East 47th Street, New York, N.Y. 10017.

This investigation, which was part of a General Electric computer study sponsored by the American Iron and Steel Institute, involved the corollary problem of service loading conditions that produce the highest stresses and the possibility of fatigue damage. Static loading, rim heating, residual stress measurements, and fatigue tests were made in the laboratory on representative B33 wheels. A series of simulated loading conditions was studied, and the resulting stresses combined by simple superposition principles. The resultant stress patterns were compared with fatigue test results using the modified Goodman relationship. In this way, the service loading conditions that produce fatigue damage could be predicted.

TDOP: 03-067

Bruner, J. P., S. Levy, R. D. Jones, and J. M. Wandrisco, "Effects Of Design Variation On Service Stresses In Wheels," ASME, 67-WA/RR-6, 345 East 47th Street, New York, N.Y. 10017.

Computer programs simulating braking and loading conditions reported in 1965 are claimed to have been applied (in this 1967 paper) to different wheel designs. No one optimum wheel design satisfies all possible service conditions. A method is proposed to optimize design for specific service conditions to safeguard against fatigue damage.

TDOP: 03-068

Byrne, R., "Railroad Axle Design Factors," ASME, 67-RR-3, 345 East 47th Street, New York, N.Y. 10017.

Railroad axle designs have been developed from an application of theoretical principles of Reuleaux combined with extensive laboratory fatigue studies supplemented, to some extent, by road service tests. The designer is presented with data on the effects of the complex forces acting on axles operating in railroad service. Criteria for wheel seat and axle body stresses, established from fatigue tests and modified by practical considerations, are discussed. This paper gives elaborate reference material for use by future investigators of axle properties and designs.

TDOP: 03-069

Canadian National Railways, <u>Dynamic Loads on Freight Car Roller Bearings</u>, by F. E. King, J. Thivierge, and K. Waranica, Research Report, June 1963, Department of Research and Development, Technical Research Branch, Montreal, Quebec.

This report gives details of the measurement and analysis of dynamic loads on freight car roller bearings. Tests were conducted on 5-1/2 in. by 10 in. Timken, Type A-P, roller bearings with the majority of readings taken on cars loaded to 80-percent capacity. Statistical techniques were used to evaluate the mean effective load for dynamic conditions. Based on data developed in this report, bearing life formulae for freight car applications are suggested for further evaluation. The effects of various parameters on dynamic loadings are also considered in some detail.

TDOP: 03-070

Eksergian, C. L., "Theory And Practice Of Wheel Control," January 22, 1952, (paper presented before Committee on Land Transportation, American Institute of Electrical Engineers, New York).

This paper discusses the theory and practice of a wheel slip control device. Retardation of a railroad car with conventional brake systems is established by the degree of adhesion between wheel and rail and the brake shoe on tread limitations. Development of disc brakes in the late 1930's opened the way for improvements in retardation. To prevent wheels from sliding when braking forces exceeded the wheel-rail adhesion, a wheel protector control device was introduced.

TDOP: 03-071

Federal Railroad Administration, "Research Study To Perform Analysis Of Railroad Car Truck And Wheel Fatigue," (FRA Contract CN-DOT-FR-20070), June 1972, IIT Research Institute, 10 West 35th Street, Chicago, Illinois 60616.

The object of this contract is to identify and investigate the load and environment factors which influence performance efficiency of freight car trucks and wheels, devise and conduct an experimental program to determine the effect of the frequency and magnitude of the dynamic loads which are imposed upon trucks and wheels in a range of freight operations, define engineering design and service life criteria for freight car trucks and wheels, investigate the interrelation between existing designs and manufacture and the degree of structural adequacy which each offers, and evaluate alternative conceptual approaches to freight car truck design on the basis of broad application of costs and benefits.

RRIS: 036354

TDOP: 03-072

Gogol, M., "Needed Improvement In Reliability Of Rotating Parts On Freight Cars," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 41-51.

This article reviews the history of wheel and axle development and suggests a factor to be considered in making selections of these components to obtain optimum performance. The article also traces the development of AAR specifications for wheels and axles, and reviews 1970 hot box performance and the development of ultrasonic inspection techniques for railroad axles. A resume of data on Southern Pacific Transportation Company's wheel and axle failures is included.

TDOP: 03-073

Harrison, F. D., "Novel Axleboxes For Express Locomotives," Railway Gazette, Vol. 92, May 12, 1950, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 539-545.

The axleboxes covered in this report are of the Athermos mechanically-lubricated type. All the accepted components of an Isothermos axlebox are incorporated, namely: a bearing ensuring copious fluid film lubrication over the whole journal, an oilflinger conveying a large volume of oil to the bush, an oil sealing ring shrunk on the axle, and a safety pad which effectively protects the oilflinger from damage. The special features peculiar to this new axlebox are: a novel guiding system consisting of forked links mounted in Silenblocks that permit the axlebox to move vertically without fore and aft deviation and a novel device allowing a controlled lateral play of 20 mm that has a marked effect on easing the running on sharp curves.

RRIS: 037897

TDOP: 03-074

Horger, O. J., "Factors Affecting Axle Stresses," ASME, 70-WA/RR-6, 345 East 47th Street, New York, N.Y. 10017.

The author claims too little use of advanced technology is being made to increase fatigue strength of axles. Means are presented to reduce stress concentration and fretting corrosion on axles and wheel assemblies.

TDOP: 03-075

Horger, O. J., "Wheel, Axle, And Rail Stress Problems Related To Higher Capacity Cars - Part 1 - Axle Problems," ASME, November 21, 1963, 345 East 47th Street, New York, N.Y. 10017.

This paper submits some fundamental considerations in the design of axles and proposes two new axle designs of 72,000 and 80,000 pounds capacity, respectively. Larger axle design standardization must satisfy a wide range of car geometry factors such as (a) center of gravity height from 72 to 94 in. and (b) wheel diameter ranging up through 40 in. The effect of these factors on axle capacity is shown by curves derived from the Reuleaux formula; serious deficiencies in this formula are also discussed. Other primary axle design factors presented are: (c) wheel seat design; (d) kind of steel and heat treatment; (e) effect of curved track; (f) effect of switches, frogs, and crossings; (g) effect of speed; and (h) effect of flat spots and shell-outs of wheel treads.

TDOP: 03-076

International Union of Railways, Maintenance Of The Wheelsets Of Trailer Stock. Tests On Passenger Coaches To Ascertain The Permissible Out-Of-Roundness And Out-Of-Balance Of The Wheels Of These For Speeds Of Between 0 And 250 KM/H, Report 9, Question B79, October 1970, Office for Research and Experiments, Utrecht, Netherlands.

Previous reports contain the results of the tests carried out with a view to determining the permissible out-of-roundness and out-of-balance values for the wheelsets of trailer-stock, for running speeds comprised between 0 and 150 km/h. New tests have been carried out for speeds between 0 and 250 km/h. These tests consisted of bench tests carried out on the SNCF dynamic test rig for vehicle suspensions at Vitry-sur-Seine, and have concerned 3 coaches (SNCF, DB, and FS). In addition, line tests have been carried out with the SNCF coach. In the first series of tests, a speed of 250 km/h was reached, while in the second series of tests, the running-speed was only 160 km/h. The results obtained have permitted the following limit values of plus or minus 0.15 mm for the out-of-roundness and 0.125 mm for the out-of-balance per wheel (static balancing would seem to suffice in most cases) to be fixed, such that the comfort obtained with the three coaches is still acceptable. This is valid up to the speed of 250 km/h in the case of DB and SNCF coaches and up to 220 km/h for the FS coach. It has also been found: a) that the test-bench used constituted a perfect reference basis, and that it seems desirable to devise a better correlation between the excitation obtained on the track and that produced on the rig; b) that, for the purpose of determining the comfort, the whole of the floor of the coach should be included in the examination, since, for speeds beyond 150 km/h, the body can present a complex vibratory system with several nodes; c) that, if a coach is to be used for high running speeds, it must be checked to see that beyond a certain speed there is no very sharp increase in all the recorded accelerations.

RRIS: 033264

TDOP: 03-077

International Union of Railways, Roller Bearing Axleboxes-Passage Of Electric Current Through Axle Roller Bearings, June 1965, Office for Research and Experiments, Utrecht, Netherlands.

One organization noted a large number of axle roller bearings damaged by grooves formed on both the rollers and the inner ring. The damage was subsequently found to be due to the passage of electric current which flowed from the locomotive down the main heating lead, through the heaters, and then across the roller bearings on its way to the rails and back to the locomotive. Quite small currents can cause roller bearing damage, particularly if the grease used has unfavorable electrical properties. The only reliable solution is to fit flexible copper connections between the coach body and truck, and to fit at least one sliding contact electrically connecting the truck to the wheelset and thereby providing a low resistance circuit so that the bearings are effectively shunted.

RRIS: 040407

TDOP: 03-078

Kaplan, A., T. K. Hasselman, and S. A. Short, "Independently Rotating Wheels For High Speed Trains," Paper No. 700841, Society of Automotive Engineers, 2 Pennsylvania Plaza, New York, N.Y. 10001.

An investigation has been made of the possible application of independently rotating wheel systems to high-speed rail systems. The objective of this application is to eliminate the hunting stability problems inherent in conventional wheel systems, and thus improve the vehicle ride comfort and safety. A study of possible wheel-rail configurations indicated that a single point contact with zero contact angle and a difference between wheel and rail contour radii of between 5 and 10 in. was preferable. An individually rotating wheel system appears to be a strong candidate for high-speed rail systems.

RRIS: 046923

TDOP: 03-079

List, H. A., "A Wheel Profile: For Better Riding And Longer Wheel Life," Modern Railroads, May 1970, Watson Publications, 5 South Wabash Avenue, Chicago, Illinois 60603, pp. 61-62.

This report states that riding qualities can be improved and wheel life can be extended by making rather modest modifications to conventional profiles. The present AAR profile has one major shortcoming in common with the conventional British profile, and one additional problem that is unique to railroading in North America. The common problem is that there are two points of contact between the wheel and the rail when flange guiding is required. The uniquely American problem is associated with the inward cant at a 1:40 angle whereas the angle of the tread face is 1:20. Because of the angle between the rail and the tread face, there is rapid tread wear immediately after wheels are turned and an attendant loss in lateral riding qualities. The proposed profile uses a basic taper of 1:40. A flange throat contour which at all points has a slightly larger radius than the head of the rail, and a short section of flange at a 70-degree angle. The region of the tread face which gets the least use is relieved at a 1:10 taper. Using a modified flange throat contour will improve the steering action of a wheelset.

Lateral movement of the wheel will be opposed by a smoothly increasing lateral restraining force, and on a curve the rolling radius of the outside wheel can increase much more than is possible with a conventional wheelset.

RRIS: 037667

TDOP: 03-080

McArd, G. W., "Axle Stresses," Railway Gazette, Vol. 73, September 27, 1940, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 323-324.

This report includes a comprehensive survey of the methods used to determine axle stresses. The axles are classed as carrying axles or driving axles. In this portion of the survey report, bending moment diagrams are shown for braked and unbraked carrying axles for both the inside and outside bearings. The bending moment diagram for a carriage axle including allowance for the overturning effect at speed is also illustrated.

RRIS: 037685

TDOP: 03-081

Menaker, E. G., "The Fundamentals Of Infrared Hot Box Detection," American Institute of Electrical Engineers, 345 East 47th Street, New York, N.Y. 10017 (paper presented at Conference of American Institute of Electrical Engineers).

Existing data on railroad car plain bearing temperatures are presented and interpreted by means of equivalent thermal circuits. Analyzing the radiation from the car truck, relating it to the characteristics of infrared measuring equipment, and discussing the handling of the resulting data suggests approaches to more efficient use of wayside hot box detectors. The effects of some ambient phenomena are presented. Further study of transient temperature conditions following lubrication failure is suggested.

RRIS: 040136

TDOP: 03-082

Nakamura, H. and C. Konishi, "On The Strength Of The Car-Axle," Railway Technical Research Institute, April 1963, Japanese National Railways, Japan (paper presented at International Wheel-Sets Congress, Bergamo 7-12, 1963).

Frequency and magnitude of dynamic stress at wheel seats were obtained. Mean values and standard deviations of bending stresses for various radii curves were calculated and shown. S-N curves for wheel seats are shown for various heat-treated specimens. The effect of flat wheels upon the bending stress of axles, stresses in the wheel, and vertical load on the wheel were measured and presented.

TDOP: 03-083

Nishioka, K., K. Ishii, and H. Komatsu, "Fatigue Strength Of Induction Hardened Railway Axles," Japanese National Railways, Tokyo, Japan.

The effect of size on the fatigue strength of induction-hardened and press-fitted axles is described, together with the effect of their tempering temperature after quenching. The measurement of superficial residual stress by an X-ray diffraction method at the fretted portions is carried out, and it is revealed that the decrease of compressive residual stress is closely related to the occurrence of minute cracks. Statements are included about the investigations of real axles in service on the new Tokaido Line where the induction hardening procedures are employed. Related to the results obtained from these examinations, fundamental experiments are performed in order to clarify the fretting fatigue phenomena connected to the occurrence of minute cracks. Finally, attention is paid to the effect of size on the relative slip and to the effect of the relative slip on the fretting fatigue strength.

TDOP: 03-084

Nishioka, K., T. Kunitake, K. Hirakawa, and H. Komatsu, "Improvement Of The Fatigue Strength Of Press-Fitted Axles," The Sumitomo Search, No. 3, May 1970, Sumitomo Electric Industries Limited, Osaka, Japan, pp. 72-84.

This paper summarizes the results of rotating cantilever bending fatigue tests made on shafts having a press-fitted outer disk member to find the most effective method to improve fatigue strength of press-fitted assemblies. Many factors, such as shape or geometry of assembly, induction hardening, nitriding, subcritical quenching, different engineering steels and heat treatments, and type of lubricant used between the mating surfaces, are treated here to demonstrate how each respectively influences the fatigue strength of fitted assemblies. After reviewing these factors, it should become apparent that great improvement in fatigue resistance against crack propagation can be obtained by introducing favorable residual stress. Against crack initiation, the effect of residual stress is somewhat limited, but local surface hardening such as tuftriding and induction hardening is rather promising.

TDOP: 03-085

Riegel, M. S., "Stresses In Wrought-Steel Wheel Rims And Their Relation To Wheel Life," ASME, 59-A-241, 345 East 47th Street, New York, N.Y. 10017.

This paper discusses laboratory investigations of residual stresses in new wheels. Service stresses were determined by laboratory testing simulating operating conditions and investigation after a period of actual service. The paper relates these results to wheel life.

TDOP: 03-086

Sherrick, J. W., "Roller Bearing Adapter Mountings For Railroad Cars," ASME, 68-WA/RR-5, December 5, 1968, 345 East 47th Street, New York, N.Y. 10017.

This paper deals with the development of an elastomeric mounting for installation between the roller bearing adapters and truck side frames of railroad cars. This mounting will provide for a controlled lateral motion between the adapter and side frame of a nominal plus or minus 5/16 in. Test installations and the results of these preliminary tests are discussed, and the advantages of a roller bearing adapter mounting are pointed out. Most of the purported advantages are supported with test results while others are predicted as a result of the test data.

RRIS: 040322
TDOP: 03-087

Temple Press Limited, "Conversion From Plain To Tapered Roller Bearings," Railway Gazette, Vol. 106, June 21, 1957, 161-166 Fleet Street, London EC4, England, p 711.

All coaches of the Morning Talisman were fitted with the Timken tapered roller bearings. Special design problems were involved for each type of coach, because the bogie designs varied. A diagram shows a Timken dual race bearing with a 4.5-in. bore and an outside diameter of 7.94 in.

RRIS: 037635
TDOP: 03-088

Temple Press Limited, "Journal Stops For Wagon-Type Plain Bearings," Railway Gazette, Vol. 109, October 3, 1950, 161-166 Fleet Street, London EC4, England, pp. 425-426.

This report points out that in a new A.A.R. journal bearing assembly, with all dimensions nominal and journals central laterally, the maximum displacement from impact is 7/16 in. Bronze stops and a cap screwed to the side wall of the journal box to prevent displacement of the journal out of its bearing were considered the most practicable means to achieve the desired stabilization. The increased bearing life obtained with the journal stops results principally from reduced end wear and the elimination of spread linings.

RRIS: 037658

TDOP: 03-089

Temple Press Limited, "Measuring Longitudinal Wear On Journals," Railway Gazette, Vol. 70, January 27, 1939, 161-166 Fleet Street, London EC4, England, p 140.

A gauge permitting the longitudinal wear on the journals of railway carriage axles to be measured was devised at the shops of the former P.O.-Midi Railways (now South-Western Region, French National Railways). It was designed to make a comparison of the standard with the actual dimensions easily possible. The difference between the two represents the amount of wear. The use of the device is described in this report, and the device is illustrated.

RRIS: 039531

TDOP: 03-090

Temple Press Limited, "Wheelset Development Reviewed By Engineers," Railway Gazette, Vol. 124, July 18, 1969, 161-166 Fleet Street, London EC4, England, pp. 550-553.

Design and wearing qualities to accommodate increased speeds and axleloads predominated in the papers delivered at the Third Wheelset Conference. Spalling of wheels experienced on North American rapid transit systems had been brought about by wheel slip, and it was recommended that a total adhesion system should be considered at the design stage of coaches. In introducing synthetic brake blocks where wheel tyre cracking existed, it was considered that tyre steels which should be used were those that did not have a hardening tendency such as soft steels of low carbon content, which were immune to heat cracking. Resilient wheels for use under rail vehicles for suburban and mainline railways, hollow-tread profiles, wrought-steel wheels, and nonalloy wheels were also briefly discussed as well as cast steel wheels.

RRIS: 037446
TDOP: 03-091

Weaver, G. R., P. A. Archibald, E. B. Brennan, and G. M. Cabble, "Investigation Of The Thermal Capacity Of Railroad Wheels Using COBRA Brake Shoes," ASME, 69-WA/RR-2, 345 East 47th Street, New York, N.Y. 10017.

This paper presents an objective evaluation of the thermal capacity of 36 in. CR wheels braked with COBRA brake shoes and relates it to that of wheels braked with cast metal brake shoes. The wheel tread conditions, hardness and macrostructure of the rim, and residual stress patterns, which developed from high speed dynamometer braking, were investigated through three progressive test series. The results indicate that the thermal capacity of wheels braked with COBRA brake shoes far exceeds limits previously established for cast metal shoes.

TDOP: 03-092

Wise, S., D. T. Catling, and R. C. Fernando, "Wheels, Tyres, and Axles," Railway Engineering Journal, July 1973, Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England.

This article presents a review of current practice in the UK and Europe concerning wheels and axles. The typical railway wheelset with wheels pressed on to the axle and tyres shrunk on to the wheels has been used on European railroads now for more than 100 years. Traditionally, the expertise in this branch of technology rests with steel makers, railway wheel shopmen, and metallurgists. We are now approaching an era in which substantial changes may have to be made in established forms of design, material, or manufacturing methods; and it may be of value to review established practice, to describe current practice, and to indicate the directions of possible future changes with particular reference to BR and LTE.

RRIS: 046083

TDOP: 03-093

Yontar, M., "Research On The Operating Stresses in PATH Rail Car Axles, Drive Systems, Wheels And Rail Joints," ASME, 66-RR-6, 345 East 47th Street, New York, N.Y. 10017.

This paper identifies the cause of premature cracking of axles in PATH drive systems as the bending mode oscillation of the axle. Test data confirming this conclusion are presented. Corrective actions proposed were under investigation by the Port of New York Authority at the time of this report.

TDOP: 03-094

Miscellaneous Component Systems

Angold, J., "Improved Freight Car Running Gear--A Necessity For The 70's," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 32-37.

This paper analyzes motions of the freight car truck in the interest of indicating how the truck action relates to wear of components such as: coupler shanks, coupler carriers, bolsters, side frames, roller bearing adapters, etc. A conical centerplate concept, which is being used in experimental applications under a series of cars operating on a western railroad, is presented. The coaxial train concept where wheel suspensions are individually installed in a continuous center sill is described.

TDOP: 03-095

Davis, F. H., "Steel Castings in Railway Engineering," Railway Engineering Journal, March 1972, Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England, pp. 7-18.

This paper reviews the development and application of steel castings to design of railway equipment and particularly to freight car trucks. The process of meeting the economic and technical requirements is described in detail.

RRIS: 048007

TDOP: 03-096

Johnson, M. R., "Development of Fatigue Standards For Freight Car Truck Components and Wheels," ASME, 74-RT-4, 345 East 47th Street, New York, N.Y. 10017.

Factors which should be considered in the development of fatigue standards for freight car truck components and wheels are discussed. These standards would be formulated to provide a desired level of operational reliability. They would be based on the fluctuating loads acting on the components in service. Typical data describing this environment are presented. Statistical considerations in establishing the margin between the environmental loads and component fatigue strength are also described.

RRIS: 054006

TDOP: 03-097

Section 4

# TRACK-TRAIN DYNAMICS AS RELATED TO TRUCK PERFORMANCE

## INTRODUCTION

The following subsections are comprised of reprinted articles, which were selected from the assembled literature concerning track-train dynamics as related to truck performance, and the TDOP bibliography for this subject.

The reports selected for reprinting are: "Application Of Rail/Vehicle Dynamic Analysis To Train Operation," by E. F. Lind, H. Ten Broeck, and H. Nations and "Vehicle/Track Interaction And Dynamic Responses To The Track," (portion of panel discussion) by L. A. Peterson. The report by Mr. Lind, Mr. Ten Broeck, and Mr. Nations deals with the results of a study conducted by a task force on track-train dynamics. The presentation by Mr. Peterson deals, in part, with a theoretical discussion of train dynamics in which the author emphasizes the role of field testing in understanding the wheel-rail interface.

The bibliography, which follows the reprints, is arranged in alphabetical order by author. Supplemental pages will be added to the "Bibliography--Track-Train Dynamics As Related To Truck Performance," as new articles become available throughout the course of the TDOP.

# APPLICATION OF RAIL/VEHICLE DYNAMIC ANALYSIS TO TRAIN OPERATION

Joint Presentation
by
Southern Pacific Transportation Company Representatives

E. F. Lind, Manager of Analytical Engineering
Howard Ten Broeck, Equipment Design Engineer
Harry Nations, District Road Foreman of Engines

at the

Association of American Railroads

Conference on
Track/Train Dynamics Interaction

December 16, 1971
Chicago

Reprinted from Proceedings of Conference on Track/Train Dynamics Interaction, December 15 and 16, 1971, Association of American Railroads, Research and Test Department, American Railroads Building, 1920 L Street, N.W., Washington, D.C. 20036.

MR. LIND:

During the past few years there's been a definite resurgence of the idea that our industry will be required to produce more freight ton miles than ever before. Possibly as high as 45% of the Nation's total by the year 1985. Railroading is a business that depends on hardware. We can survive and prosper in the future only if we have the physical facilities adequate to meet these growing demands. Our ability to develop and apply improved technology is therefore, of major importance if we are to improve the quality and quantity of our service. Marketing emphasis on certain very profitable commodities in the past decade has been responsible for developing a constant trend towards larger and heavier cars. This phenomena along with longer and faster trains has introduced serious dynamic problems that impose definite constraints on the safe operations of the railroads. After evaluating the importance of the subject of track-train dynamics one must be amazed at how little progress has been made in better understanding the dynamic interrelationship between trains and track, however, an attempt to write the equations of motion necessary to study the dynamics of even a relatively simple freight car will quickly lead to the conclusion that the simple freight car is in reality a very complex dynamic system. The fact is that until the recent development of modern high speed computers, it was impractical if not impossible to consider all the dynamic factors involved. Even now a complete train representation is not practical and attention is generally focused on segments of a train. As a result of some various mysterious situations involving derailments, our company formed a task force to study this phenomena, track-train dynamics. From the very beginning, we realized the need for sophisticated instrumentation and computers in the analysis of transient data relating to the dynamic characteristics of vehicles, track structure and their interaction if any success was to be achieved. The primary purpose in instrumenting a locomotive-freight car combination was to obtain transient data regarding train behavior during our test runs. To this end, the combination was instrumented to measure and record on analog FM tape certain forces, displacements, angles and speed vectors. At this time, Howard Ten Broeck will present (some background) information explaining the reasons for our involvement in this project.

MR. TEN BROECK:

During a 2-1/2 year period ending in February 1970, we experienced 11 major derailments on the Southern Pacific, and several of these had price tags exceeding 1-1/2 million dollars. In reviewing the details of each occurrence, it was determined that:

1. The causes of these derailments were not conclusive.
2. Most occurrences involved speeds between 50-60 MPH and speed changes prior to the point of derailment.
3. Many of these derailments occurred in territories involving reverse curves with short reversing tangents.

4. A large number of these derailments involved the trailing units of a multi-unit consist and as many as 48 cars on the head end of the train.

We also learned that during the same period of time other railroads had experienced similar type derailments. At this time, only limited investigations had been conducted to determine their causes. Initial fingers of suspicion were pointed at high horsepower 6-axle locomotives and long overhanged freight cars. At this point, our task force was formed and with the assistance and cooperation of the AAR Research and Test Department, National Castings Company and E.M.D. Our track-train dynamic study was then initiated. After reviewing previous test programs and derailment data it was decided to concentrate our investigations in the following areas:

1. Compare L/V ratios on the trailing wheels of a multiple unit consist to the L/V ratios on the lead axle of the first car behind the locomotive.
2. Compare four axle versus six axle locomotives and measure dynamic profile under loaded conditions.

One other important thing we want to evaluate is the effects of train handling on high L/V ratios. In our tests, we use E.M.D.'s locomotives, the four axle GP35 and their six axle SD45 and we are of the opinion from other tests we have conducted any 6-axle high horsepower locomotive will produce the same results as far as the wheel-rail reactions are concerned. Our basic test sites were located between Los Angeles, California and Tucson, Arizona where we made repeated runs using different combinations of locomotives and cars. All of our tests were conducted in revenue trains on expedited schedules having high horsepower per ton ratios. Additional tests were made between Los Angeles and El Paso, Texas and further tests between El Paso, and Pine Bluff, Arkansas. This represents over 5,000 miles of test data and later in the program, we will explain to you how this data was analyzed and our conclusions. It was apparent from beginning that one of the most significant factors effecting an increased tendency for derailments was incorrect train handling. I would like at this time to introduce Harry Nations who was a member of the task force. He will tell you about some of the things we have found about train handling and also what steps were taken to educate our engineers and road foremen to correct train handling practices.

<u>MR. H. NATIONS</u>:

Thank you Mr. Ten Broeck. With the advent of the high horsepower multiple unit locomotive, it's becoming increasingly apparent that the engineer can no longer handle the train by the seat of his pants method. I, as a member of the task force, and being a district road foreman of engines was primarily interested in train handling. In our tests that was made with the EMD's Test Car we determined a number of factors that were not in keeping with good train handling practices. We also determined that by careful train handling on the part of the engineer, we could keep slack action within safe limits. In one test we deliberately had the engineer go from throttle position No. 6 to dynamic braking without waiting the required 10 seconds. The load meter immediately indicated the effects of such action and we had a severe run-in. The instrumented car behind the locomotive registerd a very high L/V ratio and the truck under the car being tested actually tried to twist on the rail. We feel that if this had occurred on a curve or had it been for an extended period of time, we very possibly would have derailed, so we have taken steps now to equip our locomotives with a ten second delay. This is automatically activated when going from power to dynamic brake and from dynamic brake to power to prevent any mishandling on the part of the engineer. We also had engineers make various reductions of speeds by making use of braking. In one case, the engineer was going 57.1 MPH in curve territory. He then reduced the speed of the train by making a reduction in the airbrake pipe of 10 lbs. This reduced his speed to 43.5 MPH but in the process, he increased the throttle from No. 5 to No. 7 position. Our tests indicate that this is not a good method of train handling because we received a high L/V ratio that could also cause a derailment. We found that when using the dynamic brake we could control the slack occurring within the train by having the engineer observe the ten second delay and increase the braking force gradually so as to keep the L/V ratio within safe limits. Immediately after these tests we started a program of training our engineers. First, we held meetings with the road foremen of engines. We told them what information our tests developed and discussed train handling methods with them to insure better train handling that would avoid harsh slack action. We would like to avoid all slack action, however we cannot expect the impossible, but we can avoid severe slack action by use of proper handling practices. After these meetings we held meetings with engineers and following these meetings we rode with them for instructional purposes. We feel that because of these classes we have decreased the derailments and damage to lading materially. We recently developed a computerized form that the engineer fills out on each train separation. The report contains such information as the number of cars, the tonnages, the number of locomotive units and their location in the train and any other pertinent information. This is accompanied by another form which is filled out by the road foreman of engines who is required to investigate each and every separation stating the cause and corrective action taken. We feel that this program which is now under way will eventually help us further reduce separations and rough handling which may contribute to derailments. We also have in Los Angeles area a simulator at our enginemen's training center. We have a number of well qualified instructors at this location who teach engineers proper train handling practices by using our simulator to show them what takes place in a train. We can simulate just about any kind of condition that occurs in the real world. By using this tool an engineer can actually see what he is doing wrong and take corrective action. We feel that this will eventually elevate our engineers to a much higher level of competence. We eventually hope to have all of the engineers receive this training. At present we are giving this training to new engineers. At the training center in addition to that, we give instructions on airbrake systems, electrical systems, and diesel engines. When a man completes the course at this training center he leaves as a well qualified engineer. We have a number of young engineers that are living examples of the benefits of this program. Most recently, in the last strike we had, of course, a shortage of engineers. We gave management personnel a short training course in the simulator. They knew very little about operating rules and had never handled a train. After completing this short course, they did a fine job of handling engines during the strike. They had no difficulties in performing the duties of a helper engineer. All of this of course was due to the training they received at our training center. I would now like to tell you about an accident we had in Arizona desert prior to these tests. We had a derailment east of Yuma, Arizona on one of our expedited trains. We piled up about 35 cars and the cause of the accident was unknown. The area involved had a series of curves (reverse curves) on a 1% descending rate. I ran trains over this section of track using different methods of train handling. First I went through the area controlling the train with a light brake application, then I followed with the use of dynamic brake to control the speed of the train at 60 MPH. We did not register any high L/V ratios or any undue stresses in the track by use of either method. We made some 15 tests using every conceivable method of operation. I even changed from air to dynamic brake, dynamic back to air in the body of the curve, being very careful in each instance to control the slack action, so that there would be no possibility of derailing. Finally, a high ranking officer came out and wanted to know what we determined the problem to be. I told him that we do not know at this time. He said that he would find out and proceeded to have a conference with the engineer of test. He came out and said "I want you to handle this train as rough as you know how". I told him "we might derail the train". He said "Okay, derail the train". So I went into that curve at 60 MPH in throttle 8 and closed the throttle with one sweep of the hand and went immediately into maximum dynamic braking. Of course I got a terrible run-in and I'd thought that I did everything except derail the caboose. But without my knowledge, this officer boarded the caboose. So when I went back to the test car expecting to find the water barrels all torn off the walls, the man indicated smooth as silk -- no rough handling. We did that 2 or 3 times once I went from throttle 8 to the maximum independent brake application, the slack almost threw me into the back wall of the cab. Again he said smooth as silk. This told us that the slack action was being absorbed in that train. Many times we have had derailments that we thought were caused by slack action and the conductor said that there was no severe slack action. We thought he was lying to protect the engineer but our test indicated it was possible. We then took a train of a hundred empties, one load, and four 2500 HP units. My boss said, what are we going to do this time? I said that I am going to go into emergency in the body of the curve. I was going about 60 MPH in main body of the curve, when I went into emergency. This time I went back to the caboose and my boss said, we will not do that again. Later I was told that the caboose stopped about six times. But in these other tests, that I referred to earlier, we did get high stresses in the rail even though it was not transmitted to the caboose. So we know that slack action can cause high stresses in the rail. Because of these tests I feel that it is absolutely essential that the road foreman of engines continue constant supervision over the engineers on his district. You cannot go out and tell an engineer one time "This is what we want and this is what you are going to do". You have to make them do it. On November 29 and 30 we had a series of meetings in Los Angeles, the district road foremen were present; the rules examiner and a representative of The Enginemen's Training Center with the idea of standardizing train handling methods throughout the system. The General Manager decided that we should standardize train handling practices. We used the airbrake book published by the Airbrake Association Modern Train Handling as a base or guide to establish better train handling practices. We made a few changes that we considered particularly beneficial to our railroad. It will be published in booklet form and will be distributed to all engineers as guide for better train handling. Mr. Lind will further explain other parts of this program. Thank you.

MR. LIND:

Thank you Harry. I will now attempt to describe the analytical tools that we developed to evaluate the results of these tests.

The principal reasons that necessitated the development of new computerized techniques to evaluate real-time data were a need to:

1. Analyze the tremendous amount of analog data stored on magnetic tapes.

2. Establish what correlation exists within the data when certain conditions of interest occur.

3. Geographically locate where conditions of interest occur.

4. Eventually incorporate the results of our analysis into a mathematical simulation model.

5. Expand this basic approach to other testing areas.

In the preliminary analysis to scan preprocessed analog data, an E.A.I. 690 Hybrid Computer was used to determine critical occurrences. The samples of interest were then written on magnetic tape according to a predefined numerical format.

In evaluating these tapes four (4) conditions were used to establish when certain conditions of interest had occurred. To be more specific a period of interest was initiated when the L/V ratio on either side of the locomotive or freight car axle did exceed a given value. These values could be changed independently by the engineer while the data was being analyzed. This allowed the engineer a great deal of flexibility in determining what cutoff values were appropriate in any given analysis. It also enabled the engineer to re-cycle the data using different test criteria to compare results.

The time length of each period was a variable depending upon the velocity of the train. Thus, a window look at the data was initiated whenever an interrupt occurred. The window remained open for a time period directly proportional to the velocity of the train.

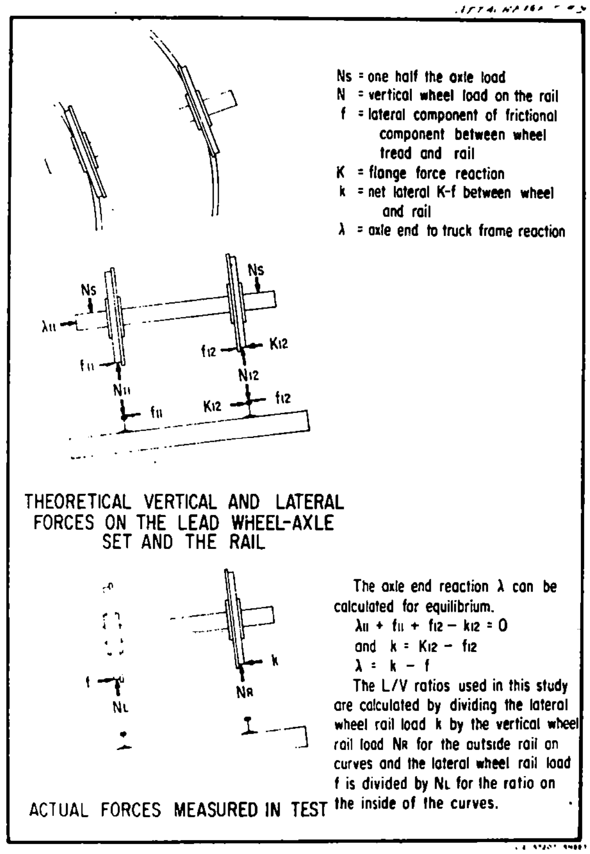
An interrupt did also occur each time a mile post signal was detected. The digital output for each interrupt consisted of eighteen (18) values of information.

In order to see directly the relationship between any two (2) variables such as the L/V ratio and drawbar force, for instance, it was desirable to weed out the effects of other variables such as speed, coupler angle and track deflection. To this end, each occurrence along the railroad track was classified as falling within the appropriate ranges of a five dimensioned matrix. These matrixes represent a population base that could be more accurately interpreted.

It also became obvious after inspecting our "Detail Computer Reports" that the use of curve-fitting techniques and graphics would be required, if any meaningful correlations were to be derived. The decision was then made to use the "Least Squares Method" in order to find a polynomial equation which passed within a specified tolerance of all points in the table. This computational method expresses the dependence of one variable on one or more variables.

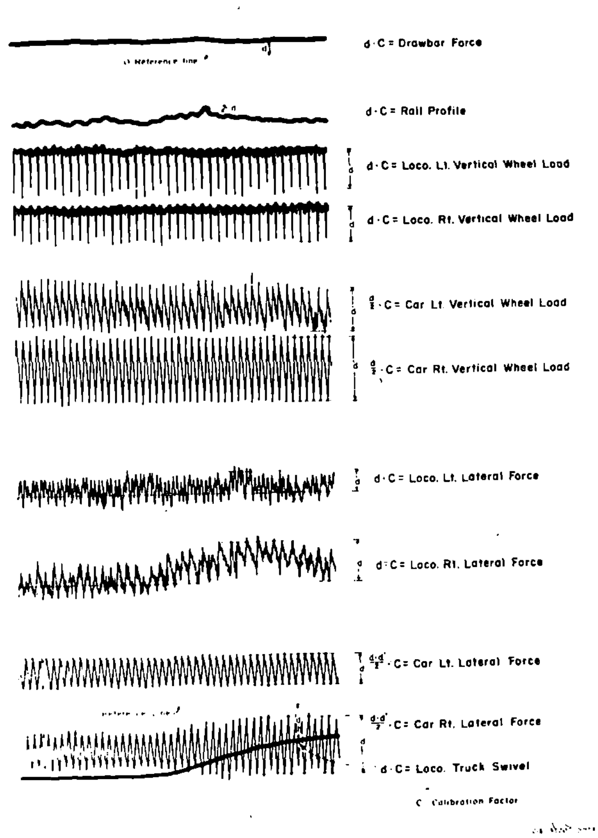
At this time I would like to describe with slides the transient data obtained from an instrumented locomotive-freight car and the programs designed to analyze the data.

Slide 1



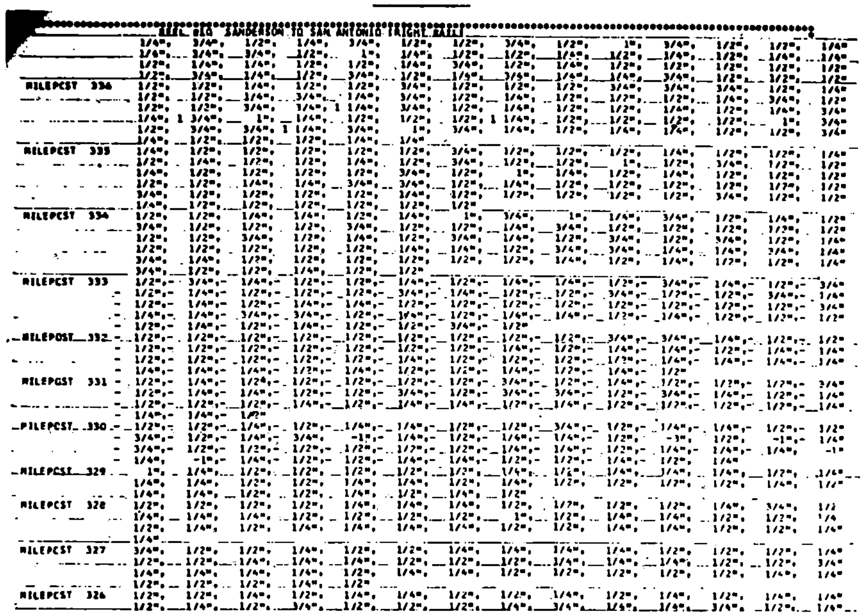

This one shows the actual forces measured in the test, such as lateral and vertical forces, coupler angle (between the locomotive and freight car), velocity of the train, and drawbar forces between the two instrumented cars. This information was recorded simultaneously and continuously over some 5, 000 miles of track.

## Slide 2




This is a copy of an Oscillograph record taken from the test. On top is the signal that represents drawbar force, next you see rail profile signal, and by the way we were using an accelerometer to obtain this measurement. We were able to read deflections directly, and we think that this is of great value to our people in the field. The next series of signals you see is the force signals taken from an instrumented locomotive-freight car. Following these signals is the truck swivel signal which was used to measure the geometric configuration of the track.

Slide 3



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| | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | | | | | | | |
| MILEPOST 334 | 1/2" | 1/2" | 1/4" | 1/4" | 1/2" | 1/4" | 1" | 3/4" | 1" | 1/4" | 3/4" | 1/2" | 1/4" | 1/2" |
| | 3/4" | 1/2" | 1/2" | 1/2" | 3/4" | 1/2" | 1/2" | 1/4" | 3/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" |
| | 1/2" | 1/2" | 3/4" | 1/2" | 1/4" | 1/2" | 3/4" | 1/4" | 1/2" | 3/4" | 1/2" | 3/4" | 1/2" | 1/4" |
| | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/4" | 1/2" | 3/4" | 3/4" | 1/4" | 1/4" | 3/4" | 1/4" |
| | 3/4" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/2" | 1/2" | 1/4" |
| | 3/4" | 1/2" | 1/2" | 1/4" | 1/2" | 1/2" | | | | | | | | |
| MILEPOST 333 | 1/2" | 3/4" | 1/4" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 3/4" | 1/4" | 1/2" | 3/4" |
| | 1/2" | 1/4" | 1/2" | 1/2" | 1/2" | 3/4" | 1/4" | 1/2" | 1/2" | 3/4" | 1/2" | 1/2" | 3/4" | 1/4" |
| | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 3/4" |
| | 1/4" | 1/4" | 3/4" | 3/4" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/4" | 1/2" | 1/2" | 1/2" |
| | 1/2" | 1/4" | 1/2" | 1/4" | 1/2" | 1/2" | 3/4" | 1/2" | | | | | | |
| MILEPOST 332 | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 3/4" | 3/4" | 1/4" | 1/2" | 1/2" |
| | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/4" | 1/4" |
| | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/4" | 1/4" | 1/4" |
| | 1/4" | 1/4" | 1/4" | 1/2" | 1/4" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/2" | | | |
| MILEPOST 331 | 1/2" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 3/4" | 1/2" | 1/4" | 1/2" | 1/2" | 1/2" | 3/4" |
| | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 3/4" | 1/2" | 3/4" | 3/4" | 1/2" | 3/4" | 1/4" | 1/2" | 1/4" |
| | 1/2" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" |
| | 1/4" | 1/4" | 1/2" | | | | | | | | | | | |
| MILEPOST 330 | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 3/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" |
| | 3/4" | 1/2" | 1/4" | 3/4" | -1" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | -3" | 1/2" | -1" | 1/4" |
| | 3/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/4" | 1/2" | 1/4" | 1/4" | 1/4" | -1" |
| | 1/4" | -1" | 1/4" | 1/2" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | | |
| MILEPOST 329 | 1" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/4" | 1/2" | 1/4" |
| | 1/4" | 1/4" | 1/2" | 1/2" | 1/4" | 1/4" | 1/2" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/2" |
| | 1/4" | 1/4" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/2" | | | | | | |
| MILEPOST 328 | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/4" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 3/4" | 1/2 |
| | 1/4" | 1/4" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 1/4 |
| | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/4" | 1/4" | 1/2" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 1/4" |
| | 1/4" | | | | | | | | | | | | | |
| MILEPOST 327 | 3/4" | 1/2" | 1/2" | 1/4" | 1/2" | 1/2" | 1/4" | 1/4" | 1/4" | 1/4" | 1/2" | 1/2" | 1/2" | 1/4" |
| | 1/2" | 1/2" | 1/4" | 1/4" | 1/4" | 1/2" | 1/2" | 1/2" | 1/4" | 1/4" | 1/4" | 1/2" | 1/2" | 3/4" |
| | 1/4" | 1/4" | 1/4" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 1/2" | 1/2" | 1/2" | 1/4" | 1/4" |
| | 1/2" | 1/2" | 1/2" | 1/4" | 1/2" | | | | | | | | | |
| MILEPOST 326 | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" | 1/2" | 1/2" | 1/4" | 1/2" | 1/4" | 1/2" | 1/4" | 1/4" |
| | 1/2" | 1/4" | 1/2" | 3/4" | 1/2" | 1/2" | 1/2" | 1/4" | 3/4" | 1/4" | 1/4" | 3/4" | 1/2" | 1/4" |

I would like to explain that a whole series of programs were developed to analyze this dynamic data. Now the first three programs developed basically analyzed dynamically loaded track profile. We used a highbred system to digitize the values and from these values we then used our large digital computers to prepare the various reports. The first report we generated was a detailed report and within this report we showed the actual deflections that occurred during a particular mile of track. One of the disadvantages of this report was the fact that we generated too much paper. Because of this we developed a statistical report. The only time we now utilize the detail report is in areas where we know the track was very rough and therefore warrants such detailed print-out.

## Slide 4

[illegible] SAN ANTONIO TO CORSICANA (RIGHT RAIL)

| BETWEEN MILEPOST | AND MILEPOST | DEFLECTIONS LESS THAN 1/2" | BETWEEN 1/2" AND 3/4" | BETWEEN 3/4" AND 1" | BETWEEN 1" AND 1-1/2" | BETWEEN 1-1/2" AND 2" | OVER 2" |
|---|---|---|---|---|---|---|---|
| 206 | 205 | 19 | 5 | 4 | 0 | 0 | 0 |
| TOTAL OCCURANCES THIS MILE 28 | | | | | | | |
| TOTAL DEFLECTION THIS MILE 13.51" | | | | | | | |
| AVERAGE DEFLECTION THIS MILE 0.48" | | | | | | | |
| 205 | 204 | 42 | 7 | 4 | 3 | 0 | 1 |
| TOTAL OCCURANCES THIS MILE 57 | | | | | | | |
| TOTAL DEFLECTION THIS MILE 28.73" | | | | | | | |
| AVERAGE DEFLECTION THIS MILE 0.50" | | | | | | | |
| 204 | 203 | 28 | 10 | 4 | 2 | 1 | [illegible] |
| TOTAL OCCURANCES THIS MILE 45 | | | | | | | |
| TOTAL DEFLECTION THIS MILE 24.36" | | | | | | | |
| AVERAGE DEFLECTION THIS MILE 0.54" | | | | | | | |
| 203 | 202 | 29 | 15 | 8 | 3 | 0 | 0 |
| TOTAL OCCURANCES THIS MILE 55 | | | | | | | |
| TOTAL DEFLECTION THIS MILE 30.00" | | | | | | | |
| AVERAGE DEFLECTION THIS MILE 0.55" | | | | | | | |
| 202 | 201 | 49 | 11 | 0 | 0 | 0 | 0 |
| TOTAL OCCURANCES THIS MILE 60 | | | | | | | |
| TOTAL DEFLECTION THIS MILE 24.47" | | | | | | | |
| AVERAGE DEFLECTION THIS MILE 0.41" | | | | | | | |
| 201 | 200 | 37 | 21 | 12 | 0 | 0 | 0 |
| TOTAL OCCURANCES THIS MILE 70 | | | | | | | |
| TOTAL DEFLECTION THIS MILE 37.60" | | | | | | | |
| AVERAGE DEFLECTION THIS MILE 0.54" | | | | | | | |
| 200 | 199 | 34 | 23 | 8 | 9 | 0 | 0 |

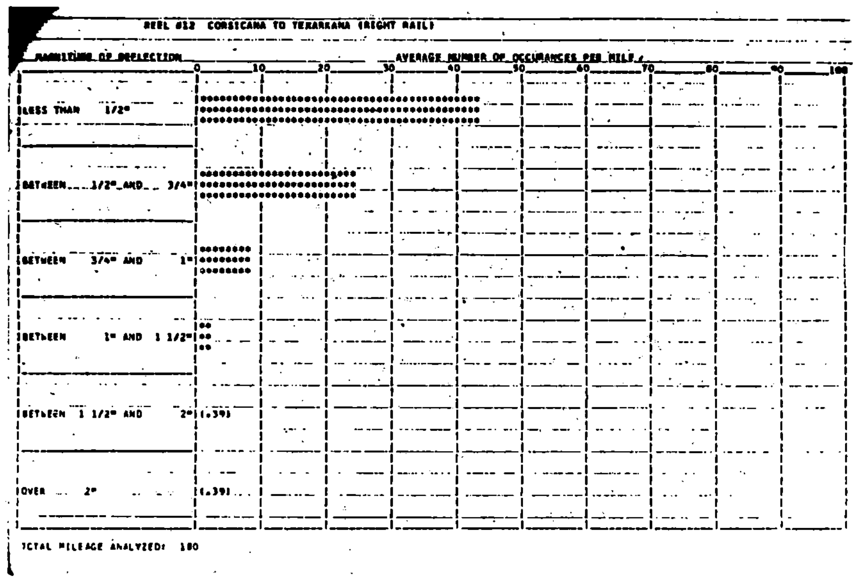
This slide depicts our statistical information which is stored on disc files and can be retrieved instantaneously, and because of this, we are able to perform a statistical analysis. This report shows milepost, total number of deflections for a particular mile of track, and average deflection of this mile. This last item is what we call a roughness co-efficient and we can utilize this in much the same manner as the Southern does to rate a particular mile of track. You also see these deflections categorized and broken down into specific ranges which are established by the user, giving us great flexibility in the report print-out.

Slide 5

```
REEL 012  CORSICANA TO TEXARKANA (RIGHT RAIL)

BETWEEN                   DEFLECTIONS        BETWEEN          BETWEEN        BETWEEN          BETWEEN
MILEPOST AND MILEPOST     LESS THAN 1/2"     1/2" AND 3/4"    3/4" AND 1"    1" AND 1-1/2"    1-1/2" AND 2"    OVER 2"

TOTAL OCCURANCES THIS MILE  60
TOTAL DEFLECTION THIS MILE  28.04"
AVERAGE DEFLECTION THIS MILE  0.47"

MILEPOST 440  3/4", 1/2", 1/2", 1/2", 1/4", 1/2", 1/4", 1/4", 1/4", 1/2", 1/4", 3/4", 1/2", 1/2"
              1/2", 3/4", 1/4", 1/2", 1/2", 1/2", 1",   1/2", 3/4", 1/2", 1/2", 1/4", 1/2", 1/4"
              1/2", 1/4", 1/2", 1/4", 1/2", 1/4", 1/2", 1/2", 1/2", 1/4", 1/2", 1/2", 1/4", 1/2"
              1/4", 3/4", 1/4", 1/2", 3/4", 3/4", 1/4", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2"
              1/2", 1/2", 1/2", 1/2", 3/4", 3/4", 3/4", 1/2", 1/2", 3/4", 1/2", 1/2", 1/2", 3/4"
                                   EQUATION: MP 440 = MP 435
              1/2", 1/2"

   435     434        30           31           2            1            0           0

TOTAL OCCURANCES THIS MILE  72
TOTAL DEFLECTION THIS MILE  35.76"
AVERAGE DEFLECTION THIS MILE  0.50"

MILEPOST 434  1/2", 1/2", 1/4", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2", 1/4", 1/2", 1/2", 1/4"
              3/4", 1",   3/4", 3/4", 3/4", 1/4", 1/4", 1/4", 1/4", 1/4", 1/4", 1/2", 1/2", 1/4"
            1 1/4", 1/2", 1/2", 3/4", 1/4", 1/4", 3/4", 1/2", 1/2", 1/2", 1/2", 3/4", 1/4", 1/2"
              1/2", 1/4", 1/4", 1/2", 1/2", 1/4", 1/4", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2"
              1/2", 1/2", 1/2", 3/4", 1/2", 1/2", 1/4", 1/2", 1/2", 1/4", 1/2", 1/4", 1/2", 1/2"
              1/2", 1/4", 3/4", 1/2", 1/2", 3/4", 1/4", 1/2", 3/4", 1/2"

   434     433        54           18           7            1            0           0

TOTAL OCCURANCES THIS MILE  80
TOTAL DEFLECTION THIS MILE  38.82"
AVERAGE DEFLECTION THIS MILE  0.49"

MILEPOST 433  3/4", 1/2", 1/2", 3/4", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2", 1/2", 3/4", 1/4", 1/2"
              3/4", 1/2", 1/2", 1/2", 1/2", 1/2", 1/4", 3/4", 1/2", 1/2", 1/2", 3/4", 3/4", 1/4"
              1/2", 1/2", 3/4", 1/2", 3/4", 3/4", 1/2", 1/2", 3/4", 1/2", 1/2", 1",   1/2", 3/4"
              1/2", 1/4", 1/4", 1/4", 3/4", 1/4", 1/2", 1/4", 1/2", 1/4", 1/2", 3/4", 1/2", 1/4"
              1",   1/2", 1/2", 1/4", 1/4", 1/2", 1/2", 1/4", 1/2", 1/4", 1/2"

   433     432        38           21           6            2            0           0

TOTAL OCCURANCES THIS MILE  67
TOTAL DEFLECTION THIS MILE  34.60"
AVERAGE DEFLECTION THIS MILE  0.92"

MILEPOST 432  1/2", 1/4", 1/2", 3/4", 1/2", 1/2", 1/4", 1/4", 1/2", 1/2", 1/2", 1/2", 3/4", 1/2"
              1/2", 1/2", 1/2", 1/2", 1/2", 1/4", 1/2", 1/2", 1/2", 1/4", 1/2", 1/2", 1/4", 1/2"
```

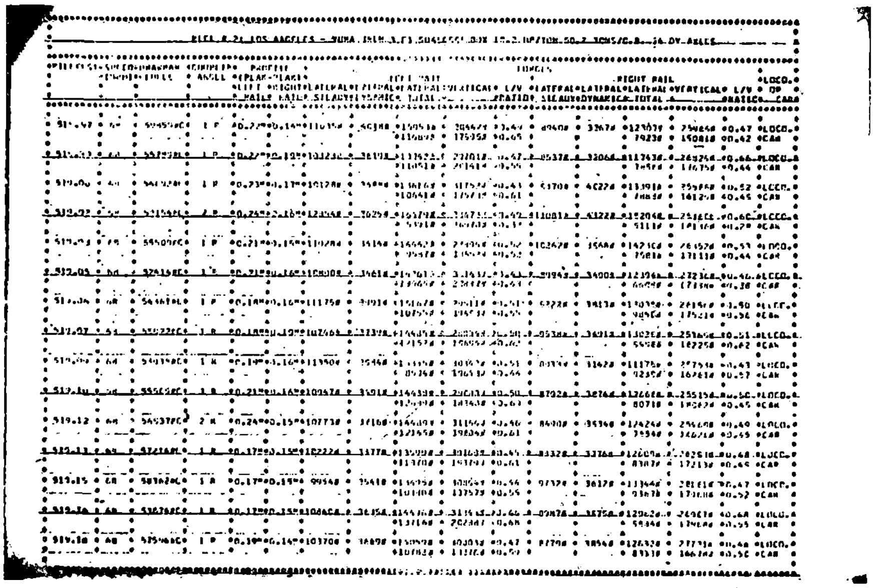
Here we developed a combination report that combines the first two reports into one. This reduces the total time required to produce the reports individually.

Slide 6



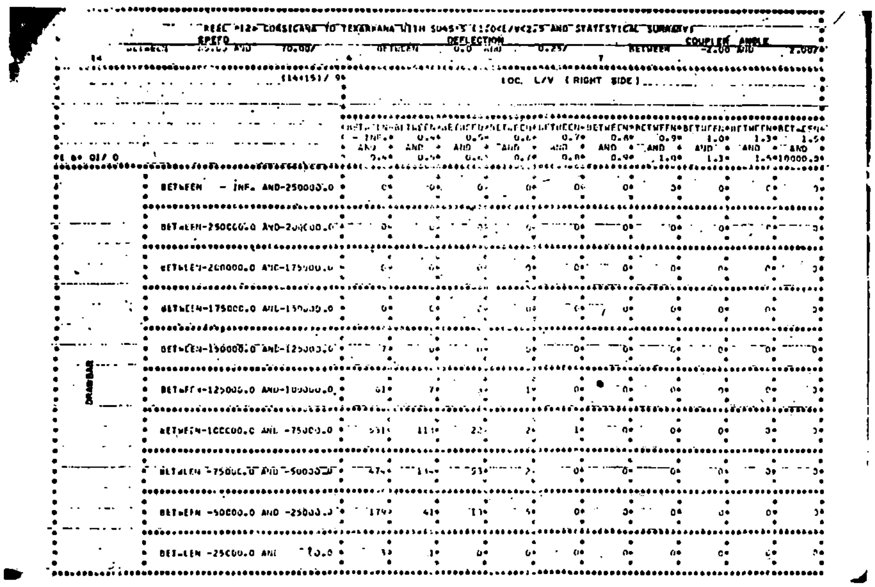

Here is probably the most useful piece of information as far as management is concerned. This is a graph that depicts the general condition of a section of track which gives you an idea of the average number of occurrences per mile within a particular area. This enables management to look at the rate of deterioration of a section of track so it can be a tremendously effective tool in assisting management to determine where our maintenance dollars should be allocated.

## Slide 7



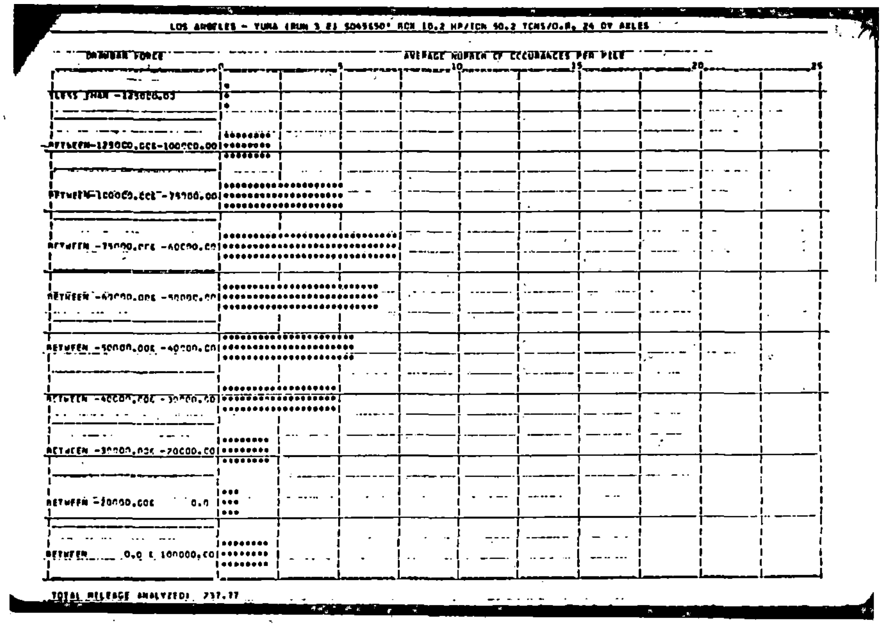
Here we have a detailed report that shows everything that occurred during a particular segment of the test; the milepost where the sample of interest had occurred; recorded the speed of the train; the drawbar forces that were actually measured between the locomotive and car; the coupler angle (between the same cars); the dynamic profile during that interval of time; the steady-state forces on the locomotive; the dynamic forces for the locomotive; the total lateral force; total vertical force and the L/V ratio for that side of the locomotive. The same type of information is given for the other side of the locomotive. Right below this you will see we have the same information for the car. This report indicates what is happening at any given point in time between a car and a locomotive.

Slide 8



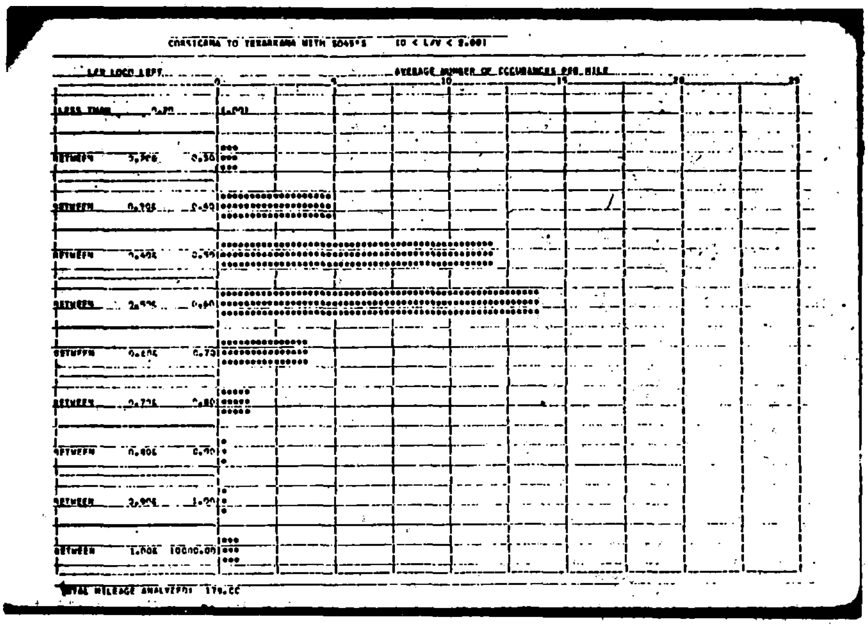

This is the fifth dimensional matrix that I had discussed earlier. We can take any five variables that have been measured during the test. In this slide we depict drawbar and L/V ratio for the right side of the locomotive. By analyzing these two variables along with speed between 65 and 70 deflections between zero and 1/4 inch and coupler angle between two degrees to the left and two degrees to the right; and are actually trying to group these occurrences in order to find out what type of correlation exists between these variables.

Slide 9



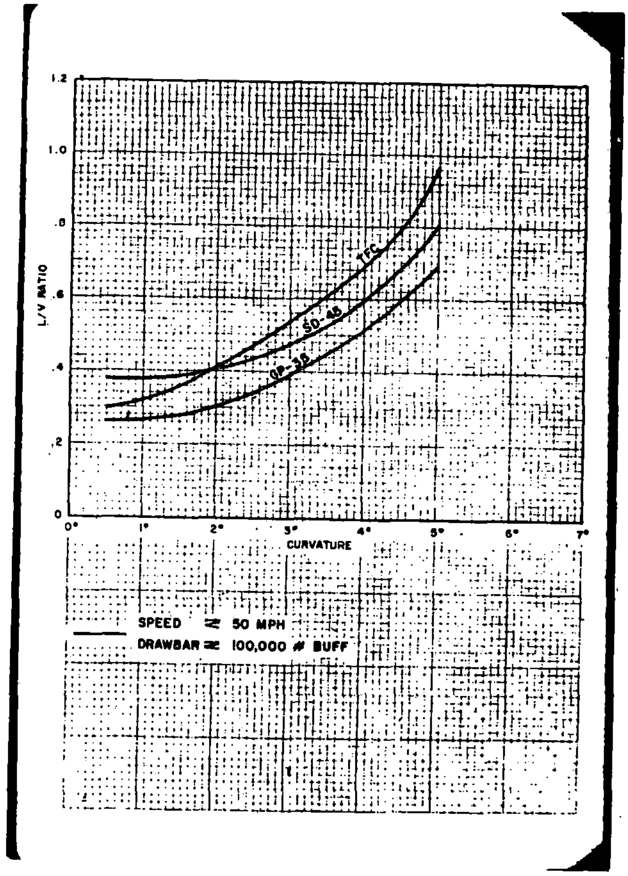

These graphs have been produced directly by the computer. We are looking at the level of drawbar forces encountered over this territory. In this particular run you can see that we are operating in the range of 60,000 to 75,000 lbs. most of the time.

<u>Slide 10</u>



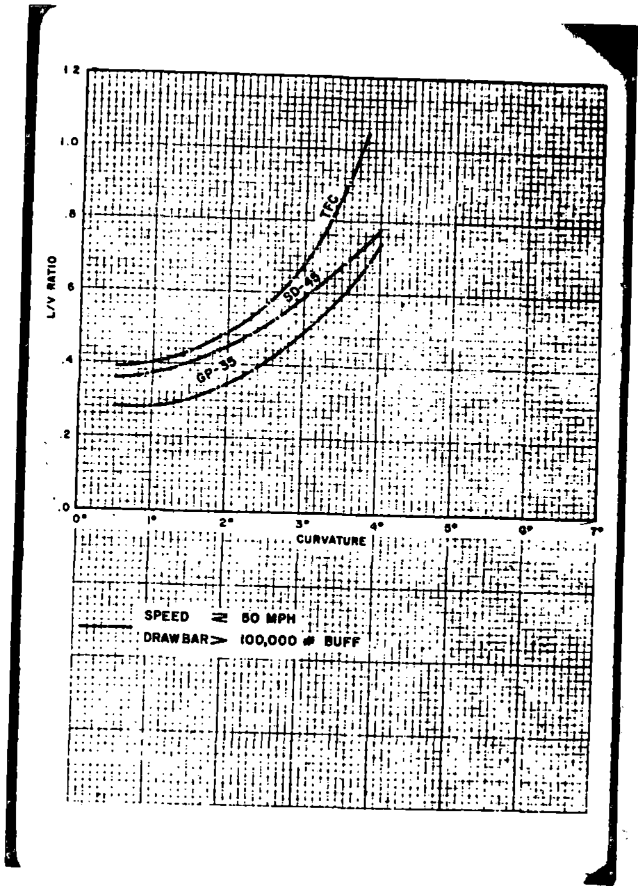
This is a similar type of report for the locomotive. During this run we were operating most of the time between .5 and .6 L/V ratios.

Slide 11



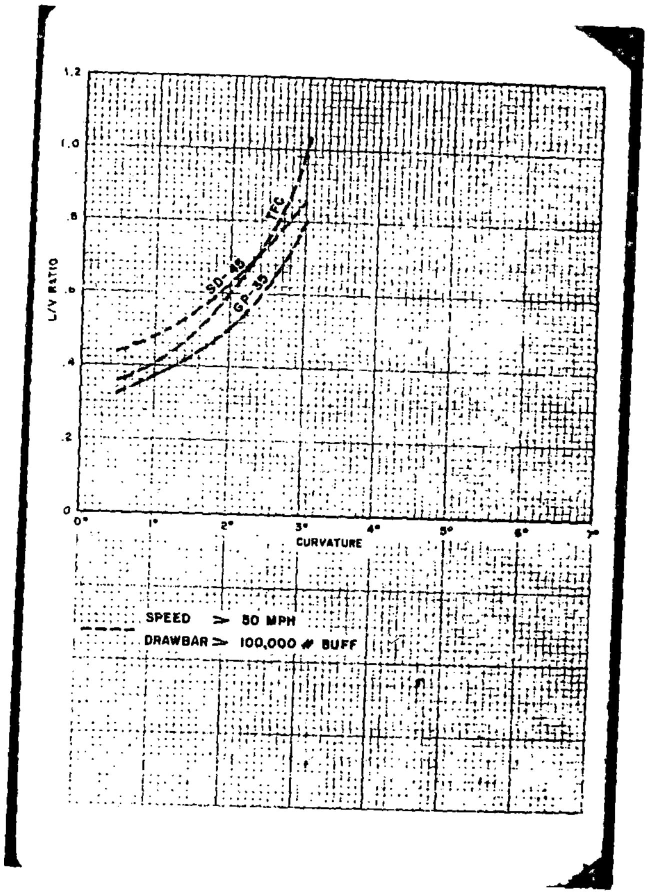

On the first set of curves we show speed equal to or less than 50 MPH and drawbar equal to or less than 100, 000 lbs. We are evaluating the L/V ratio versus curvature. The GP-35 looks pretty good which is what we anticipated. The SD-45 has somewhat higher steady stay forces, which also looks pretty good. The T FC car behind the locomotive looks pretty high.

Slide 12



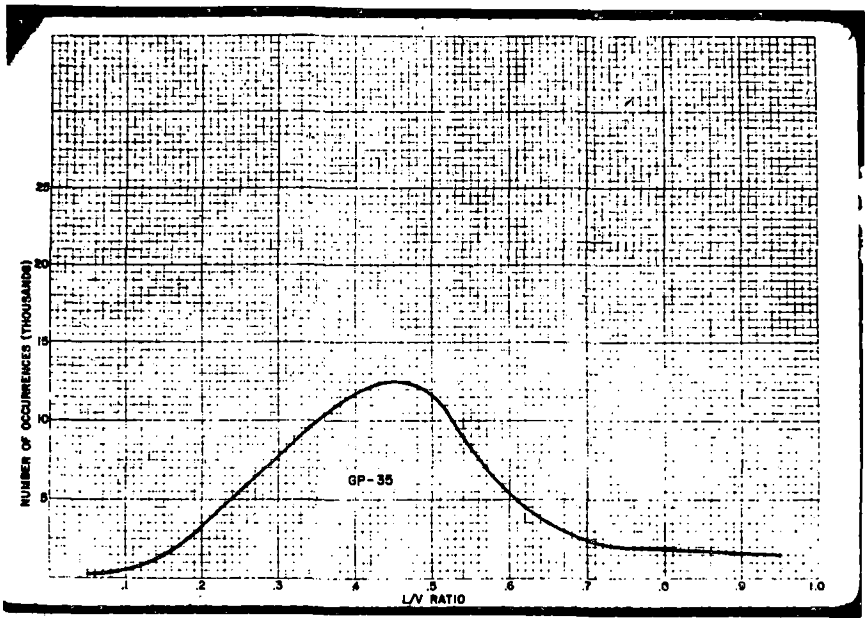

You can now begin to see the effects of speed and drawbar. These are the two most important factors effecting slack action in trains. Here we have speed less than 50 MPH but we have drawbar forces greater than 100,000 lbs. so you can see how these curves are beginning to shift to the left but none as rapidly as the TFC curve.

Slide 13



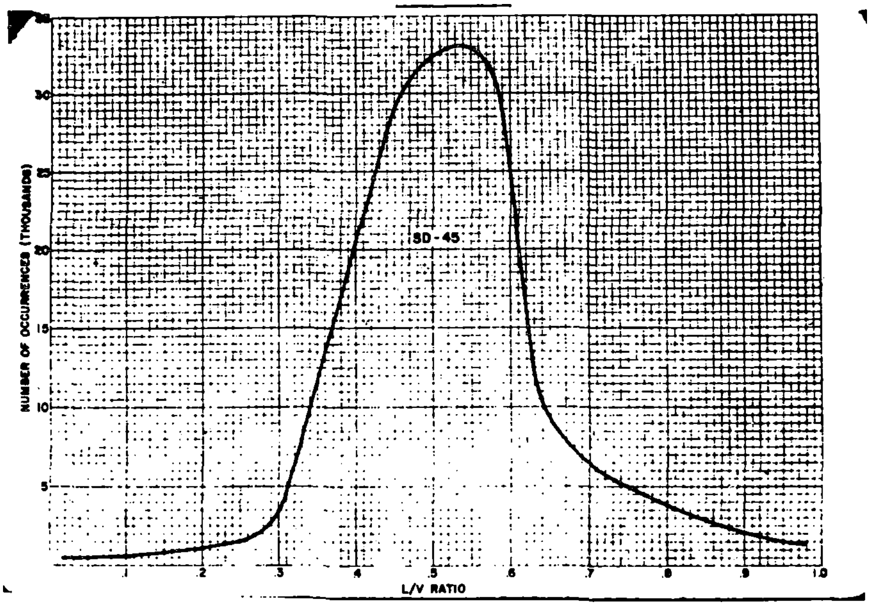

Here we have speeds in excess of 50 MPH with drawbar forces in excess of 100,000 lbs. and these curves, especially the TFC curve, are showing extremely high values which I might classify as alarming.

Slide 14



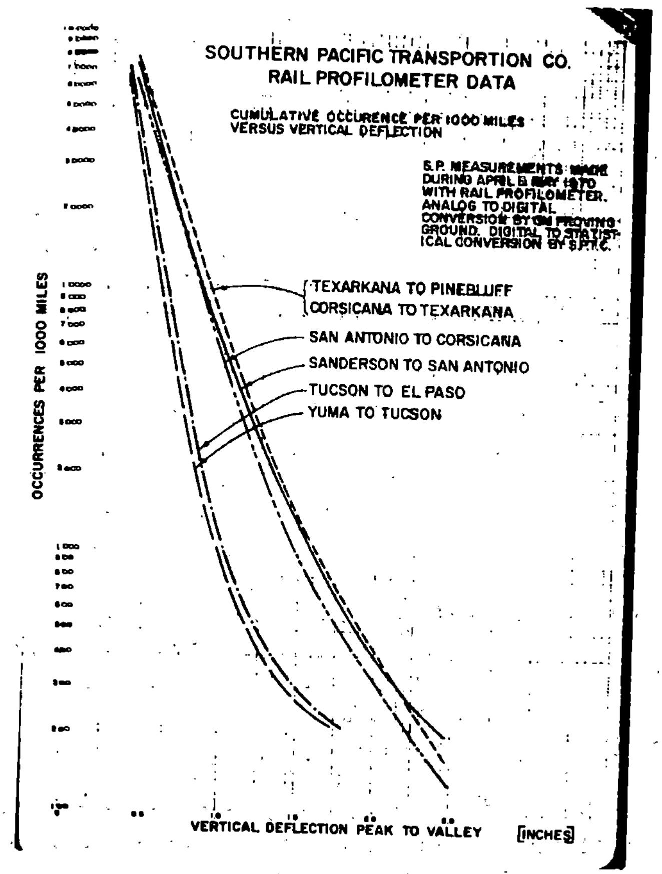

This shows the operating ranges of the L/V ratios for the GP-35.

Slide 15




This one shows the same information for the SD-45. We have more data on the SD-45 causing the amplitude of curve to appear higher, but you can see that the peak value has shifted to a value of .54.

Slide 16




Here is information relevant to our track. We took our profilometer data and actually made an analysis for each division. We used a curve fitting technique to show the relative strengths and weaknesses of each division for budgetary considerations.

We were able to use these techniques to derive the following meaningful conclusions:

1. L/V ratios for the SD-45 and the GP-35 were comparable in many respects although the SD-45 consistently had higher steady state forces on light curves (0-3 degrees). We attribute this phenomenon to the basic design characteristics of the SD truck.

2. The GP truck operating in heavy curve territories under high buff conditions in many cases shows higher dynamic lateral forces than the SD truck. A higher degree of hunting in the two axle truck is the major factor attributing to this difference.

3. The normal distribution curves show the steady state lateral forces of the SD-45 to be 10 - 15% higher than that of the GP-35.

4. SD truck has much better unloading characteristics than the GP-35 truck. This, in some cases, accounts for the better L/V ratios of the SD-45, since it is able to maintain a more evenly distributed vertical load at each wheel.

5. The two most significant factors present in very high L/V conditions are rate of change of drawbar (slack action within the train) and speed. This is predicated entirely on train handling within a given territory.

6. Vertical deflections in the rail between 0-3/4 inch has very little effect on high L/V ratios. Change in cross level or differential deflections of the rails is by far the most significant factor in track conditions attributing to high L/V ratios.

7. Vertical deflections of greater than 1 inch are very significant because the primary suspension system of the truck is unable to absorb and dampen these vertical accelerations causing vertical unloading of the truck. Softer spring systems such as the new high adhesion truck on the SD-45X could represent a significant improvement in this area. It is designed to dampen a greater percentage of the vertical accelerations at the axle without transmitting them through the truck.

8. L/V ratios on the TFC car were extremely critical in many cases. This is attributable to the resolution of the longitudinal forces into their respective lateral components. These forces were primarily caused by high buff conditions transmitted through a long-short coupler combination. This type of connection greatly increases the effects of coupler angle.

9. Much more work is required to determine how relative slack action between cars within a train affects the rate of change of the longitudinal and lateral accelerations vectors to cause critical L/V conditions to occur.

10. Rail and flange wear is primarily attributable to high lateral steady state forces and not L/V ratios. Therefore, from this standpoint the GP-35 is less severe on our tracks and equipment. Our existing policies concerning power will continue to increase our maintenance cost because of the heavy concentration of the SD-45 type of power on the point or in helper service. This means higher lateral forces sustained for a longer time period causing greater wear in the wheel-rail surfaces.

11. The TFC car showed the greatest potential of being a prime candidate in causing future derailments. Serious consideration should be given to limiting the location of this type of equipment in trains. This is especially true of light TFC cars on the front end of heavily powered trains in territories where the dynamic braking is used extensively.

12. Dynamic braking is very effective in controlling speed and does not cause excessively high lateral forces if present instructions concerning its use are observed.

13. Poorly maintained draft gears on high horsepower locomotives can be a very significant factor in producing high L/V ratios. This condition is directly related to the amount of brake force available within the extended range of the dynamic brake curve on these units.

14. Short reversing tangents in high speed territories greatly increase the L/V ratios. This is particularly true of long cars such as the TFC car.

By using numerical analysis and computer techniques, we were able to gain a better understanding of the relationships between the various measured variables. In many cases we found that when operating conditions were controlled within similar territories (grades, curves, reversing tangents, etc.) the reactions were significantly repetitive which is very important for the following reasons:

1. Probability of occurrences can be predicted under certain limited conditions.

2. This type of data can be collected and used in the development of a more accurate simulation model.

3. We can use this limited knowledge as a starting point to investigate more complex relationships.

4. This knowledge can be used in the future for car and track designs to reduce the possibility of derailments.

5. We can also use this knowledge to control train operations within questionable areas.

It is also clear that once a power-spectral analysis of rail displacements are obtained, it will be possible to predict the probability of some rail displacements occurring over a given stretch of track. A very severe limitation of this statistical technique, however, is that the lateral deflections were not measured simultaneously to obtain the combined effects of vertical and lateral displacements in the rail.

In evaluating this data, we were unable to capture any information concerning track stresses and probable damage due to high dynamic loads. This phenomenon should be incorporated in future studies.

We are now involved in the development of a computer model that would simulate the motion of two adjacent cars and a section of the track immediately beneath them using a finite time approach. In addition, this model would have the capability of handling track characteristics such as profile, speed, alignment, etc., as well as operating characteristics to calculate dynamic reactions.

The computer model would involve the combination of several idealized models of the real world. They are:

1. A train simulation model, such as program developed for our locomotive simulator.

2. A car-truck model. This would be a further sophistication of our present program to correlate such factors as car body roll, drawbar force, coupler angle, etc.

3. The truck-hunting model. This would incorporate the differential equation for truck motion developed by B. S. Cain in Dynamics of Rail and Road Vehicles. This equation relates such factors as truck configuration, truck rotation, creep and speed to derive an equation of motion. It holds true only so long as the wheel flanges do not experience impact with the rail. Such an impact

would require restarting this model with new initial conditions.

4. A modified beam model on elastic foundation to idealize the lateral deflection of the rail. This would be necessary to analyze the rail-wheel interaction at time of flange impact. It would have to be modified to handle in-plastic movement of the track structure whenever lateral force levels are excessive. It would also have to be modified to incorporate an energy equation instead of a force equation at the time of impact. This model would necessitate the most extensive field testing in order to obtain realistic constants for an idealized model.

After evaluating current state of knowledge, one must say that the development of modern computers at last enables the dynamics of trains and track to be studied in some detail, a feat before impractical because of the complexities of both vehicle and track dynamic systems. Solutions to the track-train dynamic interaction problems will:

1. Result in better time reliability for movement of freight traffic by rail.

2. Improve the safety of train operations.

3. Significantly reduce operating and maintenance costs.

It is hoped that Southern Pacific's proposal to develop a national research program on track-train dynamics for the AAR will effect a better understanding of this phenomena in order that solutions to these problems can be developed, evaluated and implemented as rapidly and as effectively as possible. Our running start does not make the job any easier. The kind of excellence we should be striving to achieve is neither easily obtained nor once obtained, easily maintained. I would hope that the AAR would want to continue their work in the area of track-train dynamics because they value the services it can perform for the industry. Perhaps in the final analysis this resurgence in the railroading science will change forever the image and profitability picture of a very essential segment of our transportation system. I would at this time like to give credit where credit's due and we have three gentlemen in our audience who were responsible for the success of this program. They are: Mr. Spencer, our Vice-President of Operations; Mr. King, our General Manager; and Mr. Williamson, our Chief Engineer.

On behalf of Harry, Howard and myself, I certainly enjoyed coming here and having the opportunity to present this paper at the conference.

Question and Answer Session:

PARTICIPANT:

(Bob Byrne, AAR) Ed, I didn't hear you mention the alignment control feature on existing high horsepower locomotives. I believe this was involved in your study. Would you comment on it's use and what are the requirements for maintaining this device to achieve maximum benefits in the area of train-track dynamics?

MR. LIND:

Yes, Bob; the alignment control was one of our first concerns, since we felt that that if it was not maintained properly, it could be a source of many problems. The standard locomotive coupler draft gear arrangement, has proven to be a big maintenance item on high horsepower locomotives. There have been recent improvements made in the alignment control device to obtain higher capacities and longer life.

The alignment control device must be maintained so that the maximum swing does not exceed 8 inch; if it goes beyond these limits, then you're going to create a jackknifing situation which can produce very high lateral forces on the rails. We have measured units with as much as 14 to 16 inches of uncontrolled lateral swing movement. Because of these conditions we initiated a very rigid maintenance inspection program for the alignment control devices on our high horsepower locomotives.

PARTICIPANT:

(Henry Marta from Electro Motive) Ed, I wonder if you would comment about one conclusion of your test and that was you indicated that the dynamic lateral loads that were measured on GP trucks were higher than SD trucks in curve territories. I wonder if these were actually measured on curved or tangent track and were they measured in low or high speed territories.

MR. LIND:

These valves were measured on both curve and tangent track and we attribute these results to the hunting characteristics of the truck. In future tests, we're going to have to obtain a better understanding of critical time duration necessary to cause a derailment at variable speeds. These tests were run primarily in high speed territories.

PARTICIPANT:

Were these curves considered sharp?

MR. LIND:

I would say basically they were light, curves ranging from $0^{o}30'$ to $3^{o}0'$ degrees.

PARTICIPANT:

(Jake Anders of North Western Railway) My question is directed to Mr. Nation, District Road Foreman. Did you develop any new techniques in handling slack action on trains ranging from 100 to 200 cars?

MR. NATIONS:

I understand you to say, did we develop new train handling techniques? Is that right?

PARTICIPANT:

Yes.

MR. NATIONS:

No, we did not develop new techniques, we just wanted to impress on the engineers the importance of planning ahead in order to control slack action.

PARTICIPANT:

Does the Southern Pacific advocate a variable operating speed of 10 - 15 MPH within the same territory, or do they have a set maximum speed limit?

MR. NATIONS:

We have a fixed maximum speed.

PARTICIPANT:

(Mr. Caldwell from Canadian National) I have four questions to put to you. In your initial studies of derailments, were there any occurrences of wide gauge on track considered as a contributing factor to the derailments? Second question is -- we are concerned with L/V ratios as a means of determining derailment criteria; I'm wondering if you have any feelings whether this is the only criteria or should L/V ratio in conjunction with the given magnitude of lateral load be considered?

MR. LIND:

In response to your first question -- yes, in fact we ran some tests in Arizona where we actually introduced wide gauge.

PARTICIPANT:

No, I'm wondering if gauge was widening under the normal operations of trains?

MR. LIND:

No, we were not measuring gauge and we were not measuring the lateral deformation of the track. There were no field reports from the engineering people indicating they were having trouble maintaining gauge. Excuse me. I thought you meant in the test itself. We have certain areas where we have to look at gauge problems very closely. We have similar conditions. We have observed where you have dynamically wide gauge due to excess tieplate cutting on the high rail or physical deformation of the spikes by bending them into an "S" shape and we attribute the cause of this to be excessive lateral loads applied to the track structure.

MR. LIND:

In answer to your second question, the L/V ratio is just one criteria. We have looked at many other things occuring simultaneously but I do think this is the only important criteria. I certainly think very high lateral force (neglecting the L/V ratio) is a very significant part of this whole problem.

PARTICIPANT:

Do you think your curves for the TFC cars show the L/V ratio to be higher for the TFC car compared to the locomotive, but the net lateral force is probably lower on the TFC car than that of the locomotive?

MR. WILLIAMSON:

Actually we're having some problems with gauge like all railroads are. Ed was not involved in this phase of the problem before the derailment study. This is one of the things that got us involved in the L/V studies. We were concerned about what heavy power was doing to our track since we had to do more gaging, even though we were using the heavier 18 inch plates. We thought that we had the gauge problem licked with 14 inch AREA plates. In our test we deliberately put wide gauge in the track to see what effects the increased attack angle of the wheel would have on the high rail and of course our tests proved exactly what's been proven in theory and that is you get a much higher L/V ratio when you widen the gauge. I would say that the gauge in the cases of our unknown derailments, didn't exceed 1/2 inch which I wouldn't consider alarming unless it was in curved territory. So gauge was not a major factor. I think the biggest thing we've found, and the one thing we corrected immediately is the very short reversing tangents on flat curves in high speed territory, because the L/V's you generate in this condition are very high. We have restricted speeds to 40 MPH in those areas where we don't have at least 200 ft. of reversing tangent between 2-3 degrees reversing curves. We will make line changes to increase the speed but as an interim measure we have put these restrictions into effect. I'm confident that we have averted some very serious derailments because of these actions.

PARTICIPANT:

My third question -- have you considered taking any measurements to define the threshold levels of force which cause track structure to fail. I know we observed yesterday that the SD trucks under steady curving conditions do apply more lateral force to the track. It's assumed that this lateral force can be absorbed by the track but I haven't seen anything to say where track fails, is there any feeling on that?

MR. LIND:

Yes, in certain areas where we had derailments previously and we ran our test over these areas repetitively; we had field crews check gauge and almost in all cases, there was no gauge widening at all. So the levels we are now experiencing are lower than the failure threshold of the track, apparently. We are concerned about a decrease in our vertical load which tends to increase the L/V ratio.

PARTICIPANT:

Is the nature of this type of track structure heavy steel and big tie plates, etc.

MR. LIND:

Yes.

PARTICIPANT:

O.K., the fourth question -- you mentioned the hunting of the locomotive -- is this hunting the dynamic instability of the locomotive truck?

MR. LIND:

Yes, we find the GP truck to be much more unstable on tangent track at high speeds and on light curves than the SD truck.

PARTICIPANT:

Is there an observed speed at which the truck goes unstable ?

MR. LIND:

As far as resonant speed we haven't been able to observe a critical speed. I feel much depends on the relative velocity of the cars immediately behind the locomotive.

PARTICIPANT:

I see, thank you.

PARTICIPANT:

I'd like to comment on that last question. The speed at which you get into resonant hunting conditions of a truck depends on the number of axles in the wheel base and with our four wheel domestic truck this is approximately 75 MPH with the standard table wheel contour. For that reason we have historically recommended that for higher speed operation that cylindrical wheels be considered.

PARTICIPANT:

Do hollow worn wheels on the locomotive affect resonant hunting conditions?

MR. LIND:

Well, the effect of the hollow worn thread magnify the amount of differential you can get with a fixed shift on the rail.

PARTICIPANT:

Does it lower the speed of hunting?

MR. LIND:

Actually with a well maintained tapered wheel thread and normal gauge to rail lateral clearances I'd say it's rare that we get into hunting -- even at the 75 MPH. 75 MPH is where our experiences indicated with worn wheel contours, we get into problems.

**L. A. Peterson,** Director of Applied Research, Bessemer and Lake Erie Railroad

Can Vehicle/Track interaction measurement help in meeting the challenge of a changing railroad environment? Changes in the railroad environment can cause problems in the future as they have in the past. I've been working for a couple of railroads which have experienced the effects of dynamic car-rail responses in a first-hand way. The pictures in Figures 1 and 2 offer proof that there are conditions under which the dynamics of a steel wheel running on a steel rail can cause problems to the track and to the car (Figure 3). There is alarming evidence to indicate that the problems now being incurred are new ones, emanating from changes in operation, car designs and track factors.

**Figure 1. Corrugated Rail on Curve**

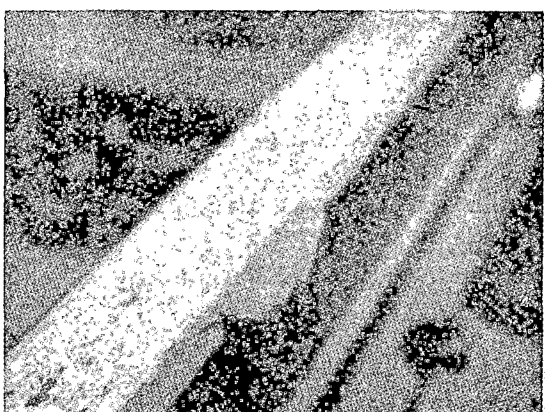


**Figure 2. Deteriorated Low Rail On Curve**

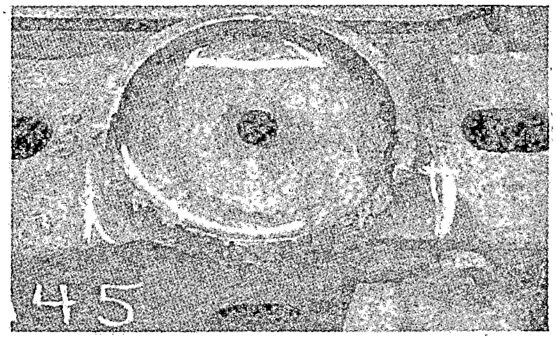




**Figure 3. Failed Truck Bolster Center Plate**

Of course, car associated problems have not escaped attention. Through the efforts of many companies, including those represented here today, there are a host of remedial type devices available for achieving improvement in particular areas of car performance. But, which device or combination of devices is best for a given set of track and operating conditions? Which of the many possible variations in car design will be most effective? *And* most economically sound? What are the dangers of improving a particular characteristic at the expense of creating detrimental effects in another place? Enlightened answers to such questions are difficult because there are staggering degrees of freedom present and it appears that an infinite number of combinations of car suspension systems and track and operating condition combinations are either available or possible.

As a more specific example, consider the modification in Figure 4.

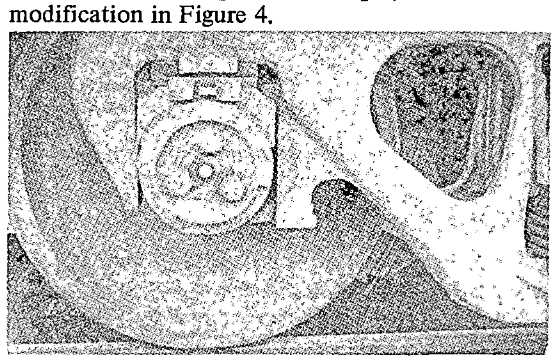


**Figure 4. Lateral Pad Installation on Roller Bearing Car**

The performance of the lateral pad pictured will normally be evaluated through a period of in-service usage. Such a process usually takes a long time and often is not conclusive – for a number of reasons, including difficulties in controlling or accounting for the influencing elements. In addition, decisions concerning the worth of the device

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seem to be all too often, predicated upon wearing and maintenance qualities of the device itself rather than what it does in influencing the dynamics of the wheel-rail system. There is a serious economic error in justifying a new component *solely* on the basis of price and how long it itself will last. Perhaps a new component's contribution to the performance of the rail-car transport system could justify a 100% (or more) increase in component purchase price – but it would be extremely foolish to pay the premium price without some factual evidence of benefit and favorable investment.

In this example, installation of the lateral pads of Figure 4 on roller bearings of cars only increases the initial cost of the car. However, it's conceivable that this action could result in savings in rail, wheel and other car component maintenance costs and possibly even decrease the probability of derailments. On the other hand, in another instance, a device might control "rock" at the expense of causing poorer curving properties; or a particular truck incorporating roller bearings may achieve success in reducing hot boxes while increasing lateral force levels.

In order to objectively assist in answering the questions which have been raised, I believe that a scientific and factual means to measure and evaluate car performance and system reactions as the car moves over representative track is needed. While I applaud the mathematical modeling which has been accomplished and I commend Mr. Cappel and Wyle Laboratories for their impressive simulation efforts, I must still caution that mathematics, simulators and computers alone can only go so far. They need facts to arm them and field data to support them so they can be properly tuned in and verified over a wide range of variable values.

I believe that the basic element in evaluating changes in car design, track structure or operating methods is more complete knowledge of the resulting dynamic reaction at the wheel-rail interface. The most sophisticated devices installed on cars are either aimed at improving this relationship or will nevertheless effect it.

Figure 5 is a simplified representation of the type of forces present at the wheel-rail interface.

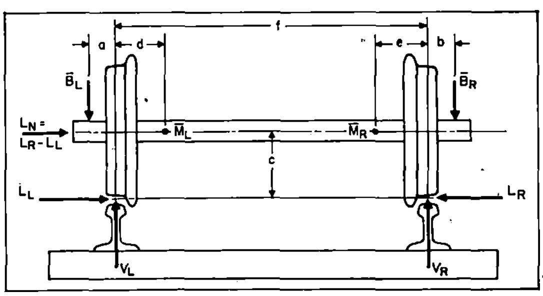


**Figure 5. Diagram of Forces Acting On An Axle-Wheel Unit**

Primarily, all that is required is to measure the lateral forces $L_L$ and $L_R$ and the vertical forces $V_L$ and $V_R$ between the wheel and the rail. For static conditions, measurement is easy, but when the wheel moves on the rail, it no longer is a trivial task – even with space age technology. To further complicate matters the technical routine of collecting and analyzing continuous measurements is itself fraught with all kinds of traps and pitfalls.

We on the Bessemer and Lake Erie Railroad, in partnership with the Quebec Cartier Mining Company have been engaged in attempting to solve such problems since late 1967. By the method which we've developed, the bending moments $M_L$ and $M_R$ and the bearing loads $B_L$ and $B_R$ are measured directly.

The desired lateral forces $L_L$, $L_R$, and vertical forces $V_L$ and $V_R$ and the net axle lateral $L_N$ are determined by solving a set of simultaneous equations. By our defined conventions, the dynamic quality $B_R - B_L$ is called "rock" and $B_R + B_L$ is designated "*bounce*". Continuous determinations of such quantities, say at the rate of 500 per second of elapsed time, obviously requires computer assistance. Figure 6 is a picture of a typical instrumented axle showing the strain gages which are calibrated to measure bending moments and the torque bridge in the middle. Leads from these gages are brought out to the end of the axle to terminate in the slip ring assembly of Figure 7. Bearing loads are measured through installation of load cells (Figure 8) on the bearing adaptors. Figures 9 and 10 are typical load cell installations, on the inner axle of a two axle truck and on each of the axles of a 3 axle truck respectively.

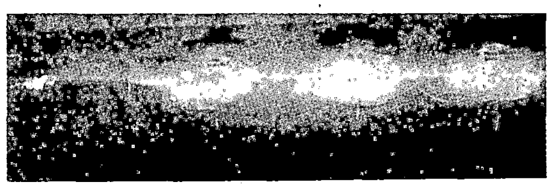

**Figure 6. Typical Strain Gaged Axle**

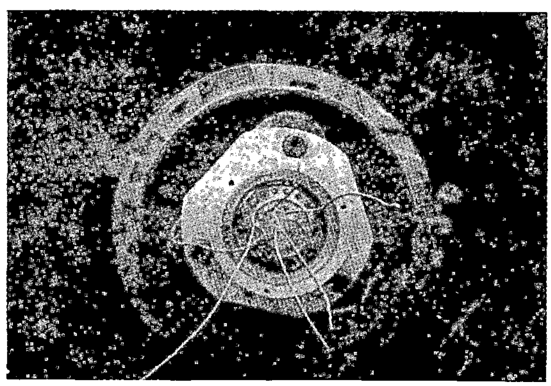

**Figure 7. Slip Ring Assembly At End Of Roller Bearing Axle**

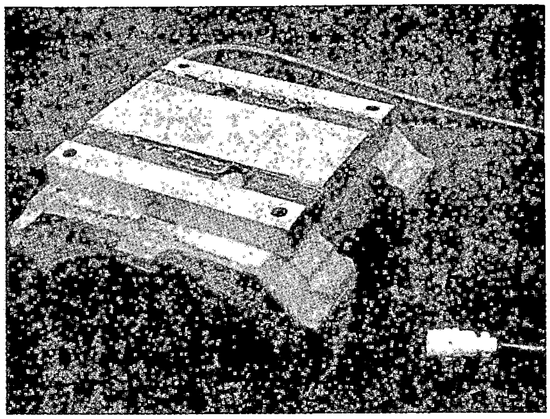

**Figure 8. Load Cell Mounted On Adaptor**

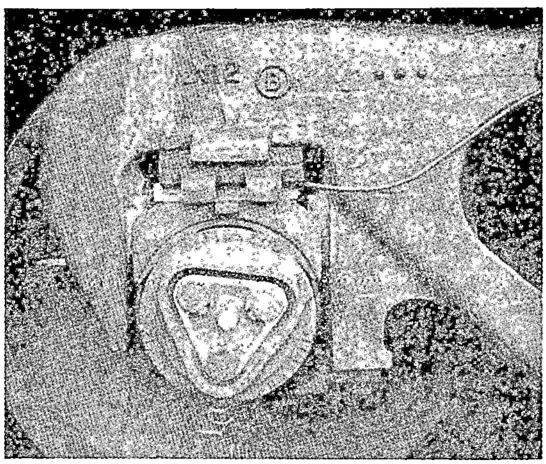

**Figure 9. Load Cell Installation On Lateral Pad Equipped Car**

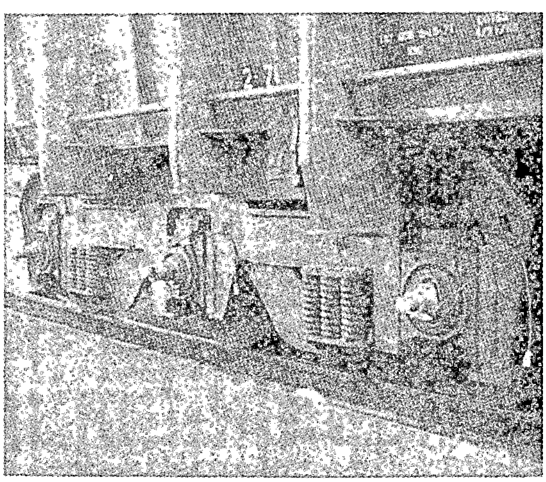

**Figure 10. Slip Ring & Load Cell Installation On Axles of 3 Axle Truck**

We have made a major commitment, not only to obtain meaningful dynamic measurements but also to provide the machinery to adequately analyze them. For this purpose, B&LE car number A283, shown in Figure 11, came into being.

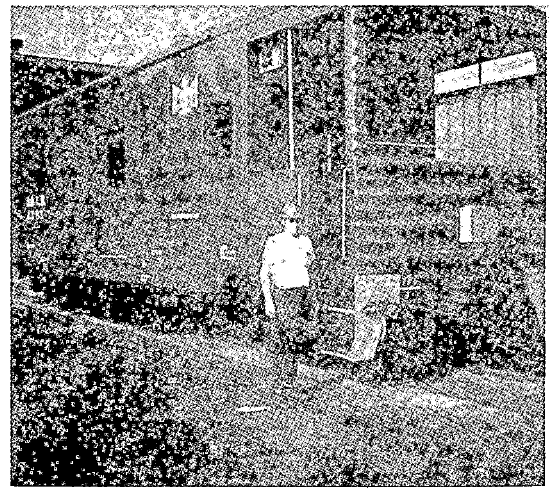

**Figure 11. Exterior View Of Research & Testing Car**

It was converted from a 40′ box car to a research and testing car early in 1971. The physical car structure was modified to include a six foot observation platform, a 48″ side access door, suitable windows, toilet facilities, dual A.C. power generators providing 17.5 KW and driven by two banks of propane fuel tanks, heating and air conditioning systems, overhead fluorescent lighting and various housings and wiring boards to handle data cables. In addition, the car is equipped with an instrumented draw bar and is weighted to a nominal weight of 50 tons.

Presently the car is used to receive and process signals from instrumented axles and/or other sensors applied to any particular car for which dynamic measurements are desired. Up to 28 channels of simultaneous data can be received, recorded, displayed and processed continually, but 14 channels have usually proved adequate. Through the "patent applied for" procedure previously mentioned, signals representing the bending moments in the axles and the dynamic loads on the bearings are sent to the heart of the system, an on-board Hewlett Packard 2116C mini-computer, (Figure 12) where the appropriate equations are solved to yield continuous dynamic determinations of the lateral and vertical wheel-rail forces as the car traverses the track. Various other quantities such as axle torque, bolster and wheel stresses, draw bar strain, etc., are also measured and can be correlated with the wheel-rail dynamics.

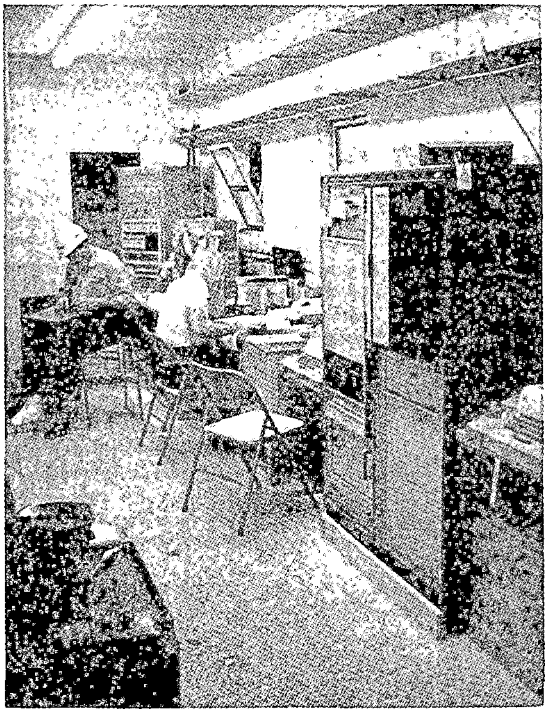

**Figure 12. Interior View Of Research & Test Car With Hewlett Packard Mini-Computer In Foreground**

Both analog and digital recording capabilites are provided. This allows maximum flexibility, back-up and verification. Special computer programs are used to convert from analog to digital, to determine continuous and instantaneous lateral and vertical wheel-rail forces, to produce combined relationships such as L/V ratios and to analyze the collected data. Hard copy printouts and CRT displays are available in real time.

This equipment is being used in conjunction with controlled tests during which representative track sections are selected for test runs and kept constant during the duration of the tests. Exact positions on curves and tangent are recorded through plates placed between the rails and located to produce a code which is picked up either by the photo-electric cell system of Figure 13 or by a metal detection type coil.

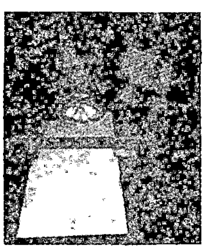

**Figure 13. Photo-Electric Cell Unit And Location Plate**

One of the objects of the tests already performed was to quantitatively appraise the difference in several types of car designs. Car designs which have been tested at speeds ranging from 15 to 50 MPH over both welded and jointed track sections with curvatures of up to 10°, include:

1. Plain bearing cars with various snubbing and springing systems.
2. Roller bearing cars with and without lateral pads.
3. Cars with lateral pads allowing various amounts of lateral and longitudinal deflection.
4. Cars with butyl and natural rubber lateral pads.
5. Cars with different types of hydraulic snubbing.
6. The third bearing or "Free-Wheeling" car which Mr. Robertson of the QCM will report on tomorrow.
7. The above "Free-Wheeling" car with lateral pads.
8. Cars with different types of three axle trucks including a prototype three point suspension model.
9. Cars with various types of center plates, including the conical type.
10. Cars with different types of side bearings.

You can easily envision the enormous amount of data which has been generated and properly conclude that we will be here all night if I attempt to cover the findings in any kind of detail. Accordingly, my subsequent remarks will be limited to pertinent general observations relating to the capability and potential of the methodology.

In this respect, the results are very encouraging from many viewpoints. They indicate that differences between car designs can be detected and quantified. As an initial step computer-produced graphs similar to Figure 14 can be displayed. Review of such charts, which you will note *are not the ordinary type strip charts,* can be very enlightening and is intuitively rewarding and invaluable in setting up analysis procedures; but there are more powerful ways of comparing car performance from measured data. The forces occurring on a given curve or other selected track section can be computer sorted into the frequency of occurrence in given force intervals and cumulative frequency charts can be prepared for more graphic comparisons. Figure 15 shows how the various resulting wheel-rail forces on the first axle of three types of cars compare on a 5° welded rail curve at 35 MPH. How do the forces vary by car design with track curvature? Graphs such as shown in Figure 16 for the net axle lateral force illustrate how meaningful comparisons can be made in this respect. Similar types of charts will depict other quantities such as L/V ratios, bounce, rock, or individual wheel-rail lateral forces.

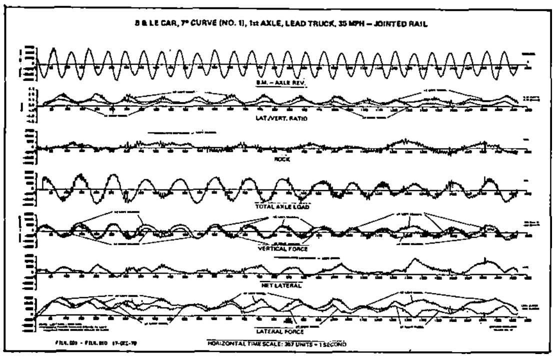


**Figure 14. Typical Computer Output Graphs For Various Forces**

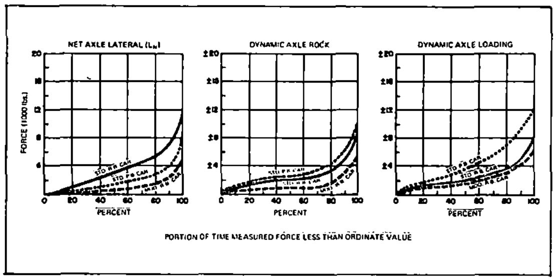


**Figure 15. Comparison Of 3 Different Types For The Forces Indicated**

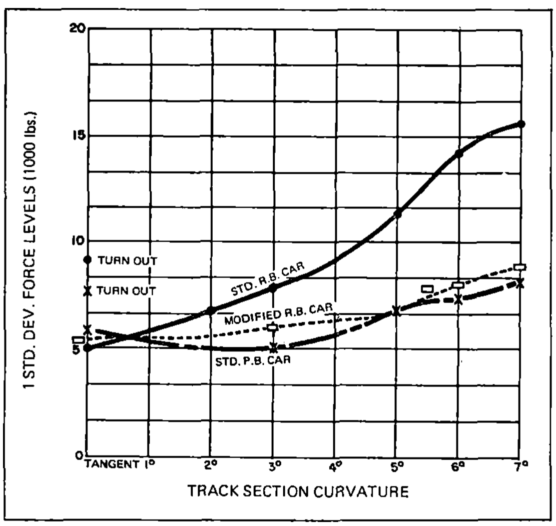


**Figure 16. One Standard Deviation Level Of The Net Lateral Axle Forces On Jointed Rail**

Furthermore, such things as the merits of welded rail can be more factually verified when comparing the performance of a selected car design, say the plain bearing car on welded vs. jointed rail (Figure 17).

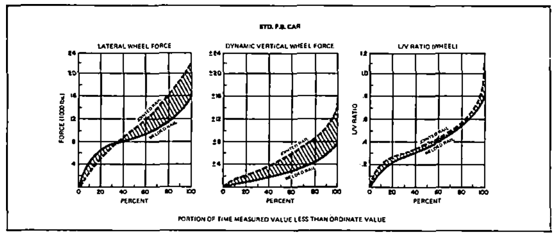


**Figure 17. Comparison Of Jointed Vs. Welded Rail For The Forces Indicated**

Other track parameters including the adequacy or inadequacy of super-elevation for various car types at different speeds can also be appraised through examination of load shifts as in Figure 18.

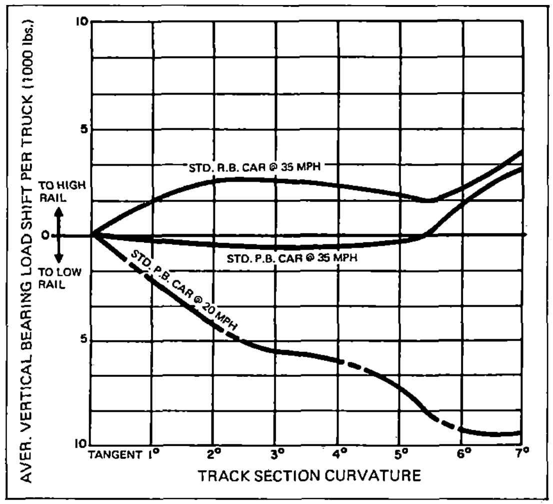


**Figure 18. Illustration Of The Effect Of Track Super-elevation At 2 Different Speeds**

I could go on and on talking about results to date with specific examples and comparisons, but as already stated there is insufficient time to cover this subject in any kind of comprehensive way. The figures referred to are merely intended to illustrate the capability of the process in more concrete terms. For those of you that are interested, I refer you to ASME paper 71-WA/RT-4 which covers this subject in somewhat more detail.

In closing, I believe that the techniques similar to those briefly described (with inevitable refinements) can, in terms of the vehicle/track interface, accomplish the following:

1. Factually evaluate the effect of changes in car design, such as in snubbing, springing, loading, lateral tolerance, bearing methods, etc., on the measured quantities so that the merits of various designs are judged scientifically and objectively under the same set of conditions.

2. Factually evaluate the effect of operating methods of the performance of various car designs, such as speed, braking, and slack action.
3. Factually evaluate the effect of track initiated input into the car being tested, such as joints, turnouts, super-elevation, alignment, etc., and allow investigation of cause-effect relationships over a range of different car designs.
4. Automatically monitor track condition by making successive measurement "runs" separated by appropriate time intervals, with an identical car over the same track sections and comparing the force frequency distributions to determine significant deviations and/or unacceptable force levels. This has been called dynamic "performance" measurement as opposed to track geometry measurement.
5. Complement existing track geometry measuring systems by quantifying the results of deviations from dimensional standards.
6. Continuously determine and study combined relationships such as L/V ratios, "rock" and "bounce", to determine their true significance and variations over a wide range of car designs and track conditions.
7. Provide valuable data to aid in car and truck design considerations and mathematical modeling and simulator verification.
8. Provide guidelines to make better combinations of track, cars, and operations possible at economical cost levels.
9. Stimulate an expansion of research and development activites on the part of suppliers toward obtaining improved car and truck designs; the incentive being the possibility of quicker investment recovery and increased sales leverage due to prompt and objective evaluation of the merit of new designs.

I'm sure that we are not the only ones working in the evaluation areas I've covered. I'm not sure about the others, but I for one am very encouraged by results to date. Perhaps the likelihood that someday we will be able to more effectively cope with our changing environment is increasing.

CHAIRMAN SMITH: Thank you... Our third speaker is Mr. J. F. Sapp, Louisville and Nashville.

Association of American Railroads, Effect of Wheel Damage And Wheel Load On Extent Of Rail Damage, by R. S. Jensen, Report No. 52440, Progress Report on the Research Program of The Joint Committee on Relation Between Track and Equipment, 3140 South Federal Street, Chicago, Illinois 60616.

This is a report on a research program to study pressures as affected by the area of contact between wheel and rail. The report covers rolling load tests of rail specimens subjected to wheels of different diameters and to various loads. In addition, strain determinations are made of specimens having static loadings applied.

TDOP: 04-001

Association of American Railroads, Investigation To Determine The Cause of "Sudden" Wide Gage On The Delaware & Hudson Railroad, Report No. ER-68, October 1966, Engineering Research Division, AAR Research Center, 3140 South Federal Street, Chicago, Illinois 60616.

The Delaware and Hudson Railroad experienced a number of derailments, the cause of which could not be definitely determined. Also, there were numerous occurrences of what was termed "sudden" wide gage where the outer rail of a curve was pushed outward as much as two inches for a length of from 10 to 40 ties. These phenomena always occurred in very cold weather. They occurred on various degrees of curvature, but not on tangent track. The purpose of this investigation was to determine the cause of this "sudden" gage widening. This investigation measured the jackknifing forces developed in conjunction with the types of diesel electric locomotives used in the territory where the gage widening occurred, and the lateral forces exerted on track by an 89-foot TTX flat car coupled to a short hopper car when operated in regular revenue trains over the territory in question. The first part of

the investigation centered on a three-degree curve. A special test train was utilized which consisted of a number of diesel units for dynamic braking and 15 loaded hopper cars, including one 100-ton car, to provide the pushing resistance at one end, an 89-foot TTX flat car and short hopper in the middle, and the pushing diesel units under test at the other end. Six running tests were made in regular trains between Oneonta, New York and South Lanesboro, Pennsylvania to determine the effects of dynamic braking and slack action on the lateral forces exerted on the track. The instrumented TTX car, the adjacent hopper car, and a recording instrument car were positioned at the front, middle, and rear of the trains. The investigation did not determine any clear cut and specific causes of the "sudden" wide gage incidents. However, it appears likely that the occurrences were due to lateral forces exerted against the outer rail as a result of locomotive jackknifing under dynamic braking. Several recommendations are included which may be helpful in reducing the magnitude of the causal forces or in increasing the resistance of the track to gage widening.

TDOP: 04-002

Association of American Railroads, Lateral Forces on Track and Equipment Due to Dynamic Braking on the Southern Pacific Lines, By Schinke and Aggarwal, Technical Report ER-69, October 1966, Association of American Railroads Research Center, 3140 South Federal Street, Chicago, Illinois 60616.

This report embraces the description and analysis of data secured during the operation of regularly scheduled freight trains on the Southern Pacific Transportation Company between Roseville, California and Sparks, Nevada. The purpose of the investigation was to determine the coupler forces and resulting lateral forces exerted on the rails by the passage of an 85-ft car coupled to a short car in a train with the locomotive using dynamic braking while operating on steep grades and curves up to 10 degrees. During the investigation, data were secured on coupler and car angles, lateral and vertical truck forces, and longitudinal accelerations of the 85-ft car with the train operating at various speeds. The analysis of data contained in this report may be summarized as follows:

- The steady or longitudinal coupler forces for both pull and push conditions, as expected, are in proportion to the weight of that portion of the train behind the point of measurement. The compressive coupler forces were reduced considerably when the train air line pressure was reduced with the locomotive under dynamic braking on the descending grades.
- The coupler angles are in direct proportion to the track curvature with the values obtained under the pulling condition slightly greater than those under the pushing condition.
- The data indicate there is a linear relationship between the track curvature or coupler angle and the lateral truck forces acting on the rail.
- The occurrence of slack action was rather infrequent due to the long and almost continuous grades, but some coupler forces as large as 90,000 lb were obtained by an application of the dynamic brakes that permitted the slack to run in.
- The sprung weight of the 85-ft car was subjected to longitudinal accelerations as large as 2.2 g, but there does not appear to be any direct relationship between the direction and magnitude of this acceleration with respect to the direction and magnitude of the slack action coupler force.

RRIS: 039991

TDOP: 04-003

Association of American Railroads, Speed of Trains Through Turnouts, AAR-ER-14, August 1961, 3140 South Federal Street, Chicago, Illinois 60616.

This report discusses testing by the AAR to determine standard turnouts that would give maximum comfort at maximum speed. Field tests used no. 24 frogs with 39-ft switch points at 50 mph and no. 20 frogs with 30-ft switch points at 40 mph. It was concluded that these frog-switch speed combinations gave the maximum lateral accelerations expected in high-speed running.

RRIS: 040000

TDOP: 04-004

Association of American Railroads, Tracking Test Of A Flat Car Trailer Carrier On A 6.5 Degrees and 12 Degrees Curve, by R. Schinke and I. Gitlin, Technical Report ER-49, October 1965, Engineering Research Division, 3140 South Federal Street, Chicago, Illinois 60616.

This report is concerned with the tracking characteristics and best location in a train of a loaded trailer-carrier flat car. Such cars have derailed on sharp curves when high drawbar pull was present, resulting in the inner rail overturning or the wheel flange climbing the inner rail. Test results show that these situations can be avoided by mitigating accelerations and high drawbar pull while in curves through car placement.

RRIS: 039700
TDOP: 04-005

Blader, F. B. and E. F. Kurtz, Abbreviated Report Of Free Lateral Oscillations In Long Freight Trains, Canadian Institute of Guided Ground Transport, February 1973, Queen's University, Kingston, Ontario K7L 3N6, Canada.

This report is an abbreviated form of a full report that has been prepared describing a study of lateral vibrations in long freight trains. A mathematical model for investigating the dynamic stability of cars in long freight trains was developed that takes into account the effects of coupler forces. The model was used to investigate the stability characteristics of a specific three-container car employed by Canadian National Railways for which dynamic stability data is available, and the model was able to predict the behavior of this car accurately. Groups containing up to 16 cars were investigated, and the results indicated that the range of train velocities at which a long freight train will exhibit stable behavior can apparently be determined with satisfactory accuracy by consideration of individual cars free of coupling forces. The model includes the effects of creep and spin forces at the rail-wheel interface; spin forces being found to have an important effect on the determination of dynamic behavior. The lozenge stiffness of freight-car trucks was shown to be an important parameter with regard to dynamic stability. It is possible to define an optimum value of lozenge stiffness for which freight trucks of the type used in North America would be stable for all speeds of interest for freight trains. More importantly, this optimum value would also result in a well damped response of freight trucks to track irregularities in nominally straight track.

RRIS: 051904

TDOP: 04-006

Blader, F. B. and E. F. Kurtz, Jr., "Dynamic Stability of Cars In Long Freight Trains," ASME, 73-WA/RT-2, 345 East 47th Street, New York, N.Y. 10017.

This report describes a method for investigating the lateral response and hunting tendency of cars in long trains. A linear model is developed representative of North American freight-car design with parameters descriptive of wheel wear and truck flexibility. Included is a simple model of the couplers assumed under tension. Transfer matrices are used to examine the behavior of groups of cars. Results are given showing that the behavior of a group of such cars under tension is no less stable than that of the single uncoupled car. The effects of variations in certain parameters on the natural response are described. Truck stiffness is shown to have a primary effect on the lateral stability of the car.

TDOP: 04-007

Blaine, D. G., "Train Brake Action And Effects On Track Structure," <u>Proceedings of Conference on Track/Train Dynamics Interaction,</u> December 1971, Association of American Railroads, Research and Test Department, Chicago, Illinois, pp. 382-407.

This paper reviews the basis of the action of train braking systems and the effects of the required retardation forces on the track structure. Major points covered include:

- A train can be considered a long link chain or a loosely jointed semi-flexible column.
- Automatic air brake functions have transmission speeds which are faster than the normal slack run speeds of 200-400 fps. Any application of dynamic or independent braking at the head end tends to bunch the slack and may cause heavy slack run in.
- Train air brake action pressure and force differentials between adjacent cars are small, not over 1/2 psi and 200 lbs.
  - However, with the common single capacity brakes, retardation response will vary with load condition of the car from 2/1 to as high as approximately 5/1. This could mean 2000 - 4500 lbs force difference between an empty and a loaded car.
  - With empty load brake equipment, the retardation response would be held to a range around 2/1 to 3/1 since loaded braking ratio is higher, and empty is lower than with single capacity brakes.
  - With variable load equipment, the retardation response could be held close to 1/1 for both empty cars and for cars loaded to any degree. Equipment of this type has been in service for many years in the mass transit field but is much more complex than simple empty-load. However, advantages would be essentially uniform retardation and stopping response on any train.
- Concentrations of tractive and retarding forces are functions of weight and adhesion.
  - 200,000 lb to 250,000 lb are practical values for any locomotive consist's tractive brake, dynamic brake, or independent brake effort.
  - A 6-axle locomotive can produce 3000 lb - 4000 lb force demand per linear foot of track.
  - A 125-ton capacity, 4-axle freight car can produce a force demand of 1000 lb - 1100 lb per linear foot of track.

- An 80-ton high performance passenger car can produce 800 lb force demand per linear foot of track.

- Areas of concern are in force inputs to cars and locomotives in buff or squeeze where the input is off center, causing lift, twisting, and lateral rail forces. The light or empty car and the short car are of greatest concern.

TDOP: 04-008

Canadian Institute of Guided Ground Transport, "Dynamics Of Freight Trains," (R&D program - E. F. Kurtz investigator), October 1973, Queen's University, Kingston, Ontario K7L 3N6, Canada, 2.8 (funded by Canadian National Railways, Montreal, Quebec, Canada, Ministry of Transport, Canada, Tower C, Place de Ville, Ottawa, Ontario, Canada).

Through this project, a mathematical model has been developed for investigating the dynamic stability of cars in long freight trains and appears able to predict accurately the stability characteristics of a specific container car employed by Canadian National Railways for which dynamic stability data are available. Groups of up to 16 cars were investigated, and the results indicate that the range of train velocities for which a long freight train will exhibit stable behavior can apparently be determined with satisfactory accuracy by consideration of individual cars free of coupling forces. The model includes the effects of creep and spin forces at the rail-wheel interface, and a consideration of spin forces was found to be important. It is possible to define an optimum value of lozenge stiffness for which the freight-car trucks would be stable as regards hunting for all speeds of interest for freight train operation.

RRIS: 054696

TDOP: 04-009

Canadian Pacific Railway, Track-Train Dynamics - Report No. 1, Report No. S-456-73, June 1973, Department of Research, Montreal, Quebec.

This volume reports on the first phase of Canadian Pacific's track-train dynamics test program. This program determined the vibrational properties of some of the heavy freight cars believed to be responsible for much of the damage to track and related structures. The natural frequencies and rate of damping of these cars were determined in both the tare and the laden conditions. The effect of the C-PEP side bearing on both the natural frequencies and damping rates has also been evaluated. Certain of the information derived from the vibration tests was intended primarily to provide back-up information that would be of use during later phases of the program.

TDOP: 04-010

Department of Transportation, Special Problems of Research on Wheel-Rail Dynamics At Office Of High Speed Ground Transportation, October 17, 1967, Washington, D. C. 20591.

This report deals with the Department of Transportation's four-car electrically-propelled research train. The self-propelled cars are coded T-1, T-2, T-3, and T-4, and each car constitutes a rolling research laboratory. There are four coordinated instrumentation subsystems:

- Track and caternary dynamics, at the wayside
- Track and wheel dynamics, on car T-2
- Truck and car dynamics, on car T-4
- Pantograph and caternary dynamics, on car T-1

TDOP: 04-011

Garin, P. V., "Truck Effect On Train Operation," Technical Proceedings - 1967 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, New York 14043, pp. 39-44.

This paper, which was part of a panel discussion on current truck performance, discusses means by which the standard A.A.R. truck could be improved. Among the points considered are:

- Should we develop and improve the design of a general purpose truck to suit all equipment or should we develop a special purpose truck suitable for specific equipment and certain loading and operating conditions?
- If the latter alternative is accepted, should we consider an entirely new truck design or a modification of the existing design?
- What is the best approach: low initial cost for an expected 14-year truck life or a higher initial cost for extended truck life?
- How will new designs affect maintenance costs for car structure and trucks, including other components such as wheels, axles, bearings, etc.?
- Will improved rideability reduce maintenance-of-way costs?
- Should special designs of trucks be compatible with existing car structures or be totally new, requiring a different interface between car and truck?

TDOP: 04-012

Gelson, W. E. and F. A. Blackwood, "Track Stress Research," Railway Gazette, Vol. 73, February 23, 1940, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 254-255.

The purpose of the work was (1) to verify the speed allowance formula, (2) to investigate the increment of stress over and above the static effect under normal conditions from track defects, (3) to investigate the conditions of support at rail joints, (4) to investigate the stability of ballast, and (5) to investigate the effects on rails and sleepers of the lateral forces set up by the hunting movement of locomotives. In regard to (1) and (2) it was found that experimental static deflections and stresses are in reasonably close agreement with calculated values. In regard to (3), it was concluded that stronger fishplates are needed to facilitate maintenance and that rail joints of inadequate strength are the cause of much of the impact effects. Increasing the number of sleepers will reduce these impact effects. In regard to (4), the shape of the sleeper was found experimentally to have no noticeable influence on its deflection under a given load, but stress distribution between the sleepers and subsoil requires investigation. In regard to (5), further investigation on curved track was recommended to confirm the consideration that design of rolling stock to reduce nosing on the sharper main-line curves would reduce the secondary stresses and thus offset the increased stress due to centrifugal effects.

RRIS: 037249

TDOP: 04-013

Giesking, P. F., "Total Car Design For Optimum Car Utilization (Relation of Car Trucks and Rail Corrugations)," Technical Proceedings - 1969 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York 14043, pp. 59-63.

This paper reports on the efforts of Canadian railroad suppliers to study and correct rail corrugation problems. Two 100-ton, Barber S-2 trucks were assembled with "third" bearings mounted within one outsized wheel hub per wheelset. Wheel treads were cylindrical with wide flanges. The trucks were tested under a 100-ton ore car, and results showed necessary drawbar pull was lowered by as much as 30 percent on curved track.

TDOP: 04-014

Hancock, R. M., "Vehicle Design Related To Track Conditions," Railway Gazette, April 1959, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 445-446.

Recommendations that may influence improvements in the safety and comfort of passenger-train rolling stock are made in this paper. The paper deals with the necessity of relating vehicle suspension and bogie design to the track conditions likely to be encountered in practice, particularly where lateral and crosslevel wave shape are concerned, as these are most likely to produce discomfort. The vehicle-response basis of systematic testing of main routes as carried out with the Western Region track-testing car has provided much of the experience from which the illustrations in the paper are drawn. The effects of coning and track shape, in relation to the riding of four-wheel vehicles, are considered with reference to an investigation of four-wheel vehicle derailments in fast trains.

RRIS: 037691
TDOP: 04-015

Henker, H., "General Considerations Concerning The Design Of Change-Of-Gradient Points," Rail International, June 1965, International Railway Congress Association, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 410-429.

This article discusses the engineering aspects of changes necessary in change of gradient points in railroad tracks to increase maximum rail speed. Vertical transition curves for change of gradient points date to early days of railways. Only recently have factors of geometry and dynamics of the vehicles been considered. The earlier determinations were done empirically without practical equipment considerations.

RRIS: 033852
TDOP: 04-016

Hillman, A. B., "Criteria For Track Geometry Design As Related To Modern Equipment," AREA Bulletin, 1970, American Railway Engineering Association, 59 East Van Buren Street, Chicago, Illinois 60605, p 414.

This paper reports on a proposal being considered for establishing standards for the minimum tangent distance between reverse points of various degrees of curves to permit negotiation of the curves by long box cars having 68-ft truck centers and coupler forces less than 200,000 lb. A list of tangent lengths with corresponding degrees of curvature are shown.

RRIS: 040043
TDOP: 04-017

Hood, C. M., II, "Articulated Container Cars (A Snake Full of Boxes)," ASME, 69-WA/RR-9, 345 East 47th Street, New York, N.Y. 10017.

This paper points out that containers are the first new, widely distributed, cargo carrier since the advent of piggyback and appear to justify a car dedicated to container service alone. Operating difficulties experienced with the long cars presently used appear to have been eliminated by a three-body articulated container car that seems to deliver significant economic benefits. This paper (which really constitutes a suggestion to carbuilders and users) describes the car and compares its performance with current designs. A comparison also is made of the economics of containers versus piggyback, with a conclusion heavily in favor of containers in general and the articulated container car in particular.

TDOP: 04-018

International Railway Congress Association, "Work Session--Section On Bogie--Suspension Systems Only," Rail International, June 17, 1968, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 1018-98.

This paper consists of a discussion of rolling stock for high-speed operation. It includes a discussion of bogie-suspension systems, locomotive design, and the relationship to track for determining speed limits. Part II, which concerns fixed installations, discusses the theoretical and experimental solutions to the problems of track design for high-speed operation. Note that as these are records of working sessions, the papers are abstracted, and there are questions and discussions of many points raised in the meetings.

RRIS: 033863
TDOP: 04-019

International Union of Railways, "Problems Of Interaction Of Vehicles And Track," by I. Van Bommel, International Union of Railways Annual Report, Question C 9, January 31, 1961, Office for Research and Experiments, Utrecht, Netherlands.

This committee report considers that the railway vehicle is made up of units that are united by elastic elements. The movements of the various units are considered, and formulae to represent phenomenon as: single mass oscillations, nonlinear vibrations, and self-sustained movement are included. The problem of hunting is defined, related factors are discussed, and the part that the track plays in the phenomenon is examined.

RRIS: 033206
TDOP: 04-020

Koffman, J. L., "The Effect of Suspension Design on Rail Stresses. The Matching of Spring Stiffness And Damper Characteristics As An Aid To Improving Riding and Reducing Rail Stresses," Rail International, September 1960, International Railway Congress Association, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 756-766.

This article considers the relationship between spring stiffness and damper characteristics of rolling stock as a way to improve comfort and to reduce stress at the rail, and also considers factors of vehicle mass, spring stiffness, damping factors of vehicles and track irregularity, sprung-unsprung weight mass, stiffness of track, and the softness of the ballast.

RRIS: 033390
TDOP: 04-021

Lind, E. F., H. Ten Broeck, and H. Nations, "Applications of Rail/Vehicle Dynamic Analysis to Train Operation," Proceedings of Conference on Track/Train Dynamics Interaction, December 15 and 16, 1971, Association of American Railroads, Research and Test Department, American Railroads Building, 1920 L Street, N.W., Washington, D.C. 20036, pp. 683-713 (joint paper presented at Conference on Track Train Dynamics).

This paper reports on the results of a study performed by a Southern Pacific Transportation Company (SPTCo) task force on track-train dynamics. The task force was set up after SPTCo experienced a number of derailments without clear causative agents. The track-train dynamics study concentrated on two areas:

- The comparison of L/V ratios on the trailing wheels of a multiple unit consist to the L/V ratios on the lead axle of the first car behind the locomotive.
- The comparison of four axle versus six axle locomotives and the measurement of dynamic profile under loaded conditions.

The report details the tests conducted by the task force and the computation tools used to analyze the test results.

Among the study conclusions reported were the following:

- The SD truck has much better unloading characteristics than the GP 35 truck.

- The two most significant factors present in extreme L/V conditions are rate of change of drawbar (slack action within the train) and speed.
- Vertical deflections greater than one inch are significant since the primary suspension system of the track is unable to absorb and dampen the resulting vertical accelerations, causing vertical unloading of the truck. Softer spring systems could represent a significant improvement in this area.

TDOP: 04-022

Lind, E. F., L. A. Peterson, J. F. Sapp, and J. T. Sullivan, "Vehicle/Track Interaction and Dynamic Responses To The Track," (panel discussion) Technical Proceedings - 1971 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York 14043, pp. 64-78.

Mr. Lind's report deals with a train dynamics program which has been undertaken by Southern Pacific Transportation Company and was funded by the A.A.R.

This study includes in its scope:

- The collection of additional information regarding train dynamics and the development of new analytical techniques
- The recognition that train dynamics includes both terminal and line operations

Mr. Peterson's paper begins with a theoretical discussion of train dynamics in which the author emphasizes the role of field testing in understanding the wheel-rail interface. A cooperative research and field testing program between the Bessemer and Lake Erie Railroad and the Quebec Cartier Mining Company is described.

Mr. Sapp's report covers the Louisville and Nashville Railroad Company's experience with rock and roll phenomena in connection with covered hopper cars. The results of various track and equipment tests are summarized.

The research program eventually caused the recommendation of the use of MDA hydraulic snubbers. Cost and performance data regarding these snubbers are included in the report.

Mr. Sullivan's paper discusses the problem of running "high-cube" cars over today's track structures. His paper ends by challenging the assembled delegates to design a car that can safely transport increased amounts of goods over existing track.

TDOP: 04-023

Magee, G. M. and W. M. Keller, "Passenger Ride Comfort on Curved Track," Bulletin 516, AREA Bulletin, American Railway Engineering Association, 59 East Van Buren Street, Chicago, Illinois 60605, pp. 125-214.

Tests were carried out to obtain data for making recommendations for the permissible speed on curves, the length of transition curves for passenger comfort, and for establishing clearance requirements on curved track. The first test was run on the Louisville and Nashville, May 10, 1950, using the Chesapeake and Ohio track inspection car and making use of 20 observers. Results of this test indicated the importance of the roll of the car body in reducing the effective elevation of the track insofar as passenger comfort was concerned. A second test on the Kansas City Southern developed gyroscope and recorder techniques to show the angle of the car body from the vertical. From the results of these tests, it was possible to establish a very satisfactory relationship between passenger reaction and the amount of lateral acceleration, so that in subsequent tests it was not necessary to use passenger observers. To obtain data on the various types of modern passenger cars being used, running tests were subsequently made on seven railroads. The tests have indicated that for types of modern equipment having soft springs and no provision for restricting the roll of the car body on curves, the present AREA limitation of 3-in. unbalance should be continued. A new and different procedure is recommended for determining the length of transition curves based on the rate of change of lateral acceleration entering and leaving the curve rather than on the rate of change of elevation. With respect to clearance, the test data gives displacement characteristics due to roll of the car body on the springs of the various types of passenger cars. The records indicated that an allowance of plus or minus 1 degree in car body roll will provide for irregularities in line and surface for representative main-line track for speeds up to 90 mph.

TDOP: 04-024

Matsubara, K., "Welded Rail Joint Fractures And Their Effect On 200 KM/H Operation," Railway Technical Research Institute, Vol. 15, No. 3, September 1964, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 21-24.

JNR conducted a series of tests to determine the effect of broken welded rail joints on trains running at high speed. A rail gap of 20 to approximately 30 mm was employed since this was considered the likely amount just after a rail fracture in winter on the New Tokaido Line. The train used for this test consisted of six, 2-axle bogie-type electric rail-cars with an axle-load of 15 tons. Items measured included: rail deflection, rail stress, stress on the fastening device, track vibration, accelerations, sleeper stress, etc. Most of these were measured using wire strain gauges. Onboard measurements included: wheel side thrust, wheel load, bogie stress, carbody vibration, axle box vibration, and similar forces. The results of the test indicate that train operation on the New Tokaido Line is judged completely safe from the point of view of possible broken welded rail joints, in that even the lateral discrepancy of ends of the broken rails and wheel side thrust at the train passing the broken point were found to be less than 1/2 of the respective maximum limits for safe train operation, and values for carbody vibration and other items were also found to be sufficiently small.

RRIS: 037229
TDOP: 04-025

Matsudaira, T., "How High Can Train Speed Be Increased? A Review of Present and Future," Japanese Railway Engineering, June 1966, Japan Railway Engineer's Association, P. O. Box 605, Tokyo Central, Tokyo, Japan, pp. 131-134.

The effect of wave propagation in air and on the rail is discussed as a theoretical limit for train speed. As a practical limit to speed, the deflection by the pantograph to the wire at the point of contact is described. By this principle, the critical speed of the New Tokaido Line (NTL) train has been calculated as 400 km/h. Adhesion force is plotted versus tractive resistance for a 12-car NTL train. Vibration limits speed to 230 km/h on straight track due to passenger comfort. The problems concerning curved track are also briefly discussed.

RRIS: 040200
TDOP: 04-026

Mauzin, Andre, "The Static and Dynamic Parameters of Railway Coaches," The Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England (paper presented at Joint Convention on Interaction Between Vehicle and Track, November 1965).

This report consists of a general discussion of railcar dynamics in combination with a synopsis of recent S.N.C.F. testing with two prototype trucks one, type A, aimed at reduction of transversally sprung masses; the other, type B, was of a more conventional design. The report concludes that, from the perspective of comfort, the body of a rail vehicle can, as a first approach, be considered as a solid linked flexibly at its ends to two trucks capable of alternating movements whose amplitude is limited by the clearance of the track and the period maintained between two extreme limits.

The author states that this concept leads to simple theories and complex practices. However, experience suggests that solutions exist and are practicable.

TDOP: 04-027

Meacham, H. D., Jr., "Modern Approach To Train-Track Dynamics," Battelle Technical Review, July 1968, Battelle Memorial Institute, 505 King Avenue, Columbus, Ohio 43201, pp. 13-21.

This report deals, in a general way, with the vehicle and rail as a dynamic system, and discusses the study of this system through computer modeling. Without considering the track and roadbed, the author considers that there are 46 important vehicle motions, and lateral and vertical track effects add 16 more degrees of freedom. The problem of setting up a computer model to reflect the phenomenon of car rocking is given as an illustration. The problem of arriving at a dynamic track structure model (one that reflects a track's reaction to a passing vehicle) is then discussed. Various research programs to improve and evolve the conventional track structure are cited, and a theoretical discussion of track behavior is included.

TDOP: 04-028

Miller, T. C. B., "Towards Higher Speeds," Railway Division Journal, Vol. 1, No. 6, 1970, Institution of Mechanical Engineers, 1 Birdcage Walk, London SW1, England, pp. 628-661.

It is essential that track design and locomotive design should be considered together as a single joint problem. Present bogie damper designs for both locomotives and coaches must be developed to the stage where they give satisfactory performance for 150,000 miles. Existing carriage brake gear is inadequate for braking from any higher speed than 100 mph and has shown itself to be unable to withstand normal service buffeting without costly maintenance. Passenger standards for comfort require future stock to be noise-proof and probably, air-conditioned. Studies of sustained higher speed let to several design changes to such things as carriage brakes, the vehicle suspension of both locomotives and carriages, the overhead contact catenary on electrified lines, and so on. They also focused attention upon the importance of the weight, both sprung and unsprung, on the axles of locomotives, and the effect their characteristics have on the permanent way and upon its maintenance.

RRIS: 033865

TDOP: 04-029

Ministry of Transport, "Ministry Of Transport Accident Report," by W. P. Reed, Railway Gazette, Vol. 100, May 28, 1954, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 613-614.

This accident report relates to the derailment of a passenger train at 55 mph on straight track near Kingsbury. It was concluded that the engine was hunting on the approach to the point of derailment and for some distance past it, and that this distorted the track. The hunting was caused by the significant variations in cross level some distance back, followed by lesser ones coinciding with the period of hunting of the engine, and was contributed to by uneven loading of the engine bogie and coupled wheels which, with the tender axle side play, made the engine less stable.

RRIS: 037954

TDOP: 04-030

Ministry of Transport, "Ministry Of Transport Accident Report," by G. R. S. Wilson, Railway Gazette, Vol. 97, July 18, 1952, Temple Press Limited, 161-166 Fleet Street, London EC4, England, p 77.

The derailment of a passenger train travelling at 65 mph near Weedon, London is described. The accident occurred as the train exited from the transition of a curve, at which time a defective bogie became derailed causing the engine to plunge down a 12-foot embankment. Several people were killed. Details are provided.

RRIS: 037960

TDOP: 04-031

Ono, K., "On The Rail Creepage," Permanent Way, Vol. 4, No. 13, December 1961, Japan Railway Civil Engineering Association, Kyodo Bldg. 18-7 Hagashi-Uyeno 2 Chome, Daito-ku, Tokyo 110, Japan, pp. 20-28.

This paper describes studies on the processes and the causes of rail creepage. Rail creepage under traffic was measured in relation to the kind of cars and the temperature of the rails. It was found that the rail creepage occurred distinctly only under bogie cars, and the sum of the rail creepages gradually increased when the temperature of the rail varied. It was concluded that rail creeps by the movement of the deflection of the rail under wheels and by the elongation or the contraction of the rail, which is caused generally by the passage of the traffic, and is a result of the releasing of the pressure or the tension in the rail.

RRIS: 033352
TDOP: 04-032

Prud-Homme, A., "The Track," French Railway Techniques, No. 2, 1970 Federation des Industriels Ferroviaires, 92 Rue Bonaparte, 75 Paris 6E, France, pp. 67-79.

Track stresses caused by future trains intended to operate at maximum speed of 300 km/h will remain within acceptable limits for the orthodox type of track, and comfort will be excellent without the necessity of maintaining a quality of high-speed track better than that already achieved. The laying of concrete slab track also is not justified from either the technical or economic point of view.

RRIS: 033848
TDOP: 04-033

Railway Research, Method of Analysis For Determining The Coupler Forces And Longitudinal Motion of A Long Freight Train In Over-The-Road Operation, by G. C. Martin and W. W. Hay, June 1967, Department of Civil Engineering, University of Illinois, Urbana, Illinois.

This report describes a method of analysis for determining the longitudinal dynamics of a long train in over-the-road operation. The method hinges upon modeling a train and its longitudinal cushioning devices as a mass-spring system. The forces acting on a train during its operation are described and applied to the model. A computer simulation is then performed using a numerical integration scheme to find the coupler forces, displacements, velocities, and accelerations for the cars in the train.

Two examples are presented in this paper as an illustration of the method.

TDOP: 04-034

Smith, F. R., "Car Rollability On Grades," paper 38110, March 4, 1957, American Railway Engineering Association, St. Louis, Missouri (paper presented before the American Railway Engineering Association Annual Meeting, 1957).

This paper defines rolling resistance as the summation of all forces opposing free rolling. The author suggests that it is essential that rolling resistance in freight cars be lessened in the interest of providing better railroad service at less cost. The following recommendations are made:

- Improve center and side bearings through good lubrication and the use of rust resistant materials
- Design brake rigging to minimize rotation or rocking of the truck
- Create a device for freeing brake shoes when air is bled out of car and handbrake is released

- Develop a truck that will prevent longitudinal translation of side frames
- Consider wheel and axle design to minimize weight variation

TDOP: 04-035

Southern Pacific Transportation Company, Track-Train Dynamics Study Project TR-14 Rail-Vehicle Interaction Study - Report No. 1, by J. P. Lynch, H. R. Ten Broeck, T. B. Wagner, E. F. Lind, and R. A. Bardwell, June 1970, 1 Market Street, San Francisco, California 94105.

As a result of various derailments, a task force was formed to investigate the dynamic forces exerted by locomotive and freight car wheels against the rail. Critical study was directed at the 3- and 2-axle trucks of high horsepower locomotives, 85-ft trailer-on-flat-car cars, and 50-ft box cars. The objective of this investigation was to determine if dynamic forces of sufficient magnitude to cause derailment were being generated by equipment, track structure, and operating practice, and to recommend any indicated corrective action.

Extensive field tests of wheel-rail interation were conducted in various territories between Los Angeles and Pine Bluff, Arkansas. An analysis of these test results led to the following general conclusions:

- Dynamic forces of sufficient magnitude to cause derailments are being generated in everyday operation of revenue trains.
- The forces are also sufficient to cause greatly accelerated wheel and rail wear, as well as rapid wear on other equipment components such as draft rigging.
- Forces of sufficient magnitude to exceed the ability of the track structure to resist permanent deformation in alignment are also being generated.

While the causes of these dynamic forces are not entirely clear, the following specific conclusions are offered:

- The 3-axle truck of the SD-45 (EMD unit) does not appear to be any more likely to cause derailment due to wheel-rail loading than the 2-axle GP-35 truck (EMD unit), but does cause considerably more wheel and rail wear.
- Long cars of the truck-on-flat-car type cause very high lateral loads at authorized speeds on existing alignments. Maintaining alignment to a high level is necessary on curves where long cars operate at high speeds.
- Heavy, high horsepower locomotives have made possible the operation of heavy trains at continuous high speeds. This increases the dynamic loads imposed on equipment and track to the extent that the usable life of these assets will be materially shortened if operating practices are not changed, or maintenance levels are not increased.
- The force levels being generated in the everyday operation of trains make it absolutely necessary that applicable rules be observed at all times. Especially critical are those concerning train handling and speed restrictions.
- Dynamic braking in higher speed ranges does not cause high lateral force levels if the instructions concerning its use are observed.

A total of 29 attachments of data, drawings, samples of oscillograph tapes, and statistical plots and charts are included in this report.

TDOP: 04-036

Southern Pacific Transportation Company, Track-Train Dynamics Study Project TR-14 Rail-Vehicle Interaction Study - Report No. 2, by E. F. Lind and N. W. Luttrell, April 1971, 1 Market Street, San Francisco, California 94105.

Following a description of the problem involved in preparing a hybrid computer program for the analysis and reduction of transient data taken in the actual field tests of the Track-Train Dynamics Study Project TR-14, reported in Report No. 1 (see TDOP: 04-036), and an explanation of the computational approach, this report includes a description of the method of interfacing the digitized data with the digital computers used to obtain an in-depth engineering analysis of the results of the tests.

Among the conclusions drawn from the computerized results were the following:

- L/V ratios for the SD-45 and the GP-35 (EMD units) were comparable in many respects although the SD-45 consistently had higher steady state forces on light curves (0 - 3 degrees).
- The GP truck operating in heavy curve territories under high buff conditions in many cases shows higher dynamic lateral forces than the SD truck.
- The SD truck has much better unloading characteristics than the GP-35 truck.
- Vertical deflections in the rail between 0 and 3/4 inch have very little effect on high L/V ratios. Change in cross level of differential deflections of the rails is by far the most significant factor in track conditions attributing to high L/V ratios.
- Vertical deflections of greater than 1 inch are very significant because the primary suspension system of the truck is unable to absorb and dampen these vertical accelerations causing vertical unloading of the truck.
- Rail and flange wear is primarily attributable to high lateral steady state forces and not L/V ratios.
- The TOFC car showed the greatest potential of being a prime candidate in causing future derailments.

Under general recommendations were given the following:

- The establishment of long-range educational programs regarding the phenomena associated with derailments in the Operating, Mechanical and Engineering Departments of the railroad.
- The formation of a select group of individuals from all departments to investigate all questionable derailments and have the sole responsibility for determining the cause.
- The suggestion of various possible alternatives for future study, which included a suggestion for the creation of a modern test car or cars for carrying on such work. Such field tests should be complemented by the use of a large digital computer to simulate the motion of two adjacent cars and a section of the track beneath them using a finite time approach.

TDOP: 04-037

Temple Press Limited, "Rail-Guided Vehicles On Concrete Track," Railway Gazette, Vol. 87, August 29, 1947, 161-166 Fleet Street, London EC4, England, pp. 237-238.

In western India, an apparently successful attempt has been made to solve the feederline problem by constructing a nine-mile Guideways track carrying pneumatic-tired passenger and goods vehicles. The track consists of a continuous lime-concrete slab, 3 ft wide and 6 in. to 9 in. deep, surfaced with cement plaster and having a central guide rail to prevent vehicles leaving the track. Great economy is claimed for Guideways, both in prime cost and maintenance. One man patrols each four-mile length, sweeping the track clear of sand and stones.

RRIS: 039477

TDOP: 04-038

Temple Press Limited, "Slightly Staggered Rail Joints," Railway Gazette, Vol. 94, February 23, 1951, 161-166 Fleet Street, London EC4, England, p 201.

The staggering of rail joints is logical because it avoids placing two weak spots, the joints, directly opposite each other. It also reduces the impact at the joints to that of a wheel load instead of an axle-load, and it produces a more uniform vertical continuity of the track. Certain railways in India experimented with short-pitch staggered joints. Graphs obtained with the Hallade track recorder showed that the running over the slightly staggered road was inferior to running over normal, square-joint track. Selection of the optimum pitch for the stagger is half the length of the wheelbase of the standard type of bogie fitted to passenger stock. This complies with the condition that the stagger must be less than the minimum wheelbase of any bogie allowed to run over the line, namely, 6 ft.

RRIS: 039473

TDOP: 04-039

Thille, M., "Adaptation Of The Methods Of Laying, Aligning And Maintaining The Permanent Way To Carry Traffic At Very High Speeds (120 KM/H and More): A) On The Straight: B) On Curves: So Far As They Affect Safety And Taking Into Account The Type Of Rolling Stock Used," Rail International, International Railway Congress Association, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 492-725.

The following topics are discussed in this paper: effects of rolling stock on track; layout of lines; points and crossings; loading gauges; distances between running lines; equipment, ballast, and track renewal of high-speed lines. Appendices include answers related to railway technology from responding members.

RRIS: 033273

TDOP: 04-040

Union Pacific Railroad, Test Of Lateral Loads Required To Spread 133 Lb. Main Line Rail Under Various Loads Statically Imposed, by P. E. Flebbe, July 1969, 1416 Dodge Street, Omaha, Nebraska 68102.

In order to determine the lateral force required to spread or overturn the 133-lb rail on a main line under static vertical load conditions, vertical loading was imposed by a freight car of known weight standing on the track with axle center-line of one pair of wheels in the vertical plane through the line of force of the jack.

Lateral forces were developed and measured by a calibrated 150-ton hydraulic jack suitably blocked between the rails at the gauge line for the zero vertical load, and with the lateral loading applied on the gauge side of head of rail through the flange of one wheel and suitable blocking on the opposite rail.

The results obtained follow:

| Test | Lateral Force (lb) | Vertical Load on Rail (lb) |
|---|---|---|
| 1 | 48,000 | 16,700 |
| 2 | 45,000 | 7,700 |
| 3 | 34,000 | 0 |
| 4 | 47,000 | 10,900 |

RRIS: 039495
TDOP: 04-041

Union Pacific Railroad, Track Gauge Widening On Tangent Track-Roller Bearing Tie Plate Tests, by P. E. Flebbe, Union Pacific Railroad Tests (File 340), March 29-April 9, 1969, 1416 Dodge Street, Omaha, Nebraska 68102.

A problem of gauge widening on tangent track, at mile posts 614.08 and 626.21, in particular, led to these tests with instrumented ties with roller bearing tie plates and an instrumented diesel-electric locomotive, which was reputedly "rough riding," and involved a total of 179 trains. The road bed was in average condition, with the track of 133-lb rail, and with a grade descending in westward direction in the amount of 0.82 percent at mile post 614.08 and 0.35 percent at mile post 626.21.

Results of the tests showed the following:

- Only 51 cars (out of the 179 trains) created lateral forces of 7,000 lbs or more, with a maximum of 15,000 lbs. Twenty-two of these instances were created by empty cars, twenty-seven by loaded cars, and two cars could not be identified.
- The maximum lateral force created by the locomotive was 10,000 lbs.
- The maximum lateral accelerations recorded did not exceed 0.1 g.
- No skewing forces were recorded even under heavy dynamic braking.

Conclusions drawn from these tests were as follows:

- No class of locomotives or cars individually were found to exert a lateral force at the test site sufficient to result in the gauge widening being experienced.
- The cars created a greater number of instances with higher values of lateral forces than the locomotives involved.
- Trains operating at higher speeds created the higher lateral forces.
- Dynamic braking of the trains produced no noticeable lateral forces in the trailing unit of the locomotive consist or the head end cars.
- Apparently, the gauge widening resulted from the high utilization of the track and high train speeds.

RRIS: 039693

TDOP: 04-042