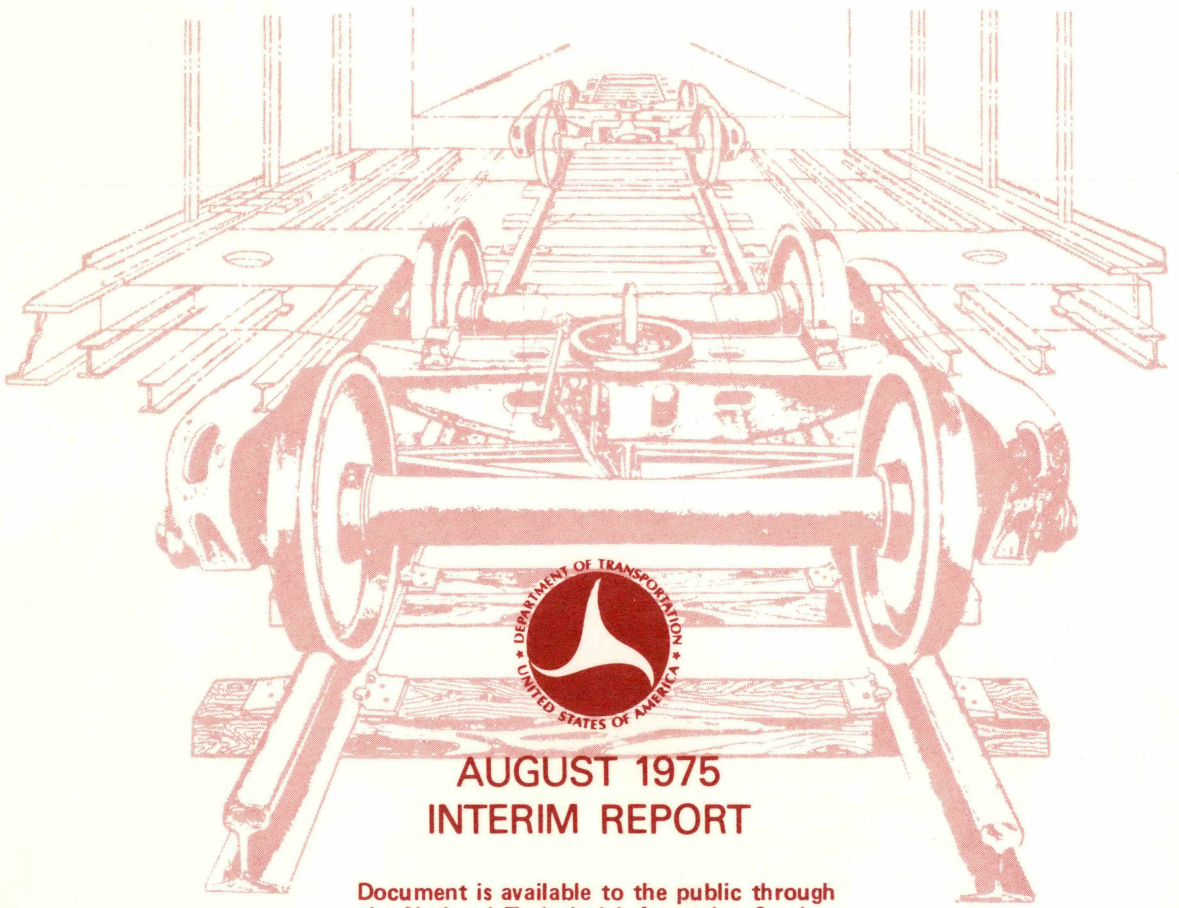
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FREIGHT CAR TRUCK DESIGN OPTIMIZATION

LITERATURE SEARCH - VOLUME III

Southern Pacific Transportation Company
Technical Research and Development Group



AUGUST 1975
INTERIM REPORT

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03 - Rail Vehicles & Components

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This interim report represents the third of a three-volume set. Bibliographical additions will be made to these volumes as new articles become available throughout the course of the Truck Design Optimization Project. See also, FRA-OR&D 75-81A & B.

16. Abstract

Volume III of the TDOP Literature Search contains the sections entitled: "Truck Performance" and "Literature Search Title Index." The section dealing with truck performance 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 Performance" is further organized into the following subsections:

- Computer Analysis of Truck Performance
- Field Analysis of Truck Performance
- Laboratory Analysis of Truck Performance

The index section contains a listing alphabetized by title of all publications included in the three-volume Literature Search.

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Section 5

TRUCK PERFORMANCE

INTRODUCTION

The following subsections are comprised of reprinted articles, which were selected from the assembled literature concerning truck performance, and the TDOP truck performance bibliography. The reports selected for reprinting are: "Car Roll And Wheel Lift Tests On Simulated Service Track: Test Conditions And Results At PRR Altoona Test Site," by L. W. Bertram; "A Criterion For The Control Of 100 Ton Hopper Car Roll Motion," by K. A. Henderson and J. Johnson; "Running Safety," by Japanese National Railways; "Lateral Loading Between Locomotive Truck Wheels And Rail Due To Curve Negotiation," by L. F. Koci and H. A. Marta; "Nonlinear Wheelset Dynamic Response To Random Lateral Rail Irregularities," by E. H. Law; "Digital Computer Simulation Of Railroad Freight Car Rocking," by A. A. Liepins; "Performance Testing Of Freight Car Trucks," by R. W. Luebke; "Problems On Hunting Of Railway Vehicle On Test Stand," by T. Matsudaira, N. Matsui, S. Arai, and K. Yokose; "Measurement And Analysis Of Wheel-Rail Forces," by L. A. Peterson, W. H. Freeman, and J. M. Wandrisco; Road Test For Truck And Component Wear Investigation, by Seaboard Coast Line Railroad; and "General Aspects Of The Lateral Dynamics Of Railway Vehicles," by A. H. Wickens.

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

- Computer Analysis of Truck Performance
- Field Analysis of Truck Performance
- Laboratory Analysis of Truck Performance

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

magnitude, you practically have to tear the insides out and start all over again. Control could be kept to a certain limit. However, these cars are being handled in such a way that the impacts will occasionally exceed 14 miles an hour.

Chairman Smith: Elmer.

Mr. Sebrell: E. H. Sebrell, Seaboard Air Line Railroad Co. I know in car impact, especially in yards, you have a human element there that is hard to control — especially about 3:00 o'clock in the morning, when it's raining and everybody is sort of sleepy like, and the cars just naturally are getting impacts exceeding eight miles an hour. However, on the Seaboard, we have these instruments that we will put in trains, and the operating engineers don't know what color these impact instruments are or where they are located and as a result, we can locate some check on the conditions of impacts in the yards and also in the trains. Through this method, we give awards for the best engineer and conductor in handling his train which does promote or have some incentive to have cars and trains in operation that will be to the advantage of the railroad and to the engineer to see which one can handle the best train.

Chairman Smith: I think we will have to move on to the next paper in order to keep our schedule. Thank you, Mr. Leriche.

(APPLAUSE)

Chairman Smith: The next paper will be given by Mr. L. W. Bertram, Resident Mechanical Engineer, The Pennsylvania Railroad Company, on "Car Roll and Wheel Lift Tests on Simulated Service Track: Test Conditions and Results at PRR Altoona Test Site." Mr. Bertram.

Mr. Bertram: Thank you, Mr. Smith. Gentlemen, I would like to talk today about the testing that we have conducted actually at our Hollidaysburg yards which is in the Altoona area.

The words "Rock and Roll" mean different things to different people. To a modern teenager, it's the music, shall we call it, of the Beatles or the Animals. To a railroader, however, it's the ominous sounds of center plates slapping and truck parts grinding as a 100-ton hopper car comes waddling down the test track like a fat old woman. These sounds have been ringing in our ears since the spring of 1964, when our new 100-ton coal hopper cars began derailing at an alarming rate on both the yard and mainline tracks. This problem is not new, it merely has been intensified in the past few years as thousands of high capacity, high center of gravity cars, have been placed in service.

The case in point, which I will talk about, is our Class H-43 — 100-Ton Hopper Car. This car has a truck center of 36'2" and a loaded center of gravity of 88" above the rail. After our rude awakening to this problem of derailment, it was quickly decided to set up a test track to simulate the rock-off phenomena and determine what modifications could be made to the car to reduce wheel lift and car roll. Except for a few of the very first tests, our testing has been conducted at the Hollidaysburg yards in order that modifications can readily be made at our Samuel Rea Shops.

All cars were tested first on a tangent track with rails shimmed ¾" to produce an 1½" total change in cross level at each 39-ft. joint over a distance of ten rail lengths at speeds in the range of ten miles per hour to 22 miles per hour at approximately one mile per hour increments or until the critical speed was exceeded. The cars were then tested on a 3° curve having a 4½" super-elevation with an 1½" change in cross level superimposed at each joint over a distance of six rail lengths. Speeds on the 3° curve did not exceed 15 miles per hour. A speed check instrument, using rail head contacts, was used on the test track sections to record the speed of the car. Visual observations were made of any lifting of the car wheels over a four rail length section of the test track. Wheel lifting less than ½" was designated as slight. From ½" to 1" as small. From 1" to 2" as medium and over 2" as large. On the surface, this would appear to be a crude way to measure wheel lift. But the same people were used to observe each test; and since there were numerous tests run during the past several years, we feel that the wheel lifts were accurately reported. The total including angle of car roll was, in most cases, determined from visual observation. In several of the later tests, car roll instrumentation was included. Motion pictures were also taken during each test and have been very helpful in reviewing the performance of various devices that were tested.

Our main test, during the summer of 1964, was to determine what could be done to our existing H-3 Hopper Cars in order to prevent further derailments. The initial test proved, very quickly, that as the car approached the speed of approximately 15 miles per hour, the track input at the low rail joints was synchronized with the natural frequency of swaying of the car on the truck springs, thus increasing the amplitude of force input by the springs back into the car body. The car sway would then build up to the point where the load on opposite wheels became so unbalanced that the wheel would lift. This wheel lifting would occur after the car would pass over about the fourth shimmed joint and would build up to a maximum within the ten rail lengths of shimmed track. After the car passed this section of

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track, the rolling of the car would slowly dampen itself out.

After testing 57 combinations of changes in the original Class H-43 Trucks — which included springing, side bearings, center plates, journal bearings, bolster lug clearances and other features, it was decided to adopt the following truck modifications: — 1. Replace the 2½" travel springs with 3 11/16" travel springs. The softer spring rate reduces the amplitude of rail input into the car body and changes the natural frequency of car body roll. Also, it allows more travel of the snubbing elements as car roll begins to increase; 2. Add two friction snubbers to each spring group to absorb energy input into the spring group; 3. Move the side bearings in from 50" centers to 46" centers. As car roll becomes extreme, the car body slams down on the side bearings on one side of the car. In fact, there are times when the center plate separates completely from the truck and the entire body rests on two side bearings. Shifting the side bearings in toward the center line of car distributes more load to the wheel which is tending to lift; 4. Machine the bolster gibs to permit more bolster lateral freedom. This allows the bolster to shift in the direction of rotation of the car body, thus reducing the total car body inertia. Also, additional gib clearance at the side frame column reduces the possibility of the bolster gibs binding on the side frame at the time of maximum spring compression, which would tend to lift up on the side frame and aggrevate wheel lift.

There were many other truck designs and modifications tested by the PRR and several arrangements gave an improved performance on the test track as compared to our present standard truck with the four modifications as I have reviewed. General Steel Industries Aligned Truck with the D5 Spring, 50 per cent increased capacity in the friction block side springs, an 1" thick x 6" wide friction plate ring around the center plate and horizontal stabilizers between the truck frame and bolster produced a maximum car roll of 5° with no wheel lifts. The Standard Car Truck, Barber Truck with D5 outer and D2 inner springs, 50 per cent increased capacity in the friction block side springs and a pre-loaded resilient side bearing produced a maximum car roll of 7° with no wheel lifts. The Stucki Company's Hydraulic Snubber, which fits between the top of the bolster and the truck side frame, was applied to a Barber Truck having D5 springs, 46" side bearing spacing and ½" increase clearance on the bolster outer gibs. The test car was instrumented to measure side bearing reaction, axle load at roller bearing adapters, pressure level of hydraulic device and the car roll angle. The results of the tests showed marked improvement over the present springing arrangement with wheel lifts eliminated and car roll reduced from 7° to 4½°. Also, maximum side bearing reactions were reduced from 160,000 lbs. to 70,000 lbs. — that's the reaction at one side bearing.

Another interesting fact, which developed during the test of the Stucki device, was that even with wheel lifts for an unmodified H-43 Car, there was always a positive reaction at the roller bearing adapter indicating that the journal at the lifting wheel never becomes completely unloaded. Maximum and minimum axle loads measured during these tests of the standard truck were 60,000 lbs. and 7,000 lbs., respectively. The reactions measured during the tests of the hydraulic snubber were 38,000 lbs. and 26,000 lbs. indicating a much more favorable distribution of load on the wheels. By adjusting the pressure level in the hydraulic device, it was determined that any loads under 14,000 lbs. on one end of the axle could produce a slight wheel lift. The Motion Dampening Assembly, a hydraulic snubber developed by Railroad Dynamics, Inc., was tested on a H-43 Car equipped with ASF Ride Control Trucks having D5 springs, 46" side bearing spacing and ½" increase clearance on the bolster outer gibs. This hydraulic device mounts between the car body and the truck side frame at the outside of the spring seat. Force dampening is accomplished in both directions of piston stroke. This device also showed a definite improvement over present standard truck in precluding excessive car oscillation associated with the critical speed. Wheel lifts were eliminated and car roll was reduced from 7° to 4½° on tangent track. We plan to equip some of our H-43 Cars with this device for service testing to determine what sort of service life we may expect before maintenance is required. We are not satisfied that our present truck modification is a solution to the problem. However, it has proven itself as a means to keep the wheel lift and car roll within tolerable limits.

As a matter of interest, we have started to conduct tests in collaboration with our Maintenance-of-Way Department to determine to what extent, if any, our track specifications governing variations in cross level need to be changed in order to eliminate the conditions which cause both modified and unmodified 100-ton capacity cars to rock off of the track.

We must also consider the fact that even though a truck modification eliminates wheel lift on the test track, we are not sure just how close the wheel is to becoming unloaded. Also, we do not know how

close to unloading the wheel can come under varying conditions of curves, grades and train handling before a derailment condition actually exists. There is no doubt that further improvement is required. Not only to prevent derailments to the rock-off, but to control the car sway which is repeatedly adding higher stresses to the car body, the truck components, wheels and axles and the rail itself.

If you can bear with me for a few minutes, I would like to show you a short film of some of our recent car roll tests at Hollidaysburg. (Film shown.)

This about concludes my talk this morning. I would like to point out that in addition to the PRR sponsored tests that I have been talking about, the Association of American Railroads and outside firms have had various devices and truck arrangements tested on our tracks at their own expense.

If there are any test results or comments that anyone has on our tests or other tests that have been conducted, we would be most happy to hear from them during the discussion period. Thank you, gentlemen.

Chairman Smith: We have time for a short discussion before lunch and we can continue the discussion later this afternoon. Bill.

Dr. Manos: *W. P. Manos, Pullman-Standard. What does the PRR recommend now for truck modification to improve the ride.*

Mr. Bertram: The modifications that I have outlined are still the modifications that we follow. We can't recommend anything further than this right now because we are going to have to get much more service experience with the new devices. As I pointed out, you can't duplicate everything on a test track. You may find that even though you do not experience a wheel lift on this test track, the right condition of curve, and of run-in in a train may cause a wheel lift. We hope that we can see further improvements on truck modifications. Some of these look very promising. As you know, in the railroad industry, you look a little cautiously at hydraulic devices and so do we. We are not saying that this is the answer. In fact, only after we get some service experience, will we know that these hydraulic devices can actually operate in extreme heat and cold along with the other conditions that you face on the railroad. Right now, we are still relying on these four modifications. There are several ways to improve the condition, and that would be a change in the design of the truck or a modification of existing trucks. There is a lot to be said about the present design of the existing four-wheel truck which has a lot of experience and history behind it. Very possibly, there may be something basic in the truck design that can be changed to improve this condition. Right now, we are still falling back on the utilization of a standard four-wheel truck with these modifications and, of course, testing on other types of devices.

Mr. Koci: *L. F. Koci, Electro-Motive. This 1½" out of cross level, was that as measured with no load on the rail.*

Mr. Bertram: Before each test, the cross level is measured with a load on the rail. We follow behind the loaded car so that we are sure that with the car weight on the rail we have this 1½" change in the cross level. On the track standardization tests that we are presently running, we use cork gauges for measuring total deflection of the track. As you know, the road bed can change considerably between tests and this was one of our fall backs on the first tests, in that sometimes we did not always have the 1½" change in cross level that we thought we had. It could have been more or less which would, of course, influence the results of the tests. We are zeroing in more accurately on the condition of track before we actually test and we are finding out some very interesting things. Something less than an 1½" change in cross elevation can produce a wheel lift on a standard type of car. If you have enough rail joints at ¾", you can produce wheel lift. You can't say that this is the limit. Presently, we don't know what it is. This is what we are trying to find out. It is not just four joints at 1½" change in cross level. A combination of less cross elevation over a longer span or one or two joints in the track that do not quite come up to present specifications could produce wheel lifting.

Mr. Koci: *Well, what about your regular track? Your present standard on the Pennsylvania is the 1½" limit, is that without load on it?*

Mr. Bertram: As far as I know, it is.

Mr. Koci: *Does anyone know, roughly, what it is out of cross level under load on your regular track?*

Mr. Bertram: Well, no, I can't give you any accurate figures on that. I do know what we have done on the test track. We have checked it. Of course, the condition of ballast governs just how much change in cross level you do have. This may have to be part of our new specification. We are working on it now.

Chairman Smith: Elmer.

Mr. Sebrell: *E. H. Sebrell, Seaboard Air Line R.R. Co. This test that you conducted, were these cars loaded?*

Mr. Bertram: Yes, they were fully loaded to 100-ton capacity with coal, and we checked our center of gravity calculations with the Test Department by pushing on the side to make sure that we had

the figure 88" accurately located when the car was loaded with coal.

Mr. Sebrell. *Have you made any tests with a load that would be probably unbalanced?*

Mr. Bertram: Do you mean unbalanced from side to side?

Mr. Sebrell: *That is right. Maybe with a load heavier on one side of the car than on the other.*

Mr. Bertram: Well, that is a good point. Usually, when we load our open hoppers with 100 tons of coal, running about 55 lbs. per cubic foot, we have to put a good heap on the top in order to get our load limit. We did find on testing of covered hopper cars loaded with lime that the lower center of gravity will get you out of the problem. If you load it with wheat or something lighter, building up the load to the full volume, you get back into this center of gravity problem. We did not try to load the cars unevenly. I don't know whether that could be done with a coal car or not. You have to have the weight in here. This is all a factor — weight and center of gravity all affect this phenomena. Every car has its own natural sway. You find that some can have a critical speed of less than 15 miles per hour, and we have had cases where we didn't get into problems until we reached 20 to 22 miles per hour. As far as I know, we have always tested the car with the load evenly distributed.

Mr. Sebrell: *Now, I have one more question. I understand that these tests were made with $3^{11}/_{16}$" spring travel. Is that right?*

Mr. Bertram: Yes. We have tested all types of spring travel.

Mr. Sebrell: *I was getting ready to ask if you had tried the 2½".*

Mr. Bertram: Yes, In fact, the cars were originally equipped with the standard 90-ton grouping of 2½" travel springs. During our testing, we tried the D3, D4 and D5 springs, and our best results were obtained with the $3^{11}/_{16}$" travel springs. Every once in awhile we check on ourselves by running a couple of these tests again to confirm this fact.

Mr. Sebrell: Thank you.

Chairman Smith: Mr. Parker.

Mr. Parker: C. W. Parker, Canadian Pacific Railway Company. In response to Mr. Sebrell's question about the unbalanced load, a lot of our derailments with 100-ton capacity hopper cars did occur with cars that had an unbalanced load. When we tested the things, we found what you might imagine — they will go over farther on the heavy side but less on the light. The total amplitude is just about the same. In respect of the spring groups, I think that there has been a general assumption that what is good for one railroad's cars is good for another, and this is not the case,

Mr. Bertram: I agree with that.

Mr. Parker: We found that it was better in the case of our cars to kick out the D5's and the D4 springs and go back to D3s. We couldn't find any improvement at all with the reduced spacing of the side bearings. We found that if you increase the clearance of the bolster gibs that it was totally ineffective. So, you see what I mean, what is good for one set of cars is not necessarily good for another.

Mr. Bertram: True. My previous boss pointed out several years ago at one of these conferences that the solution to our problem may not be the solution to yours.

Mr. Parker: There is only one more point to make and that is about the speed range. We saw what the L&N was doing and saw that while they attempted to test below 20 miles per hour, there was one car that was run at 22 miles an hour and it was lifting its wheels. So, we carried on tests in the 30 mile an hour range and found that the empty cars particularly tend to rock more above 20 miles an hour. You could divide them into groups depending on the truck centers. The cars with the longer truck centers rocked more at the higher speeds than the cars with the shorter truck centers. In the case of the loaded cars, whereas the maximum wheel lift occurred on one group of cars between 16 and 20 and on another group between 19 and 26 miles per hour, the corresponding cars, emptied, rocked most between 22 and 26 and some as high as 32 miles per hour.

Mr. Bertram: Well, here again, the center of gravity is lower on the lighter car which is acting like a pendulum, seated on the center plate. If you lower the center of gravity, I think your time of swaying would cut down.

Chairman Smith: Last question. Mr. Christie.

Mr. Christie: *H. C. Christie, Chicago, Rock Island & Pacific R. R. Co. Getting away from the car on this subject. Several railroads are now laying yard track in panels with a so-called square joint. Have your company and your maintenance-of-way people found any improvement in this situation if a track is laid with square joints?*

Mr. Bertram: Here again, I think this was brought up last year, if I remember right. I think if you come up with square joints you are going to face some other problems with track alignment. You may also get into a galloping condition. I think the main thing — I am no expert on laying track — believe me — I think the one thing that Mr. Hammond was most concerned about was track alignment, keeping track alignment at square joints.

Chairman Smith: The last one.

Mr. Petrie: *R. Petrie, Erie-Lackawanna Railroad Company. Were some of those tests conducted on welded rail?*

Mr. Bertram: No, not that I know of. All the tests at Hollidaysburg were on the same track, number 14 and it had the staggered 39′ joints on tangent, 33 on curve.

Chairman Smith: We can have a further discussion on this paper later this afternoon. Thank you Mr. Bertram. Meeting is adjourned for lunch.

AFTERNOON PROCEEDINGS
2:00 P. M.

Chairman Smith: I want to announce that Mr. Hart is here. He will be able to present his paper during the session following this afternoon's intermission.

The next paper is entitled, "Future Requirements for Coupling of Freight Cars" by Mr. R. D. Timpany, Assistant Vice President — Operating Administration, New York Central System. Mr. Timpany.

Mr. Timpany: Thank you very much, Loren. First, my compliments to you and your company for making possible this free exchange of information between the railroad's customer and the manufacturers. Secondly, let me say that I am completely overwhelmed by the technical competence of the people in this room. Let me hasten to say that since I know very little about the subject, I can qualify as an expert and speak with authority. Let me say also, that in our company in recent years we have been trying very hard to convert our younger people to problem solvers instead of problem staters. Today, I appear as a problem stater.

I would like to take the liberty, too, in defining as a coupler all the appurtenances that are associated in physically connecting two cars together, and as long as I have the liberty, I may go a little farther than that. It seems to me, that this coupler we have has been around for about one hundred years. It is one of the more fragile parts of a car, tends to break, tends to open, it constitutes some five to eight per cent of the weight of a car, up to 25 per cent of the cost of the car and perhaps up to 15 per cent of the total cost of repair. I notice over here we have some different type couplers. I understand there is an "E" Type Coupler that has been around for many, many years. We had a "F" Type Coupler. While it tended to solve some problems, it created some others, particularly on humps as pin pullers had trouble getting the pin. One solution is to spring the carrier iron which increases the cost of installation and also increases the maintenance.

A few years ago we bought some locomotives. We wanted to use those locomotives for general purpose so we were concerned with the lateral movement in the coupler. We bought a locomotive of that type, equipped it and it didn't take long to find out that when you hook four or five of those locomotives together and try to back a train, you tend to derail them. So we didn't solve the problem, we simply modified what we set out to do initially and prevented the coupler from moving as far laterally as it was intended to move.

I guess ever since we have had a coupler, it has had a definition of being automatic. They do pass. The incidence of passing couplers with a so-called long shank coupler has been considerably aggravated. One solution to that is a centering device. I believe you saw a model of one that is under consideration. Here again, additional expense for installation, and it would appear, some additional cost for maintenance. The long coupler too has exaggerated the problem at the hump because now we have even more trouble in getting the pin. One solution to that problem is to hinge the coupler and spring that coupler so that it can move both laterally and vertically. What are we doing, here again, to the cost of equipment and expense of maintenance?

Some of you in the East, particularly north of us, have found that when you have large blocks of cars with an energy absorbing device, you are running those things on a train, and all of a sudden you find that train out in a corn field. I think this typifies one of the problems we have in this industry. Too many times we purchase equipment before we fully understand the environment in which that equipment has to operate. We have a research project going now with a Midwestern university, and we have asked a group of youngsters out there to tell us what happens to the stresses between cars in a 200 car train. As you walked into this room this morning, you passed a computer, a little black box. These youngsters had a computer, a size again larger than the one you saw. Because of the variables in the problem, they were only able to calculate stresses in a ten car train. We were able to get them a larger computer. We hope one of these days, we will know a great deal more about what happens between these cars in a train, and at this point, we are only considering level tangent track. It has been our experience on our own railroad, particularly with a train that has several loads on the rear, and that is not unusual on any railroad, on straight tangent track to find sufficient buff action that we derailed up to 12 cars. I was amused last night in hearing of some of the old arguments between the track supervisors and master mechanics as to causes of

A.S.M.E. Technical Paper

A Criterion for the Control of 100 Ton Hopper Car Roll Motion

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March 27th, 1968.

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INTRODUCTION:

A program of experimental investigation was undertaken by the Canadian National Railways with the object of preventing 100 ton hopper cars from derailing. It has been realized for several years that 100 ton hopper cars have a tendency to derail at low speeds and that the derailments are associated with excessive car rolling. The solution to the problem requires that the roll motion be kept within tolerable limits under service conditions. The control which is needed to limit the roll motion can be achieved by appropriate modifications to the suspension system. The selection of the most suitable modification involves an engineering decision based upon an understanding of the environmental conditions. An empirical approach to the problem was taken by the Canadian National to provide the environmental data necessary to evaluate the performance of any car, with or without modifications to the suspension. This paper discusses data derived from this experimental work and presents a criterion for the control of the roll motion.

THE TEST FACILITY:

It is of paramount importance in an experimental investigation of this nature that careful attention be given throughout the test program to minimizing experimental error; for example, accurate speed control is essential.

The severity of the roll motion which can build up depends upon train speed. A critical speed exists at which the frequency of traversing rail joints corresponds to the frequency at which the car body rolls naturally. At this natural frequercy, the roll motion can be so great as to alternately lift the wheels at each side of the car every rail length. It is most important that the behaviour of the car be examined at the critical speed. A deviation of one mile per hour from the critical speed can cause a reduction of the roll angle by twenty-five percent. Therefore, poor control of speed can be responsible for apparent large differences in car performance.

Test work must be carried out on track which has been deliberately altered in order to induce large rolling motion. Most investigators have used a track shimmed to produce maximum cross-level variations of three-quarters of an inch per rail length for ten rail lengths. It was decided to use a test track of this type to relate test findings with the work of other investigators.

Examination of test results from different sources reveals many contradictions. This is due in part to difficulties involved in controlling the variables such as train speed and track surface and also to the fact that in many cases, changes in behaviour of the car motion attributed to suspension modifications are so marginal as to be within experimental error. However much larger discrepancies are associated with the choice of measurements. Comparative evaluations of various modifications have been based upon the value of the roll angle and on the incidence and magnitude of wheel lift. The work to follow will illustrate that neither of these measurements should be used for comparing one modification with another. If these parameters alone are used contradictory results will be unavoidable.

The Canadian National program of test work was carried out on track number 30 in the Turcot Yard, Montreal. This track is straight and level for its 2,500 foot length and is able to accommodate two test sections. The first was shimmed to three-quarters of an inch cross-level variation for ten rail lengths. A second

THE TEST FACILITY: (Cont'd)

section of eight rail lengths was adjusted to different profiles during the tests. A transition section of six rail lengths was left between the two sections and was not altered in any way.

Up to the present, all tests have been conducted on straight track. In the future, tests involving promising modifications will be carried out on both straight and curved track.

THE TEST VEHICLE, INSTRUMENTATION, AND TEST PROCEDURE:

All measurements were taken at the leading end of the car to allow the limited number of data channels available to record the maximum number of parameters. Test work was carried out in two separate programs; the first in the spring of 1967, the second in the autumn of 1967 and winter of 1968. The following parameters were recorded during the first test series:

1) Train speed.
2) Wheel lift of the leading wheel at each side of the car.
3) Roll angle of the car body at the center of gravity.
4) Lateral displacement of the car body at the center of gravity.
5) Vertical displacement of the car body at the center of gravity.
6) Spring displacement at each side.
7) Relative vertical displacement between the car and truck bolsters.
8) Lateral truck bolster movement relative to one side frame.
9) Vertical load on each side frame.
10) Lateral load on each side frame.
11) Lateral acceleration of the car body at the center of gravity.

Figure No. 1 shows the location at which measurements were made. A typical chart record from one test run is shown as Figure No. 2.

The test vehicle used during the first test program was car N.A.H.X. 51441, loaded with one hundred tons of potash. Table No. 1 lists the characteristics of this test vehicle, as well as others used in later tests. The same trucks were used under all vehicles.

At the beginning of each day of testing, the track profile was measured under load and the track was shimmed to the required surface. The initial test runs were made at and about the critical speed immediately following shimming. This allowed the important runs to be made with minimum deterioration of the shimmed track. In all, a speed range of ten to twenty-five miles per hour was investigated.

SUMMARY OF OBSERVATIONS FROM THE FIRST TEST PROGRAM:

The program of the first phase of test work was concerned with modifications to the truck suspension and the rail profile. Only truck suspension modifications which had shown promise during previous tests were investigated. A summary of the results is presented as Table No. 2 in terms of maximum roll angle and wheel lift. A typical graph of roll angle versus train speed is shown as Figure No. 3. This table and figure illustrate the type of information which has usually been collected during roll motion tests. It can be used to compare the results of Canadian National tests with results from other sources.

SUMMARY OF OBSERVATIONS FROM THE FIRST TEST PROGRAM: (Cont'd)

General conclusions can be drawn from this form of presentation of results, but comparison between the various modifications becomes difficult. For example, the maximum roll angle which can be tolerated depends upon the nature of the modification. Long travel springs and friction snubber side bearings permit a larger roll angle before the springs go solid. It will be shown later that the magnitude of wheel lift does not have much significance when considered as a single parameter. When wheel lift does not occur, a summary of results, such as given in Table No. 2 does not indicate how close the car is to wheel lift. Specifically, it does not provide information concerning the degree of protection offered by any modification.

A technique of analysis was developed following the completion of the first test program which can provide information of a quantitative nature for making comparison of relative improvement and of the sufficiency of any particular modification.

METHOD OF ANALYSIS:

The analysis is concerned with the behaviour of the car during the particular cycle of oscillation for any test run which exhibits the maximum roll angle. During this cycle, one or more instants exist at which the forces applied to the side frames are a maximum. The analysis is based upon the forces acting on the car at these instants.

The reaction between the wheel flanges on one side of the car and the rail creates a fulcrum and moments are induced about the rail head. The weight of the car and the roll oscillation of the car cause forces to be applied to the unsprung portion of the trucks. These dynamic forces have been measured in both the vertical and lateral directions for both side frames of the leading truck. This measured data is the raw information used in the analysis.

Each of the loads applied to the unsprung components of the truck and the unsprung weight, contribute to moments about the rail head. There will be moments tending to overturn the car about the rail head and moments tending to stabilize the car. The ratio $\frac{\text{total overturning moments}}{\text{total stabilizing moments}}$, which is called the Moment Ratio, is of considerable interest and forms the basis of the analysis. The maximum numerical value of this ratio for each series is a meaningful measure of instability; for example, if this ratio is greater than one, the wheels will tend to lift off the rail.

Figure No. 4 is a schematic sketch of the unsprung portion of the truck. This sketch shows the forces acting on the side frame and also the moment of each force about the rail head. The method of calculating the Moment Ratio is shown by the sample calculation sheet of Figure No. 5.

The first series of tests were analysed by selecting at least three runs from each series, representing a run at the critical speed, a run at a stable speed and also a run exhibiting a behaviour between the first two. The values of the peak vertical and lateral forces acting on the side frames were measured from the chart records during the cycle of maximum roll angle for each run and these values were substituted in the calculation sheet of Figure No. 5. All force measurements must be made at the same instant. When more than one instant of peak force exists, as illustrated by Figure No. 2, the Moment Ratio is calculated for each and the instant yielding the highest Moment Ratio is used. Each vertical force in this calculation is the

METHOD OF ANALYSIS: (Cont'd)

static weight plus the measured dynamic load.

The technique used to determine the Moment Ratio can best be illustrated by means of an example. The test series representing the evaluation of the car in an "as received" condition, but with an additional three-eighth inch outer bolster gib clearance will be used for this purpose. Figure No. 5 shows a completed calculation sheet for one test run from this series.

A calculation of this type can be used to determine the magnitude of the component moments at any value of the Moment Ratio. This can be accomplished by first preparing a graph showing the distribution of each individual moment versus the Moment Ratio for all runs from a test series. Such a graph is shown as Figure No. 6 and is simply a visual presentation of the results of Figure No. 5, but including all the test runs. This graph allows the distribution of moments to be examined at any value of the Moment Ratio. For example, at the value of this ratio equal to 0.5, the magnitude of each component moment can be determined. The composition of these moments at this ratio can best be shown by means of a bar graph.

Figure No. 7 is a bar graph of this type which shows the composition of the moments for a Moment Ratio equal to 0.5, 1.0, 1.25, 1.50, 1.75 and for the highest measured value.

A bar graph, such as shown in Figure No. 7, is useful for several purposes. The maximum value of the Moment Ratio is a measure of the performance of the modification tested and its numerical value provides a measure of the propensity of the car to wheel lift. If this ratio is less than one, there will be little likelihood of the wheels lifting. When the ratio is greater than one, wheel lift will likely occur. The smaller the numerical value of the maximum ratio, the better the performance of the car.

The value of each component moment, as shown by the bar graph, is also of interest. It indicates the relative importance of the stabilizing and overturning effect due to vertical side frame forces and similar effects for the lateral forces transmitted to the side frames from the bolster. A modification to the suspension can be judged in terms of its overall improvement and it can also be determined if the improvement occurs as a result of equalizing the spring forces, or from minimizing the lateral forces, or from both. The value of presenting results in this way is made more evident when various test conditions are compared.

SUMMARY OF RESULTS FROM THE FIRST SERIES OF TESTS:

Figure No. 8 shows the composition of the stabilizing moments and overturning moments at the highest measured value of the Moment Ratio for several of the suspension modifications tried. The numerical value of the highest ratio shows that three modifications produced considerable improvement. They are the use of 3-11/16 inch travel springs with double springs under the friction wedges, the use of two friction snubbers per nest and a friction damper as a side bearing. However, even for the two modifications which prevented wheel lift, the Moment Ratio was measured to be approximately one and the wheels must have been on the verge of lifting. The performance of the modifications tried during the first phase of testing showed that not one offered the degree of improvement required of the "solution".

SUMMARY OF RESULTS FROM THE FIRST SERIES OF TESTS: (Cont'd)

It is interesting to consider the influence of track surface on the Moment Ratio. This information is given as Figure No. 9. Stability of the car can be greatly increased by improving the surface of the track as can be seen by the reduction in the Moment Ratio achieved by five-eighth and three-eighth inch cross-level variation with mid-point stagger. An improvement was also found by using a seventy-eight foot rail length with quarter-point stagger instead of a thirty-nine foot rail with mid-point stagger. Third-point stagger offered an improvement over mid-point stagger, but the improvement was judged insufficient. This result is in agreement with the theoretical studies carried out by Messrs. W.P. Manos and J.C. Shang[1].

THE SECOND SERIES OF TESTS:

Experience gained while analyzing results from the first series of tests led to instrumentation revisions for the second series. Side frame force measurements were made at two locations for each vertical and lateral force with strain gauges located on each side of the bolster pocket. Also, the static car weight was determined by weighing each end of the car. This allowed for a more accurate measurement of the static load at the instrumented end. This additional accuracy, combined with the more detailed instrumentation used to measure forces allowed the Moment Ratio to be calculated with greater precision for the second series of tests, especially at large roll angles.

The result of the additional precision is a reduction of the maximum Moment Ratio when compared with results from the first series of tests and an improved correlation of wheel lift and Moment Ratio. However, the ranking of relative improvements offered by the modifications tested during the first series of tests is valid. Several of the modifications tried during early tests were repeated during the second series of tests with the more detailed force measurements.

The following parameters were recorded during the second phase of test work:

1) Train speed.
2) Wheel lift of the leading wheel at each side of the car.
3) Roll angle of the car body at the center of gravity.
4) Lateral displacement of the car body at the center of gravity.
5) Vertical load on each side frame (measured at two locations).
6) Lateral load on each side frame (measured at two locations).
7) Displacement of hydraulic shock absorber.
8) Damping force developed by hydraulic shock absorber.

The location of the measuring instruments was identical to that shown as Figure No. 1 except that the instruments measuring shock absorber characteristics were mounted on the shock absorber.

The profile of the track was kept at three-quarter inch cross-level variation for both test sections. The technique of shimming the track and the test procedure was similar to the first phase of test work.

1. Dynamic Analysis of Rolling Freight Cars. A.S.M.E. Paper 65-WA/RR-8.

PRESENTATION OF RESULTS FROM THE SECOND SERIES OF TESTS:

The second phase of testing was concerned with the evaluation of hydraulic shock absorbers as well as with friction snubbers. Two test vehicles were used, car CN 370640 and CN 373041 with characteristics as listed in Table No. 1. These cars represent the two types of 100 ton hopper cars operated on Canadian National lines. Both cars behaved similarly during the tests, hence discussion here will be limited to the tests carried out on car CN 373041 which is the three hopper design, similar to the N.A.H.X. car tested earlier.

A more thorough analysis was made using the results from the second series of tests. In particular, the value of the Moment Ratio was calculated for each run and plotted against train speed. Figure No. 10 illustrates typical results and shows a graph of Moment Ratio versus Train Speed for three different suspension systems. Each point represents a test run and any run during which wheel lift occurred is identified. It is obvious from these graphs that wheel lift is associated with a Moment Ratio of one and that wheel lift can occur even with 3-11/16 inch travel springs and double coils under the snubber wedge. The likelihood of wheel lift occurring is reduced, as evidenced by the smaller values of the Moment Ratio and the narrower band of speed during which the Ratio of one is exceeded.

The relationship between Moment Ratio and Train Speed is an important one and serves as a measure by which the critical speed range and likelihood of wheel lift can be determined for any suspension system. These curves represent maximum points for individual test runs and are similar to the steady state resonance curves of vibration theory. They do not provide information concerning the rate of build-up of the Moment Ratio for any test run. The rate of build-up is also important because it gives a measure of the number of consecutive bad rail joints required to cause wheel lift.

For each of the three suspensions of Figure No. 10, the test run which exhibited the most rapid build-up of the roll angle was analyzed in detail. The Moment Ratio was calculated for each cycle of oscillation throughout the test section and in this way, the rate of build-up of the Moment Ratio was determined. Figure No. 11 is a plot of the Moment Ratio versus the Number of Rail lengths from the start of the test section.

This information can be of particular importance when two or more modifications are being considered which yield similar maximum values of the Moment Ratio, especially if the maximum values are in the order of one. Figure No. 11 shows that all three suspensions require two and one-half rail lengths of poor track, with three-quarter inch cross-level variation, before wheel lift will occur at the critical speed. With long travel springs the Moment Ratio builds up less rapidly than with the standard two and one-half inch stabilized suspension, but the 3-11/16 inch springs with single side springs yields a higher value of Moment Ratio than the conventional two and one-half inch springs. The 3-11/16 inch travel springs with double side springs approached a value of one within the same number of rail lengths, but tended to stabilize at that value and only occasionally exceeded a value of one.

The likelihood of derailment as a result of severe rolling would definitely be reduced with 3-11/16 inch travel springs and double side springs but this modification does not offer a sufficient improvement to rule out completely the possibility of derailments.

PRESENTATION OF RESULTS FROM THE SECOND SERIES OF TESTS: (Cont'd)

Contradictions have been reported from different sources from tests involving modifications such as the long travel springs with double side springs. It would appear that when contradictions occur, the modification is likely yielding a Moment Ratio in the order of one. This is certainly the case with the above modification. When the ratio approaches one, wheel lift may or may not occur. Test data has shown that with this particular modification, occasionally wheel lift has occurred on only one of the two test sections. Table No. 3 summarizes the results from a test run with long travel springs and double side sorings. It can be seen from this Table that wheel lift did not occur with a Moment Ratio of 1.02 and a roll angle of six and one-half degrees but the wheels lifted one inch, two rail lengths later, when the roll angle was six and three-quarter degrees and the Moment Ratio was 1.03. The forces developed changed little, yet the performance as judged by wheel lift alone would be significantly altered. This Table illustrates the fact that when the Moment Ratio is in the order of one, the magnitude of wheel lift has little significance as a comparative measure and can only lead to contradictions from one test to another.

Examination of the build-up of roll angle, wheel lift and Moment Ratio for each cycle of oscillation during a test run often reveals a phenomenon in which the motion builds up, decays and builds up again within ten rail lengths of the test section. This behaviour is evidenced in the curves of Figure No. 11. This behaviour is likely due to the fact that when wheel lift occurs, the car body takes a different time period to complete a cycle of oscillation than when the car is rolling but not lifting wheels. The most severe rolling occurs at the speed when the frequency of traversing rail joints corresponds to the frequency at which the car body rolls naturally on its springs. When wheel lift occurs, the complementary relationship between the body rolling and the rail joints is disturbed and the track irregularities are temporarily prevented from causing further build-up.

THE USE OF HYDRAULIC SHOCK ABSORBERS:

Hydraulic shock absorbers have been used to control the build-up of 100 ton hopper car roll motion with considerable success. It was therefore decided to investigate the performance of hydraulic dampers during the second phase of test work. This aspect of the study is still in progress, however, one part of this test work will be reported here.

Adjustable hydraulic shock absorbers were mounted at each corner of the test vehicle between the side frame and the car body. The shock absorbers were mounted vertically and were used on a truck with 3-11/16 inch travel springs.

Tests were conducted with three settings on the shock absorbers, called soft, mid and hard. The Moment Ratio was calculated for each test run and these results are presented as Figure No. 12, which is a series of curves showing the relationship between Moment Ratio and Train Speed for the three shock absorber settings. Again the rate of build-up of the Moment Ratio was determined for the test run which exhibited maximum build-up of the roll angle. This result is shown graphically as Figure No. 13.

The degree of improvement obtained with the shock absorbers on the hard setting shows clearly that the problem of excessive rolling of hopper cars can be controlled on straight track with commercially available shock absorbers. The maximum Moment Ratio measured with the shock absorbers on the hard setting was 0.74 with a three-quarter inch cross-level deviation and the roll angle was kept to a maximum value of four degrees.

THE USE OF HYDRAULIC SHOCK ABSORBERS: (Cont'd)

Damping force and shock absorber displacement were recorded during the test runs and were analyzed during the cycle of maximum build-up of roll motion. The hysteresis loops are presented as Figure No. 14 which shows the energy dissipated at each corner by the shock absorbers on the soft, mid and hard settings.

Care must be taken in interpreting the results because the shock absorbers actually dissipated more energy when on the soft setting. This occurs because a much larger roll angle is permitted to build up before peak amplitudes are reached. For example, the maximum roll angle recorded for the soft, mid and hard settings was seven, five and one-quarter and four degrees respectively. The maximum shock absorber displacements were four and five-eighths, four, and two and three-quarter inches respectively.

It appears that in order to be most effective, the shock absorber should develop its peak damping force within a small travel and should have a relief feature which prevents excessive build up of damping force. This would allow the most efficient use to be made of the small travel by providing a rectangular shaped hysteresis loop.

When the soft and mid settings were used on the shock absorber, the hysteresis loop deviated from the rectangular shape and tended to approach a shape more typical of viscous damping. Although the shock absorber developed high damping forces at these settings the high forces were not available for the full travel. The absorbers only became effective when relatively large motion was already present and were of little use for small roll angles.

The result is the apparent anomaly that a shock absorber on the soft setting dissipates as much or more energy per cycle as a shock absorber on the hard setting. It must be remembered however, that in the example given, the shock absorber maintained a maximum Moment Ratio of 0.74 and roll angle of four degrees on the hard setting and allowed for values of 1.19 and seven degrees respectively on the soft setting.

FUTURE WORK:

It would be desirable if meaningful generalizations could be made regarding the requirements of hydraulic shock absorbers. Much more empirical data is required before this can be attempted, and to this end the Canadian National plan to continue the investigation of hydraulic damping in detail. Specifically, absorbers will be tried across the main springs and between the side frame and car body for trucks with two and one-half inch and 3-11/16 inch travel springs.

Test work will be carried out on curved track with superelevation as well as on straight track. In addition tests will be run to evaluate the influence of promising modifications throughout the full operating speed range.

While test work is time consuming and expensive, it is desirable that proprietory items be tested and evaluated. Other railroads and agencies are presently cooperating in carrying out experimental work. However, a standardized test procedure should be adopted and results should be analyzed in the same way. The criterion presented here could form the basis of a standardized test which would allow results from different sources to be compared directly.

CONCLUSIONS:

The main purpose of this paper is to introduce a technique of analyzing the roll motion of heavy cars. The Moment Ratio is presented as being a realistic yardstick which can easily be calculated from measurements which are reasonably simple to make. The Moment Ratio, as a criterion, can be used for the following purposes:

1) The highest value of the Moment Ratio can be used as a basis for ranking the effectiveness of various suspension modifications. The highest value of Moment Ratio can also be used to judge the sufficiency of any modification.

2) The speed range over which wheel lift can occur can be determined by examining the relation between train speed and Moment Ratio. The critical speed is the range during which the Moment Ratio is greater than one.

3) The number of consecutive track irregularities needed to cause wheel lift can be found by analyzing the build-up of Moment Ratio throughout the test track. This can be used as a measure of the likelihood of wheel lift occurring in service.

4) Examination of the composition of the stabilizing and overturning moments used to calculate Moment Ratio can provide information concerning how an improvement was achieved.

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Mr. A. Charenko, Engineer, Technical Research Branch, Research and Development.
Mr. D. Lawless, Engineering Assistant, Technical Research Branch, Research and Development.

Table 1

CHARACTERISTICS OF THE TEST VEHICLES

	Test Vehicles		
	KAHX 51141	GN 370640	GN 373041
Light Weight (lbs.)	62,100	57,600	58,100
Total Weight (lbs.)	260,860	259,420	260,900
Load Weight (lbs.)	198,760	201,821	202,800
Sprung Weight (lbs.)	247,000	245,500	247,000
Sprung Weight at Instrumented End (lbs.)	* 120,000	126,000	130,000
Truck Center Distance	41.3	41.3	41.3
Number of Compartments	3	4	3
Density of Potash Load (lbs./ft.3)	72	74	74
Center of Gravity of Sprung Weight from Top of Rail (ins.)	94	91	95

* Estimated for KAHX car.

- Table 3

SUMMARY OF RESULTS FROM ONE TEST RUN

3-11/16 inch Spring Travel, Double Side Springs under Snubber Wedges.

*Cycle Number	Roll Angle (degrees)	Wheel Lift (inches)	Moment Ratio
1/2	2-1/2	0	0.31
1-1/2	4-1/2	0	0.53
2-1/2	5	0	0.80
3-1/2	6	0	0.89
4-1/2	6-1/2	0	1.02
5-1/2	6-1/2	Verging	1.03
6-1/2	6-3/4	1	1.03
7-1/2	7-1/2	1-1/4	1.10
8-1/2	7-1/2	1-3/4	1.15
9-1/2	6-1/2	1	1.01
10-1/2	3-1/2	0	0.55

*Number of roll motion cycles from start of test section.

TABLE 2

SUMMARY OF RESULTS

TEST OF 100 TON HOPPER CAR NARX 51111

DESCRIPTION OF CAR		DESCRIPTION OF TRACK	CRITICAL SPEED (M.P.H.)	WHEEL LIFT	MAXIMUM ROLL ANGLE
Test Series "A"					
"As Received" but with 3/8" increase in outside gib clearance.		3/4" cross level	18	multiple lifts 2-1/2" maximum	8-1/2°
Test Series "B"					
2-1/2" travel springs, double side springs under snubber wedges, outside bolster gib clearance increased 3/8".		3/4" cross level	19	multiple lifts 2-1/4" maximum	7-1/2°
Test Series "C"					
2-1/2" travel springs, two friction snubbers per nest, outside bolster gib clearance increased 3/8".		3/4" cross level	19	multiple lifts 2" maximum	7°
Test Series "D"					
3-11/16" travel springs, double side springs under snubber wedges, outside bolster gib clearance increased 3/8".		3/4" cross level	18	nil	5°
Test Series "E"					
3-11/16" travel springs, two friction snubbers per nest, outside bolster gib clearance increased 3/8".		3/4" cross level	19	multiple lifts	5-1/2°
Test Series "F"					
3-11/16" travel springs, single side spring under snubber wedges, outside gib clearance increased 3/8".		3/4" cross level	16	multiple lifts 2" maximum	8-1/2°
Test Series "G"					
2-1/2" travel springs, friction side bearing, outside gib clearance increased 3/8".		3/4" cross level	12.5	nil	8°
Test Series "H"					
2-1/2" travel springs, single side spring under snubber wedges, normal bolster gib clearance.	H-1	39' rail, 1/2 point stagger, 3/4" cross level	18.5	multiple lifts 2-1/2" maximum	8°
	H-2	78' rail, with 1/4 point stagger, 3/4" cross level.	20.5	nil	4°
	H-3	39' rail, 1/2 point stagger, 3/8" cross level.	18.5	nil	4°
	H-4	39' rail, mid point stagger, 5/8" cross level.	20	several slight lifts.	4-1/2°
	H-5	39' rail, 1/3 point stagger, 3/4" cross level.	19	slight lifts.	5°

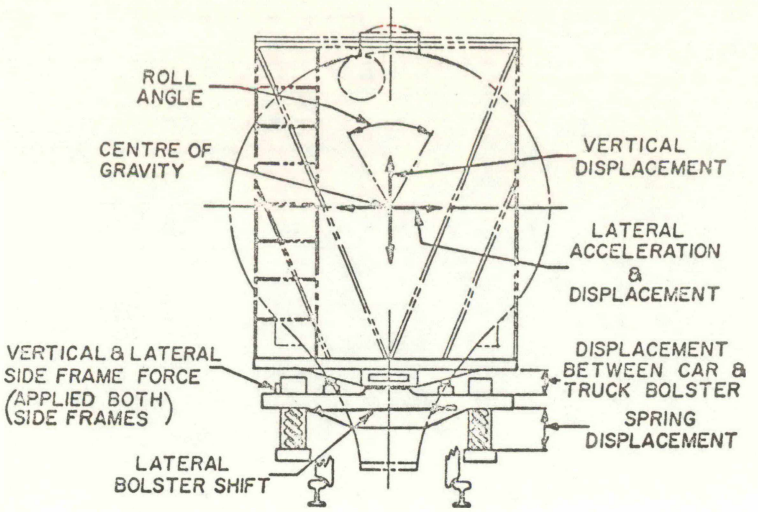
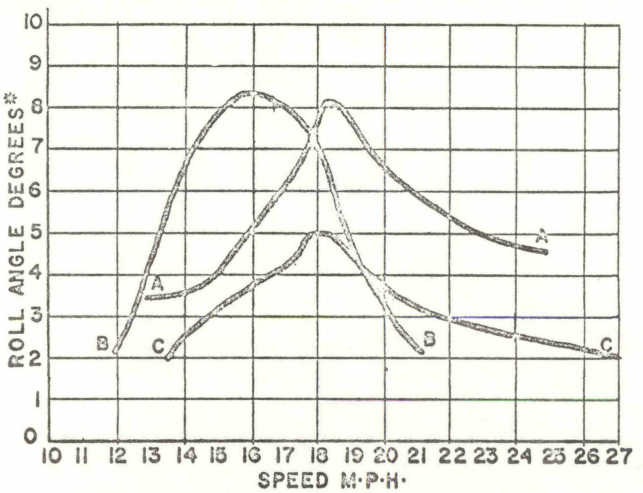


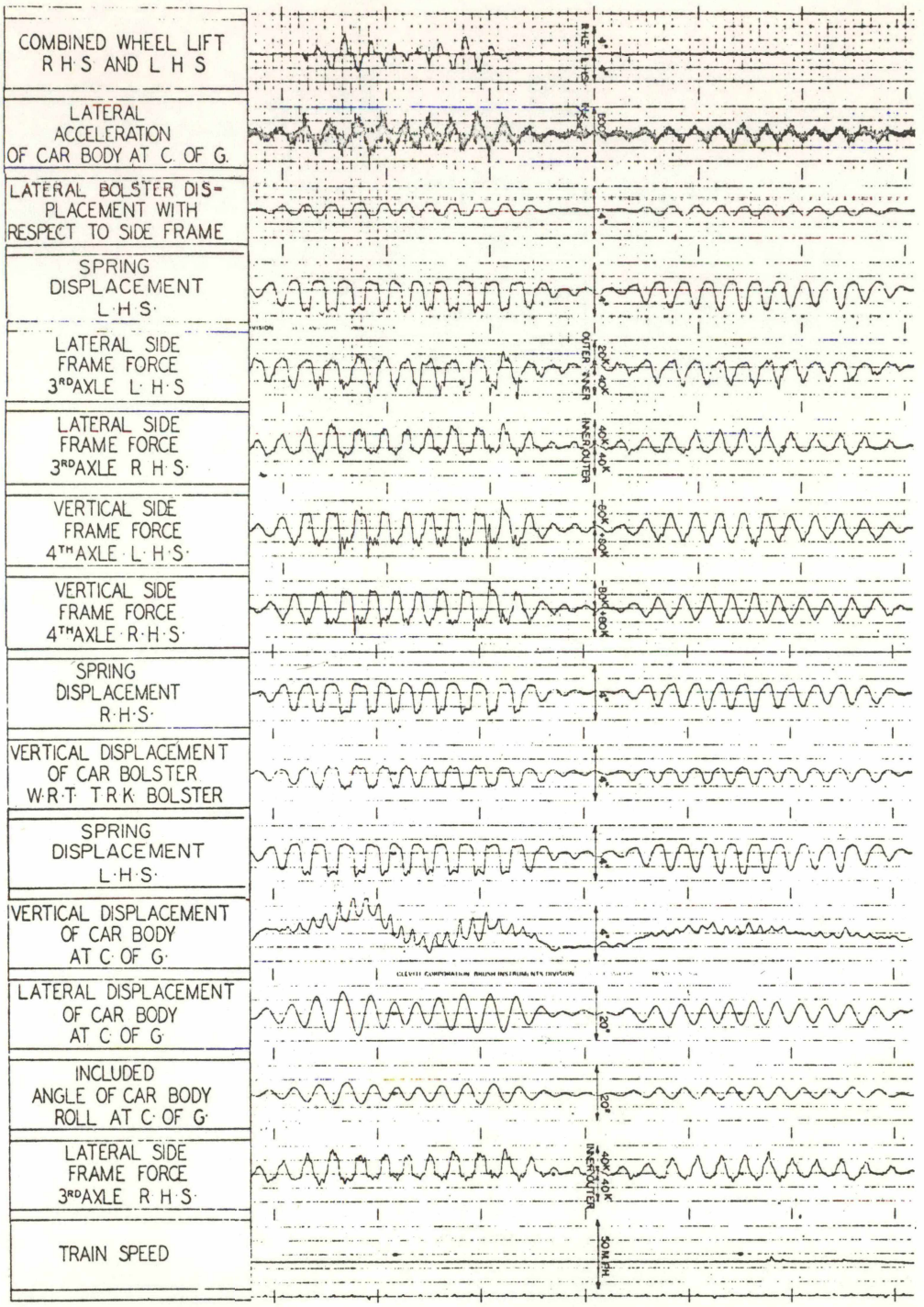
Fig. 1 LOCATION OF MEASUREMENTS.



"A" $2\frac{1}{2}$" TRAVEL SPRINGS, SINGLE SIDE SPRINGS·
"B" $3\frac{11}{16}$" TRAVEL SPRINGS, SINGLE SIDE SPRINGS·
"C" $3\frac{11}{16}$" TRAVEL SPRINGS, DOUBLE SIDE SPRINGS·

* INCLUDED ANGLE

Fig. 3 TYPICAL GRAPH OF ROLL ANGLE VERSUS TRAIN SPEED



NAHX 51441 TEST SERIES H-4

— TRUCK DESCRIPTION —

$2\frac{1}{2}$ INCH TRAVEL SPRINGS
SINGLE COIL SIDE SPRINGS
STANDARD BOLSTER GIB CLEARANCE

— TRACK DESCRIPTION —

	1st SECTION	2nd SECTION
RAIL LENGTH	39 FEET	39 FEET
RAIL STAGGER	MID POINT	MID POINT
CROSS LEVEL DEVIATION	$\frac{3}{4}$ INCH	$\frac{5}{8}$ INCH

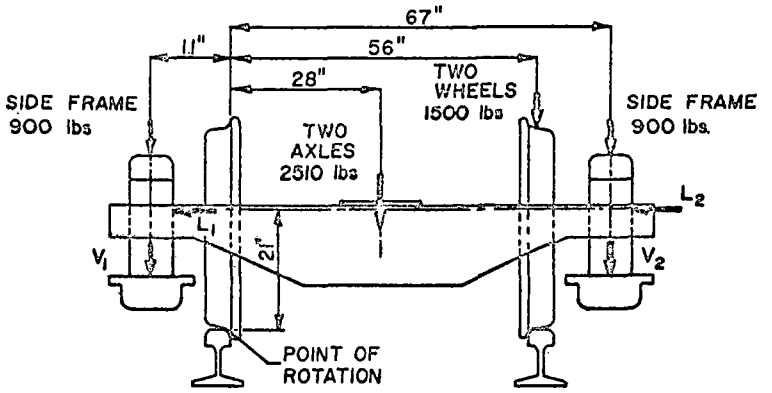


Fig. 4 SKETCH OF UNSPRUNG PORTION OF TRUCK.

MOMENTS ABOUT THE RAIL DURING THE CYCLE OF MAXIMUM ROLL ANGLE

TEST SERIES A RUN NO. 13

OVERTURNING MOMENTS — IN. LBS.		STABILIZING MOMENTS — IN. LBS.	
SIDE FRAME 900X11"	9,900	VERTICAL FORCE (V_2-STATIC LOAD)X67" = 60000-42,000 X67"	1206,000
LATERAL FORCES L_1 X 21" = 56000 X21" =	756,000	SIDE FRAME 900 X67"	60,300
L_2 X21" = 42,000 X21" =	126,000	AXLES 2,510 X25"	62,750
VERTICAL FORCE (V_1+STATIC LOAD)X11" = 60000+42,000X11"	1,122,000	WHEELS 1,500 X56"	84,000
TOTAL	2013,900	TOTAL	1413,050

MOMENT RATIO = $\frac{\text{TOTAL OVERTURNING MOMENTS}}{\text{TOTAL STABILIZING MOMENTS}}$

$= \frac{2,013,900}{1,413,050}$

$= 1.43$

Fig. 5 SAMPLE CALCULATION SHEET

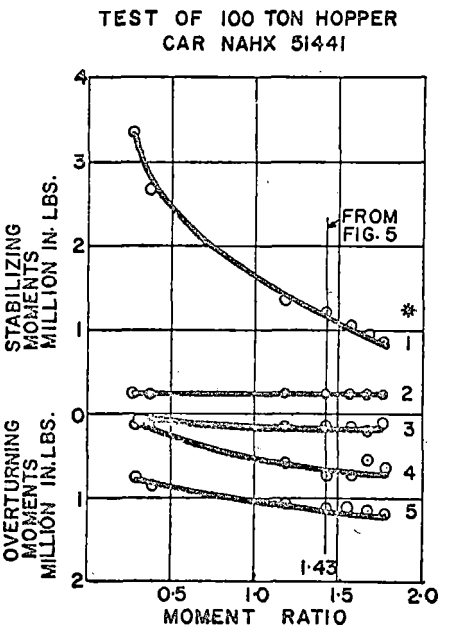


Fig. 6 COMPOSITION OF STABILIZING AND OVERTURNING MOMENTS

* SEE LEGEND ON PAGE 14

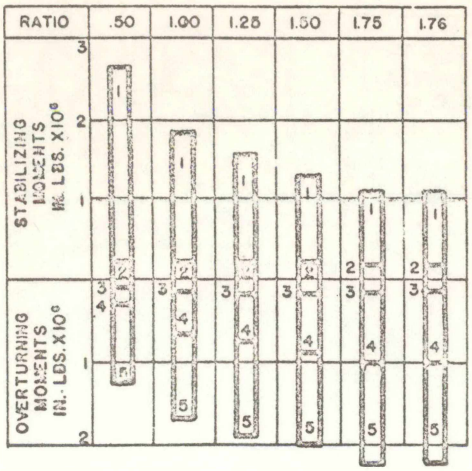


Fig. 7 BAR GRAPH SHOWING COMPOSITION OF STABILIZING AND OVERTURNING MOMENTS

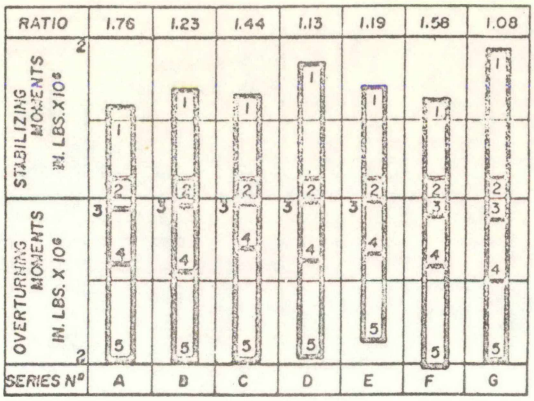


Fig. 8 BAR GRAPH FOR VARIOUS SUSPENSION MODIFICATION

LEGEND

1. REPRESENTS MOMENTS DUE TO STABILIZING SPRING FORCES
2. REPRESENTS MOMENTS DUE TO THE WEIGHT OF TRUCK COMPONENTS
3. REPRESENTS MOMENTS DUE TO LATERAL FORCES ON THE UNLOADED SIDE FRAME
4. REPRESENTS MOMENTS DUE TO LATERAL FORCES ON THE LOADED SIDE FRAME
5. REPRESENTS MOMENTS DUE TO OVERTURNING SPRING FORCES

SERIES NUMBERS ARE AS DESCRIBED IN TABLE 2 ON PAGE 10

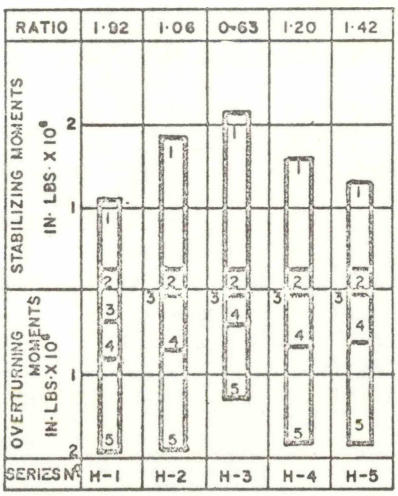
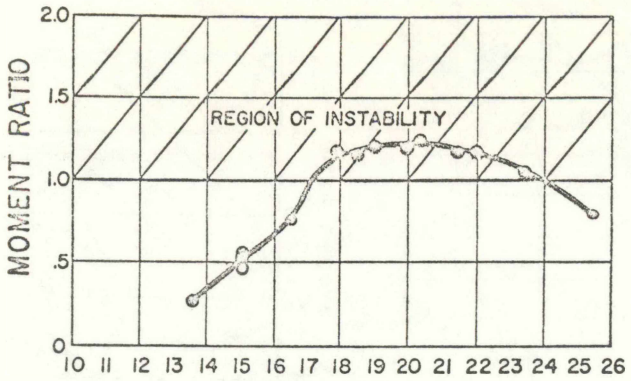
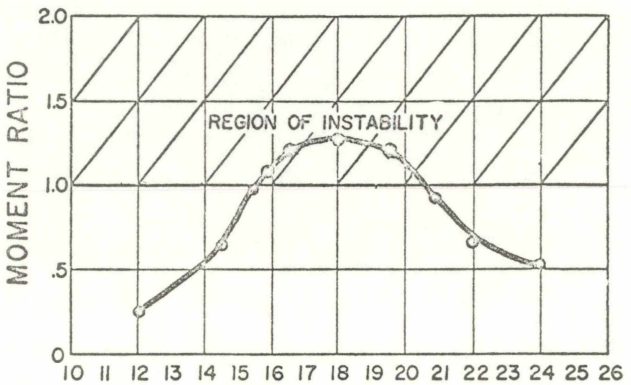


Fig. 9 BAR GRAPH FOR TRACK MODIFICATIONS

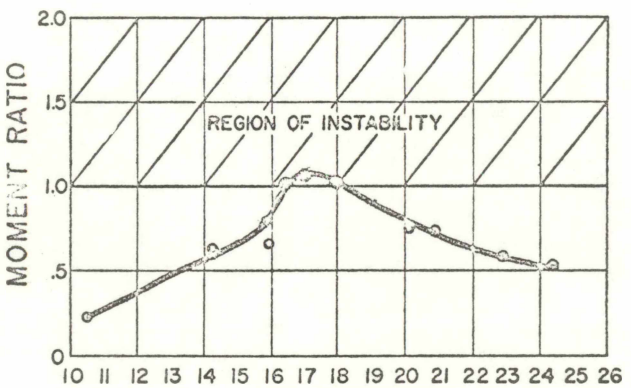
TEST OF 100 TON HOPPER CAR CN 373041



$2\frac{1}{2}$" TRAVEL SPRINGS, SINGLE SIDE SPRINGS.



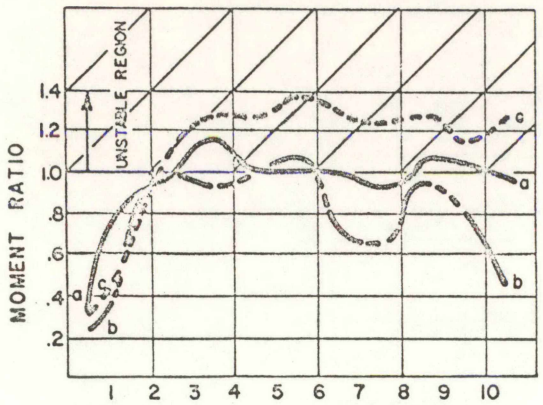
$3\frac{11}{16}$" TRAVEL SPRINGS, SINGLE SIDE SPRINGS.



$3\frac{11}{16}$" TRAVEL SPRINGS, DOUBLE SIDE SPRINGS.

Fig. 10 GRAPHS OF MOMENT RATIO VERSUS TRAIN SPEED FOR THREE SUSPESIONS.

O – INDICATES WHEEL LIFT OCCURED.



a - 2 1/2" TRAVEL SPRINGS SINGLE SIDE SPRINGS
b - 3 1/16" TRAVEL SPRINGS DOUBLE SIDE SPRINGS
c - 3 1/16" TRAVEL SPRINGS SINGLE SIDE SPRINGS

Fig. 11 GRAPH OF MOMENT RATIO BUILD-UP FOR THREE SUSPENSIONS.

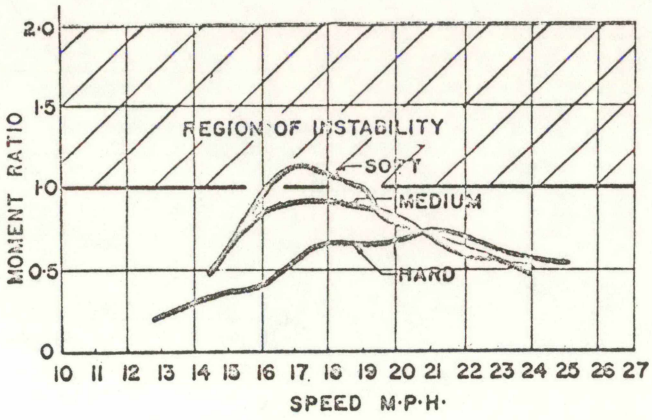


Fig. 12 GRAPH OF MOMENT RATIO VERSUS TRAIN SPEED FOR THREE HYDRAULIC SHOCK ABSORBER SETTINGS.

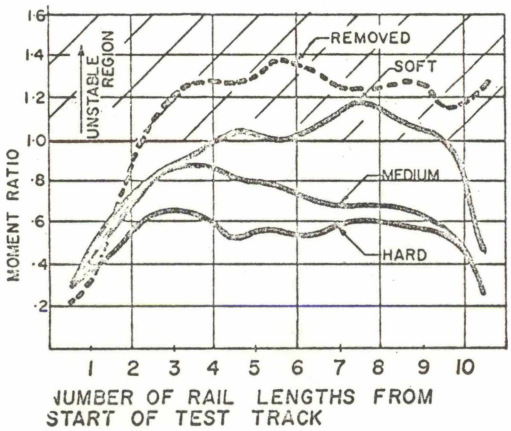
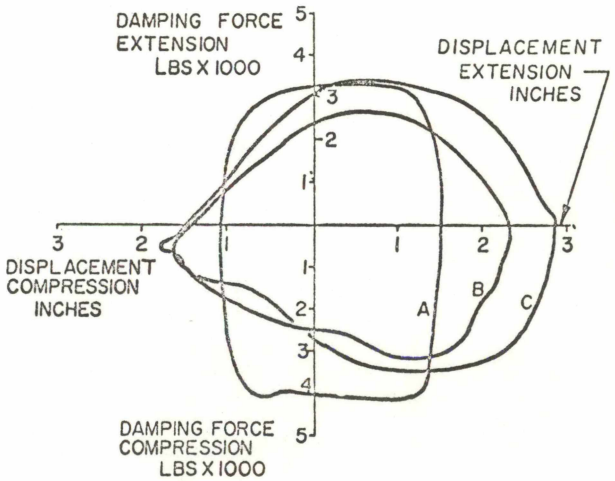


Fig. 13 GRAPH OF MOMENT RATIO VERSUS TRAIN SPEED USING HYDRAULIC SHOCK ABSORBERS



A-SHOCK ABSORBER ON HARD SETTING 17,260 IN. LBS.
B-SHOCK ABSORBER ON MID SETTING 15,700 IN. LBS.
C-SHOCK ABSORBER ON SOFT SETTING 21,200 IN. LBS.

Fig. 14 HYSTERESIS LOOPS FOR THREE SETTINGS OF THE HYDRAULIC SHOCK ABSORBER

There are no regulations in force pertaining to the methods under Rules or Laws. The methods generally employed at present will be explained here.

The wheel load and lateral pressure of rolling stock are measured by the wire strain gauge and read out from oscillograph. The various measured values given below are determined by the limits as explained below.

The first limit is the value indicating the necessity of controlling so it shall not be exceeded; the second limit is the target value desired to be attained through further improvement.

1. Allowable limit for the decrease in static wheel load

The allowable limit for the decrease in static wheel load, $\Delta P a \ell$, of a non-bogie car, attributable to torsion of the track, is determined for the average wheel load, $\overline{P}$, as follows:

$$\frac{\Delta P a \ell}{\overline{P}} = 0.6$$

This decrease in wheel load is attributable largely to the horizontal bias of the car's center of gravity (including its load), friction of the suspension spring and torsion of the track and the rolling stock. The sum of all the decreases seen in wheel load must be kept within the above-mentioned limit.

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2. Allowable limit for the decrease in dynamic wheel load

The allowable limit for the decrease in dynamic wheel load caused by the rolling stock vibration and others is set as shown in Fig. 1.

3. Allowable limit for lateral pressure of rolling stock

When the lateral pressure and wheel load that the rolling stock acts on the track are assumed to be Q_1, Q_2 and P_1 and P_2, respectively, as shown in Fig. 2, and when Q_1 and Q_2 are called as the lateral pressure of the wheel, $Q_1 \sim Q_2$ (taking the direction indicated by the arrow to be the positive one) as the lateral pressure of wheel axles, and these two together as the lateral pressure of the rolling stock, the following limits need be observed, taking track fracture into consideration:

For the lateral pressure of wheel --

The first limit:

$$Q \leqq 2.9 + 0.30P \text{ (with tieplates)}$$

$$Q \leqq 1.8 + 0.30P \text{ (without tieplates)}$$

The second limit:

$$Q \leqq 1.9 + 0.30P \text{ (with tieplates)}$$

$$Q \leqq 1.2 + 0.30P \text{ (without tieplates)}$$

For the lateral pressure of wheel axles --

The first limit:

$$Q_1 \sim Q_2 \leqq 1 + 0.35 (P_1 + P_2)$$

The second limit:

$$Q_1 \sim Q_2 \leqq 0.85 \{1 + 0.35 (P_1 + P_2)\}$$

These relations are illustrated in Figs. 3 and 4.

4. Allowable limit for the ratio of lateral pressure and wheel load

For the allowable limit for the ratio of lateral pressure and wheel load (Q/P) against derailment, the standard shown in Fig. 5 is adopted in consideration of climbing or jumping of wheel-flange on rail.

The value of Q/P can be sought by measuring the lateral pressure Q and wheel load P. At present, there are two ways of measuring Q. The standards applicable are decided according to the respective cases.

The application standard (I) shown in Fig. 6 is for the measuring of lateral pressure continuously. In this, the acting time of lateral pressure is taken into consideration, and the lateral pressure acting momentarily below 1/20 second (the boundary time between the derailments due to climbing and jumping of wheel-flange on rail shown in Fig. 5) is discarded. A maximum value of Q/P is then taken for every section of 500 m (one in the section in case of a curved track less than 500 m or a turnout included) and this is applied when the cumulative frequency is expressed for every 0.05.

The application standard (II) shown in Fig. 7 is for the measuring of lateral pressure intermittently (twice during one wheel revolution). The Q/P value is sought for a lateral pressure exceeding a given value and for the wheel load falling below a given value, and this is used when the frequency per km is expressed for every 0.05. In this case, the shock-like lateral pressure is included.

5. Allowable limit for the wheel-rail colliding speed

The allowable limit for the wheel-rail colliding speed is determined by derailment caused by the jumping of wheel-flange on hitting the rail. The limit is set as is shown in Fig. 8.

6. Allowable limit for the horizontal vibration acceleration of the carbody

The horizontal vibration acceleration of the carbody is measured with an acceleration gauge, 9 to 12 Hz in natural frequency, on the surface of the front or the rear axle of a non-bogie car.

The acceleration will have to be as follows:

0.6g in total amplitude

0.4g in total amplitude when only a vibration of less than 2 Hz is read out

When this standard is to be applied, the limit is set as shown in Fig. 9, taking the frequency distribution of the vibration into consideration.

7. For the running speed

In seeking an ordinary vibration acceleration of the rolling stock tested and its relation to the running speed, the maximum value of the vibration acceleration for every 100 m or 500 m and the running speed at the time are read out, as in the case of lateral pressure and Q/P, and a graph is made of the readings of the vibration acceleration on the axis of ordinate and those of running speed on the axis of abscissa. Furthermore, to

make them numerically plain, the mean value of the vibration acceleration and the standard deflection are sought for each running speed to be shown in a graph for the running speed.

8. Deliberation on the wave form

The vibration acceleration of the testing car can be grasped as explained in the preceding paragraph. Should, however, the vibration acceleration be large, it will become necessary to see whether it has the characteristic of affecting the running safety, and if it has, what sort of vibration is accountable for this characteristic. What is to be done for this is to carefully observe the wave form. Thinkable as one way of doing this is frequency analysis.

In this analysis, those waves similar in amplitude and linked together continuously are picked out from the vibration wave form at a similar cycle, and their frequencies, amplitudes and running speeds at the time are read out. Then, as shown in Fig. 10, the frequencies are graphed on the axis of ordinate and the running speeds on the axis of abscissa. The amplitudes are illustrated in the Fig. by circles to indicate their sizes.

This enables us to find out the natural frequencies of a carbody and the truck and the vibration caused by wheel rotation and to determine the steps to be taken for improvement.

9. Judgment on riding quality

The riding quality in connection with vibration depends upon the vibration frequencies and the vibration acceleration. At present, the riding quality standard is set as shown in Fig. 11,

in which (1) is rated excellent, (2) good, (3) ordinary, (4) bad and (5) very bad.

In determining the riding quality from the vibration records, those that are similar in frequency and amplitude and linked together continuously are picked out, and the readings of their frequencies and amplitudes are used to determine the riding quality based on the criteria shown in Fig. 11. The amplitudes in Fig. 11 are one-side amplitudes.

10. Others

Besides the points explained so far, scrupulous studies are made on the center of gravity of the carbody, the location of the flexibility center, spring constant of carbody support, carbody deflection, truck rotation resistance and damping.

Fig. 1 Allowable Limit for the Dynamic Decrease in Wheel Load

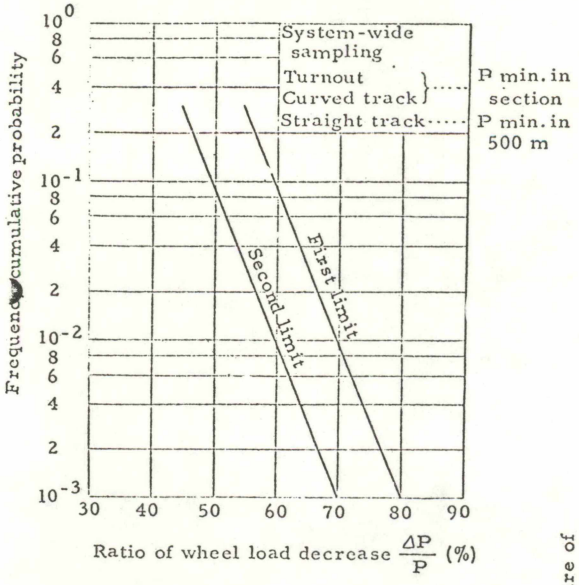


Fig. 2 Force between Wheel and Rail

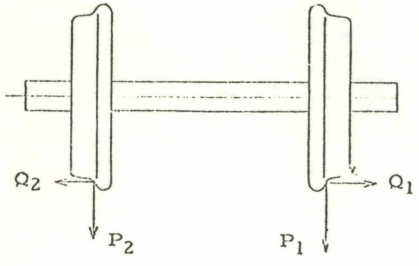


Fig. 3 Limit for the Lateral Pressure of Wheel when Gauge is Enlarged

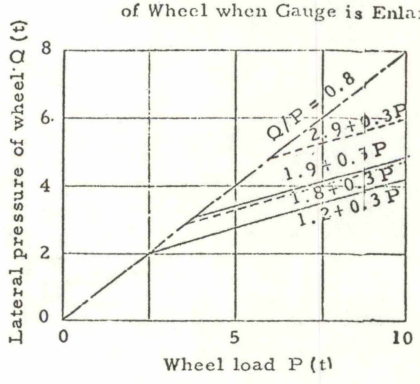


Fig. 4 Limit for the Lateral Pressure of Wheel Axles in Case of Sudden Line Irregularity

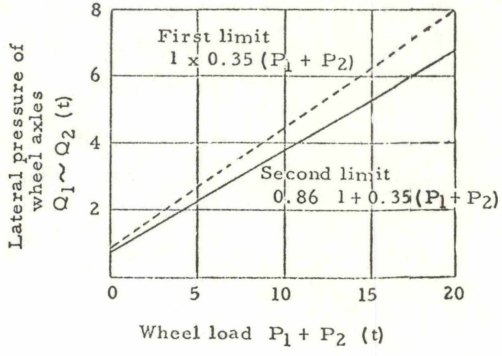


Fig. 5 Allowable Limit for Q/P

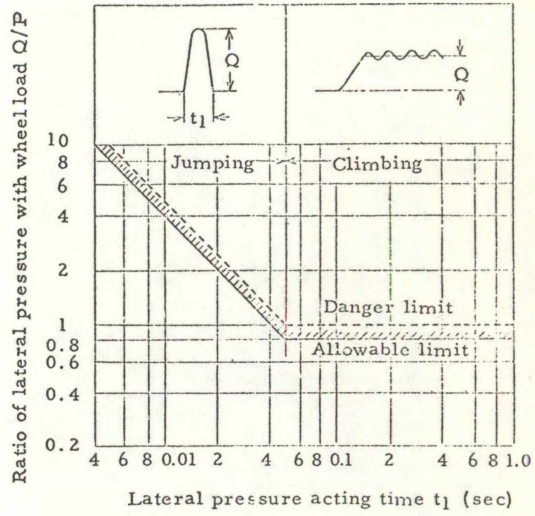


Fig. 6 Standards for Application of Q/P (I)

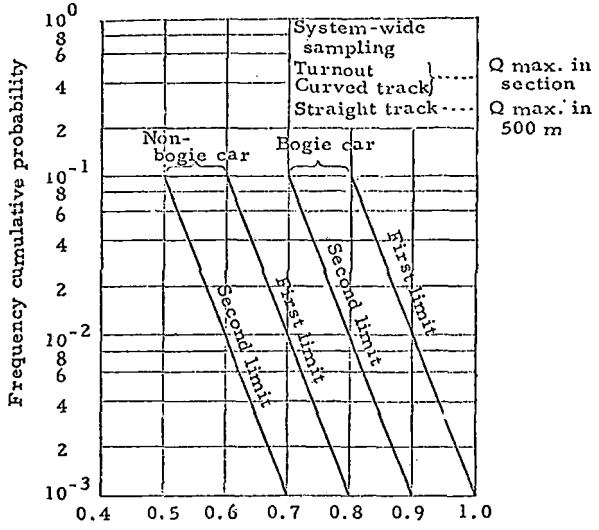


Fig. 7 Standards for Application of Q/P (II)

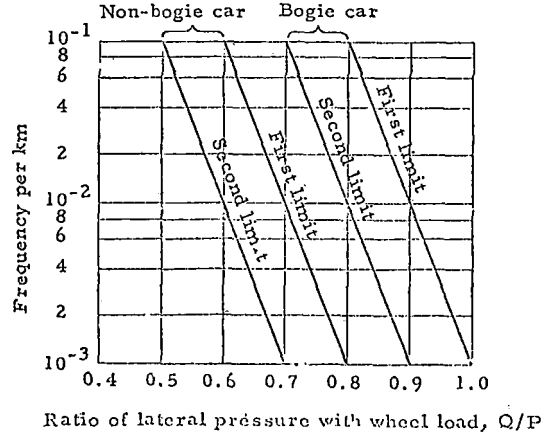


Fig. 8 Allowable Limit for Wheel-Rail Colliding Speed

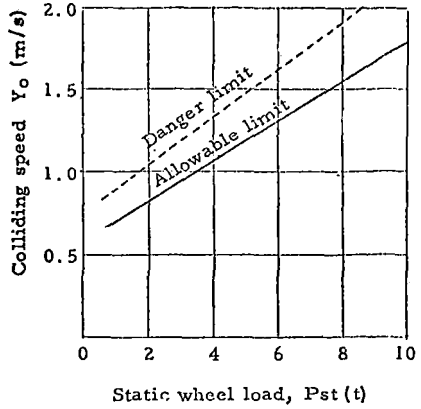


Fig. 9 Allowable Limit for Horizontal Acceleration of Carbody

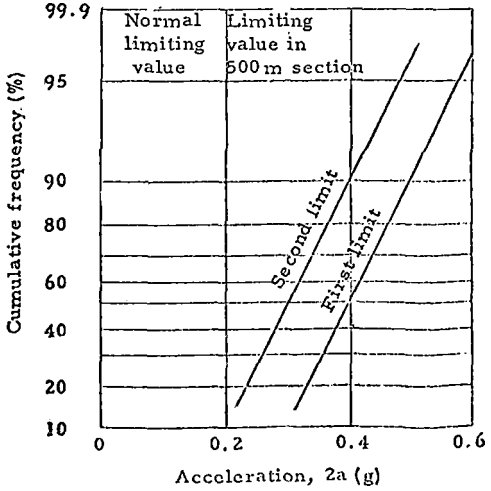


Fig. 10 Circle Analysis

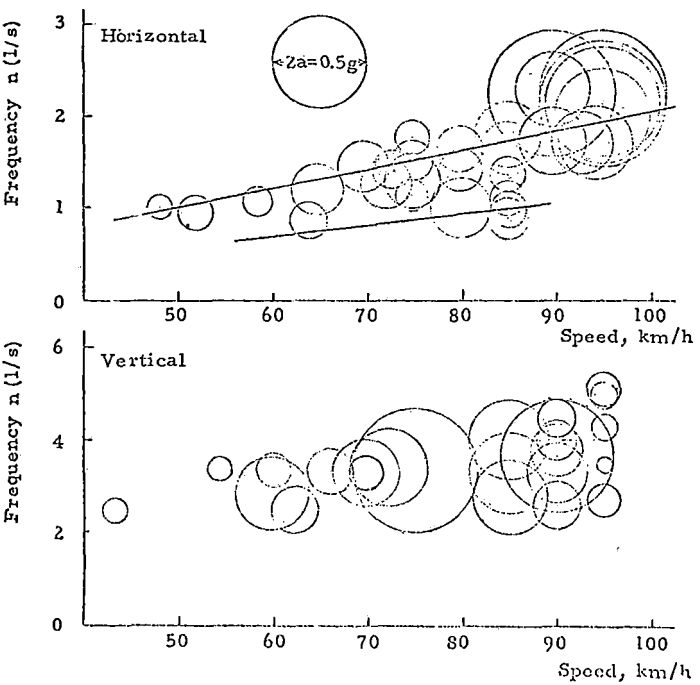
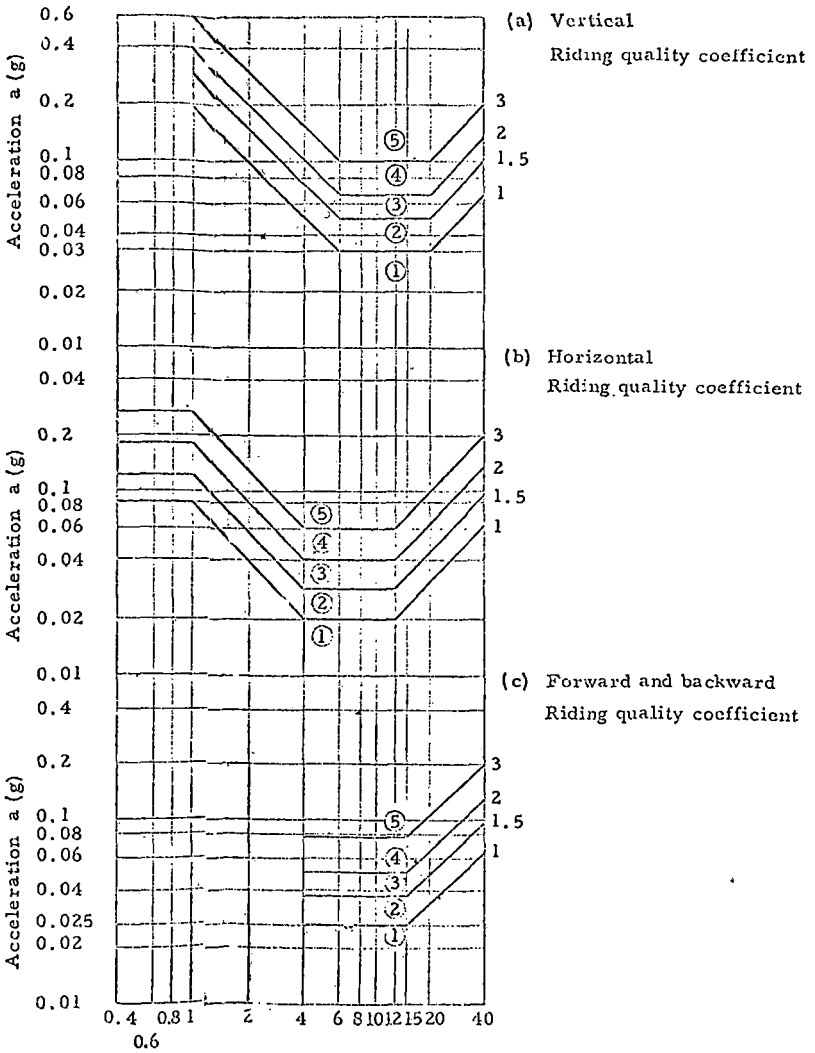


Fig. 11 Classification of Rolling Stock Vibrations for Riding Quality



Lateral Loading Between Locomotive Truck Wheels and Rail Due to Curve Negotiation

LUDVIK F. KOCI **HENRY A. MARTA**

NOMENCLATURE

- d = distance between axle and friction center
- F = resultant frictional force between wheel tread and rail in direction of resultant slip
- f = transverse component of F
- k = flange-force reaction
- k = net lateral load reaction K-f between wheel and rail
- l = wheelbase
- N = vertical wheel load on the rail
- N_s = one half the sprung load on each axle
- ρ = curve radius in feet (R in inches)
- α = misalignment angle between wheel and rail
- μ = friction coefficient between wheel tread and top of rail
- μ_e = effective transverse-friction coefficient between wheel tread and top of rail
- λ = axle end to truck frame reaction

Subscripts

- m = axle number; m = 1, 2, 3, or 4 where m = 1 is the leading axle
- n = inner or outer rail sides of the track; n = 1 or 2, respectively

When a rail vehicle having a number of axles held parallel or in fixed relation to each other passes through a curve, the directions and magnitudes of the resulting friction forces between wheel and rail cause what may be referred to as curve-negotiation phenomenon. The general nature of this phenomenon has been noted and reported long ago (8) and (10).[1] Recently, more detailed theoretical analysis of the relations existing during curve negotiation have been made allowing more complete understanding of the mechanics involved. However, to be meaningful and useful the theoretical analysis must be supported and supplemented by experimental work, not only for practical demonstration and corroboration, but also to determine the behavior and effect of such empirical factors as the effective friction - creep relation between railroad wheels and rail in actual operating conditions. In some instances, the theoretical analysis has been partially supported or supplemented by experimental work. However, most previously published experimental investigations have consisted either model work on a greatly reduced scale or measurements of such indirect rail load indications as axle-to-frame thrust loads. The former is meaningful in studying and supporting the general nature of curving mechanics but cannot provide with certainty any quantitative data because of scale factors. The latter indirectly give an indication of curving phenomenon but are so far removed from the specific area of interest or are dependent upon so many other variables that separating and assessing this one factor is difficult and of limited reliability. No thorough testing over extended periods of operation and including significant changes both in operating parameters (such as wide curvature variations) and in truck dictated parameters (number of axles, creep-coefficient relation, conditions of constraint) appeared to exist. The existence of three Electro-Motive truck designs using the identical wheel-axle set assembly, but with two, three, and four axles, and related varying wheelbase dimensions provided a unqiue opportunity to study curving mechanics and in particular, the creep effects under these varying conditions. Recording of net lateral forces between leading wheels and rails and between axles and truck frames under operating conditions was performed. The results provided the effect of creep relations upon the lateral loads present and confirmation of the phenomena predicted analytically.

This paper describes the testing performed and discusses the test results in addition to a brief and general review of curve-negotiation mechanics. Curve-negotiation mechanics, instrumentation and test equipment, testing conditions, test results, and conclusions are discussed in the following.

The scope of this paper includes only the phenomena occurring as a result of the relations between wheel and rail in passage of a rail vehicle over curved track and under coasting conditions. This is not the result of any lateral force applied to the truck or vehcile externally, such as acceleration forces or centerplate reaction from coupler angling and drawbar force; in fact, the phenomena discussed would occur in the same manner if the vehicle were standing still and

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

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the rail was moving past it. Moreover, although dynamic lateral loadings caused by track irregularities such as rail joints are related indirectly to curving mechanics. They are not discussed in detail nor are the effects of demand for driving or braking adhesion between wheel and rail during curve negotiation. In all these areas theoretical and experimental analysis can be, and has been, performed. However, they are beyond the scope of the subject matter discussed herein.

CURVE-NEGOTIATION MECHANICS

The friction forces between rails and wheel treads are the result of longitudinal and lateral slip. The longitudinal component is due to the difference in rolling distances on onner and outer rails, and the lateral slip component is due to the misalignment angle between wheel and rail. The resultant direction of these forces varies with the position of the truck within the rails. Consequently, the truck assumes that position which places the friction forces present into equilibrium with lateral flange reactions at one or more of the wheels.

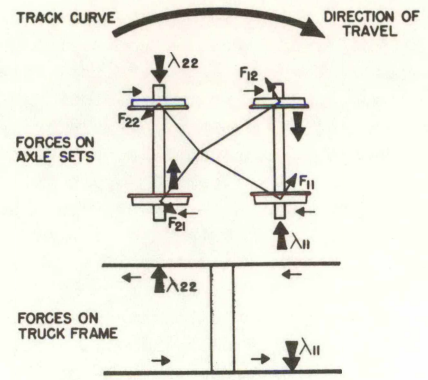
When the curvature is low enough and lateral clearances great enough, the generally common position for most vehicles is with the leading outer wheel-flange bearing against the rail creating a guiding force K_{12}, and with no other flanges in contact. The trailing axle is disposed toward the inner rail with commonly used tread contours. (This is one of the reasons the taper tread cannot and does not fulfill the function of a differential in curve negotiation, and its effect is commonly neglected in calculations.) The flange-guiding force and associated "angle of attack" α between the wheel and rail are of interest relative to wheel flange and rail wear. The algebraic resultant of the flange force and lateral friction force at a wheel k_{mn} is of interest from the standpoint of axle loading, force on rail, and associated rail stress and wheel climbing tendency. The algebraic resultant of the flange force and the lateral friction forces at both wheels of the affected axle gives the net lateral load on an axle, λ_{mn}, and is of interest in regard to truck frame loading and shifting of the entire track structure (17).

When increasing curvature (decreasing radius) causes the relative lateral clearance between vehicle frame and wheels to be taken up, the trailing inner wheel comes into flange contact with the inner rail. This condition is defined as the first degree of constraint, while the previously described position with flange contact at one wheel only is referred to as free curving. The combination of flange load and angle-of-attack prevalent in the first degree of constraint are not undesirable. However, as curvature increases further an inner wheel on an intermediate axle (where such exists) may come into flange contact. This second degree of constraint causes an increase in lateral flange forces. Beyond this condition, final constraint with end axle flanges against outer rail and intermediate flanges against inner rail occurs and is a limiting condition.

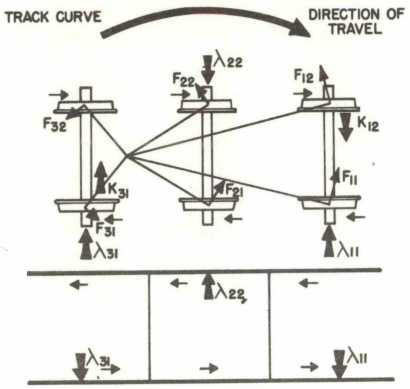
The relation between the longitudinal and lateral slips involved is such that the directions of the resultants, and correspondingly of the friction forces, at both wheels of a set act about a common point. It is on that particular radius of the curve which is parallel to the axle. Moreover, under zero traction conditions, this point is at the intersection of that radius with the longitudinal truck centerline, and can be referred to as the apparent center of friction. When all axles of a truck are parallel to one another, they share a common point as a friction center. Knowing this and the available lateral clearance, the equilibrium position of a railroad vehicle or truck can be analytically determined. Considerations of minimum energy conditions, for example, allow determination by graphical methods (18). However, particularly when account for the effects of such factors as various lateral clearances between wheels and rail and between axles and truck frame, or of externally applied forces, it is generally most convenient to solve mathematically.

Theoretical analysis has generally assumed a constant friction force acting with the presence of any amount of relative slip between wheel and rail in the direction of that slip, and of magnitude equal to the product of the normal force and the friction coefficient. However, reasonable considerations, early test indications, and practical experience with such factors as location and amount of rail wear, all indicated a force dependence on the amount, not only the presence, of slip occurring (it would appear, for example, unreasonable to expect similar load levels regardless of curvature). Recent investigations in many fields have established the presence of creep phenomena in which the friction forces present between bodies in rolling contact are a function of the relation of slipping velocity to rolling velocity. It is apparent that this creep relation is an important factor in the force levels occurring in curve negotiation.

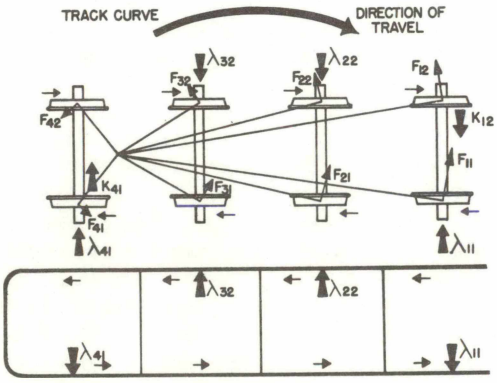
Although many of the recent investigations have developed creep relation data, they have generally involved laboratory specimens or specific products such as rolling element bearings or



(a) Two-axle truck



(b) Three-axle truck



(c) Four-axle truck

Fig. 1 Typical reactions on a truck frame and wheel-axle sets in curve negotiation under coasting conditions

automotive tires. Not only do the railroad wheel and rail involve unusual relations in configuration and size, but full curving mechanics description would demand complete knowledge of the creep relation for slipping direction at any possible angle to rolling direction. This complex relation involving the friction forces and their affects on equilibrium position and reactions is one of the factors which made it appear desirable to obtain test results reflecting the end result of these relations in a manner which could be used to understand and predict curving mechanics phenomena without full and thorough knowledge of the entire creep relation. Similarly, the compatibility of deflections and clearances between the track and vehicle structure affect the load distribution. This is particularly so with resilience in the axle lateral suspension which under certain conditions could allow, for example, both first and second axles in a 3-axle truck to bear against the outer rail. Again, it was necessary to obtain the results of these factors under actual operating conditions.

INSTRUMENTATION AND TEST EQUIPMENT

A typical railroad truck experiences transverse-load reactions at several locations: between the locomotive carbody and the truck bolster at the centerplate, between the truck bolster and frame at the lateral connection of these two components, between the truck frame and axles through the journal boxes, and between the wheel-axle sets and the rails at the flanges and tread surfaces. These lateral loads may be the result of transverse track inputs because of rail curvature or rail irregularities, or the result of forces transmitted to the truck through the centerplate from either transverse components of coupler-drawbar loads or acceleration forces.

Of primary importance in these tests were the lateral-load reactions between the leading wheel-axle set of a truck and the rail. However, the instrumentation was arranged to indicate not only the net lateral load on each of the two leading wheels but also the axle-to-truck frame reactions at all truck axles, the vertical wheel load at both leading wheels, the coupler-drawbar load, and coupler and truck swing relative to the locomotive carbody. Data on the above mentioned items were not only necessary to check the equilibrium condition of transverse forces but valuable in determining the load levels occurring at such locations and whether forces were of a stable or transient dynamic nature.

Fig.1 indicates resultant forces acting on wheel-axle sets and truck frames of two, three, and four-axle trucks negotiating a curve under coasting conditions and at balance speed. Under such curving conditions of no tractive effort,

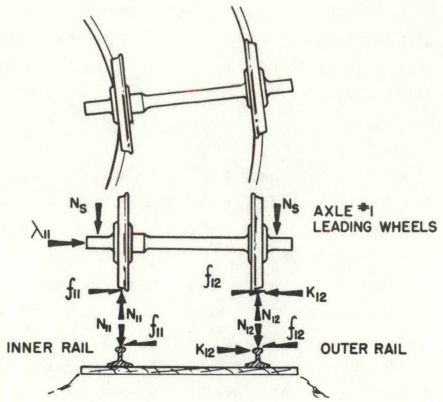


Fig. 2 Vertical and transverse reactions on the leading wheel-axle set and the rail

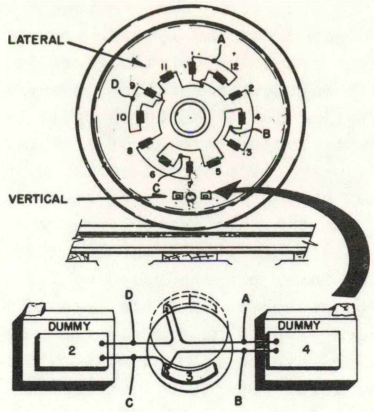


Fig. 3(a) Strain gage location and connections of the lateral and vertical load transducers

coupler-drawbar load, or centrifugal force, there are no forces transmitted between the truck and locomotive carbody. The only reaction at the centerplate is a frictional torque, which is a result of truck rotation while entering and leaving the curve; with the geometry involved this torque has a negligible effect on loadings present. The torque is not a factor while within the curve, since no truck rotation relative to carbody is taking place. Therefore, all lateral reactions to the frame come through the axle ends; these forces are of the same magnitude as the algebraic summation of the lateral reactions between the rail and the wheel-pair of the corresponding axle.

Fig.2 indicates the lateral reactions that act on the leading wheel-axle set of the truck. The outer wheel experiences a net force resulting from a guiding flange load and the transverse component of the frictional reaction between that wheel and the outer rail, i.e., $K_{12} - f_{12} = k_{12}$. The inner wheel senses the transverse component f_{11} of the frictional reaction F_{11} between that wheel and the inner rail. The equilibrium condition of the wheel-axle set requires that $\lambda_{11} = k_{12} - f_{11}$; all of these reactions were measured during the tests by suitable transducers discussed below. Considering that forces f_{11} and f_{12} are approximately equal in magnitude, it is possible to evaluate the flange load K_{12} by measuring the net lateral loads f_{11} and k_{12} occurring at the two leading wheels, i.e., $K_{12} = k_{12} + f_{11}$.

At the remaining wheel-axle sets the axle-to-frame reactions, λ_{mn}, were measured. Although reactions between the individual wheels and rails were not measured at these axles, the test results and analysis indicate that reliable data on the wheel-to-rail reactions at these locations could

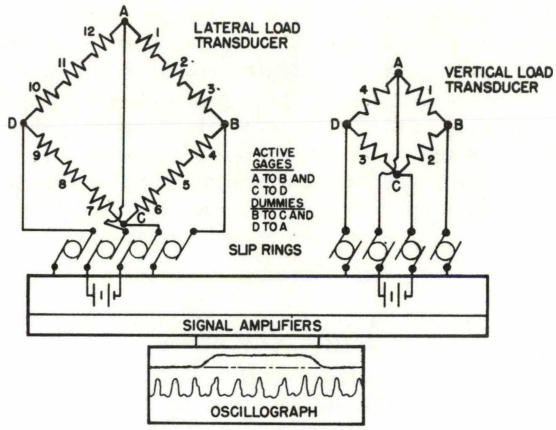


Fig. 3(b) Instrumentation line diagram

be developed from the λ_{mn} obtained at intermediate axles and reactions measured at the leading wheel-axle set.

The instrumentation of two 40-in-dia wheels into lateral and vertical load transducers was carried out through an extensive investigation at the EMD development laboratory. A test setup was arranged to provide vertical and/or lateral load application to a wheel-axle set placed on rails fixed to the test floor simulating loading conditions in service. The object was to determine strain-gage locations on the wheel plate which would produce satisfactory amounts of strain from either a lateral or vertical load and be substantially insensitive to the other. A number of gages were placed in a radial direction along a line between wheel hub and rim fillets for the investigation.

A method similar to one previously reported was investigated for the lateral load transducer (2). It was found that strain gages located on a circle of a particular constant radius from the axle centerline and oriented in radial directions showed a satisfactory amount of strain from lateral loads only. The strain varied with the central angle between the gage and load locations as measured from the axle centerline. The summation output of six gages located at equal central angles on the inner plate surface of the wheel and properly connected in opposite legs of a wheatstone bridge circuit gave a continuous strain signal under transverse loading reasonably independent of the wheel rotation. Two wheels were prepared with lateral load transducers and carefully calibrated for k reactions of up to 35,000 lb between wheel and rail, Fig.3.

For the vertical load measurement, to obtain insentivity to lateral load, it was found necessary to drill a hole $1\frac{1}{4}$ in. diameter in the wheel plate and place two strain gages along the inner surface of the hole 180 deg apart, and perpendicular to a line joining the axle and hole centerlines, Fig.3. These gages were sensitive to vertical loads only once per revolution when the hole was directly between the axle and the rail; their output was therefore a spike signal. Thus, by proper connection into separate circuits, it was possible to provide one vertical-load transducer on each side of the same wheel-axle set.

The axle to truck frame reactions λ_{mn} required axial-load transducers. These cylindrical tube-type transducers, used by EMD in past testing, were calibrated for loads of up to 35,000 lb and then placed between each axle and journal box housing of the instrumented truck. The two transducers at both ends of each axle were connected into a Y circuit, to give a common signal responding in opposite directions to reactions λ_{m1} and λ_{m2} occurring at the inner and outer rail sides of the axle.

The transverse load component at the truck centerplate was not measured directly. However, to determine such load levels under substantial drawbar loads, the coupler adjacent to the instrumented truck was provided with two separate transducers, one having an output proportional to the drawbar load between that locomotive unit and the remainder of the train, and the other output proportional to the coupler swing angle relative to the locomotive carbody. By recognizing that the transverse-centerplate load, at balance speeds is a function of coupler-drawbar load, swing angle, and locomotive geometry, it was possible to determine approximate lateral load levels at the centerplate under these conditions.

The truck swing relative to the carbody was measured to approximately confirm the degree of curve specified in the track profiles and to aid in recognizing and analyzing data either within or outside the curve.

The aforementioned instrumentation was applied to each of three different trucks and cabled to the EMD test car. The outputs were monitored through proper amplifier channels to an 18-channel oscillograph recorder. This arrangement made it possible to record the outputs of all the transducers simultaneously, giving a measurable output signal from stable or transient dynamic load reactions occurring at the instrumented truck. The wheel transducer signals were monitored through two slip-ring arrangements applied at both ends of the No.1 axle (where lateral wheel loads are maximum under normal conditions(in combination with the axial-load transducers at those locations.

The trucks used for the tests consisted of two, three, and four-axle units having different wheelbases but uniform axle loading. Unsprung components were identical. The two-axle truck had a wheelbase of 108 in., the three-axle truck 163 in., and the four-axle truck 205.5 in. All three trucks were standard gage and had the same nominal lateral clearance between axles and truck frame at all locations.

TESTING CONDITIONS

The lateral loading tests were initiated on a four-axle flexicoil truck as part of new product field tests, and expanded to include tests on a two-axle swinghanger truck and a three-axle flexicoil truck. Although a portion of these tests were performed at EMD yard track on sharp curves of about 16 and 19 deg, the major portion of the program was carried out on mainline track of the Union Pacific Railroad between Omaha, Nebraska and Los Angles, California.

Testing conditions on the railroad covered a wide range of curves, varying between one and eleven deg, and grades of up to 2.2 percent. The instrumented locomotive was used in various positions of a multi-unit consist. Although this paper is more specifically intended to present results of lateral loading between wheels and rail under coasting conditions with little or no drawbar loads, test runs were also recorded under a wide range of drawbar pull and buff loads. In all cases, however, the train travel was at approximately the balance speed. Under these conditions resultant load components as a result of centrifugal force and superelevation were nulli-

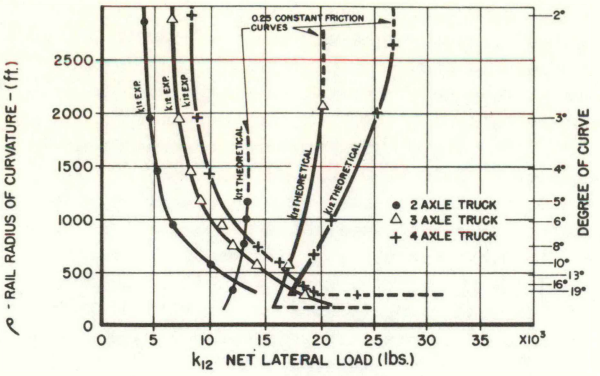


Fig. 4(a) Relationship of theoretical constant friction and experimental results for the net lateral load on the leading outer wheel as a function of curve radius

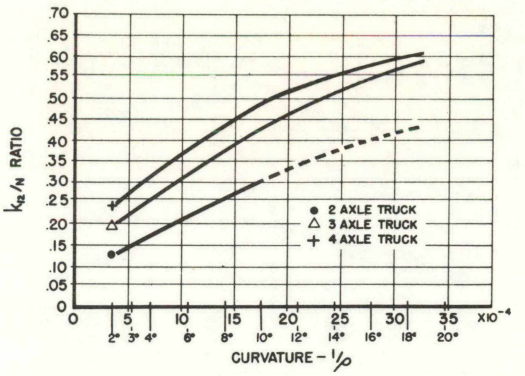


Fig. 4(b) Experimental net lateral load coefficients on the leading outer wheel as a function of track curvature

fied by one another and are not considered in the analysis. It may be noted that outward force such as centrifugal force resulting from traveling over the balance speed has by no means an essential influence on the adjustment of a vehicle in a curve or on the lateral pressure (11) with commonly occurring force levels.

The testing procedure consisted of oscillograph recording of all instrumented components for overall truck performance data giving a history behavior of truck reactions in negotiating either curves or rail irregularities on tangent track. A number of test runs were recorded for each size curve at various locations.

Within the curve range under which the three trucks were tested, the truck position was generally in free curving or the first degree of constraint. The two-axle truck is in free curving on curves of about 1150 ft radius, and in first degree of constraint on sharper curves. The three-axle truck is in free curving on curves above 2050 ft radius and in first degree of constraint on curves between 2050 and 193 ft radius. The four-axle truck is in free curving for curves above 2650 ft radius, and on curves of 2650 to 307 ft radius the truck is in the first degree of constraint.

The direction of travel was such that the instrumented wheel pair was at the leading position on the truck.

TEST RESULTS

In presenting the results of lateral load reactions between locomotive truck wheels and the rail, primarily on the leading wheel pair, some of the theoretical calculated results based on constant 25-percent friction coefficient are shown for purposes of comparison. The substantial difference which exists between the indicated theoretical and experimental data is primarily because of the assumed constant wheel-to-rail friction coefficient used in the calculations, while the experimental data reflect the result of the effective transverse-friction coefficient which varies with the angle misalignment between wheel and rail and is much lower than 25 percent of wheel load for very small angles.

Fig. 4a shows both theoretical and experimental results of the net lateral reactions between the leading outer wheel and rail as a function of track radius-of-curvature for the two, three, and four-axle trucks. The net lateral loads represent the difference between the flange guiding force and transverse-frictional force between the same wheel tread surface and outer rail. The theoretical results are based on an analysis of the static equilibrium condition of a truck on a curved track at a given instant, assuming a constant friction coefficient of 25 percent of the wheel load for reactions between wheel tread surface and rail (regardless of the creep coefficient), and flange reactions at the leading outer wheel and/or the trailing inner wheel. All these conditions act about an apparent center of friction as shown by the wheel-axle sets of the three trucks in Fig. 1. The friction center is determined by the truck geometry within the outer and inner rails and the radius of curvature. With the truck in coasting conditions on a curve, the apparent center of friction is usually on the longitudinal truck centerline on either side of the trailing axle depending on the wheelbase, the number of axles, and degree of constraint resulting from the curvature. It is evident that theoretical load levels, assuming a fixed friction coefficient for

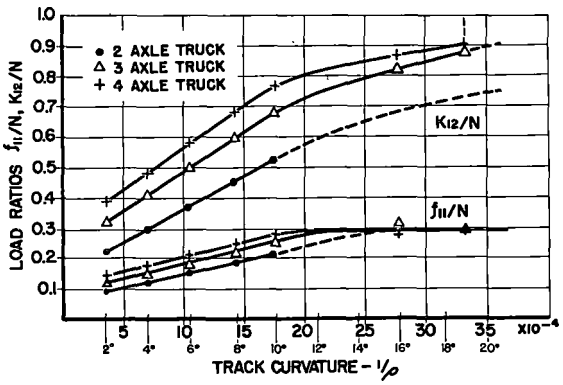


Fig. 5 Transverse reaction coefficients between wheel and rail at the leading axle as a function of track curvature

small curves, are excessive and misleading because they are higher than on sharper curves. Experience with rail wear alone correctly indicates otherwise.

The experimental results show a number of interesting comparisons among the three trucks and with the theoretical data of k_{12}: (a) The actual data bear out the low level of loading within the lower curvature range; in this range the experimental load levels amount to about one-half or less than those of the theoretical constant friction coefficient values on the corresponding trucks. (b) The load levels vary to a great extent as a function of curvature as shown in Fig. 4(b). Between zero and about a 10 deg curve, the load levels change directly proportional to the change in curvature. For sharper curves, the rate of change decreases, because of the limiting value of the friction coefficient with increasing creep, provided the vehicle maintains the same degree of constraint. (c) The transverse friction between the wheel treads and rail is substantially lower than 25 percent of the vertical wheel load for curvature ranges below approximately 8 deg. (d) The ratio of the lateral loading on the other trucks to that on the two-axle truck decreases with sharper curves. (e) As expected, the loading of the four-axle truck is not substantially higher than the three-axle truck; this is the result of the design considerations for a truck arrangement providing minimum wheelbase compatible with other important factors such as allowable lateral clearances and suitable suspensions. (f) Experimental results confirm mathematically the commencing of second degree of constraint for the four-axle truck at about a 19-deg curve.

The experimental data in the Fig. 4 curves represent the average mean load levels measured for each size curve. The variation of the mean from the average values was about $\pm$ 2500 lb or $\pm$ 7.5 percent of the wheel vertical load. This variation may be attributed to slight variations from the balance speed, transverse-load component of minor amounts of drawbar load, or minor traction demands between wheel and rail.

The transient dynamic-type lateral-load reactions act over and above the net lateral load produced by curve negotiation. As mentioned before, these loads are of very short durations and are caused by sudden inputs such as track irregularities and rail joints. The disturbances are sensed at the flange of the leading outer wheel. Since the leading outer wheel lateral-load transducer measures the difference between the guiding flange force and the transverse-frictional reaction at the wheel tread surface, any short duration signals sensed by the same transducer amount to the transient dynamic loads on the wheel while being reacted by the same amount on the rail. It was found that these forces are to a great extent a function of the unsprung mass at each axle and substantially, although not totally, independent of the truck wheelbase. For any size curve or tangent track the average transient dynamic load was about 6000 lb, and two or three times that amount was not uncommon on any of the trucks. According to theoretical analysis done at EMD on the same trucks, a 6000-lb force can be caused by a 5/16 to 3/8-in. transverse-rail irregularity forcing a motion of the type y_o (1- cos wt) traversed during 1/5 sec, where y_o = 5/32, or 3/16 in. On this basis a transient dynamic force of 18,000 lb results from about 1-in. rail irregularity traversed in 1/5 sec or 1/4-in. disturbance traversed in 1/10 sec.

Fig. 5 shows the remaining transverse reactions that occur between wheel and rail. This experimental data shows the transverse-frictional reactions between the tread surfaces of the leading wheels and the rail, and the apparent guiding force at the leading outer wheel. The frictional forces f_{11} were measured at the leading inner-wheel transducer since that is the only reaction acting on the wheel at that location in the transverse direction. By assuming that $f_{11} = f_{12}$ the flange loading was determined by the summation of $k_{12} + f_{11}$.

The friction forces f_{11} in Fig. 5 increase directly with curvature up to a value of about 29 percent of the vertical wheel load as indicated by data obtained on the three-and-four-axle trucks. It was evident at the time of the analysis of this experimental data that such behavior should be directly related to the friction-creep phenomena. Further investigation confirmed that

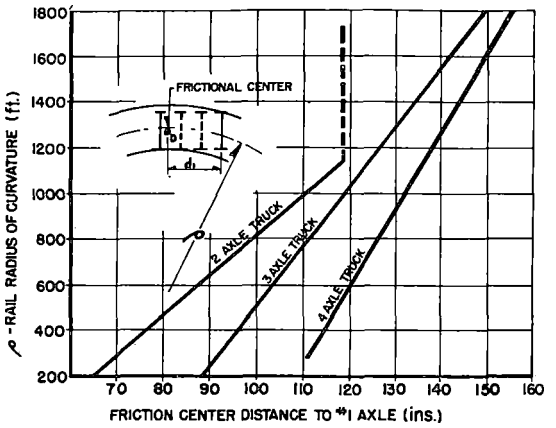


Fig. 6 Relationship between the apparent friction center distance from the leading axle and curve radius for the three trucks

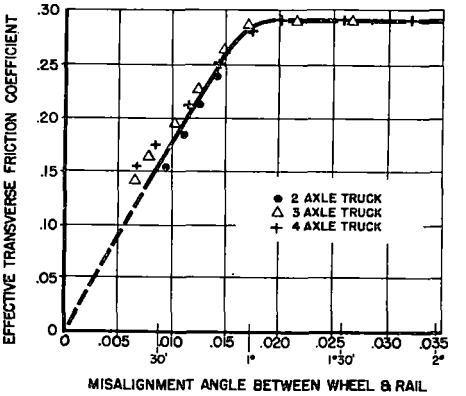


Fig. 7 Apparent effective transverse-friction coefficient as a function of transverse creep

these frictional loads were substantially a direct function of the misalignment angle between the leading wheels and rail within the increasing range of f_{11}, and further that the limiting value of f_{11} = 29 percent of the wheel load occurred at the same apparent values of wheel-to-rail misalignment angles for the three trucks. Because of the small value of these angles it is assumed here that the angle is equal to the sine of the angle and thus, a misalignment angle expressed in radians is equal to the resultant transverse creep between the wheel and rail.

The apparent flange-force curves in Fig.5 show a similar behavior to k_{12} and f_{11}. The load levels increase directly with curvature up to a 10-deg curve for the four-axle truck, up to 12-deg curve for the three-axle truck, and up to a 15-deg curve for the two-axle truck. For sharper curves, the rate of increase in K_{12} decreases substantially as the limiting friction coefficient at the leading wheel is attained. Note also that at any size curve, the summation of the frictional loads at the leading wheel-pair represents between 67 and 85 percent of the total flange load; this shows that the frictional reactions at the intermediate and trailing wheels are substantially lower than f_{11} or f_{12} by virtue of the transverse friction - creep phenomena at those locations.

In relating the frictional loads f_{11} and creep at the leading wheel pair it is possible to determine an effective transverse friction-versus-creep curve for locomotive wheels on rail by plotting the ratio f_{11}/n as a function of wheel-to rail-misalignment angle; such an angle was assumed to be equal to the ratio of the apparent friction center distance d_1 from the leading axle (as given in curves of Fig.6) to the curve radius of the track. Such a curve is shown in Fig.7, and is a result of data measured at the leading axle inner wheel of the three different trucks. Note that the data used for this friction-creep curve, or any of the experimental results, were measured under service conditions of wheel-to-rail operation regardless of weather conditions or frictional-surface contaimination between the wheel tread surface and the rail.

A comparison of this friction-creep curve with similar published theoretical or laboratory experimental curves (3) and (15) indicates the same type of behavior except for the slope of the curve at very small angles and the creep value at which the transverse friction coefficient μ_e reaches a limiting value. The slope in Fig.7 is two to three times those in the cited references, and the knee of the curve occurs at a value of about 1-deg misalignment angle (i.e., 0.0175 creep) compared to approximately one-half that value shown in similar curves. However, the limiting friction coefficient value of 29 percent of the vertical loads is within the expected level and is nearly equal to values obtained on small model equipment in the laboratory regardless of surface contamination with water or oil, or the amount of vertical load on the wheel (3).

A confirmation of the validity of the friction-creep relation shown in Fig.7 was also possible for the remaining wheels in each truck. Data on the intermediate wheel-axle sets, where such existed, were more conclusive for this purpose because the axle-to-frame reactions at those locations are high enough, and substantially due to the transverse-frictional reactions within free or first degree of constraint curve negotiation

of a truck (i.e., $\lambda_{m2} = f_{m1} + f_{m2}$). It was confirmed that the axle-to-frame reaction λ_{m2} is equal to approximately the sum of the calculated transverse-frictional reactions at the affected wheels as determined by the use of Fig.7 and the calculated value of the apparent friction center. At the trailing axle, it was possible to apply the same type of check for a truck in free curving only, where the absence of a guiding flange force at the inner wheel gives a similar equilibrium condition as on the intermediate axles, i.e., $\lambda_{m1} = f_{m1} + f_{m2}$.

An effect of high tractive effort and coupler-drawbar loads on lateral loading was quite evident. The general tendency was a decrease in the net lateral load at the leading outer wheel with increasing pull because of the increase in longitudinal adhesion between each wheel and rail and because of the lateral component of the coupler load. On the other hand an increase in braking effort and coupler-drawbar buff did not increase the net lateral loads appreciably for two main reasons: (a) The alignment control feature in the coupler application tended to reduce the lateral component of the buff load; and (b) the truck tended to be positioned against the outer rail, thereby reducing the transverse frictional loads at the leading wheel-pair which are proportional to the angle of attack for very small angles, consequently, resulting in a decreased flange force.

CONCLUSIONS

The test results of transverse curving forces between locomotive wheels and rail confirm the direct relationship to the friction-creep phenomena. Under coasting conditions for curving, the transverse reactions at the tread surface or the flange, and the net lateral loads on the wheel or rail can be predicted with reasonable accuracy from Fig.7 and a calculated apparent center of friction for known constraint conditions of any truck. This is possible assuming Fig.7 represents the resultant friction-creep phenomena regardless of direction of creep; by so doing, the resultant frictional forces at the tread surface can be determined, leading to the factors aforementioned.

Further investigation would be desirable, however, to explain the difference between friction-creep phenomena obtained in the field and in the laboratory where models are used. Also, further tests are essential to determine the effect of definite amounts of traction demand on the curving forces.

Acknowledgments

The authors wish to express their appreciation to Mr. D. Clifton and his department at EMD who are responsible for providing and operating the test instrumentation, to the Union Pacific Railroad for their continuous cooperation during the testing on Union Pacific mainline track, and to General Motors for allowing the publication of these results.

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Nonlinear Wheelset Dynamic Response to Random Lateral Rail Irregularities

The nonlinear equations of motion for a railway vehicle wheelset having profiled wheels and contact of the wheel flange with flexible rails are presented. The effects of spin creep and gyroscopic terms are included. The rails are considered to have random lateral irregularities which are described by prescribed power spectra. The equations of motion are integrated numerically and the effects on the dynamic response of quantities such as speed, track roughness, wheel wear, flange clearance, and lateral stiffness of the rails are investigated.

Introduction

Secondary hunting of railway axles and trucks places constraints on the maximum safe speed for railway vehicles. Secondary hunting of railway axles and trucks is a coupled lateral and yawing oscillation of the axle (or axles, for a multi-axle truck) and truck frame. This motion is usually weakly coupled to car body motions at high speeds. The mode of motion wherein strongly coupled lateral motions of the truck and car body occur is called primary hunting and usually occurs at a lower speed than does secondary hunting. Above a certain value of speed, disturbances of the truck (such as due to track irregularities) will result in secondary hunting wherein the axles slam from rail to rail. This results in damage to wheels, rails, and truck components and may lead to derailment if unchecked.

Results of model experiments by Hobbs [1][1] and Matsudaira [2] indicate that worn wheels and rails having curved cross-sectional profiles strongly influence the character of the secondary hunting motion. Matsudaira also reports that wheel flange/rail contact strongly influences secondary hunting. Gilchrist, et al. [3] have investigated theoretically and experimentally the effects of such nonlinearities as wheel slip, wheel profile curvature, and flange contact on the dynamics of two types of four-wheeled British Rail vehicles. During track tests, Gilchrist measured the variation with speed of the ratio of lateral to vertical suspension force for a wheelset for several types of vehicles. This ratio was found to increase with speed. His results also indicate that once the critical speed for secondary hunting is exceeded, the motion of the wheelset takes the form of a limit-cycle oscillation which is largely independent of alignment irregularities. Cooperrider [4, 5] has investigated theoretically the effects of flange contact and wheel slip on the dynamics of a six-degrees-of-freedom truck model. The analytical portions of the foregoing investigations have required the use of digital or analog computers to integrate the equations of motion.

Recently, Law and Brand [6] utilized the method of Krylov-Bogoliubov [7] to examine the effects on wheelset dynamic response of wheel profile curvature and wheel flange/rail contact. Due to the change in effective coning angle of the wheels with lateral wheelset displacement and the contact of the wheel flanges with the rails, the resulting dynamic motions of the wheelset were shown to be significantly different from those predicted by linear analyses.

The analytical portions of the preceding investigations have generally considered the track to be perfectly aligned, straight, level, and without variation in cross level or gauge. Thus these investigations have considered the stability characteristics of the unforced wheelset. Actual rails for nominally straight or tangent track are characterized by random irregularities in vertical and lateral profile. Typically, this random nature of the track profile is characterized by power spectra.

Stassen [8] investigated the effects of random lateral rail irregularities on the lateral dynamics of a dual axle truck with rigid primary suspension between truck frame and wheelsets. He considered linear creep forces and spin creep effects and used the method of statistical linearization [9, 10] to obtain linear estimates of the nonlinear wheel/rail kinematic relations. These linear estimates are functions of the rms value of the lateral displacement of the wheelset relative to the irregular rails, the flange clearance, and the mean value of the difference between the actual gauge and a nominal gauge. He also considered the rails to be rigid. Stassen derived analytically relations for the critical speed and the hunting wavelength. He also utilized an analog computer to determine the response of the nonlinear model to random lateral rail inputs produced by noise generators and shaped by appropriate filters. The nonlinear wheel/rail kine-

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

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matic relations were mechanized on function generators and linear creep force relations were used. However, a check was made to determine whether the system was operating in a state such that the linear creep formulation was valid. Based on the results of this check, Stassen concludes that "...the combined creep and spin components can be given by a linear function of the reduced tangential forces. Hence to make a more adequate model there is no point in introducing nonlinear functions." Stassen also conducted some experiments using an experimental truck. The effects on the dynamic response of various parameters were investigated. Among those parameters were: rms lateral rail displacement, gauge, spin creep, and forward speed.

In this paper, results are reported of a digital computer simulation of the general nonlinear equations of motion of a wheelset having curved wheel profiles and wheel flange/rail contact. The effects of the lateral flexibility of the rails are modeled. The response of the wheelset to random lateral irregularities of the individual rails as described by typical power spectra is calculated. The effects on dynamic response of various quantities such as speed, track roughness, wheel profile curvature, flange clearance, and lateral rail stiffness are investigated. In addition to other outputs, rms and maximum values of the ratios of lateral to vertical force of the rail on the wheel (L/V) for speeds both above and below the linear critical speed are shown.

Yokose [11] has advanced a theory that relates derailment to the steady state levels of L/V and wheel/rail geometry. It is relatively common to try to relate derailment to levels of L/V but very little is known about the relation between dynamic levels of L/V and the tendency of a wheelset to derail. Matsudaira [12] describes actual running tests where time histories of L and V (and hence L/V) were measured during hunting. He also presents a plot of L/V versus period of the motion on which both a prescribed "safety limit" and the measured values are recorded. The origin and rationale of the safety limit are not discussed.

The objectives of this investigation were (a) to develop an analytical method for predicting the response of a general nonlinear wheelset to random track irregularities, (b) to estimate the influence of various parameters on the dynamic response, and (c) to estimate the levels of L/V that occur for worn wheelsets on actual track at speeds above and below the linear critical speed.

Method of Analysis

Model Used in Analysis. The wheelset is assumed to be traveling at a constant speed, V, with zero traction or braking torques on track that is nominally straight or tangent. The lateral profile of each wheel is curved as a result of initial design or as a result of

Nomenclature

A_1 = nondimensional amplitude of q_1 for stationary oscillation
a = one half track gauge, ft
a_0, a_1 = coefficients in roll angle constraint equation (1)
b_1 = coefficient in vertical displacement constraint equation (2)
c_1 = coefficient in vertical displacement constraint equation (2)
F'_{RAIL} = nondimensional rail spring force, equation (23)
$G(f)$ = power spectral density, in^2/cycle/ft)
$G(\omega)$ = power spectral density, in^2/(rad/nondim sec)
f_{11} = lateral creep coefficient, lb/wheel
f_{12} = lateral/spin creep coefficient, lb ft/wheel
f_{22} = spin creep coefficient, lb ft^2/wheel
f_{33} = longitudinal creep coefficient, lb/wheel
g = acceleration due to gravity, 32.2 ft/sec^2
I_1 = moment of inertia of wheelset about axle centerline, slug ft^2
I_2 = moment of inertia of wheelset about centroidal axis normal to axle, slug ft^2
J_1 = nondimensional moment of inertia, $J_1 = \mathrm{I}_1/(\mathrm{ma}^2)$
J_2 = nondimensional moment of inertia, $J_2 = \mathrm{I}_2(\mathrm{ma}^2)$
k_R = lateral spring constant of rail, lb/ft
k_x = suspension lateral spring constant, lb/ft
k_θ = suspension yaw spring constant ft lb/rad
$M_{\mathrm{R}y}'$ = nondimensional moment exerted by rail on wheelset due to flange contact, equation (24)
m = wheelset mass, slugs
q_1 = nondimensional lateral displacement, $q_1 = x/a$
q_2 = nondimensional yaw displacement, $q_2 = \theta$
q_3 = nondimensional vertical displacement, $q_3 = y/a$
r_L, r_R = instantaneous rolling radii of left and right wheels, respectively, ft
r_0 = wheel radius at equilibrium, ft
t = time, sec
V = forward speed of wheelset at equilibrium, fps
V_0 = linear critical speed of wheelset, fps
W = axle load, lb; or, frequency response function
x = lateral displacement of axle centroid, ft
y = vertical displacement of axle centroid, ft
y_1 = vertical displacement of left rail, ft
y_2 = vertical displacement of right rail, ft
z_1 = lateral displacement of left rail, ft
z_2 = lateral displacement of right rail, ft
γ' = nondimensional weight on wheels, $\gamma' = (W + mg)a/(mV_0^2)$
δ = nondimensional flange clearance, $\delta = \delta'/a$
δ_L = angle between left wheel contact plane and axle, rad
δ_R = angle between right wheel contact plane and axle, rad
δ' = flange clearance, ft
ϵ_1' = nondimensional suspension lateral spring constant, $\epsilon_1' = k_x a^2/(mV_0^2)$
ϵ_4' = nondimensional suspension yaw spring constant, $\epsilon_4' = k_\theta/(mV_0^2)$
ϵ_5 = nondimensional rail lateral spring constant, $\epsilon_5 = k_R a^2/(mV_0^2)$
ζ_1 = nondimensional lateral displacement of left rail, $\zeta_1 = z_1/a$
ζ_2 = nondimensional lateral displacement of right rail, $\zeta_2 = z_2/a$
η_1 = nondimensional vertical displacement of left rail, $\eta_1 = y_1/a$
η_2 = nondimensional vertical displacement of right rail, $\eta_2 = y_2/a$
θ = yaw displacement of wheelset about vertical axis, rad
μ_{11} = nondimensional lateral creep coefficient, $\mu_{11} = 2f_{11}\,a/(mV_0^2)$
μ_{12} = nondimensional lateral/spin creep coefficient, $\mu_{12} = 2f_{12}/(mV_0^2)$
μ_{22} = nondimensional spin creep coefficient, $\mu_{22} = 2f_{22}/(maV_0^2)$
μ_{33} = nondimensional longitudinal creep coefficient, $\mu_{33} = 2f_{33}\,a/(mV_0^2)$
ν = speed ratio, $\nu = V/V_0$
ρ = nondimensional wheel radius, $\rho = r_0/a$
τ = nondimensional time, $\tau = (V_0/a)t$
τ_1, τ_2 = nondimensional time constants
ϕ = roll angle of wheelset, rad
Ω = equilibrium angular velocity of wheelset, rad/sec
ω = frequency, rad/nondimensional seconds

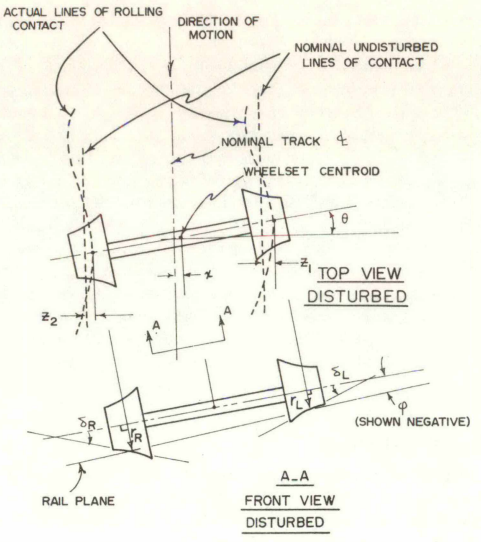


Fig. 1 Schematic of wheelset

wear. A rigid axle which connects the wheels is mounted in flexible axle bushings which provide the primary suspension. The car body (or truck frame) which is isolated from the axle by these bushings is assumed to translate uniformly down the track. The wheelset is free to move in the truck frame subject to the constraints imposed by the yaw and lateral compliances of the axle bushings.

The nondimensional lateral displacement and the yaw displacement of the wheelset from the equilibrium condition are given by $q_1 = x/a$ and q_2, respectively, as shown in Fig. 1. The movement of the point of contact over each wheel tread with respect to the wheelset is due to the wheelset moving with respect to the track centerline and the lateral irregularities of each rail. The lateral displacements of the left and right rails, z_1 and z_2, are positive as shown in the figure.

Wheel/Rail Interaction Forces. The creep forces and moments exerted by the wheels on the rails are formulated following the linear formulation of Kalker [13] as explained by Hobbs [1]. As the wheelset moves along the track and oscillates in yaw and lateral displacement, both the wheel and rail will deform elastically at the contact region. However, each will experience a different strain rate. Creep forces and moments arise due to the difference in the strain rates of the wheel and rail.

Linear creep theory asserts that creep forces and moments exerted on the wheel by the rail are directly proportional to the product of $(1/V)$ and the differences in the linear and angular velocities of the wheel and rail at the point of contact. These nondimensional ratios of linear velocities are called lateral or longitudinal creep depending on the direction of the relative linear velocity. The term spin creep refers to the ratio of the relative angular velocity about the normal to the contact area and the forward speed, V. The constants of proportionality are called creep coefficients. Generally, the vector creep force is resolved into lateral and longitudinal components in the plane of the contact area between wheel and rail. The lateral component is comprised of a term due to the lateral creep and a term due to the spin creep. The former is called the lateral creep force while the latter is called the lateral/spin creep force. The longitudinal component of the vector creep force consists of a single term due to the longitudinal creep force. The vector moment about an axis normal to the contact area is comprised of a term due to the lateral creep and a term due to the spin creep. The former is called the lateral/spin creep moment and the latter is called the spin creep moment.

When the relative rates of strain in the contact area become too large, gross sliding of the wheel over the rail occurs and the creep forces and moments are no longer proportional to the creepages but are determined by the level of adhesion. Linear creep theory does not predict this gross sliding and thus cannot be used to investigate effects associated with this type of behavior.

A simplified representation for the force exerted on the wheelset due to contact of the wheel flange with the rail is used. This force is identical to that used by Cooperrider [4, 5], Gilchrist [3], and Law and Brand [6]. The flange force is modeled as a linear spring which resists lateral displacement of the wheelset once the flange clearance is taken up.

Kinematics. As discussed in [6], the roll angle, ϕ, and vertical displacement, q_3, of the wheelset are uniquely determined by the lateral displacement and yaw angle of the wheelset when the rails do not deflect. When the rails do deflect, the roll angle and vertical displacement are determined by the lateral displacement and yaw of the wheelset plus the lateral and vertical displacement of each rail. In [6], these were given as:

$$\phi = (r_L - r_R)/2a = a_0 q_1 + a_1 q_1^3 \tag{1}$$

$$q_3 = (r_L + r_R - 2r_0)/2a = b_1 q_1^2 - c_1 q_2^2 \tag{2}$$

If the left and right rails are displaced vertically, y_1 and y_2, and laterally, z_1 and z_2, respectively, then

$$\phi = (r_L - r_R + y_1 - y_2)/2a \tag{3}$$

and

$$q_3 = (r_L + r_R - 2r_0 + y_1 + y_2)/2a \tag{4}$$

Since the wheel radii at the contact points, r_L and r_R, depend on q_1, q_2 and the rail displacements, these expressions for ϕ and q_3 may be written in terms of q_1, q_2, and the nondimensional rail displacements as given in the Appendix. The wheel and rail head contours determine the functional form of these expressions which are generally nonlinear.

The angles of inclination to the axle of the contact planes between wheel and rail, δ_L and δ_R, are also determined by the lateral wheelset displacement and the rail displacements for given wheel and rail head contours. The difference of these contact angles, $\delta_L - \delta_R$, appears explicitly in the equations of motion. This quantity is generally nonlinear and is discussed in detail in the Appendix.

Equations of Motion. The nondimensional equations of motion for a wheelset model as described in the previous sections are (with the dot notation indicating differentiation with respect to nondimensional time, τ)

$$\ddot{q}_1 + (\mu_{11}/\nu)\dot{q}_1 + \epsilon_1' q_1 - \mu_{12}(\delta_L - \delta_R)/2\rho + \gamma' \partial q_3/\partial q_1 - F'_{\text{RAIL}} + \dot{q}_2[\mu_{12}/\nu - (J_1\nu/\rho)\partial\phi/\partial q_1] - \mu_{11} q_2 = -(\mu_{11}\rho/\nu)[(\partial\phi/\partial\zeta_1)\dot{\zeta}_1 + (\partial\phi/\partial\zeta_2)\dot{\zeta}_2 + (\partial\phi/\partial\eta_1)\dot{\eta}_1 + (\partial\phi/\partial\eta_2)\dot{\eta}_2] \tag{5}$$

and

$$J_2\ddot{q}_2 + [(\mu_{22} + \mu_{33})/\nu]\dot{q}_2 + (\epsilon_4' - 2c_1\gamma' + \mu_{12})q_2 + [(J_1\nu/\rho)\partial\phi/\partial q_1 - (\mu_{12}/\nu)]\dot{q}_1 + \mu_{33}\phi/\rho - \mu_{22}(\delta_L - \delta_R)/2\rho - M_{Ry}' = \mu_{33}(\eta_1 - \eta_2)/2\rho - [(J_1\nu/\rho) - (\mu_{12}\rho/\nu)][(\partial\phi/\partial\zeta_1)\dot{\zeta}_1 + (\partial\phi/\partial\zeta_2)\dot{\zeta}_2 + (\partial\phi/\partial\eta_1)\dot{\eta}_1 + (\partial\phi/\partial\eta_2)\dot{\eta}_2] \tag{6}$$

The individual terms are defined in the Nomenclature. These equations differ from those analyzed in [6] in that they include the effects of spin creep (terms involving μ_{12} and μ_{22}), gyroscopic effects (terms involving J_1), and rail irregularities (terms involving ζ_1, ζ_2, η_1, and η_2). As discussed previously, ϕ and q_3 also depend on the rail irregularities. The nondimensional force and moment on the wheelset due to flange contact are F'_{RAIL} and M_{Ry}', respectively. The forces and moments due to creep are those

involving μ_{11}, μ_{12}, μ_{22}, and μ_{33}. The suspension restoring force and moment are $\epsilon_1' q_1$ and $\epsilon_4' q_2$, respectively, while the terms previously called gravitational stiffness effects [14] are $\gamma' \partial\ q_3/\partial q_1$ and $-2c_1\gamma' q_2$. The nondimensional quantity, ν, is the ratio of the forward speed, V, to a reference speed, V_0. This reference is the value of linear critical speed calculated by neglecting gyroscopic and spin creep terms. Although suspension damping was not considered, it is expected that only the quantitative results would be influenced while the qualitative aspects of the dynamics would be unchanged.

These equations are discussed in greater detail in the Appendix.

Rail Irregularities. It was desired to determine the dynamic response of the wheelset to random rail irregularities. Most data for rail alignment are given as plots of the power spectral density of lateral alignment as expressed in in.^{2}/(cycle/ft) versus the spatial frequency, f, in cycle/ft or the spatial wavelength in ft/cycle [15, 16]. As the equations of motion of the wheelset are nonlinear and must be integrated numerically, time histories of the nondimensional alignment of the left and right rails, ζ_1 and ζ_2, are required as inputs to the equations.

It is shown [17] that the power spectral density of a series of equally spaced impulses of random magnitude is approximately constant with frequency up to $1/(2\Delta t)$ where Δt is the time interval between impulses.

Actual experimental data as shown in Fig. 2 indicate that the lateral alignment spectra may be approximated by

$$G(f) = K_2/f^n \text{in.}^2/(\text{cycle/ft}) \tag{7}$$

over a range of f from about 0.01 to 0.1 cycles/ft where n is usually between 2 and 3. As data are not available in [15, 16] below approximately 0.01 cycles/ft, it was assumed for the purposes of this investigation that the power spectrum of the lateral alignment for each rail was of the following form

$$G(f) = \begin{cases} K_0 f^2, & 0 < f < 0.001 \text{ cycles/ft} \\ K_1, & 0.001 \leq f \leq 0.01 \text{ cycles/ft} \\ K_2 f^{-2}, & 0.01 \leq f < \infty \text{ cycles/ft} \end{cases} \tag{8}$$

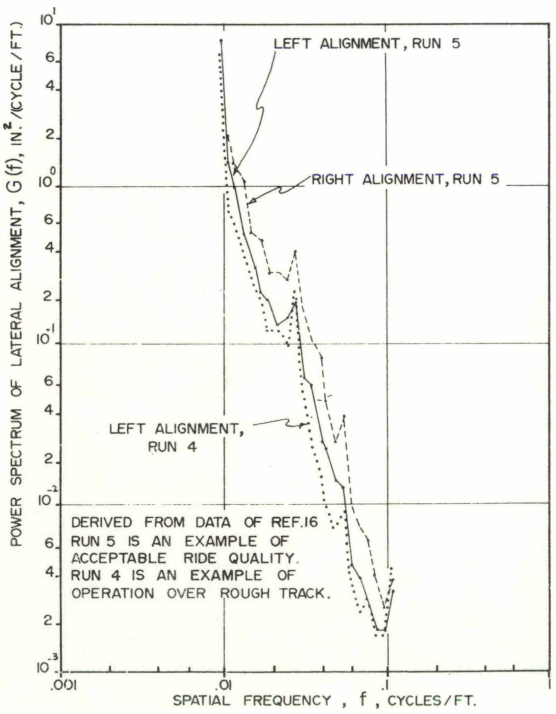


Fig. 2 Power spectra of lateral alignment (from reference [16])

Table 1 Nominal dimensional design parameters [18]

r_0 = 1.33 ft (.405 m)	V_c = 446.0 fps (135.9 m/s)
a_0 = 0.0454	k_θ = 1.25 x 10^6 lb ft/rad (1.69 x 10^6 N-m/rad)
a_1 = 7.92	k_x = 5.0 x 10^4 lb/ft (7.297 x 10^5 N/m)
b_1 = 2.9549	a = 2.5 ft (0.762 m)
c_1 = 0.0227	δ' = 0.03 ft (0.00914 m)
m = 62.5 slugs (912.1 kg)	k_R = 1.0 x 10^6 lb/ft (14.59 x 10^6 N/m)
I_1 = 100 slug ft^2 (135.5 kg m^2)	f_{11} = 2.49 x 10^6 lb/wheel (1.108 x 10^7 N/wheel)
I_2 = 348 slug ft^2 (471.5 kg m^2)	f_{12} = 2.05 x 10^4 lb ft/wheel (2.779 x 10^4 N-m/wheel)
W = 36950 lb (1.6435 x 10^5 N)	f_{22} = 258 lb ft^2/wheel (106.6 N-m^2/wheel)
V_0 = 460.8 fps (140.5 m/s)	f_{33} = 2.75 x 10^6 lb/wheel (1.223 x 10^7 N/wheel)

NOMINAL NON-DIMENSIONAL PARAMETERS

J_1 = 0.256	μ_{11} = 0.93781
J_2 = 0.891	μ_{12} = 0.00309
ρ = 0.532	μ_{22} = 0.15547 x 10^{-4}
ϵ_1' = 0.02354	μ_{33} = 1.0357
ϵ_4' = 0.09416	γ' = 0.00734
ϵ_5 = 0.47079	λ = 0.012

A series of impulses with a Gaussian amplitude distribution and nominally zero mean was generated by the digital computer for each rail. The imposed functional form of $G(f)$ for $0 < f \leq .001$ cycles/ft served as a high-pass filter to remove any d-c component due to the finite length of the pulse train.

The power spectral density of the output of an element with frequency response function $W(\omega)$ is given by

$$G_0(\omega) = |W(\omega)|^2 G_i(\omega) \tag{9}$$

Since the pulse train had a power spectrum $G_i(\omega)$ that was approximately constant, the frequency response function $W(\omega)$ required to obtain the characteristics of $G(f)$ as described by equation (8) is

$$W(\omega) = \frac{j\omega\tau_1}{1 + j\omega\tau_1} \cdot \frac{1}{1 + j\omega\tau_2}$$

$$= \frac{j\omega\tau_1}{(1 - \omega^2\tau_1\tau_2) + j\omega(\tau_1 + \tau_2)} \tag{10}$$

where

$$\tau_i = 1./(2\pi a \nu f_i) \text{ nondimensional sec}$$

and

$$f_1 = 001 \text{ cycles/ft}, f_2 = 0.01 \text{ cycles/ft}$$

When $\omega\tau_1, \omega\tau_2 \ll 1$

$$|W(\omega)|^2 \simeq (\omega\tau_1)^2 \tag{11}$$

and when $\omega\tau_1, \omega\tau_2 \gg 1$

$$|W(\omega)|^2 \simeq 1/(\omega\tau_2)^2 \tag{12}$$

The standard deviation of the series of impulses was adjusted to obtain the desired rms values of ζ_1 and ζ_2.

A straight line approximation was fitted to the data for the left alignment spectrum of Run 4 (corresponding to rough track) of [16]. Thus, the spectrum was assumed to vary as f^2 over $0 < f \leq 0.001$ cycles/ft, to be constant over $0.001 < f \leq 0.01$ cycles/

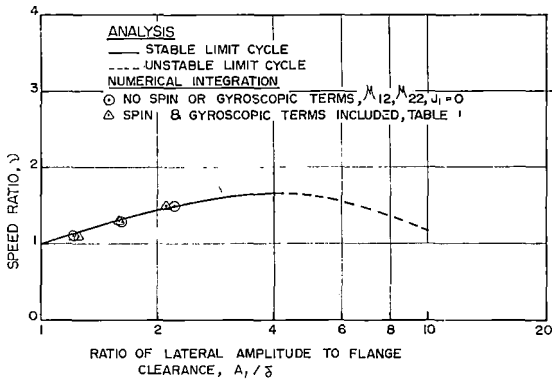


Fig. 3 Effects of spin creep and gyroscopic terms on amplitudes of limit cycles

ft, and to be given by $1 \times 10^{-6}/f^3$ over $0.01 \leq f \leq \infty$ cycles/ft. The rms lateral alignment was then calculated to be approximately 0.1195 in. Hence a range of the rms lateral alignment of 0.05 to 0.5 in. should encompass physically meaningful values for actual track, if the assumed spectrum shape is appropriate for $f < 0.01$ cycles/ft.

Discussion of Results

The equations of motion (5) and (6) were integrated numerically using a fourth order Runge Kutta routine. The IBM 360 Continuous System Modeling Program (CSMP) was used. Time histories of various outputs such as q_1, q_2, and L/V for each wheel were obtained as the wheelset traversed approximately one mile of nominally tangent track. The lateral displacement of each rail was considered to be a random function as previously described. The vertical displacement of the rails was assumed to be zero. The design parameters of the wheelset are those of a wheelset considered in [18] and are presented in Table 1. The linear critical speed for this wheelset is high compared to operating speeds of conventional trains. However, it is to be expected that conventional wheelsets having lower critical speeds would exhibit similar response characteristics within the same range of values of ν based on their critical speeds.

As mentioned earlier, the only work known to this writer that has attempted to relate dynamic levels of L/V to operational safety is that of Matsudaira [12]. Matsudaira presents a plot of L/V versus "working period of side thrust" of wheel against rail. Working period of side thrust as defined by Matsudaira is apparently the time duration (from rise to fall) of the side thrust of the wheel against the rail. If the inverse of this quantity is called frequency, the "safety limit" for L/V is given as 0.8 for frequencies up to 20 Hz. Above 20 Hz, the safety limit increases linearly with frequency to a value of 4.0 at 100 Hz. Matsudaira does not discuss the origin or rationale behind the proposed safety limit. The largest values of L/V measured during tests of two trucks over the speed range from 112 to 155 mph all fell below the safety limit. A value of L/V of about 1.1 or 1.2 was measured at a frequency of about 15 Hz during hunting.

Matsudaira also notes that the observed hunting was largely affected by track condition in that it did not appear on a rainy day and hardly occurred when the track was "well-arranged." Thus he concludes that hunting is strongly affected by (*a*) creep or the coefficient of friction, and (*b*) the magnitude of the disturbance due to "out-of-alignment" track.

Koci [19] has stated that since the dynamic loads are very short in duration "...they are of interest from the standpoint of rail batter and shock loads locally affecting the wheel and rail..." and thus, "...they are not of primary interest from the standpoint of tracking or derailments since they are not long-lived enough for any significant displacement or change in location to take place." However, Gilchrist [3] takes the view that "...a large ratio of lateral to vertical suspension force must necessarily increase the risk of derailment." He goes on to state that for two types of British Rail freight cars, one (called the "pallet van") had a significantly worse derailment record than the second (called the "covered van"). The pallet van was worse because it developed higher ratios of L/V at a given speed and it had a lower lateral frequency than the covered van. The lower lateral frequency allowed the adverse L/V conditions to persist longer.

As Gilchrist indicates [3], a comprehensive derailment theory must start from a realistic model of wheel/rail interaction under flange contact conditions. This will necessitate: (1) the introduction of nonlinear creep effects, (2) consideration of a range of contact angles, δ_L and δ_R, such that small angle assumptions are no longer valid, (3) a realistic dynamic model of the rail structure, and, (4) detailed consideration of the shape of the cross-sectional wheel and rail head contours. If such a model were devised, derailment could be predicted by observing the time history of the height of the wheel. When the wheel height exceeds the flange height, derailment is imminent.

In the present analysis, linear creep theory is used and δ_L and δ_R are assumed sufficiently small that small angle assumptions are valid. The effects of rail compliance are modeled by a linear rail spring with deadband as indicated in equations (23) and (24). This implicitly assumes that the rail natural frequencies are much higher than any forcing frequencies imposed by the wheelset. It also assumes that the rail only moves with the wheel once the flange clearance is taken up. Thus use of this model assumes that the predominating effect of wheel flange to rail contact is that due to the lateral component of the force of the rail on the wheel that is normal to the wheel/rail contact area. This is probably a reasonable assumption during flange contact. The model will not accurately predict vertical wheel displace-

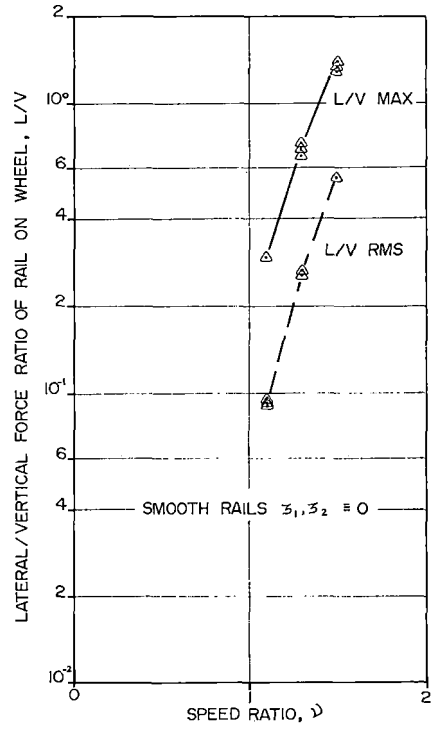


Fig. 4 Effect of speed on L/V for smooth, perfectly aligned track

ments under conditions of flange contact but should predict trends and general levels of forces between wheel and rail during flange contact conditions. Thus, the main value of the current work is to indicate the sensitivity of L/V ratios to changes in various parameters such as speed, rms lateral rail displacement, wheel profile curvature, flange clearance, and rail stiffness.

As discussed, Matsudaira has indicated a safety limit of $L/V = 0.8$ for frequencies up to 20 Hz, increasing to $L/V = 4.0$ at 100 Hz. The predominant frequencies measured in the current investigation ranged from about 8 Hz at $\nu = 0.6$ to about 24 Hz at $\nu = 1.2$. It is interesting to note that the "usual" linear variation with speed of frequency of the secondary hunting mode does not occur. The frequency of 24 Hz at $\nu = 1.2$ is substantially higher than the value of about 16 Hz that one might expect if frequency is proportional to speed. Matsudaira also noted that the frequency of vibration during hunting was significantly larger than the value calculated on the basis of "kinematical nosing." He attributed the discrepancy to the effects of the "repulsion of rail against wheel flange."

Koci [19] indicates that for steady state conditions, a value of L/V of about 0.9 should "...start being of concern," although he indicates that steady values of 1.5 may be reached before wheel climbing will occur. Gilchrist [3] calculates a critical value of 1.47 on the basis of quasi-static theory of derailment. He indicates on the basis of other studies that this value may be exceeded by a considerable amount during hunting.

Effects of Spin Creep and Gyroscopic Terms. The analytical method of [6, 20] was applied to determine the amplitudes and stability of any limit cycle oscillations of the wheelset on perfectly smooth track. In the application of this method, spin creep terms and those involving gyroscopic effects are neglected. Thus all terms involving μ_{12}, μ_{22}, and J_1 were neglected in equations (5), (6). The right hand sides of these equations were also set to zero to investigate limit cycle behavior on smooth track. The variation of the speed ratio, ν, with the amplitude of the limit cycle is shown in Fig. 3. The linear critical speed (i.e., the speed at which the equilibrium point becomes unstable) for this case is 460.9 fps. The analysis was checked by integrating the equations of motion numerically for various initial conditions.

To determine the effects of gyroscopic terms and spin creep, these terms were included in the equations of motion with the values of μ_{12}, μ_{22}, and J_1 as listed in Table 1. The linear critical speed with gyroscopic terms alone increased to 498.7 fps while the linear critical speed with spin creep and gyroscopic terms is 446 fps. Thus gyroscopic effects are stabilizing while spin creep effects are destabilizing for this wheelset.

After solving the eigenvalue problems for the linear critical speeds for these cases, the equations were integrated numerically to determine the amplitudes of the limit cycles. As shown in Fig. 3, the inclusion of spin creep and gyroscopic terms had slight effect on the amplitudes of the limit cycles.

This result should be approached with caution for higher values of A_1/δ since the detailed flange shape was not considered and linear creep theory is not valid for the high values of creep that occur during severe hunting motions.

Effects of Speed on L/V for Smooth Rails. The effects of speed on the ratio of lateral/vertical force of the rails on the wheels for smooth, perfectly aligned rails are shown in Fig. 4. As sustained stationary or limit cycle oscillations are not possible for this configuration for speeds less than $V_c = 446$ fps ($\nu = 0.967$), the data shown correspond to limit cycle conditions. As can be seen, the rms value of L/V increases from about 0.09 at $\nu = 1.1$ to 0.55 at $\nu = 1.5$. At the same time, the maximum value of L/V increases from about 0.3 to 1.4 as ν increases from 1.1 to 1.5.

Thus, although flange contact does occur, it does not appear that derailment is imminent for values of ν less than about 1.4 (taking $L/V = 1.0$ as the critical value) for theoretically *smooth* track. However, sustained flange contact conditions will contribute to increased wear and tear on wheels and rails as well as increased vibration levels in the car.

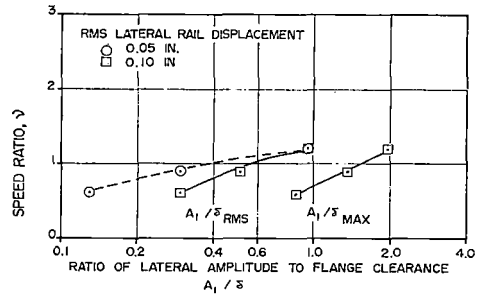


Fig. 5 Effects of rms lateral rail displacement on lateral amplitude of wheelset

Effects of Lateral Rail Irregularities. The effects of lateral rail irregularities (as measured by rms lateral rail displacement) on wheelset lateral amplitude and L/V are shown in Figs. 5 and 6. The lateral rail displacements, ζ_1 and ζ_2, are random variables which have prescribed power spectra as described previously. Although ζ_1 and ζ_2 have the same rms value, they are independent of one another.

For low values of the speed ratio ν, $(A_1/\delta)_{\text{rms}}$ depends strongly on the rms lateral rail displacement. As ν increases to 1.2 (such that the linear critical speed is exceeded), $(A_1/\delta)_{\text{rms}}$ is essentially independent of rail roughness. As expected, both the maximum and rms values of A_1/δ increase with ν. If one assumes that the limit cycle motion of the wheelset on smooth track is approximately harmonic, then from Fig. 3, $(A_1/\delta)_{\text{rms}} \cong (0.707)\ (1.4) = 0.99$ for $\nu = 1.2$. This corresponds approximately to the value for rough track from Fig. 5. However, the value of (A_1/δ) max = 2 on rough track exceeds that on smooth track, (A_1/δ) max = 1.4.

As shown in Fig. 6, L/V increases with rms lateral rail displacement. For values of $\nu \lesssim 1$, the rms levels of L/V are only slightly affected by ν. As rms lateral rail displacement increases over 0.1 in., the curves of $(L/V)_{\text{rms}}$ for ν both above and below 1 become closer. Thus it is important to operate not only with

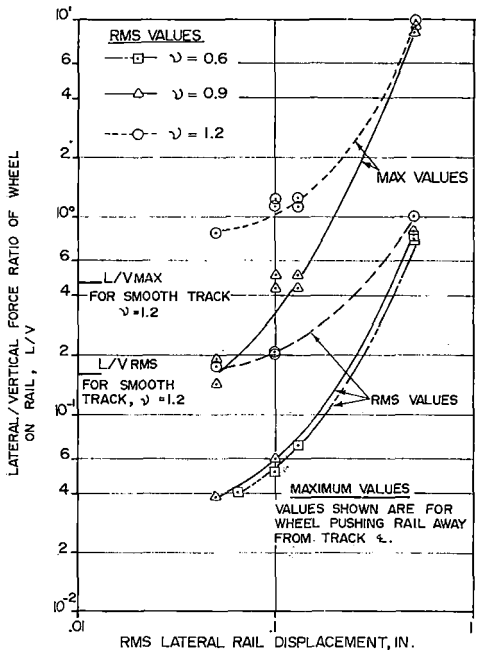


Fig. 6 Effects of rms lateral rail displacement on L/V

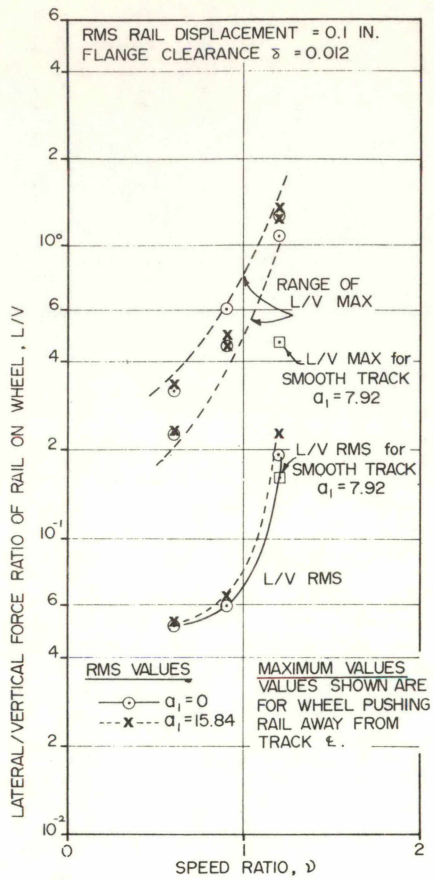


Fig. 7 Effect of wheel profile curvature parameter a_1 on L/V

$\nu < 1$ but with as smooth a track as possible. The maximum values of L/V for $\nu = 0.9$ do not offer serious concern with regard to derailment for values of rms lateral rail displacement less than about 0.2. However, for $\nu = 1.2$ the maximum value of L/V reaches 1 when the rms lateral rail displacement is 0.1 in. and it increases quite rapidly thereafter.

If the value of rms lateral rail displacement of 0.1 in. is taken as representative of "rough" track as suggested by calculations based on the data of [16], it is seen that the rms values of L/V for rough track do not increase very much over those for smooth track when the wheelset exceeds the linear critical speed. Thus this corroborates Gilchrist's observation [3] that "...when the vehicle has exceeded the critical speed the lateral motion ...takes the form of a limit-cycle oscillation which is largely independent of alignment irregularities."

However, if very rough track is encountered at $\nu < 1$ such that the rms lateral rail displacement significantly increases, the levels of L/V and A_1/δ may increase to values close to those for $\nu > 1$ on smooth track. Thus, one may think that the wheelset is hunting in the classical sense. Matsudaira [2] indicates this possibility. The "...occurrence (of hunting) depends largely on the magnitude of disturbance owing to the track out-of-alignment."

Effects of Wheel Profile Curvature. The effects of the wheel profile curvature parameter, a_1, on L/V for rms lateral rail displacement equal to 0.1 in. are shown in Fig. 7. Generally, increasing values of a_1 correspond to increasing wheel wear. When $a_1 = 0$, the wheel has a straight taper. The value of $a_1 = 15.84$ corresponds to double the nominal value. For low values of ν, it does not appear that a_1 materially affects L/V. However, as ν increases, the rms level of L/V for $a_1 = 15.84$ is higher than that for $a_1 = 0$. Thus wheel wear leads to higher rms L/V levels. The effects of a_1 on the maximum levels of L/V are inconclusive. Thus, although it is generally accepted that worn wheels lead to an increased propensity of the wheelset to hunt, this is probably due more to an increased value of a_0 with wheel wear than to an increase in a_1. The linear critical speed, V_c, decreases with increasing a_0. A reasonable approximation is that $V_c^2 \sim 1/a_0$. Although a_0 should be small (but not zero) to insure stability, the importance of the contribution of a "good" choice of a_1 should not be overlooked. It is desirable to select a wheel profile (and hence a combination of a_0 and a_1) that is stable with respect to wear. Thus the desired dynamic performance will not deteriorate with increasing wheel wear.

Effects of Flange Clearance. The effects of flange clearance, δ, on L/V for a value of rms lateral rail displacement equal to 0.1 in. are shown in Fig. 8. The nominal value of $\delta = 0.012$ corresponds to $\delta' = 0.36$ in. while the other value shown, $\delta = 0.024$, corresponds to $\delta' = 0.72$ in. When a value of $\delta' = 0.18$ in. was used, the magnitude of the lateral rail irregularities were such that there was no flange clearance whatsoever. For values of $\nu < 1$, it appears that doubling the nominal flange clearance leads to a slight decrease in the rms levels of L/V. However, as ν increases over 1, this trend is reversed and at $\nu = 1.2$ the larger flange clearance of $\delta = 0.024$ gives $(L/V)_{rms} = 0.5$, more than twice the value of $(L/V)_{rms} = 0.205$ for $\delta = 0.012$. This behavior might have been expected as it was shown in [6, 20] that for

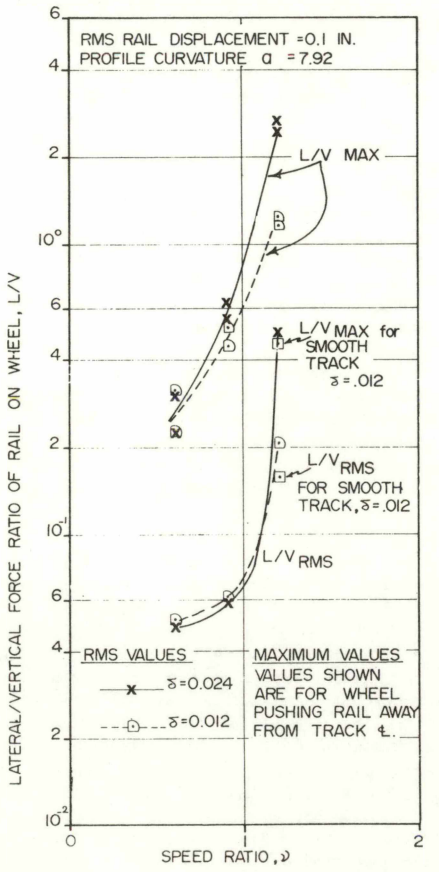


Fig. 8 Effect of nondimensional flange clearance δ on L/V

values of ν greater than critical, larger flange clearances lead to more violent hunting. When $\nu \lesssim 1$ and the wheelset is stable, it would be expected that flange contact would occur less often for the case of larger flange clearance. Hence the values of L/V should be less.

The maximum levels of L/V are about the same for $\nu = 0.6$. As ν increases to 0.9 and higher, the wheelset with the larger flange clearance develops higher maximum levels of L/V. At $\nu = 1.2$, it is questionable whether the value of (L/V) max $\cong 2.5$ for $\delta = 0.024$ would be acceptable.

Effects of Lateral Rail Stiffness. The effects of lateral rail stiffness, ϵ_5, on L/V for a value of rms lateral rail displacement equal to 0.1 in. are shown in Fig. 9. The nominal value of $\epsilon_5 = 0.4708$ corresponds to $k_R = 1.0 \times 10^6$ lb/ft which is close to the value of 40 ton/in. (or 0.96×10^6 lb/ft) used in [3] as the "... best available estimate of lateral track stiffness."

When ϵ_5 was decreased by a factor of 10 to 0.04708, the wheelset became unstable for $\nu = 1.2$. Thus an insufficiently stiff track cannot sustain the large lateral forces imposed by the wheelset during hunting conditions.

For $\nu < 1$, lower values of L/V were obtained for the rails having lower values of ϵ_5. As ν increased, the levels of L/V increased. The rms levels of L/V for the rails with $\epsilon_5 = 4.708$ were approximately triple those of the nominally stiff rails. The maximum levels of L/V for the rails with $\epsilon_5 = 4.708$ were almost five times those of the nominally stiff rails at $\nu = 1.2$ ($L/V_{\max} \cong 5$ for $\epsilon_5 = 4.708$ as opposed to $L/V_{\max} \cong 1.2$ for $\epsilon_5 = 0.4708$ at $\nu = 1.2$). It is very probable that these high levels of $L/V_{\max}$ would be unacceptable.

Thus a middle course must be steered in choosing values of lateral rail stiffness. If too low a value is chosen, the rail will not be sufficiently stiff to sustain wheelset hunting. If too high a value is chosen, the levels of L/V (in particular the maximum values) will be too high.

Conclusions

The results discussed previously indicate that random lateral track irregularities contribute to higher levels of L/V than those predicted for "smooth" track. For a given track, L/V increases with speed. The rate of increase of L/V becomes larger as the linear critical speed is exceeded. For relatively "smooth" track (rms lateral rail displacement less than 0.1 in.), the rms levels of L/V are only slightly higher than those for "perfectly smooth" track at $\nu = 1.2$.

At a constant speed, L/V increases with rms lateral rail displacement. The rate of increase becomes larger above an rms value of about 0.1 in. For speeds below the linear critical speed, the values and variations of L/V with rms lateral rail displacement are only slightly affected by speed.

As the wheel profile curvature parameter, a_1, is increased, rms values of L/V are slightly increased. It is suggested that the primary effect of wheel wear for wheels having an initial straight taper is to increase a_0 and a_1. The decrease in linear critical speed and increased levels of L/V at a given speed are then due primarily to the increased a_0.

Below the linear critical speed, the rms values of L/V decrease slightly with increasing flange clearance, δ. However, above the linear critical speed, these values increase sharply with increasing flange clearance.

Values of L/V are greatly affected by lateral rail stiffness, k_R. For low values of k_R (1.0×10^5 lb/ft), the rails are not stiff enough to sustain the large forces prevalent during wheelset hunting. At large values of k_R (1.0×10^7 lb/ft), both rms and maximum values of L/V are much larger than those for a nominal value of k_R (1.0×10^6 lb/ft).

Acknowledgment

The author extends his appreciation to the Department of Engineering Mechanics of Clemson University for financial support of this investigation.

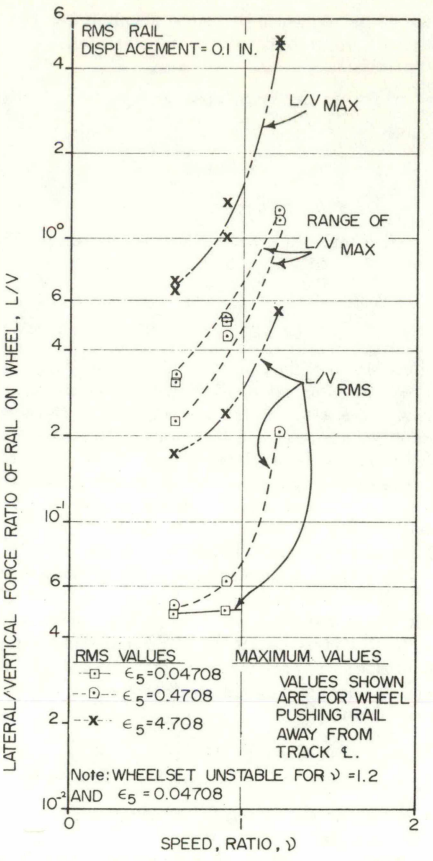


Fig. 9 Effect of lateral rail stiffness ϵ_5 on L/V

APPENDIX

If it is assumed that for perfectly smooth rigid rails the roll angle and vertical displacement of the wheelset centroid are given by equations (1) and (2), then

$$r_L/a = a_0 q_1 + b_1 q_1^2 + a_1 q_1^3 - c_1 q_2^2 + \rho \tag{13}$$

$$r_R/a = -a_0 q_1 + b_1 q_1^2 - a_1 q_1^3 - c_1 q_2^2 + \rho \tag{14}$$

Since the lateral displacement of the left rail, ζ_1, is measured parallel to the original rail plane (as is q_1), the change in r_L/a due to $\zeta_1 > 0$ is essentially the same as the change due to a negative q_1 of the same magnitude. Thus, to a good approximation

$$r_L/a = a_0(q_1 - \zeta_1) + b_1(q_1 - \zeta_1)^2 + a_1(q_1 - \zeta_1)^3 - c_1 q_2^2 + \rho \tag{15}$$

Similarly, for a lateral displacement of the right rail, ζ_2,

$$r_R/a = -a_0(q_1 - \zeta_2) + b_1(q_1 - \zeta_2)^2 - a_1(q_1 - \zeta_2)^3 - c_1 q_2^2 + \rho \tag{16}$$

These expressions when substituted in equations (3) and (4) yield equations for ϕ and q_3 that relate the roll and vertical displacement of the wheelset to the yaw and lateral displacement of the wheelset as well as the lateral and vertical displacement of each rail. The angle of inclination to the axle of the contact plane for the left rail is approximately

$$\delta_L \cong d(r_L/a)/d\Delta_L' \tag{17}$$

where Δ_L' is the nondimensional displacement of the contact point relative and parallel to the axle for the left wheel. Similarly,

$$\delta_R \cong -d(r_R/a)/d\Delta_R' \tag{18}$$

To the first order

$$\Delta_L' = q_1 - \zeta_1 + \rho\phi \cong q_1 - \zeta_1 \tag{19}$$

$$\Delta_R' = q_1 - \zeta_2 + \rho\phi \cong q_1 - \zeta_2 \tag{20}$$

neglecting terms of magnitude ρa_0 with respect to unity. Thus, using equations (15) and (16)

$$(\delta_L - \delta_R)/2 = 2b_1q_1 - b_1(\zeta_1 + \zeta_2) + (3a_1/2)[(q_1 - \zeta_1)^2 - (q_1 - \zeta_2)^2] \tag{21}$$

The nondimensional equations (5) and (6) were derived by Lagrangian techniques as explained in [20] under the assumptions described previously. In the development of these equations various terms are neglected that are small compared with unity. The dimensional lateral creep force is $(2f_{11}/V)(\dot{x} + r_0\dot{\theta} - V\theta)$. The terms on the right-hand side of the lateral force equation (5) are due to the term $2f_{11}r_0\dot{\phi}/V$ in the lateral creep force. The yaw moment due to longitudinal creep is $(2f_{33}a^2/V)[\dot{\phi} + (r_L - r_R)\Omega/2a]$ and that due to lateral/spin creep is $(-2f_{12}/V)(\dot{x} + r_0\phi - V\theta)$. The yaw moment due to the gyroscopic effect of $\dot{\phi}$ is $I_1\Omega\dot{\phi}$. When the term $\mu_{33}(\eta_1 - \eta_2)/2\rho$ is transferred to the left-hand side of (6) and combined with $\mu_{33}\phi/\rho$, the result corresponds to $(2f_{33}a^2/V)(r_L - r_R)\Omega/2a$. The remaining terms on the right-hand side of (6) are due to $(I_1\Omega - 2f_{12}r_0/V)\dot{\phi}$. As ϕ depends on q_1, ζ_1, ζ_2, η_1, and η_2, $\dot{\phi}$ depends on the first time derivatives of these quantities.

$$\dot{\phi} = (\partial\phi/\partial q_1)\dot{q}_1 + (\partial\phi/\partial\zeta_1)\dot{\zeta}_1 + (\partial\phi/\partial\zeta_2)\dot{\zeta}_2 + (\partial\phi/\partial\eta_1)\dot{\eta}_1 + (\partial\phi/\partial\eta_2)\dot{\eta}_2 \tag{22}$$

The partial derivatives are obtained from equation (3) after substitution of (15) and (16).

As a first attempt at analysis of this problem, the rails were assumed to be rigid but have lateral irregularities as described by the variations in ζ_1 and ζ_2 along the track. The "effects" of lateral rail compliance were represented by the terms F'_{Rail} in (5) and M_y' in (6).

$$F'_{\text{RAIL}} = \begin{cases} -\epsilon_5(q_1 - \delta - \zeta_1), & q_1 > \delta + \zeta_1 \\ 0, & q_1 < \delta + \zeta_1;\ q_1 > -(\delta - \zeta_2) \\ -\epsilon_5(q_1 + \delta - \zeta_2), & q_1 < -(\delta - \zeta_2) \end{cases} \tag{23}$$

$$M_{Ry}' = \begin{cases} \epsilon_5[q_1 - (\delta + \zeta_1)]q_2; & q_1 > \delta + \zeta_1 \\ 0, & q_1 < \delta + \zeta_1;\ q_1 > -(\delta - \zeta_2) \\ \epsilon_5[q_1 + \delta - \zeta_2]q_2; & q_1 < -(\delta - \zeta_2) \end{cases} \tag{24}$$

Thus, once the flange clearance, δ, plus the lateral displacement of the rail was exceeded, a force proportional to displacement opposed lateral movement of the wheelset and exerted a yaw moment M_{Ry}' on the wheelset.

In this investigation, vertical rail irregularities were not considered. Thus $\dot{\eta}_1$ and $\dot{\eta}_2$ are identically zero in equation (22). The lateral rail irregularities ζ_1 and ζ_2 were given by the outputs of the elements described by equation (10) and $\dot{\zeta}_1$ and $\dot{\zeta}_2$ were given by the derivatives of these outputs.

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68—RR-3

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Digital Computer Simulation of Railroad Freight Car Rocking

A mathematical model for the simulation of railroad freight car rocking is presented. The equations of the model are developed into a digital computer program. The model response is validated by two series of test results, and the model is considered reliable for engineering predictions.

Introduction

THE railroads have experienced excessive car rocking in the operation of high capacity (100 tons) freight cars with high centers of gravity. One way to control car rocking is to install energy dissipating devices in addition to the friction damping normally provided in the trucks. The design of such devices is enhanced by the data obtainable from a reliable mathematical model of the rocking freight car. Such a model may be used to obtain the amount and location of additional damping necessary to reduce car rocking to specified levels. It may also be used to evaluate a promising device prior to hardware development and full-scale testing.

The present paper is concerned with the development of a mathematical model for rocking freight cars and its verification by test results. The response of the model is obtained on a digital computer.

Model

The railroad freight car, consisting of a car body and two trucks, is a complicated, three-dimensional dynamic system. Rocking of this system is excited by the periodic occurrence of staggered rail joints which are usually softer than any other point on the rail. When the truck spacing coincides with rail length, the motions of the front and rear trucks are approximately the same, and the significant motions of the car body and trucks are in a plane perpendicular to the track. Thus, in modeling this system, we may restrict the motion of the model to a plane. Such a model is shown in Fig. 1.

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In this model the truck structure between the seat of the suspension springs and rails is assumed to be rigid. Thus, the change in elevation of the seat of the springs corresponds to the change in elevation of the rails, denoted by ζ_R and ζ_L. Similarly, the lateral displacement of the seat of the springs corresponds to the lateral displacement of the rails, denoted by ξ. The vertical action of the truck suspension springs is modeled by two vertical springs each with a stiffness of k, representing the stiffness of two spring groups acting in parallel. The horizontal action of the truck suspension springs is modeled by a horizontal spring with a stiffness of k_H, representing the stiffness of four spring groups acting in parallel. The truck friction dampers act in parallel with the springs with a force of F_T. Each friction damper represents the action of four friction shoes. Also acting in parallel with the springs is a pair of viscous dampers. These may be used to represent inherent structural damping in the truck and additional viscous damping devices to be installed on the truck.

A rigid body is mounted on top of the vertical springs and dampers. This body represents the mass, m, and mass moment of inertia, i, of two truck bolsters. The car body with a mass, M, and mass moment of inertia, I, is located on top of the bolster.

A pair of friction and viscous dampers acts between the car body and the truck. These dampers represent a device under development by American Steel Foundries. The viscous dampers may also be used to represent structural damping due to working of loose parts, and hysteretic and plastic deformation of parts.

The car body may transmit loads to the bolster through the center plate, represented by two point contacts, or the side bearings, or both, depending on the configuration during motion. These configurations are shown in Fig. 2. In Configuration 1 the car body and trucks act as one. Configurations 2 and 3, 4 and 5, 6 and 7 are mirror images. Configurations 2 and 3 represent the situations when the centerplate has ceased to transmit a moment, but there is still a clearance between side bearings and car. Configurations 4 and 5 differ from 2 and 3 in that the side bearing

Nomenclature

b_1 = location of car center of gravity above center plate, in.
b_2 = location of car center of gravity above side bearings, in.
c_1 = location of bolster center of gravity below center plate, in.
c_2 = location of bolster center of gravity below side bearings, in.
c_c = viscous damping coefficient for car, lb sec/in.
c_T = viscous damping coefficient for truck, lb sec/in.
d = radius of center plate, in
e = distance from king pin to side bearings, in.
= distance from king pin to center of spring groups, in.
h = distance between bolster center of gravity and midheight of friction shoes, in.
i = mass moment of inertia of bolsters, lb-in. sec^2
k = truck vertical spring constant, lb/in.
k_H = truck horizontal spring constant, lb/in.
m = mass of bolster, lb-sec^2/in.
s_1 = direction of friction force in truck, right
s_2 = direction of friction force in truck, left
s_3 = direction of friction force in car
s_4 = direction of lateral force in truck
w_B = track gage, in.
y_c = horizontal displacement of car center of gravity, in.
y_T = horizontal displacement of bolster center of gravity, in.
z_c = vertical displacement of car center of gravity, in.
z_T = vertical displacement of bolster center of gravity, in.
D_L = viscous damper load on car, left, lb
D_R = viscous damper load on car, right, lb
F_c = friction damping force on car, lb
F_T = friction damping force on truck, lb
H_1 = top of side bearing, loaded car, above rail, in.

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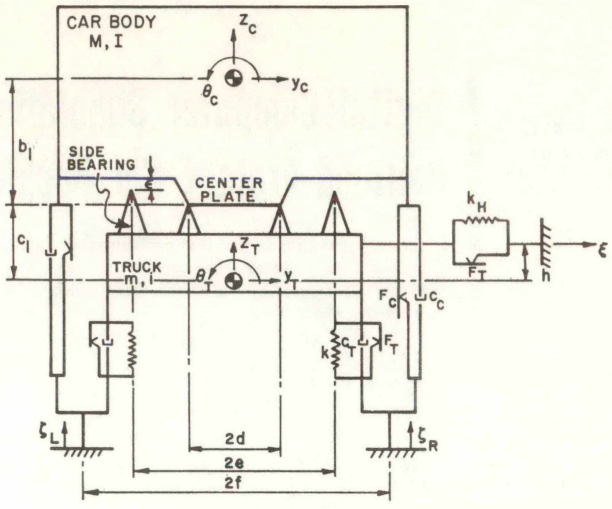


Fig. 1 Model

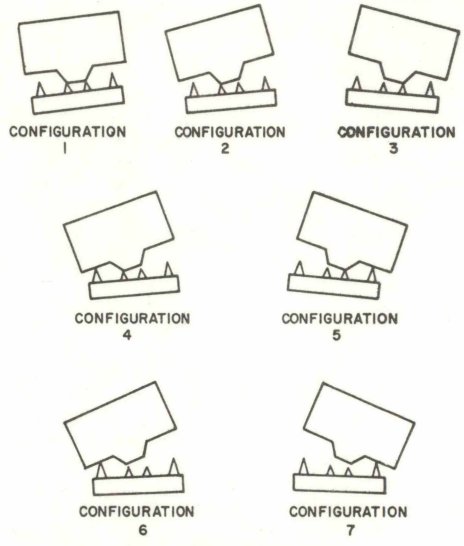


Fig. 2 Model configurations

clearance has been closed. Configurations 6 and 7 represent the situation when the car body is up on side bearings, and the center plate has separated from the truck.

The assumptions and limitations of the model are:

1 An unlimited travel of the vertical springs is assumed. Thus, spring "bottoming" is not considered.

2 An infinite gib clearance is assumed. Note, however, that zero gib clearance can be modeled by making k_H equal to the lateral stiffness of the side frames.

3 Inherent structural damping of car body and trucks, distributed throughout these structures, is represented by equivalent "concentrated" viscous damping devices.

4 The model should not be expected to represent accurately the response of a car whose truck spacing is different from the spacing of the rail joints.

5 The model is valid for small motions. Therefore, the rocking double amplitude should not exceed, say, ten deg.

Derivation of Equations

The motion of the model is described by six coordinates: θ_c, z_c, y_c, θ_T, z_T, y_T (see Fig. 1). The kinetic and potential energies, the Rayleigh dissipation function, and the external work are written in terms of these coordinates below:

Kinetic energy

$$T = \frac{1}{2} M(\dot{y}_c^2 + \dot{z}_c^2) + \frac{1}{2} I\dot{\theta}_c^2 + \frac{1}{2} m(\dot{y}_T^2 + \dot{z}_T^2) + \frac{1}{2} i\dot{\theta}_T^2 \quad (1)$$

Potential energy

$$V = \frac{1}{2} k(z_T + f\theta_T - \zeta_R)^2 + \frac{1}{2} k(z_T - f\theta_T - \zeta_L)^2 + \frac{1}{2} k_H(y_T - h\theta_T - \xi)^2 \quad (2)$$

Rayleigh dissipation function

Nomenclature

I = mass moment of inertia of car, lb-in. sec^2
M = mass of car, lb-sec^2/in.
R_H = horizontal wheel load, lb
R_L = left wheel load, lb
R_R = right wheel load, lb
W = weight of two trucks, excluding bolsters, lb
ϵ = side bearing clearance, in.
ζ_L = left rail profile, in.
ζ_R = right rail profile, in.
θ_c = rotation of car, radian
θ_T = rotation of bolster, radian
λ_1 = vertical center plate load on extreme right, lb
λ_2 = vertical center plate load on extreme left, lb
λ_3 = vertical side bearing load, right, lb
λ_4 = vertical side bearing load, left, lb
λ_5 = horizontal center plate or side bearing load, lb
ξ = horizontal rail profile, in.

Matrices

x = response column matrix
C = damping matrix
F = forcing column matrix
K = stiffness matrix
M = mass matrix

Dots above a symbol indicate differentiation with respect to time.

$$D = \frac{1}{2} c_c(\dot{z}_c + f\dot{\theta}_c - \dot{\zeta}_R)^2 + \frac{1}{2} c_c(\dot{z}_c - f\dot{\theta}_c - \dot{\zeta}_L)^2 + \frac{1}{2} c_T(\dot{z}_T + f\dot{\theta}_T - \dot{\zeta}_R)^2 + \frac{1}{2} c_T(\dot{z}_T - f\dot{\theta}_T - \dot{\zeta}_L)^2 \quad (3)$$

Increment of external work

$$\delta W_e = -Mg\delta z_c - mg\,\delta z_T - s_1F_T\delta(z_T + f\theta_T - \zeta_R) - s_2F_T\delta(z_T - f\theta_T - \zeta_L) - s_3F_c\delta(2f\theta_c - \zeta_R + \zeta_L) - s_4F_T\delta(y_T - h\theta_T + \xi) \quad (4)$$

Note that friction damping of the car is effective only against rocking. The signs of the friction forces are determined as follows:

$$s_1 = \begin{Bmatrix} +1 \\ 0 \\ -1 \end{Bmatrix} \text{ if } (\dot{z}_T + f\dot{\theta}_T - \dot{\zeta}_R) = \begin{Bmatrix} + \\ 0 \\ - \end{Bmatrix} \quad (5)$$

$$s_2 = \begin{Bmatrix} +1 \\ 0 \\ -1 \end{Bmatrix} \text{ if } (\dot{z}_T - f\dot{\theta}_T - \dot{\zeta}_L) = \begin{Bmatrix} + \\ 0 \\ - \end{Bmatrix} \quad (6)$$

$$s_3 = \begin{Bmatrix} +1 \\ 0 \\ -1 \end{Bmatrix} \text{ if } (2f\dot{\theta}_c - \dot{\zeta}_R + \dot{\zeta}_L) = \begin{Bmatrix} + \\ 0 \\ - \end{Bmatrix} \quad (7)$$

$$s_4 = \begin{Bmatrix} +1 \\ 0 \\ -1 \end{Bmatrix} \text{ if } (\dot{y}_T - h\dot{\theta}_T - \dot{\xi}) = \begin{Bmatrix} + \\ 0 \\ - \end{Bmatrix} \quad (8)$$

Since there are more coordinates than degrees of freedom, the following constraints are specified:

$$f_1 = z_c + d\theta_c - z_T - d\theta_T + \underline{\frac{1}{2} b\theta_c^2} = 0 \quad (9)$$

$$f_2 = z_c - d\theta_c - z_T + d\theta_T + \underline{\frac{1}{2} b\theta_c^2} = 0 \quad (10)$$

$$f_3 = z_c + e\theta_c - z_T - e\theta_T + \underline{\frac{1}{2} b\theta_c^2} + \epsilon = 0 \quad (11)$$

$$f_4 = z_c - e\theta_c - z_T + e\theta_T + \underline{\frac{1}{2} b\theta_c^2} + \epsilon = 0 \quad (12)$$

$$f_5 = y_c + b\theta_c - y_T + c\theta_T = 0 \quad (13)$$

where for Configurations 1 through 5, $b = b_1$, $c = c_1$ and for Configurations 6 and 7, $b = b_2$, $c = c_2$.

Note that no more than three constraints apply to any one configuration and f_5 applies to all configurations. The underlined term is retained only for use in the work expression.

The reduced potential energy is obtained using the constraints and five Lagrange multipliers λ_i:

$$\bar{V} = V - \sum_{i=1}^{5} \lambda_i f_i \quad (14)$$

Use of the Lagrange equation

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) + \frac{\partial D}{\partial \dot{q}_i} + \frac{\partial \bar{V}}{\partial q_i} = Q_i \quad (15)$$

leads to the equations of motion and expressions of λ_i. The λ_i are the loads transmitted by the center plate and side bearings.

The matrix equation of motion for each configuration is:

$$M\ddot{x} + C\dot{x} + Kx = F \quad (16)$$

$$x = \begin{Bmatrix} \theta_T \\ y_T \\ z_T \\ \theta_c \end{Bmatrix} \quad (17)$$

Equation (16) is integrated numerically.[1]

The computation starts with the model in Configuration 1. Then the solution of the equations of motion for the model in Configuration 1 is advanced by one integration step; and the center plate, side bearing, wheel, and damper loads are determined (see Fig. 3). The configuration of the model for the next integration step is determined from the center plate and side bearing loads and the side bearing clearance. Then the solution is advanced another integration step. This process is continued until the solution is carried out to a predetermined time.

These computations were programmed in Fortran IV for the IBM 7094 II and the Stromberg-Carlson 4020 off-line plotter.

Validation of Model

The model is validated by simulating the rocking of a particular car and then comparing the simulated and experimental results. The test car is a 100-ton loaded hopper on two American Steel Foundries Ride Control trucks. Experimental results for this car were obtained by American Steel Foundries in a series of tests near Hollidaysburg, Pennsylvania.

The experimental data include continuous traces of car body rock angle and vertical and lateral dynamic rail profiles for the test section of the track. The rail profiles provide the forcing functions of the model ζ_R, ζ_L, and ξ.

The geometry and inertia properties of the car and the stiffness of the suspension springs are known. These properties of the car, translated into parameters of the model, are presented in the Appendix.

Although the normal load on the friction shoes is known, the effective damping forces developed by the friction shoes are not known because the coefficient of friction under operating conditions is unknown. The inherent structural damping in the trucks and the car body under operating conditions is also unknown. Hence, the amounts of damping in the trucks and the car body were estimated by trial and error computation; the measured and computed car body rocking amplitudes at 15.0

[1] Crandall, S. H., *Engineering Analysis*, McGraw-Hill, New York, 1956, p. 177.

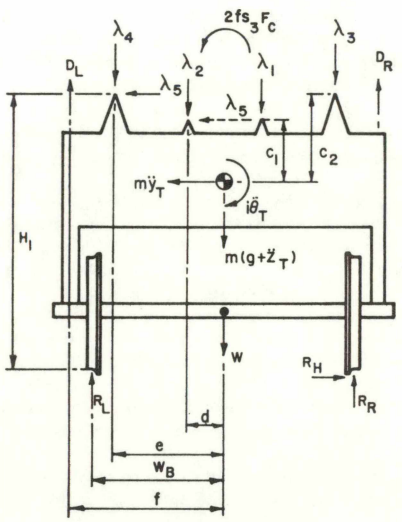


Fig. 3 Dynamic loads on truck

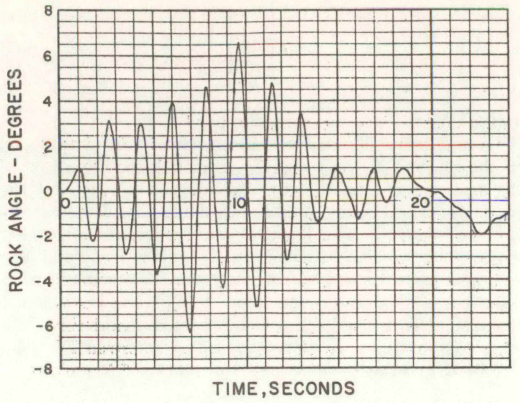


Fig. 4 Computed car body rocking at 16.5 mph

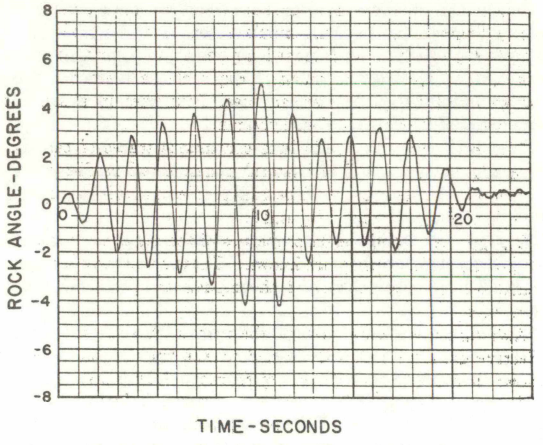


Fig. 5 Measured car body rocking at 16.5 mph

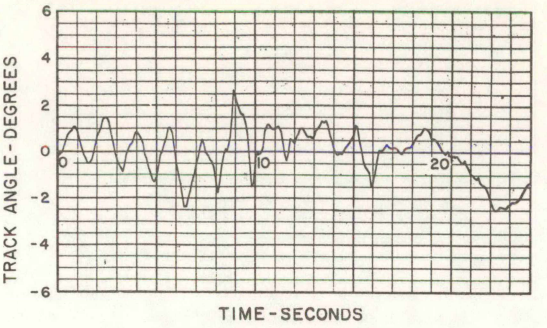


Fig. 6 Measured track angle at 16.5 mph

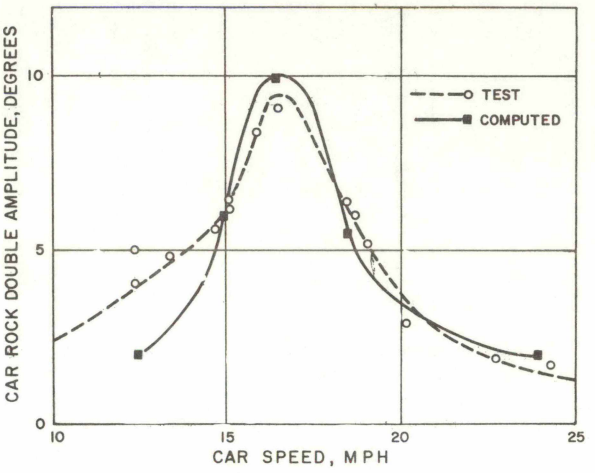


Fig. 7 Comparison of computer simulated and measured car body rocking amplitudes

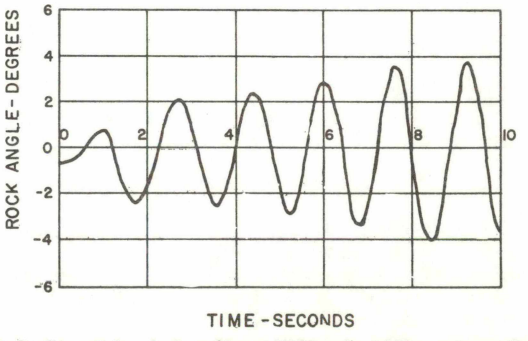


Fig. 8 Computed car body rocking at 17.03 mph, stabilizer not operating

mph were matched. Validity of the model at this speed is therefore reflected in the agreement between model and test frequency and waveform.

The damping so determined was used to calculate car body rocking at car speeds of 12.4, 16.5, 18.5, and 24.0 mph. The computed and measured car body rocking and the measured track angle for the resonant speed of 16.5 mph are shown in Figs. 4, 5, and 6, respectively. The track angle at a cross section of the track is the angle which a line over the top of the rails makes with the horizontal. It was computed from the measured vertical track profiles. The strong influence of the track angle on computed car body rocking is apparent from a comparison of Figs. 4 and 6.

The computed and measured car body rocking double amplitudes versus car speed are shown in Fig. 7. The computer simulation is in good agreement with test results. The model appears to be slightly overdamped at car speeds below the resonant speed. Possibly, this could be corrected by varying the amount of viscous damping in the car body, c_c, with car speed, providing the maximum c_c at resonance. Such a variation in c_c is reasonable because some of the phenomena that contribute to c_c can be expected to be more pronounced at resonance. For example, plastic deformation of parts from impact loads which contribute to c_c is expected to be larger at resonance than above and below resonance.

Wheel lift was computed at car speeds of 15.0 and 16.5 mph. Rail profile measurements indicate wheel lift at 16.5 and 18.5 mph.

For additional verification of the model, rocking of the test car equipped with a stabilizing device under development by American Steel Foundries was calculated. This device acts only against car body rocking. It is represented in the model by F_c. The remaining damping parameters were those used in the first series of simulations.

For this series of simulations the available rail profiles were incomplete. They were artificially completed using track data from the first series of simulations. Furthermore, it had to be assumed that the rail profiles do not change with car speed. However, track data used in the first series of simulations show a considerable variation of the rail profile with car speed.

Responses were calculated for selected stabilizer settings and car speeds. The calculated car body rocking for the unstabilized

car at 17.03 mph and one stabilizer setting at 21.42 mph is shown in Figs. 8 and 10. The corresponding measured car body rocking is shown in Figs. 9 and 11.

The maximum calculated and measured car body rocking double amplitude versus car speed and stabilizing device setting is shown in Fig. 12. This figure also shows the effectiveness of the stabilizing device in the control of car body rocking. We see that the model predicts the trend of the response with increasing stabilizer control action. The model appears to be overdamped. This may be due to the fact that in the experimental device the load is not developed instantaneously, whereas in the model it is. Therefore, more energy per cycle is dissipated in the model than in the actual experimental device. The wide disagreement between calculated and measured response for the unstabilized car at 18.09 mph is attributed to incomplete track data.

Conclusions

A mathematical model for the calculation of railroad freight car rocking has been derived. The resulting equations have been programmed for a digital computer.

The model has been validated for two configurations: (*a*) standard friction damping derived from friction shoes on the trucks, and (*b*) standard friction dampers on trucks and control of car body rocking derived from the American Steel Foundries stabilizing device. In both configurations, structural damping in trucks and the car body was considered.

Calculated and measured car body rocking is in good agreement for the first configuration. For the second configuration, although the agreement is, in general, quite good, some sizeable discrepancies exist. The probable cause of these discrepancies is the lack of complete track inputs for the calculation of the second configuration.

The model is considered reliable for engineering predictions and comparative evaluation of designs.

APPENDIX

Parameters of the Model

Geometry

$b_1 = 70$
$c_1 = 6.7$
$b_2 = 65.7$
$c_2 = 11.0$
$d = 7$

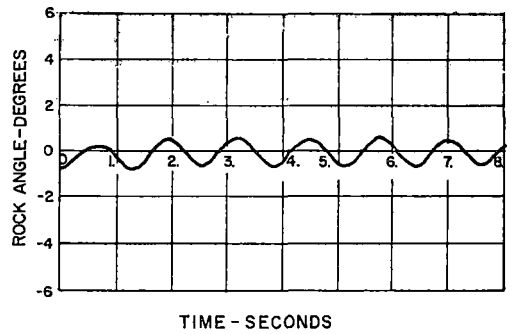


Fig. 10 Computed car body rocking at 21.42 mph, stabilizer operating

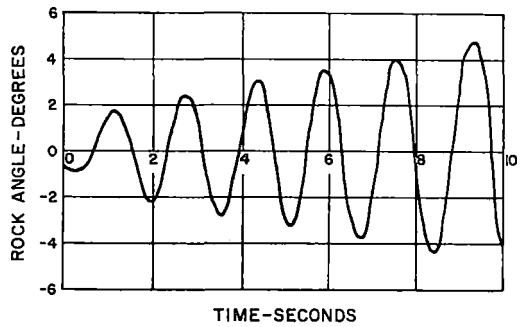


Fig. 9 Measured car body rocking at 17.03 mph, stabilizer not operating

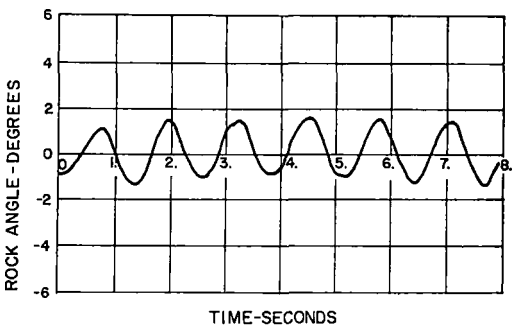


Fig. 11 Measured car body rocking at 21.42 mph, stabilizer operating

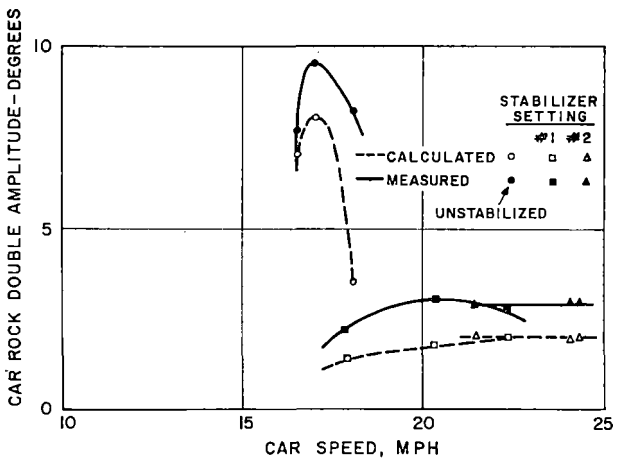


Fig. 12 Comparison of computer simulated and measured car body rocking amplitudes of car with selected stabilizer settings

e = 25
f = 39.5
h = 2.7
H_1 = 28.3
ϵ = 0.25
W_B = 28.25

All values are in inches.

Inertia Properties

M = 615 lb-sec²/in.
m = 5.12 lb-sec²/in.
I = 1,860,000 lb in. sec²
i = 3400 lb in. sec²
W = 20,000 lb

Spring Constants. The truck suspension consists of two spring groups, each with seven inner and seven outer $D5$, $3^{11}/_{16}$-in. total travel springs. The stiffness of each of these springs is[2]

outer 133.7 lb to deflect 1/16 in.

inner 66.9 lb to deflect 1/16 in.

Therefore

$$k = (133.7 + 66.9)(16)(7)(2) = 45{,}200 \text{ lb/in.}$$

The lateral stiffness of the suspension springs is calculated by the method of footnote 3.[3]

$$k_H = 40{,}000 \text{ lb/in.}$$

Acknowledgment

This study was sponsored by American Steel Foundries. The author thanks Messrs. C. E. Tack, L. D. Davis, R. A. Evans, and G. Tennikait of American Steel Foundries; Mr. H. L. Decker, consultant to American Steel Foundries; Dr. F. C. Nelson and Messrs. T. R. Yoos and P. Manglis, of Dynatech Corporation, for their assistance and discussions.

[2] Association of American Railroads Manual of Standards and Recommended Practices, D-73-1965.

[3] Sparing, W. H., "How Much Force to Deflect a Spring Sideways?" *Product Engineering*, Jan. 5, 1959.



ROBERT W. LUEBKE, Research Engineer Mechanical System, Chesapeake & Ohio Railroad

Mr. Luebke graduated from the University of Colorado in June, 1959 with honors in Aeronautical Engineering. After graduation, Mr. Luebke entered the U.S. Navy, during which time he was involved in the evaluation of guided missile weapons systems. He was also responsible for the test and evaluation of guided missiles and their components. Upon completion of his tour of active duty, Mr. Luebke was employed by the Pratt & Whitney Aircraft Division of United Aircraft Corporation to design, test and evaluate new power and propulsion concepts including fuel cells and nuclear powered rocket engines.

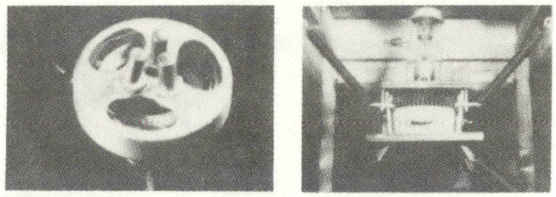
Mr. Luebke came to the C & O/B & O railroads in June of 1966 as Research Engineer–Mechanical Systems. Since coming to the combined railroads, Mr. Luebke has been actively engaged in the test and evaluation of the "Rock and Roll" problem and the Rock Car Windage problem. He has also been active in the evaluation of the performance of freight car cushioning units and the evaluation of the performance of freight car trucks.

Mr. Luebke is a member of AREA, the AAR Ad Hoc Committee on Automatic Coupler and the Ad Hoc Committee on Truck Performance. His work has been published by the ASME and the USDOT. He presently holds five U.S. and foreign patents.

"Performance Testing Of Freight Car Trucks"

The need for continued development and improvement of freight car equipment results from the continuing change in the commodities we carry and the ways in which they are packaged for in-transit shipment. These changes from wooden crates down to loose fill packaging in cardboard containers and increased car size and tonnage capacities continually change the environment to which products are subjected. The net results of these changes are increased L&D bills for the railroads and shippers alike.

The primary emphasis to date has been to develop better and more effective ways to cushion the load from impact damage in yards and terminals. This is entirely proper since I think we all concur that this type of damage is the largest single factor in L&D. Because of this a large amount of effort in recent years has been expended in the development and application of performance testing to cushioning units and load dividing equipment. Much has been learned and will continue to be learned. However, one major question still remains to be answered, "How much cushioning is required for a particular commodity service?" Is it 5", 10", 20", or 30"? This question is under active investigation now but it will still be some time before a complete answer is available. The reason I bring the question up now is that the same question directly applies to the performance evaluation of freight car trucks. How "good" must the ride be for a particular commodity service? The answer obviously rests in the data which will be accumulated by the performance testing of freight car trucks.



Vibration damage.

While impact damage is still the largest single factor in the generation of damage to commodities in-transit, hidden damage due to car vibrations caused by anything less than perfect track runs a close second. This type of damage generally results when the track input frequency is such that the truck suspension system amplifies rather than cushions the vibration input. This condition, called resonance, also occurs when the packaging material in the carton is subjected to its appropriate frequency and proceeds to damage the product inside the container. These photographs show a perfect example. Here an expensive radio transmitter tube perfectly packaged to avoid impact or dropped carton damage is susceptable to vibration resonance damge at 3 cps. Since 3 cps is a very common frequency in a freight car, the tube resonated on its cushion and smashed itself against the inside of the box. Since nearly all frequencies are available in a freight car to excite resonance in products, the potential for hidden damage is quite large. The basic problem then is to define which frequencies are most predominant and by how much. Since different truck suspension designs would ideally change the predominant frequencies, and consequently the commodity mix safely

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handled, performance testing of freight car trucks is essential. Based on the requirements of obtaining the information needed to substantially reduce the hidden damage due to freight car vibrations, the following list of objectives for performance testing of freight car trucks is proposed:

1. To define the vibration environment for packaging designers so that they can reduce their products susceptibility to vibration damage.
2. To provide data for the design of higher performance trucks.
3. To establish an objective and measurable performance specification for freight car trucks.
4. To evaluate the performances of various suspension system designs.
5. To determine the suitability of various truck designs for particular commodity service.

When the data required to fulfill these needs has been developed, we can rightly expect to be in a position to reduce hidden damage due to vibration resonance as much if not more than we've been successful in doing with the objective use of cushioning and load dividing equipment.

Although on the surface it would appear that we are just beginning to consider performance testing of freight car trucks, the science of truck performance testing has been developing for quite a few years.

The most comprehensive testing to date was accomplished during the period 1948 to 1950 by the AAR and a joint C&O–Army Signal Corps project. These initial tests evaluated the then newly developed friction damped trucks against the standard undamped truck equipped with 1-5/8 inch travel AAR 1915 springs. The tests were run over "normal track conditions" and the data consisted primarily of vertical acceleration counts, (i.e.) the number of times the vertical accelerations esceeded 0.25, 0.35, 0.5, and 0.75 G's. As a result of these tests, friction damped trucks equipped with 2-1/2 inch travel springs were made standard with a significant improvement in freight car ride quality.

In 1956, the C&O undertook a series of tests to evaluate the optimum friction damping level and to evaluate the total vibration environment within the freight car. As a result of these tests, friction damping force levels were developed to minimize damaging vibrations and packaging engineers were made aware of the environment. These tests were again run over "average" or "good" roadway and most of the data was again obtained in the form of "G" counts because of the difficulties in performing harmonic analysis with the equipment available at that time.

These tests constituted the basic body of knowledge of freight car truck performance. This data, though limited by the lack of modern vibration instrumentation, has been valuable and has provided a way to evaluate basic suspension system designs. The data is, however, lacking in many ways. For example, the tests were run over "average" or "good" roadway, consequently little was known about the effect of track or wheel irregularities on the vibration environment. Since the greatest majority of the data is in the form of "G" counts, little was known about the frequencies involved.

Consequently, because of the limitations imposed by the then available instrumentation, the data from these tests can not be used to meet the basic requirements outlined for freight car truck performance testing. This is particularly true with respect to fully defining the environment, providing design data, and evaluating the suitability of various truck designs for cars in different commodity services.

Fortunately, the instrumentation, recording, and data reduction equipment field has seen tremendous improvement in the past few years and consequently data can be obtained to fulfill all of the requirements of freight car truck performance testing.

The availability of this modern equipment and the desire of both C&O/B&O and the FRA to define the vibration environment and the effects of its major variables led to our joint performance testing program which was completed last year.

This program allowed us to take advantage of new instrumentation and automated analysis techniques.

These tests were rather extensive and included the evaluation of the effects of load, speed, track irregularities, flat wheels, snubbing force level, primary spring rate, variable rate springs, flat wheels and truck design on the vibration environment in a 50-foot, 70-ton box car. The results and details of these tests are outlined in the report "Investigation of 50-Foot, 70-Ton Box Car Vibrations", which is available from the government clearing house.

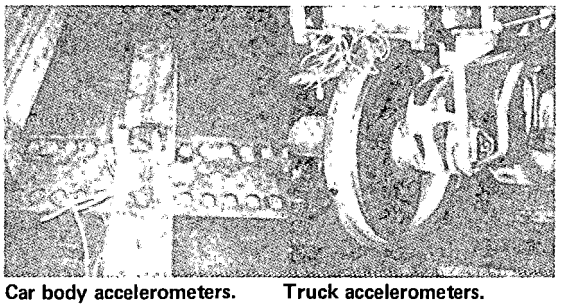
The significance of these tests with respect to the performance testing of freight car trucks is that they gave us the opportunity to use, develop and evaluate new instrumentation and analytical techniques.

It is the instrumentation and analytical techniques used in these tests, (since they allow us to now meet performance testing objectives), that I wish to discuss in more detail.

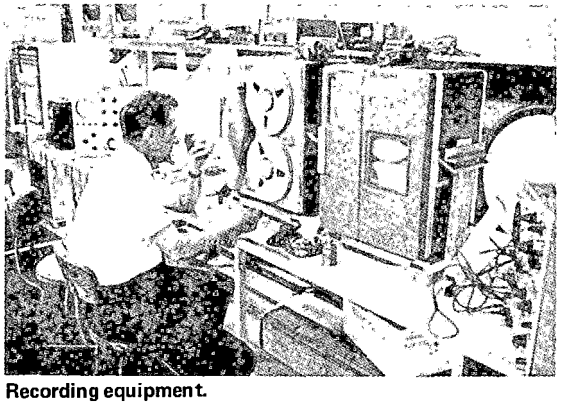
The instrumentation used comprised a vertical & lateral accelerometer mounted on the truck side frame over the roller bearing adapter and on the car floor over the center plate of the test box car. The output from these accelerometers was fed to a magnetic tape recorder and then to a brush ink pen

recorder. This series connection insured that any loss of signal, of which there were a few, would be immediately detected. The magnetic tape records were then processed to develop frequency spectrograms. The spectrogram is a plot of acceleration amplitude versus frequency and basically maps out the vibrational characteristics of the car and truck system. These spectrograms formed the basis for the detailed vibration analysis.

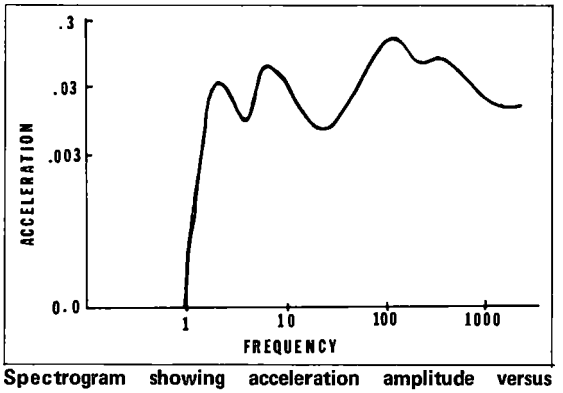
Instrumentation.



Car body accelerometers. Truck accelerometers.



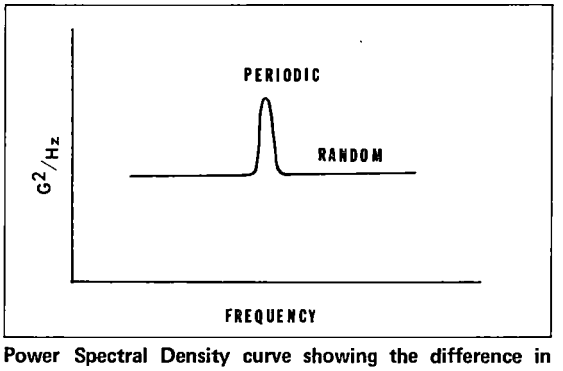
Recording equipment.



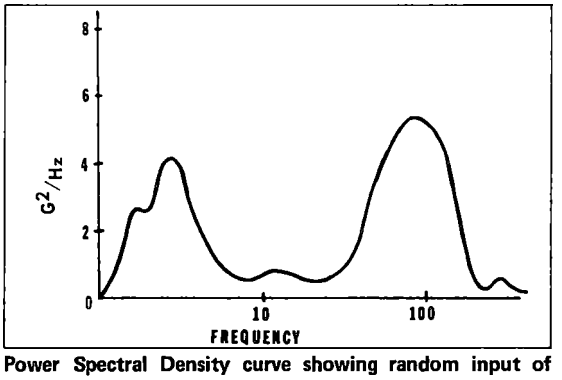
Spectrogram showing acceleration amplitude versus frequency.

While the spectrogram shows the packaging engineer which frequencies are present on the floor of the freight car, it does not tell him which frequencies are important. Since continuous vibration can cause resonance while random input is easily cushioned, we must be able to separate one from the other. The power spectral density curve which can be easily developed from the spectrogram does this nicely.

Power spectral density which is expressed in G^2/H_Z is basically a technique for describing the statistical properties of a random vibration. It describes how the power contained in the various frequency components of the vibration signal is distributed over the frequency spectrum. Since the PSD curve for random vibration is a straight horizontal line and for periodic vibrations is a vertical peak at the pure frequency, the PSD technique provides an excellent method for differentiating between the two types of vibrations present in freight cars. Another slide shows a typical power spectral density curve for freight car vibrations.



Power Spectral Density curve showing the difference in appearance of random and periodic vibrations.

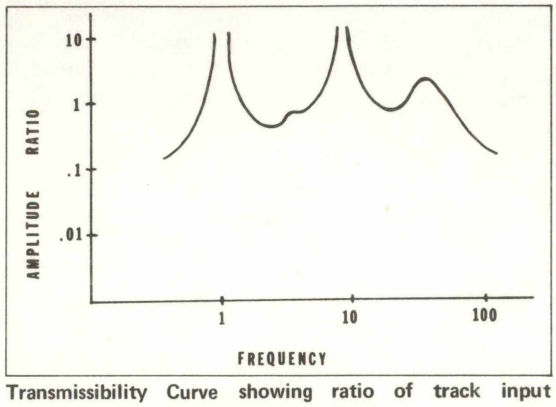


Power Spectral Density curve showing random input of vibrations separated from continuous vibrations, as present on floor of the freight car.

The use of the spectrogram and its associated PSD curve then fully meets the needs of defining the environment to which a shipping package will be subjected and will allow packaging engineers to design a proper cushion within the carton to protect their products.

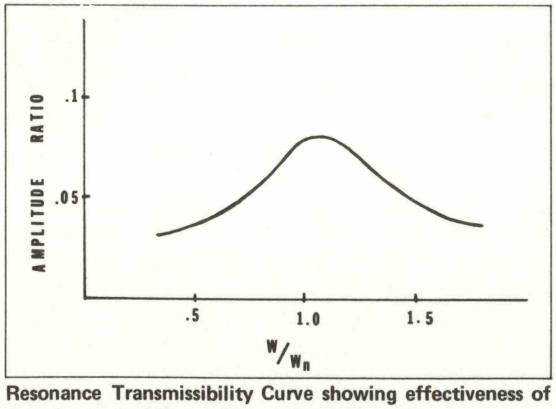
The spectrogram and the PSD curve are fine for evaluating the environment but the truck designer needs to have additional data if he is to develop a better truck. He needs to know how well and under what conditions his design will minimize the input from the track and hence reduce the car body vibration levels. This information is readily obtained by the development of a transmissibility

curve which basically shows the ratio between the amplitude of the track induced input to the truck and the resultant amplitude in the car body as a function of frequency. In effect, this curve shows how effectively a suspension system design isolates the car body from track inputs. This form of analysis also points out which input frequencies are transmitted directly or are amplified to the car body and consequently require design modification.



Transmissibility Curve showing ratio of track input amplitude to car body response amplitude.

The final piece of information required by the designer is the resonance transmissibility curve which shows the effectiveness of the damping or snubbing units of this suspension system. This curve is normally plotted as the ratio of car body to truck vibration amplitude as a function of the ratio of input frequency to truck suspension system natural frequency. With the generation of the spectrograms, PSD curves, and transmissibility curves, the needs of the designer are fully met. This same data once generated for all present designs could also serve as the basis for an objective and measurable performance specification for freight car trucks against which new designs could be properly evaluated.

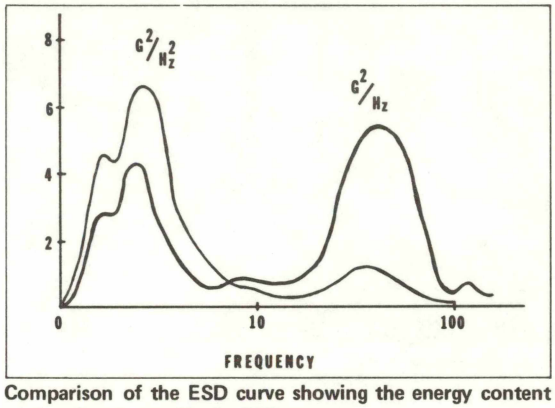


Resonance Transmissibility Curve showing effectiveness of damping or snubbing units on suspension system.

The final objective of performance testing is that of obtaining an overall measure of truck performance and of determining the suitability of a truck for particular commodity services. These objectives require a meaningful number based on physical measurements which will be unique for each truck system. The ride index that we have used for many years has served us well but is based on a rather arbitrary assignment of damage associated with a particular "G" level.

The equivalent of the present ride index would be the area under the PSD curve which is known as the mean square acceleration on the car floor available to do damage. We know, however, that acceleration by itself does not create damage. Power or the rate at which work is done on an object, however, is a good measure of the ability of a vibrating source to cause damage.

A measure of the power available in a vibrating source is easily obtained by developing what is called the energy spectral density (ESD) curve for the vibration signal. The ESD is basically the next analytical step above the PSD curve. It can be generated automatically and is expressed in $G^2/H_Z{}^2$. It shows the energy content of the vibration as a function of frequency.



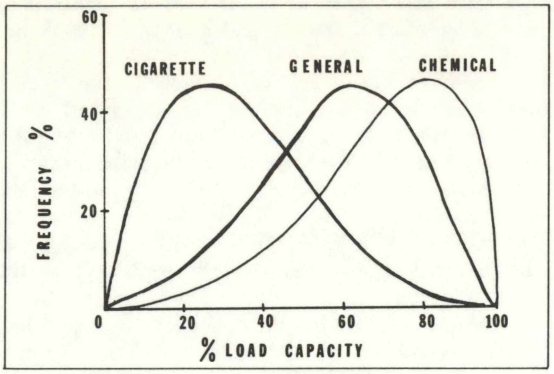
Comparison of the ESD curve showing the energy content of the vibrations with the PSD curve.

The area under the ESD curve is a direct measure of the mean power available on the car floor to do damage to the commodities carried there on.

The mean power transmitted to the car body is therefore an overall measure of the ability of a truck to isolate a car body from track inputs. It can also be measured with good repeatability and used to evaluate truck designs for various commodity services.

To give you an example of the application of the mean power or ESD technique, 3 trucks have been evaluated for service in 3 different commodity groups. The graph shows the load distribution of each of the 3 groups. The cigarette pools are lightly loaded while the chemical pools are heavily loaded. General merchandise lies in between. The next figure shows the mean power transmitted by each of the 3 trucks at light, 1/2 and full load. Using the load distribution of each group to weight the mean powers according to load

frequency, the trucks can be grouped according to which yields the minimum overall mean power for each commodity group. As you can see different trucks would be better for different groups.



Load distribution.

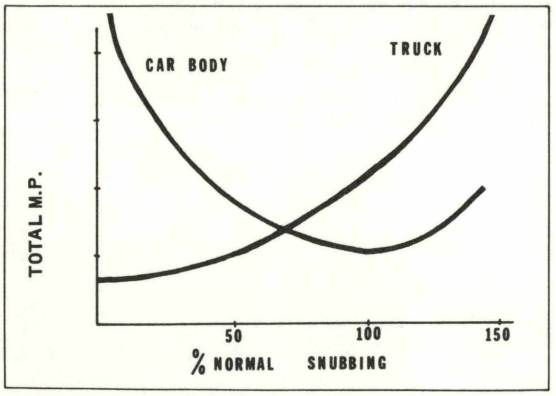
LOAD	TRUCK-A	TRUCK-B	TRUCK-C
EMPTY:	13650	20750	14400
HALF:	5050	5050	5550
FULL:	3280	1480	3300

Car body mean power.

POOL	TRUCK-A	TRUCK-B	TRUCK-C
CIGARETTE:	6088	7868	6510
GENERAL:	4480	3820	4670
CHEMICAL:	4050	2960	4165

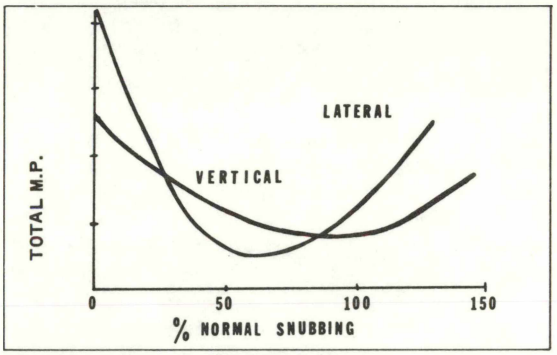
Service evaluation.

In addition to yielding a sound basis upon which to evaluate truck designs for different commodity groups, the ESD or mean power technique also provides a sound method for evaluating the effectiveness of truck snubbing levels. The next figure shows the vertical & lateral car body and truck energy levels as a function of snubber force level. Note how the minimum level occurs at present normal damping as we would expect after years of snubbing improvement.



Snubbing evaluation.

However, breaking this analysis down to vertical and lateral damping separately, we find that for vertical vibrations, normal damping is optimum. But for lateral vibrations, about 50% of normal is optimum. This technique then, can therefore be used by designers to quickly evaluate the overall performance of their truck design and the optimum damping levels required to generate best performance.



Optimum snubbing.

In summary, then, the basic requirement for performance testing of freight car trucks is the need to reduce L&D payout by a reduction of vibration caused damage. To accomplish this reduction we need to have accurate data which will define the environment in terms usable by packaging engineers. We also need data to aid in the design and evaluation of new suspension concepts, to establish reasonable and objective performance specifications, and to allow railroads to evaluate the suitability of various designs for the service to which they are intended.

The joint C&O/B&O tests while providing us the opportunity to use and evaluate the newly available techniques does not fill all of the performance testing requirements. These tests did, however, prove out the techniques described earlier and gives us all an insight into what can be accomplished and learned by their use.

With refinement and expanded use of these techniques by all concerned, the objectives of freight car truck performance testing can be met and the railroad industries loss and damage bill reduced.

Thank you.

DELEGATE COMMENT: Could you please describe how we would go about obtaining the spectrogram and calculating the Power Spectral and Energy Spectral density curves?

MR. LUEBKE: Probably the easiest way to do it is to draw a picture. You have a vibration signal from an accelerometer. You then put this through a filter bank. Let's say this filter passes frequencies from zero to one cycle, and the next from one to two, from two to three, and from three to four, etc. What you end up with is something like a histogram or a bar graph, which says that between zero and one you have a certain G level, between one and two you have another one, and between two and three you have another one, and so forth. Connecting these, you then end up with a spectrogram. To get a power spectral density curve, the only thing you have to concern yourself with is the frequency difference or band width of each filter. You square the acceleration level in each filter and divide it by the frequency difference of the filter. Completing this for all filters gives you the power spectral density curve. To get the energy spectral density curve, all you do is divide the power spectral density value of each filter again by the band width of the filter you are using. This gives you the energy spectral density curve, which is again a bar graph in effect. Taking the area under this curve gives you the mean power. The area under the power spectral density curve gives you the mean square amplitude or acceleration. All this can be done automatically on present day electronic equipment, so it would be a matter of recording the data on tape, putting it through a filter bank and letting the equipment do the work and come out with curves that you can work with directly. It saves a lot of time and it gives you a good deal more information.

DELEGATE COMMENT: Did you collect your data in such a way that you could prepare an occurance histogram?

MR. LUEBKE: The data from these tests were developed by running the test over particular test sections. We were not trying to collect environmental data per se. We were trying to find out what vibrations would be generated by a smooth track, by a normal type track and by a very bad track. We artificially shimmed opposite the joints and jacked the track over to produce the track inputs.

DELEGATE COMMENT: Would you repeat the number and name of that report from the clearing house?

MR. LUEBKE: The report is entitled, "Investigation of 50-Foot, 70-Ton Boxcar Vibrations," PB192801 available through the clearance house for technical and scientific reports, Washington, D.C. The best thing to do is to write to Mr. Ken Lawson at Department of Transportation, and he can get it for you.

DELEGATE COMMENT: With this mechanism, Bob, we will come out with a ride factor for the design of trucks. How about a wear factor?

MR. LUEBKE: Unfortunately, this won't give it to you.

CHAIRMAN SMITH: Thank you, Bob. Our next speaker is Mr. R. W. Carman, Manager Equipment Research Engineering, Southern Railway System.

Paper No.
68—WA/RR-2

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Problems on Hunting of Railway Vehicle on Test Stand

The stand test in which a vehicle is set on a roller rig to simulate its running state is available as one of the effective means for experimental study on the hunting of vehicle. Here several questions arise: Whether or not a scale model test can substitute for an actual vehicle test; whether or not a bogie with a dummy load can represent a whole car in a stand test; and whether or not a stand test produces a similar result to a field test. The present paper summarizes the results of the studies about these questions conducted at the Railway Technical Research Institute of the Japanese National Railways.

1 Introduction

PREVENTION of hunting in railway vehicle is one of the most important problems to be solved for their high-speed operation. It is for this reason that this problem has been and is still being studied both theoretically and experimentally throughout the world.

Experimental studies are necessitated for the purposes of checking the results of theoretical research or investigating the influences of factors defying theoretical treatment, such as the nonlinearity involved in the characteristics of the running gear or the body support of a vehicle. They can be performed at the test stand or in the field, using scale models, full-size models, or actual vehicles.

It would be the most practical method of experimentation to operate an actual vehicle on the track, but great hazards would be involved with a high-speed operation of an actual vehicle if it develops a heavy hunting. Besides this would involve great costs, time, and labor, and accordingly it would not be advisable to carry out a test on hunting under different combinations of numerous factors. Thus, it becomes imperative to resort to a stand test.

The Railway Technical Research Institute of the Japanese National Railways began about 1950 to test systematically 1/10–1/5 models of wheel-sets and vehicles on the test stands (see Fig. 1), and clarified the basic property of hunting, and following this series of tests, it proceeded around 1960 to a series of tests on the newly built test stand (see Fig. 2) using full-size test bogies. The results of these experimental studies contributed very much, together with the theoretical research, to the successful designing of the bogie for the new Tokaido line vehicles.

The following questions will naturally be raised about these studies using the test stand:

(1) Can the scale model test substitute for the full-size test?

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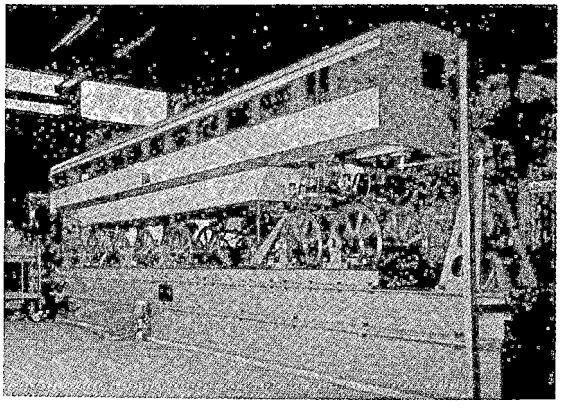


Fig. 1 Test stand for the 1/5 scale model experiment

(2) Can the single bogie test substitute for the whole vehicle test?

(3) Can the stand test and the field test produce the same results?

Here are to be summarized what we have learned through our experience about these questions, in the belief that it may prove useful for any future experimental study on the hunting of vehicle.

2 Some Problems in Model Test

For a stand test, it would be far cheaper and more convenient to employ a scale model than to employ a full-size model or an actual vehicle.

A stand test using a scale model is found very effective for elaborate observation of hunting behaviour, investigation into the basic nature of hunting and search for its prevention method, but when the critical hunting speed or the lateral stability of vehicle

Nomenclature

a = half length of wheel base
b = half distance between contact points of wheel treads and rails in lateral direction
b_0 = half spacing of bolster anchors in lateral direction
b_1 = half spacing of suspension mountings for a wheelset
b_2 = half spacing of bolster springs in lateral direction
c_1, c_2, c_2' = damping coefficients corresponding to k_1, k_2, k_2'
f = creep coefficient
h_B = height of body center of gravity above axle center
h_T = height of bogie frame center of gravity above axle center
h_1 = height of bolster spring center above bogie frame center of gravity
h_2 = height of body center of gravity above bolster spring center
I_{Bx} = moment of inertia of half body in roll
I_{Bz} = moment of inertia of half body in yaw around pivot
I_{Tx} = moment of inertia of bogie frame in roll
I_{Tz} = moment of inertia of bogie frame in yaw
I_W = moment of inertia of wheelset

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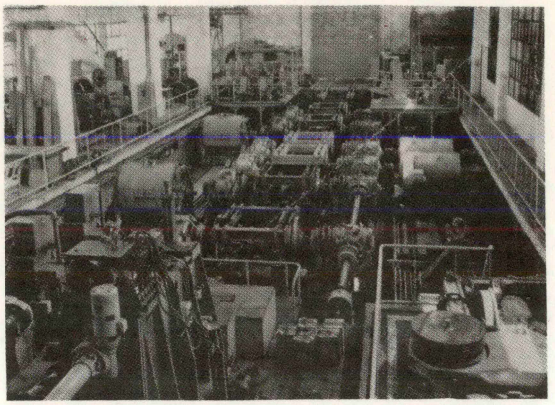


Fig. 2 Test stand for the experiment with an actual vehicle having six sets of rail-wheels, the central two of which were newly added for 350 km/h test, the rest being the ones provided from the first for 250 km/h test

for an actual vehicle is to be estimated from the results of the model test, there is some doubt about the accuracy of such estimate. In this connection let us first consider the similarity of models.

The model should be made similar in construction to an actual vehicle dynamically as a whole and even geometrically in some parts. Especially, the tread profile of both wheel and rail must be carefully made similar.

When the model is made geometrically similar to an actual vehicle, the weight ratio between them will be

$$W_m/W_a = \gamma_m l_m^3/(\gamma_a l_a^3) = \gamma_m/\gamma_a \cdot 1/R^3 \tag{1}$$

where R is the dimensional ratio of the actual to the model vehicle; γ the specific gravity of material; l the typical length; the suffixes m and a, respectively, denote the model and the actual one.

Assuming the model is made of the same material as the actual vehicle, we have

$$\gamma_m = \gamma_a$$

accordingly

$$W_m/W_a = 1/R^3 \tag{2}$$

Meanwhile, for the sake of insuring dynamic similarity at the wheel-rail contact, there should be the same values of E/σ, i.e., ratio of the Young's modulus to stress, and of μ, i.e., frictional coefficient, of wheel and rail between model and actual vehicle. Therefore, if the same material is employed, E will become equal and in consequence σ, too, ought to be equal. To satisfy this condition, the following should hold:

$$W_m/l_m^2 = W_a/l_a^2$$

accordingly

$$W_m/W_a = 1/R^2 \tag{3}$$

which turns out different from the condition of (2).

To avoid this discrepancy, such materials must be chosen as fulfill the following condition between model and actual car:

$$\gamma_m/\gamma_a \cdot E_a/E_m = R \tag{4}$$

In actuality, such convenient material is not available; the only alternative will be to adopt the same material as actual vehicle, attach an additional weight to multiply the apparent specific gravity of model by R, thereby ensuring the following condition:

$$\gamma_m/\gamma_a = R \tag{5}$$

But as a matter of fact, even this would be extremely hard to realize, and it would be next to impossible in a less than 1/5 scale model to produce perfect similarity to the smallest detail of creeping between wheel and rail.

This is the most controversial point in this kind of model experimentation. In the model experiments which have been made in our Institute so far, the model weight was designed for convenience's sake to meet the condition of (2), (attaching an additional weight if necessary), and the obtained data just as they are would not be numerically applicable to the actual vehicle. Qualitatively speaking, however, they would be applicable without any serious error.

Between the spring constants k_m in the model and k_a in the actual vehicle, the following relation has to be maintained:

$$k_m/k_a = W_m/W_a \cdot l_a/l_m = W_m/W_a \cdot R \tag{6}$$

Therefore, the following will hold about the natural frequency n, in view of the proportionality of n to $\sqrt{k/W}$:

$$n_m/n_a = \sqrt{R} \tag{7}$$

Meanwhile, for the necessary condition that the nondimensional speed v/nl be the same between model and actual vehicle, the following should hold between the model speed v_m and the actual vehicle speed v_a:

$$v_m/v_a = 1/\sqrt{R} \tag{8}$$

Using the above formula, the critical speed for the hunting of an actual vehicle can be predicted from that of a model one.

In addition to the above, there are other difficulties in this kind of model experiment: The difficulty of simulating in a model the damping of the suspension system, and also the elasticity, including the slightest play around the axlebox support and side bearers of an actual bogie; the excessiveness of the lateral rigidity of supporting wheels of rail-wheel as compared with that of actual rail; influences of supporting wheel diameter; and others.

3 Comparison of a Bogie Test With a Vehicle Test

It is, of course, ideal to use a whole vehicle for the test stand, but it would be more economical if a single bogie could substitute for a whole vehicle for this purpose. Thus, it is necessary to explore the possibility of such substitution. To this end, a comparison was made by numerical calculation both of a vehicle and

Nomenclature

i_{Bx} = $\sqrt{I_{BX}/m_B}$ = radius of gyration of half body in roll

i_{Bz} = $\sqrt{I_{BZ}/m_B}$ = radius of gyration of half body in yaw around pivot

i_{Tx} = $\sqrt{I_{TX}/m_T}$ = radius of gyration of bogie frame in roll

i_{Tz} = $\sqrt{I_{TZ}/m_T}$ = radius of gyration of bogie frame in yaw

i_W = $\sqrt{I_W/m_W}$ = radius of gyration of wheelset in yaw

k_0 = bolster anchor stiffness

k_1 = axle spring stiffness for half bogie

k_2 = bolster spring stiffness for half bogie

k_2' = bolster spring lateral stiffness for half bogie

k_{Wx} = longitudinal supporting stiffness of axle

k_{Wy} = lateral supporting stiffness of axle

k_t = gravitational stiffness for wheelset due to difference between normal forces of wheels in lateral direction

l = half length of bogie center distance

m_B = half mass of body

m_T = mass of bogie frame

m_W = mass of wheelset

r = wheel tread circle radius

r_T, r_R = radii of wheel and rail-wheel, respectively

(*Continued on next page*)

of a bogie with a dummy load corresponding to a half body, and a pivot around which the load can freely turn as a vertical axis, and as a longitudinal axis at the center of gravity of the body.

The details of this calculation are shown in Appendix 1, and its result is summarized in the following table.

Table 1

Conditions	Critical speed of hunting m/s (Frequency cps)	
	Body hunting	Bogie hunting
Whole vehicle	...	91 (2.6) 107 (2.8)
Front half of vehicle	...	125 (3.2)
Rear half of vehicle	lower 22 (0.9) upper 47 (1.3)	58 (1.9)
Fixed body	...	110 (3.2)

It can be seen from the table that though the body hunting does not occur in the whole vehicle test, it clearly appears in the rear half vehicle test. This indicates that the whole vehicle test cannot be replaced by the rear half vehicle test so far as the body hunting is concerned. In the front half vehicle test, the body hunting does not occur as in the whole vehicle test, but as can be seen in the characteristic curves in Figs. A2 and A4 there is naturally some difference in characteristics of both cases. Thus, it is concluded that, regarding the body hunting, substitution of a half vehicle for a whole vehicle in the stand test is inappropriate.

The bogie hunting occurs in each case, but the critical speed is different depending on the case. The critical speed of the bogie hunting of a whole vehicle lies between that of the front half and that of the rear half and is somewhat lower than that of the fixed body. This suggests that, as to the bogie hunting, a half vehicle test is able to substitute for a whole vehicle test, provided that an adequate spring or damping device is added between the dummy load and the outside fixed base.

4 Effect of Rail-Wheel Diameter Upon Wavelength of Hunting

One of the most important differences between the stand test and the field test is that the rail-wheel of the test stand has a certain curvature within the vertical plane. Therefore, the effect of the rail-wheel diameter upon the hunting behavior should be taken into consideration when the stand test is carried out.

The wavelength of kinematic hunting of a wheel-set run on the rail-wheel of the test stand is given by the following formulas. Their derivation is shown in Appendix 2.

$$S = 2\pi \sqrt{br_T/\gamma_e} \tag{9}$$

and

$$\gamma_e = \gamma \frac{1 + (\rho_R/\rho_T)(r_T/r_R)}{1 - (\rho_R/\rho_T)} \tag{10}$$

where S represents wave length, and other nomenclatures are listed in Appendix 1.

As can be known from (10), the effective slope of a tire increases by $\{1 + (\rho_R/\rho_T)(r_T/r_R)\}$ times that on the straight rail, and accordingly, the wave length of hunting on the test stand decreases by $\{1 + (\rho_R/\rho_T)(r_T/r_R)\}^{-1/2}$ times that on the straight rail.

Taking the following values in an actual numerical example,

$$r_T = 0.431m, \quad r_R = 0.526m$$
$$\rho_T = 0.120m, \quad \rho_R = 0.050m$$
$$b = 0.549m, \quad \gamma = 0.097$$

the effective tire slope becomes, from (10)

$$\gamma_e = 0.097 \times \frac{1 + (0.05/0.12)(0.431/0.526)}{1 - (0.05/0.12)}$$
$$= 0.223$$

The wavelength becomes, from (9)

$$S = 2\pi \sqrt{\frac{0.549 \times 0.431}{0.223}}$$
$$= 6.47m$$

This value is 16 percent smaller than the corresponding value 7.49 m on the straight rail.

For this reason, when the wheel tread is worn, the wavelength of hunting being affected by the curvature of rail-wheel will considerably diminish, and in consequence, the hunting behavior will change. If the wheel tread has a straight gradient, this effect may practically be ignored.

Another effect of the rail-wheel having a curvature is the difference of the wheel-to-rail contact area from that of the straight rail. This difference emerges as a change in the creep coefficient, but it will not usually be so large as to affect the hunting.

5 Comparison of Results Between Field Tests and Stand Tests Using Actual Vehicles

Our greatest concern is whether or not a stand test using an actual vehicle can produce the same results on hunting as a running test in the field. Therefore, we compared the results between the field test and the stand test immediately after we built the test stand.

These tests were performed using the same electric railcar. In both cases the bogie hunting occurred under the relation between frequency of vibration and speed as illustrated in Fig. 3. The results of these tests are compared as follows:

Table 2

	Field Test	Stand Test
Critical speed of hunting	90 km/h	100 km/h
Frequency of hunting motion	4.5–6.0 cps	4.0–4.5 cps

Nomenclature

v = forward speed of vehicle

y_T = lateral displacement of bogie frame (suffixes $T1$, $T2$ mean front bogie frame and rear one, respectively)

y_W = lateral displacement of wheel-set (suffixes $W1$, $W2$ mean front axis and rear axis, respectively)

γ = angle between horizontal plane and contact plane of wheel and rail at neutral position

γ_e = effective value of γ

ρ_T, ρ_R = radii of curvature of wheel and rail-wheel at the contact point

ψ_B = angular displacement of body in yaw

ψ_T = angular displacement of bogie frame in yaw (suffixes $T1$, $T2$ mean front bogie frame and rear one, respectively)

ψ_W = angular displacement of wheelset in yaw (suffixes $W1$, $W2$ mean front axis and rear axis, respectively)

ϕ_B = angular displacement of body in roll

ϕ_T = angular displacement of bogie frame in roll (suffixes $T1$, $T2$ mean front bogie frame and rear one, respectively)

Note: Bogie frame means the intermediate parts of bogie between body and wheelset.

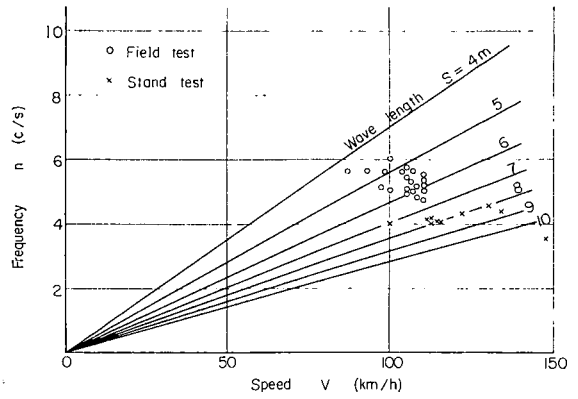


Fig. 3 Relation between speed and frequency of hunting

The critical speed did not much differ in both tests. The difference in frequency between field and stand tests may be chiefly attributed to the differences in the rail condition, such as rail profile and lateral elasticity.

After we were fully convinced of satisfactory agreement of results through this comparison, we took to a testing on the hunting of the bogie for the new Tokaido line vehicles using this stand.

Another example is a comparison of a field test and a stand test on the two prototype vehicles for the new Tokaido line, which developed bogie hunting in running tests on the test track section of the new Tokaido line. The results are compared as follows:

(*a*) For Type A bogie, the bogie hunting with the frequencies of 4–5 cps developed at the speed of over 180 km/h in field test, on the contrary, the body hunting with 1.5–1.7 cps developed at over 170 km/h and no bogie hunting occurred up to 250 km/h in stand test.

(*b*) For Type B bogie, the violent bogie hunting with about 4.5 cps developed at over 240 km/h in field test, but in stand test the body hunting of 1.7–1.9 cps developed at over 220 km/h and the bogie hunting did not occur up to 250 km/h.

These results show a wide discrepancy between the natures of hunting on an actual track and on a test stand; this discrepancy is, as explained in Section 3, mainly attributable to the use of only one bogie in the stand test.

6 Concluding Remarks

The stand test is recommendable as a means of experimental research on the hunting of vehicle, but it comprises several points to be checked. The results of our studies about these points can be summarized as follows:

(*a*) For reasons of similarity, it is difficult to make accurate estimates of the critical hunting speed and the lateral stability of vehicle in an actual vehicle from the results of a scale model test. Nevertheless, the model test is found very effective for elaborate observation of hunting behavior, investigation into the basic nature of hunting, and search for its prevention method.

(*b*) A stand test using an actual vehicle may be utilized as a substitute for a field test. Particularly, it has the merit of being far easier and safer at high speeds than the field test.

(*c*) A stand test using only a single bogie would not be suitable as a test on the body hunting. Nevertheless, as a test on the bogie hunting it would be useful, provided appropriate conditions of restraining the body are selected.

(*d*) Preliminary attention must be paid to the effect of the rail-wheel diameter which affects the wave length of the hunting when the stand test is carried out.

APPENDIX 1

Hunting Calculation for a Whole Vehicle and a Half Vehicle With Pivot at Body Center

Stand tests for a whole vehicle and for a bogie with a dummy load corresponding to a half body are schematically shown in Fig. A.1. Hunting calculation in both cases will be described in the following.

1 Equation of Motion for a Half Vehicle With Pivot at Body Center

$$m_W\ddot{y}_{W1} + k_{W_y}(y_{W1} - y_T - a\psi_T - h_T\phi_T) + k_t y_{W1} = -f\left(\frac{\dot{y}_{W1}}{v} - \psi_{W1}\right) \quad (A.1)$$

$$m_W\ddot{y}_{W2} + k_{W_y}(y_{W2} - y_T + a\psi_T - h_T\phi_T) + k_t y_{W2} = -f\left(\frac{\dot{y}_{W2}}{v} - \psi_{W2}\right) \quad (A.2)$$

$$I_W\ddot{\psi}_{W1} + b_1^2 k_{W_x}(\psi_{W1} - \psi_T) = -bf\left(\frac{b}{v}\dot{\psi}_{W1} + \frac{\gamma}{r}y_{W1}\right) \quad (A.3)$$

$$I_W\ddot{\psi}_{W2} + b_1^2 k_{W_x}(\psi_{W2} - \psi_T) = -bf\left(\frac{b}{v}\dot{\psi}_{W2} + \frac{\gamma}{r}y_{W2}\right) \quad (A.4)$$

$$m_T\ddot{y}_T - k_{W_y}(y_{W1} - y_T - a\psi_T - h_T\phi_T) - k_{W_y}(y_{W2} - y_T + a\psi_T - h_T\phi_T) + 2k_2'(y_T - h_1\phi_T \mp l\psi_B - h_2\phi_B) = 0 \quad (A.5)$$

$$I_{T_z}\ddot{\psi}_T - b_1^2 k_{W_x}(\psi_{W1} - \psi_T) - ak_{W_y}(y_{W1} - y_T - a\psi_T - h_T\phi_T) - b_1^2 k_{W_x}(\psi_{W2} - \psi_T) + ak_{W_y}(y_{W2} - y_T + a\psi_T - h_T\phi_T) + 2b_0^2 k_0(\psi_T - \psi_B) = 0 \quad (A.6)$$

$$I_{T_x}\ddot{\phi}_T - h_T k_{W_y}(y_{W1} - y_T - a\psi_T - h_T\phi_T) - h_T k_{W_y}(y_{W2} - y_T + a\psi_T - h_T\phi_T) - 2h_1 k_2'(y_T - h_1\phi_T \pm l\psi_B - h_2\phi_B) + 2b_2^2 k_2(\phi_T - \phi_B) + 2b_1^2 k_1\phi_T = 0 \quad (A.7)$$

$$I_{B_z}\ddot{\psi}_B \mp 2lk_2'(y_T - h_1\phi_T \mp l\psi_B - h_2\phi_B) - 2b_0^2 k_0(\psi_T - \psi_B) = 0 \quad (A.8)$$

$$I_{B_x}\ddot{\phi}_B - 2h_2 k_2'(y_T - h_1\phi_T \mp l\psi_B - h_2\phi_B) - 2b_2^2 k_2(\phi_T - \phi_B) = 0 \quad (A.9)$$

where the upper sign of the double sign stands for the front half of vehicle and the lower sign stands for the rear half of vehicle.

Assuming the stiffness between axles and bogie frame is very large, the following relations hold:

$$y_{W1} - y_T - a\psi_T - h_T\phi_T = 0 \quad (A.10)$$

$$y_{W2} - y_T + a\psi_T - h_T\phi_T = 0 \quad (A.11)$$

$$\psi_{W1} - \psi_T = 0 \quad (A.12)$$

$$\psi_{W2} - \psi_T = 0 \quad (A.13)$$

and the forces between axles and bogie frame are given from (A.1) to (A.4) as follows,

$$k_{W_y}(y_{W1} - y_T - a\psi_T - h_T\phi_T) = -m_W\ddot{y}_{W1} - k_t y_{W1} - f\left(\frac{\dot{y}_{W1}}{v} - \psi_{W1}\right) \quad (A.14)$$

$$k_{W_y}(y_{W2} - y_T + a\psi_T - h_T\phi_T) = -m_W\ddot{y}_{W2} - k_t y_{W2} - f\left(\frac{\dot{y}_{W2}}{v} - \psi_{W2}\right) \quad (A.15)$$

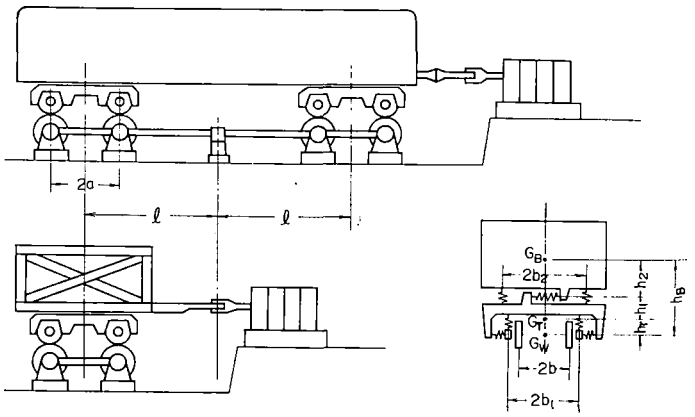


Fig. A1 Full scale test stand for a vehicle and a bogie

$$b_1^2k_{Wx}(\psi_{W1} - \psi_T) = -I_W\ddot{\psi}_{W1} - bf\left(\frac{b}{v}\dot{\psi}_{W1} + \frac{\gamma}{r}y_{W1}\right) \quad (A.16)$$

$$b_1^2k_{Wx}(\psi_{W2} - \psi_T) = -I_W\ddot{\psi}_{W2} - bf\left(\frac{b}{v}\dot{\psi}_{W2} + \frac{\gamma}{r}y_{W2}\right) \quad (A.17)$$

Further, for simplicity, putting h_T as zero, we can simplify the equations (A.5) to (A.9) to;

$$(m_T + 2m_W)\ddot{y}_T + 2f\left(\frac{\dot{y}_T}{v} - \psi_T\right) + 2k_ty_T + 2k_2'(y_T - h_1\phi_T \mp l\psi_B - h_2\phi_B) = 0 \quad (A.18)$$

$$(I_{Tz} + 2I_W + 2a^2m_W)\ddot{\psi}_T + 2f\left(\frac{a^2 + b^2}{v}\dot{\psi}_T + \frac{b\gamma}{r}y_T\right) + 2a^2k_t\psi_T + 2b_0^2k_0(\psi_T - \psi_B) = 0 \quad (A.19)$$

$$I_{Tx}\ddot{\phi}_T - 2h_1k_2'(y_T - h_1\phi_T \mp l\psi_B - h_2\phi_B) + 2b_2^2k_2(\phi_T - \phi_B) + 2b_1^2k_1\phi_T = 0 \quad (A.20)$$

$$I_{Bz}\ddot{\psi}_B - 2b_0^2k_0(\psi_T - \psi_B) \mp 2lk_2'(y_T - h_1\phi_T \mp l\psi_B - h_2\phi_B) = 0 \quad (A.21)$$

$$I_{Bx}\ddot{\phi}_B - 2b_2^2k_2(\phi_T - \phi_B) - 2h_2k_2'(y_T - h_1\phi_T \mp l\psi_B - h_2\phi_B) = 0 \quad (A.22).$$

2 Equations of Motion for a Vehicle

By adding up the equations for front and for rear half of vehicle, we can get the equations for a whole vehicle as follows;

$$\bar{m}_T\ddot{y}_{T1} + 2f\left(\frac{\dot{y}_{T1}}{v} - \psi_{T1}\right) + 2k_ty_{T1} + 2k_2'(y_{T1} - h_1\phi_{T1} - y_B - l\psi_B - h_2\phi_B) = 0 \quad (A.23)$$

$$\bar{I}_{Tz}\ddot{\psi}_{T1} + 2f\left(\frac{a^2 + b^2}{v}\dot{\psi}_{T1} + \frac{b\gamma}{r}y_{T1}\right) + 2a^2k_t\psi_{T1} + 2b_0^2k_0(\psi_{T1} - \psi_B) = 0 \quad (A.24)$$

$$I_{Tx}\ddot{\phi}_{T1} + 2b_2^2k_2(\phi_{T1} - \phi_B) + 2b_1^2k_1\phi_{T1} - 2h_1k_2'(y_{T1} - h_1\phi_{T1} - y_B - l\psi_B - h_2\phi_B) = 0 \quad (A.25)$$

$$2m_B\ddot{y}_B - 2k_2'(y_{T1} - h_1\phi_{T1} - y_B - l\psi_B - h_2\phi_B) - 2k_2'(y_{T2} - h_1\phi_{T2} - y_B + l\psi_B - h_2\phi_B) = 0 \quad (A.26)$$

$$2I_{Bz}\ddot{\psi}_B - 2b_0^2k_0(\psi_{T1} - \psi_B) - 2b_0^2k_0(\psi_{T2} - \psi_B) - 2lk_2'(y_{T1} - h_1\phi_{T1} - y_B - l\psi_B - h_2\phi_B) + 2lk_2'(y_{T2} - h_1\phi_{T2} - y_B + l\psi_B - h_2\phi_B) = 0 \quad (A.27)$$

$$2I_{Bx}\ddot{\phi}_B - 2b_2^2k_2(\phi_{T1} - \phi_B) - 2b_2^2k_2(\phi_{T2} - \phi_B) - 2h_2k_2'(y_{T1} - h_1\phi_{T1} - y_B - l\psi_B - h_2\phi_B) - 2h_2k_2'(y_{T2} - h_1\phi_{T2} - y_B + l\psi_B - h_2\phi_B) = 0 \quad (A.28)$$

$$I_{Tx}\phi_{T2} + 2b_2^2k_2(\phi_{T2} - \phi_B) + 2b_1^2k_1\phi_{T2} - 2h_1k_2'(y_{T2} - h_1\phi_{T2} - y_B + l\psi_B - h_2\phi_B) = 0 \quad (A.29)$$

$$\bar{I}_{Tz}\ddot{\psi}_{Tz} + 2f\left(\frac{a^2 + b^2}{v}\dot{\psi}_{T2} + \frac{b\gamma}{r}y_{T2}\right) + 2a^2k_t\psi_{T2} + 2b_0^2k_0(\psi_{T2} - \psi_B) = 0 \quad (A.30)$$

$$\bar{m}_T\ddot{y}_{T2} + 2f\left(\frac{\dot{y}_{T2}}{v} - \psi_{T2}\right) + 2k_Ty_{T2} + 2k_2'(y_{T2} - h_1\phi_{T2} - y_B + l\psi_B - h_2\phi_B) = 0 \quad (A.31)$$

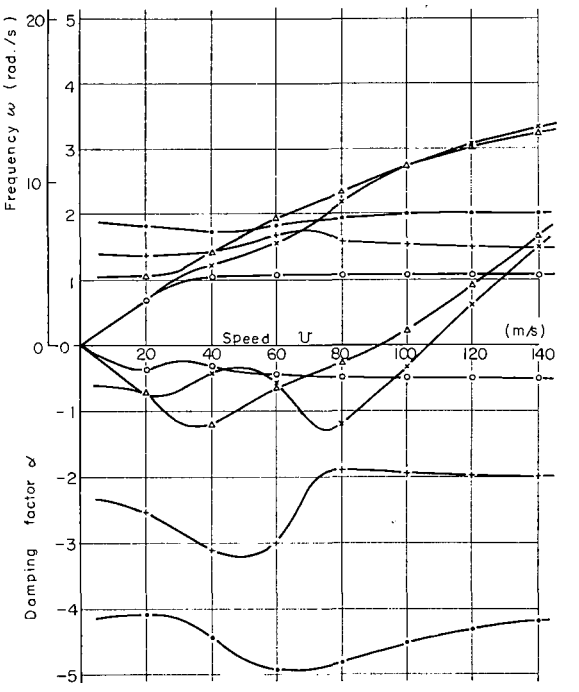


Fig. A2 Characteristic curves (variation of frequency and damping with speed) for a vehicle

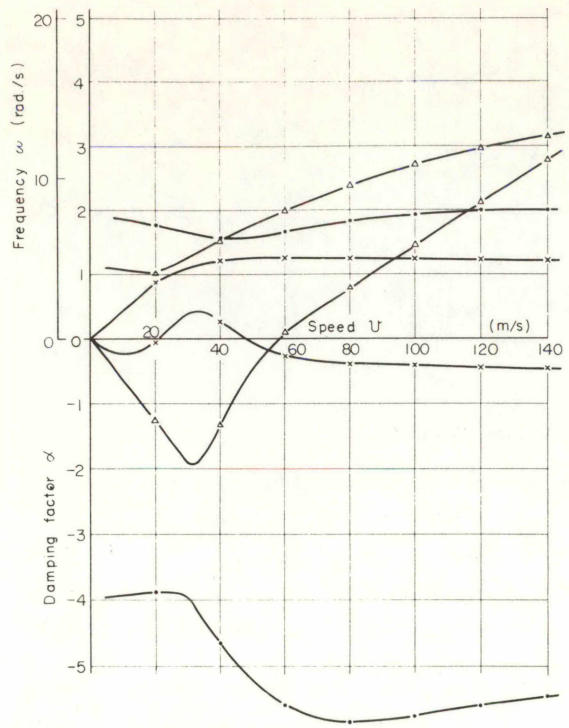


Fig. A3 Characteristic curves for rear half of vehicle

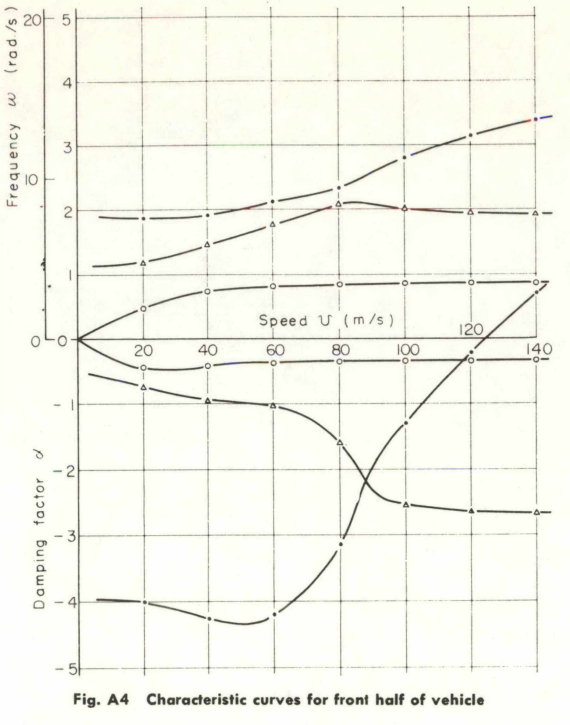


Fig. A4 Characteristic curves for front half of vehicle

where,

$$\bar{m}_T = m_T + 2m_W \tag{A.32}$$

$$\bar{I}_{Tz} = I_{Tz} + 2I_W + 2a^2 m_W \tag{A.33}$$

4 Numerical Calculation and Its Results

Substituting $y = Ye^{pt}$, $\psi = \Psi e^{pt}$, $\phi = \Phi e^{pt}$, . . ., where $p = \alpha + j\omega$, $j = \sqrt{-1}$, into the equations of motion, and putting the determinant of a coefficient matrix of Y, Ψ, Φ, . . ., as zero we get a characteristic equation, and solving it by a computer, α and ω can be obtained for various speeds of vehicle.

The numerical data used for the calculation are as follows, they are similar to those of the existing new Tokaido line vehicle except k_0.

$m_B \cdot g$ = 19,000 (kg),	$m_T \cdot g$ = 6,500 (kg),	$m_W \cdot g$ = 1,750 (kg)
i_{Bz} = 7.24 (m),	i_{Tz} = 1.18 (m),	i_W = 0.73 (m)
i_{Bx} = 1.3 (m),	i_{Tx} = 1.00 (m),	f = 1,315 (t)
h_B = 1.295 (m),	h_2 = 0.915 (m),	h_1 = 0.18 (m)
l = 6.875 (m),	a = 1.25 (m),	b = 0.75 (m)
b_0 = 1.42 (m),	b_1 = 1.05 (m),	b_2 = 1.25 (m)
r = 0.455 (m),	γ = 1/40,	k_0 = 100 (kg/mm)
k_1 = 259 (kg/mm),	k_2 = 50 (kg/mm),	k_2' = 36 (kg/mm)
c_1 = 40 (kg-s/cm),	c_2 = 22 (kg-s/cm),	c_2' = 50 (kg-s/cm)

The results of the numerical calculation are shown in Fig. A.2 to Fig. A.5.

These figures show the characteristic curves of hunting, i.e., the variation of frequency and damping of lateral motion of a vehicle with speed. The upper group of curves gives the frequency ω and the lower one the damping factor α.

The lateral motion of a vehicle is stable for the negative values of α, but it is unstable for the positive values of α. Therefore, it gives the critical speed of hunting when the value of α becomes zero.

APPENDIX 2

Derivation of Theoretical Formula for Wavelength of Hunting on Test Stand

In Fig. A.6 the points A and B represent the contact points of wheel and rails at a neutral position; A' and B' the contact points after the wheelset moves laterally by a small amount $y_c (\simeq y_A \simeq y_B)$.

Assuming that the lateral displacement of each wheel is composed of the angular displacement θ_1 around the center of curvature O_1 of the wheel profile and the angular displacement θ_2 around the center of curvature O_2 of the rail profile, we have the following relations between θ_1, θ_2, y_c and ϕ (roll angle of wheel-set).

$$\theta_1 - \theta_2 = -\phi \tag{A.34}$$

$$y_c = \rho_T \theta_1 - \rho_R \theta_2 \tag{A.35}$$

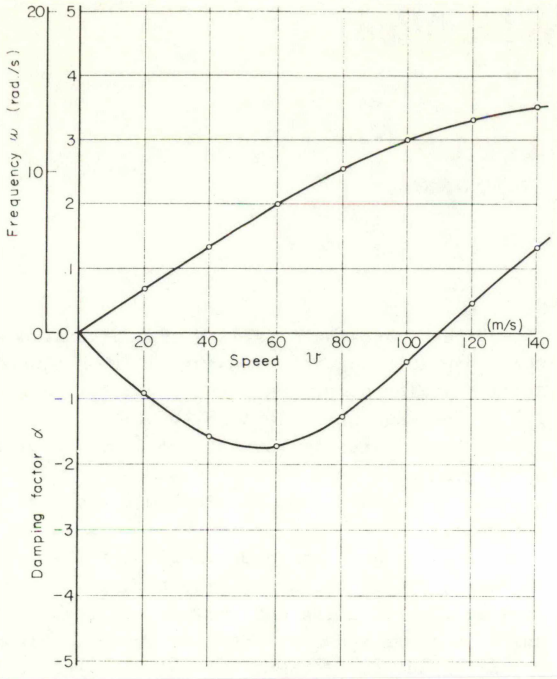


Fig. A5 Characteristic curves for a bogie with the body spatially fixed

$$\left.\begin{aligned}\theta_1 &= \frac{1+(\rho_R/b)\gamma}{\rho_T-\rho_R}\,y_c\\ \theta_2 &= \frac{1+(\rho_T/b)\gamma}{\rho_T-\rho_R}\,y_c\end{aligned}\right\} \tag{A.39}$$

Substituting (A.39) in (A.37) and (A.38), we obtain

$$\Delta r_{TA} = -\Delta r_{TB} = \frac{1+(\rho_R/b)\gamma}{1-(\rho_R/\rho_T)}\cdot\gamma y_c \tag{A.40}$$

and

$$\Delta r_{RA} = -\Delta r_{RB} = \frac{\rho_R}{\rho_T}\cdot\frac{1+(\rho_T/b)\gamma}{1-(\rho_R/\rho_T)}\cdot\gamma y_c \tag{A.41}$$

Assuming that there is no slip between wheels and rails, the yaw velocity of wheel-set must be equal to the difference between the peripheral speeds of both wheel and rail-wheel. Thus

$$\dot{\psi} = \frac{1}{b}\{(r_T+\Delta r_{TA})\omega_T - (r_R-\Delta r_{RA})\omega_R\} \tag{A.42}$$

Taking the relation

$$r_T\omega_T = r_R\omega_R = v \tag{A.43}$$

into consideration, (A.42) becomes

$$\dot{\psi} = \frac{\Delta r_{TA}}{r_T b}\,v\left(1+\frac{\Delta r_{RA}}{\Delta r_{TA}}\cdot\frac{\omega_R}{\omega_T}\right) \tag{A.44}$$

where ω_T and ω_R are angular velocities of wheel and rail-wheel.

Substituting (A.40), (A.41) into (A.44), we get

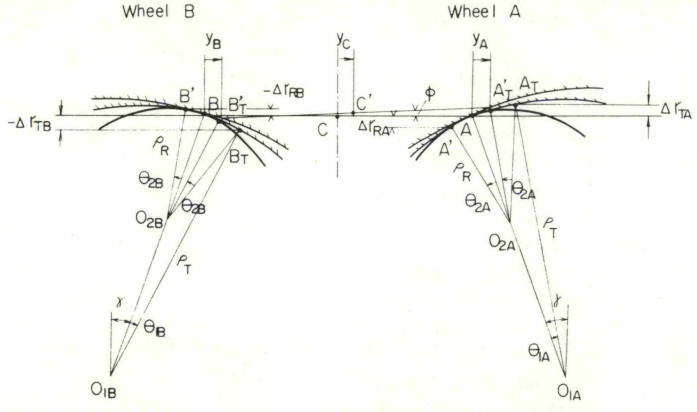


Fig. A6 Relation of lateral displacement of wheel-set to contact points of wheels and rails

$$\phi = \frac{y_c}{b}\gamma \tag{A.36}$$

When the displacement as well as the slope of tire profile is sufficiently small, the increment of wheel radius at the contact point caused by the lateral displacement of the wheel can be expressed as

$$\Delta r_{TA} = -\Delta r_{TB} = \rho_T\theta_1\gamma \tag{A.37}$$

Similarly the decrement of rail-wheel radius at the contact point is expressed as

$$\Delta r_{RA} = -\Delta r_{RB} = \rho_R\theta_2\gamma \tag{A.38}$$

From (A.34), (A.35) and (A.36), we get

$$\dot{\psi} = \frac{v}{br_T}\cdot\frac{1+(\rho_R/b)\gamma}{1-(\rho_R/\rho_T)}\left\{1+\frac{(\rho_R/\rho_T)+(\rho_R/b)\gamma}{1+(\rho_R/b)\gamma}\cdot\frac{r_T}{r_R}\right\}\gamma y_c$$

$$= \frac{v}{br_T}\cdot\gamma_e y_c \tag{A.45}$$

where

$$\gamma_e = \gamma\frac{1+(\rho_R/b)\gamma}{1-(\rho_R/\rho_T)}\left\{1+\frac{(\rho_R/\rho_T)+(\rho_R/b)\gamma}{1+(\rho_R/b)\gamma}\cdot\frac{r_T}{r_R}\right\} \tag{A.46}$$

γ_e represents the effective slope of wheel tread; in ordinary cases, neglecting the term $(\rho_R/b)\gamma$ in (A.46), it can be simply written without any significant error in the following form:

$$\gamma_e = \gamma\frac{1+(\rho_R/\rho_T)(r_T/r_R)}{1-(\rho_R/\rho_T)} \tag{A.47}$$

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Measurement and Analysis of Wheel-Rail Forces

L. A. PETERSON **W. H. FREEMAN** **J. M. WANDRISCO**

INTRODUCTION

There are compelling reasons for wanting to know what occurs at the wheel-rail interface as a railroad freight car moves along a track. These reasons assume a more practical than theoretical importance when a railroad is plagued with specific maintenance or operating problems, such as severe and accelerated rail deterioration or a relatively high ratio of derailments. The desire to initiate remedies for such problems invariably provokes many "why" questions which cannot be acceptably answered through mutual experience comparisons or surveys. The dynamic responses between the wheels of a car and the rails on which they ride is a function of many variables or degrees of freedom associated not only with track characteristics and car design, but also greatly influenced by operating methods.

The relentless and continuing quest to achieve higher speeds, to carry heavier loads, and to reduce derailments while decreasing car and track maintenance costs is creating problems which can best be solved by a scientific, factual and "systems" approach. As an example, the ability to set "track standards" for a given unit train hauling operation or for a defined high-speed passenger run is highly dependent upon knowledge of the dynamic response of the component cars which, in turn, is significantly affected by such things as the dimensional relationships and characteristics of the bearings, springing, and snubbing in the suspension system of the cars.

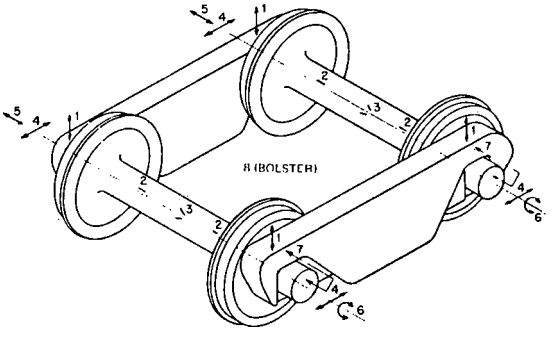
Continuous measurement of the vertical and lateral wheel-rail forces of a moving railroad car is a fundamental starting point in understanding the dynamic relationships between various components of the car and the track structure. By changing or modifying individual components of the system in small degrees, while holding the other variables constant, maximum cause-effect information can be extracted.

BACKGROUND

Practical considerations prompted the initiation of this study in 1968. Severe rail corrugations, head checking, spalling, and excessive wear were regularly occurring on curves of the Quebec Cartier Mining Company Railroad, which is a high volume ore hauling railroad with unit train operations. This paper deals with the findings of field tests conducted in 1968 on the QCM and continued during 1969 and 1970 on the Bessemer and Lake Erie Railroad.

Beginning as an investigation of the cause of rail defects, the study necessarily evolved to include consideration and performance testing of the more comprehensive car-track system. Since the significance of tight lateral constraints in standard roller bearing cars was an unknown factor which could conceivably influence wheel-rail reactions, cars equipped with modifications to give different degrees and quality of lateral movement were tested.

During the analysis of these tests, the intimate interaction between the wheel and the rail under dynamic conditions was quantified.



NOTATION	INSTRUMENT	LOCATION	MEASUREMENT
1	Load Cells	Between Bearing Cap & Side Frames	Vertical Force at Bearings
2	Strain Gages	Two Points On Each Axle	Axle Bending Moments
3	Strain Gages	Center of Each Axle	Axle Torque
4	Accelerometer	End of Each Axle	Longitudinal Acceleration
5	Accelerometer	End of Each Axle	Lateral Acceleration
6	Tachographs	End of Each Axle	Torsional Velocity
7	Displacement Transducers	Between End of Each Axle & Side Frame	Relative Lateral Displacement
8	Strain Gages	On Bolster at Two Locations	Bolster Bending Stress

Note: The Strain Gage Channels were Wired into Chevron Bridge Connections with the Lead Wires Passing thru Drilled Holes and Terminated at Slip Rings on the End of each Axle

Fig. 1 Truck instrumentation

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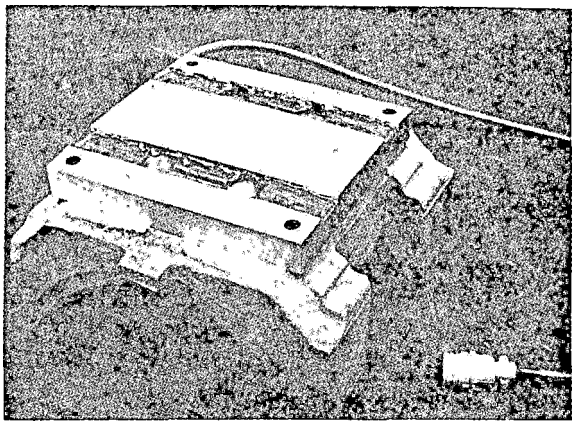


Fig. 2 Bearing reaction load cell for roller bearing car

METHOD OF INSTRUMENTATION AND MEASUREMENT

Car Instrumentation

Two basic cars, with five different car-truck configurations, were instrumented during the tests. Each truck tested was instrumented as shown in Fig. 1.

Track Instrumentation

Trackside instruments were located on a 6- and a 5-deg curve. The instrumentation consisted of linear variable differential transformers (LVDT's) attached to the field side of the heads of both the high and low rails at the same longitudinal locations to measure lateral movement of the rail head (both outward and inward). A photoelectric switch and light source at the same location were used to indicate the passage of each wheel, and a telemetry system transmitted the data onto magnetic tape in the recording car of the test train.

Lateral and Vertical Wheel Force Measurements

The means of determining the lateral and vertical force reactions at each wheel is a particularly critical part of the measurement. The vertical force transmitted from the car to the axle at each bearing was directly obtained by installation of specially made[1] bearing reaction load cells (Fig. 2). But, because of obvious direct measurement difficulties, determination of the vertical and lateral forces at the wheel-rail contact area was handled by an indirect procedure which relies upon the application and calibration of pairs of strain gages at known locations on the axle to measure instantaneous axle bending moments.

[1] Designed and assembled to specifications by Brewer Engineering Laboratories, Marion, Mass.

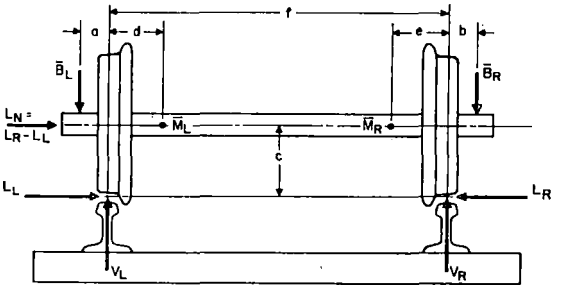


Fig. 3 Free body diagram of axle

Continuous measurement of the bending moment at any one point in a revolving axle is difficult. Oppositely placed (180 deg) strain gages in a single plane will yield a sine curve output as the point undergoes maximum tension at the top of the axle and maximum compression at the bottom with zero points at the two mid points between these extremes. Of course, this output is due to gage position and is not a direct measure of bending moment, except at the extreme points (twice per revolution). More accurate intermediate axle bending moment measures can be secured through installation of additional pairs of strain gages in rotative planes at the location. In addition, true intermediate bending moment values can be computed by trigonometric relationships involving the angle of revolution determined from the number of evenly spaced finite samples taken in each 90 deg of axle revolution.

Once appropriate axle bending moments are determined, they can be introduced as known values in free-body based equations of the wheel and axle unit using the forces and moment arms of Fig. 3.

Pertinent equations for equilibrium conditions are:

$$\bar{M}_L = L_L c + \bar{B}_L (a + d) - V_L d \qquad (1)$$

$$\bar{M}_R = L_L c + \bar{B}_L (a + (f - e)) - V_L (f - e) \qquad (2)$$

and

$$\bar{M}_R = L_R c + \bar{B}_R (b + e) - V_R e \qquad (3)$$

$$\bar{M}_L = L_R c + \bar{B}_R (b + (f - d)) - V_R (f - d) \qquad (4)$$

where $\bar{M}_L$, $\bar{M}_R$, $\bar{B}_L$, and $\bar{B}_R$ are known measured quantities and a, b, c, d, e, and f are physical dimensions.

These equations can be solved simultaneously,

Table 1 Test Car Features and Dimensions

Item	B. & L. E	Q. C. M.
Capacity	200,000 lbs.	199,000 lbs.
Load Limit	209,600 lbs.	199,400 lbs.
Lightweight	53,400 lbs.	51,500 lbs.
Cubic Capacity	2,775 cu. ft.	1,282 cu. ft.
Length over Strikers	41' 8"	31' 3 1/2"
Truck Centers	31' 8"	20' 6"
Wheel Base	68"	70"
Height Above Rail	10' 8 3/8"	8' 7"
Wheel Type	33" MW	36" MW
Test Wheel Tape Size	165	200
Test Wheel Circ.	104 5/8"	109"
Test Wheel Contours	Full Flange	Full Flange
Axles	6 1/2" x 12" RWS	6 1/2" x 12" RB
Axle Bearings	Plain	Roller
Spring Clusters	8-D-2 Outer 6-D-2 Inner	8-D-3 Outer 4-D-3 Inner
Spring Cluster Travel	1 5/8"	2 1/2"
Ride Control	None	Yes
Bolster Gib Clearance	1 3/4" Total	1/4" Total
Nominal Static Wheel Load	32,000 lbs.	32,000 lbs.

in pairs, to give expressions for calculating the desired V_L. V_R, L_L, and L_R values. The equations so derived are:

$$V_L = \frac{\bar{M}_L - \bar{M}_R}{f - d - e} + \bar{B}_L \qquad (5)$$

$$V_R = \frac{\bar{M}_R - \bar{M}_L}{f - d - e} + \bar{B}_R \qquad (6)$$

$$L_L = \frac{\bar{M}_L - \bar{M}_R}{f - d - e}\left(\frac{f - e}{c}\right) + \frac{\bar{M}_R}{c} - \bar{B}_L\left(\frac{a}{c}\right) \qquad (7)$$

$$L_R = \frac{\bar{M}_R - \bar{M}_L}{f - d - e}\left(\frac{f - d}{c}\right) + \frac{\bar{M}_L}{c} - \bar{B}_R\left(\frac{b}{c}\right) \qquad (8)$$

and since $L_N + L_R = 0$, then

$$L_N = L_R - L_L \qquad (9)$$

The desired lateral and vertical force values are obtained by substituting the known dimensions and measured quantities for the appropriate terms. A resultant computed value having a negative sign indicates that the force vector is in the opposite direction to that shown in Fig. 3.

Although the physical dimensions, a, b, c, d, e, and f, represent known pre-measured constants, it is recognized that force application loci, such as the vertical rail-wheel locations, actually do shift on the wheel tread as the car rolls on the rails and that the force vectors, in general, are not always oriented or acting in as simple a manner as illustrated in the diagram. However, the magnitude of error introduced by treating the moment arms as constants, appears to be well within acceptable limits. There was close agreement with measurements obtained by using an alternate method, similar to that described by Olson and Johnsson,[2] for determining lateral forces.

It is immediately apparent that the calculations as outlined in equations (5) through (9) could not be manually performed for even relatively small amounts of collected data. Consequently, all calibration and measurement data were recorded onto magnetic tape in analog form. This not only allowed immediate generation of strip chart displays, but also provided a mechanized way of Analogue to Digital (A to D) conversion of the data — achieving digital output on magnetic tape. These digital tapes were subsequently processed by computer where solution of the equations and comprehensive analysis were accomplished.

TEST PROCEDURES

The series of test runs were made over a 15-mile section of track which included six miles of tangent, curves from 1 to 8 deg, and grades up to 0.8 percent. The test track structure included welded 140-lb rail, and jointed 155- and 131-lb AREA rail sections laid on wooden ties, well bal-

[2] Olson, P. E., and Johnsson, S., "Lateral Forces Between Wheels and Rails," ASME-AIEE Railroad Conference, ASME Paper No. 60-RR-6, Pittsburgh, Pa., April 1960.

Table 2 Frequency Distributions for Selected Measurements

Modified R. B. Car on Tangent Track
1st Axle, Trailing Truck
35 MPH, Welded Rail

Freq. Dist.	Measurement	Force Level Ranges (100 LBS.) -51 To -75	-26 To -50	-0 To -25	0 To 25	26 To 50	51 To 75	76 To 100	Average (100 Lbs.)	Std. Dev. (σ) (100 Lbs.)
1	Dynamic Part Of Vert. Force (V'_L)		6%	41%	48%	5%			-1	17
2	Dynamic Part Of Vert. Force (V'_R)		6	45	46	3			1	17
3	Dynamic Total Axle Force	1	15	33	36	13	2		0	25
4	Dynamic Axle Rock ($Rock_R$)		4	31	59	7			3	17
5	Lateral Force (L_L)				9	75	16		39	13
6	Lateral Force (L_R)				21	53	25	1	39	17
7	Net Axle Lateral Force (L_N)		2	47	46	5			1	17

		L/V Ratios 0 To .25	.26 To .50	Average	Std. Dev. (σ)
8	L_L/V_L Ratio	91%	9%	.16	.07
9	L_R/V_R Ratio	79	21	.18	.10

lasted with blast furnace slag.

Most test runs were made at a constant speed of either 20 or 35 mph, with several short runs being made over portions of the same section of track at 20, 25, 30, 35, 40, and 45 mph.

The usual consist of the test train included eight 100-ton hopper cars loaded to capacity with ore. The instrumented cars were centrally located. Comparative dimensions and components of the two cars tested as normally operated are listed in Table 1.

All the instruments were calibrated prior to the tests and checked after each test run. The output signals of all instruments, as well as a signal marking the passage of each milepost, were simultaneously recorded on magnetic tape by two 14-channel FM tape recorders. Synchronization of the two tapes was accomplished by simultaneously recording the milepost interrupt signals on both.

Test runs were made in both northbound and southbound directions, and data were obtained with the instrumented trucks in both the leading and trailing positions.

The QCM car, normally equipped with trucks having tapered roller bearings, was tested in four different configurations:

1 Standard ride-control trucks
2 Lateral pad equipped trucks
3 Lateral pad equipped trucks with cylindrical roller bearings
4 Standard ride-control trucks with cylindrical roller bearings.

During all the tests, limited data were simultaneously recorded for the B&LE car equipped with plain bearings for control purposes. This relationship was reversed during tests when measurement of the performance of the B&LE car was the prime objective.

One special series of runs was made with flat spots on the wheels, this condition having been simulated by intentional grinding of $1\frac{1}{2}$- to 2-in. flat surfaces on each wheel of one axle.

In the duration of the tests, 28 channels of simultaneous data were recorded for over 500 miles of test train travel.

ANALYSIS

Vast amounts of detailed data were generated. Preliminary data reduction efforts pointed out the shortcomings of attempting to make conclusions based solely upon strip chart measurements or on simple averages and/or maximum values. The analytical machinery developed and used was able to examine dynamic actions sequentially, to distrib-

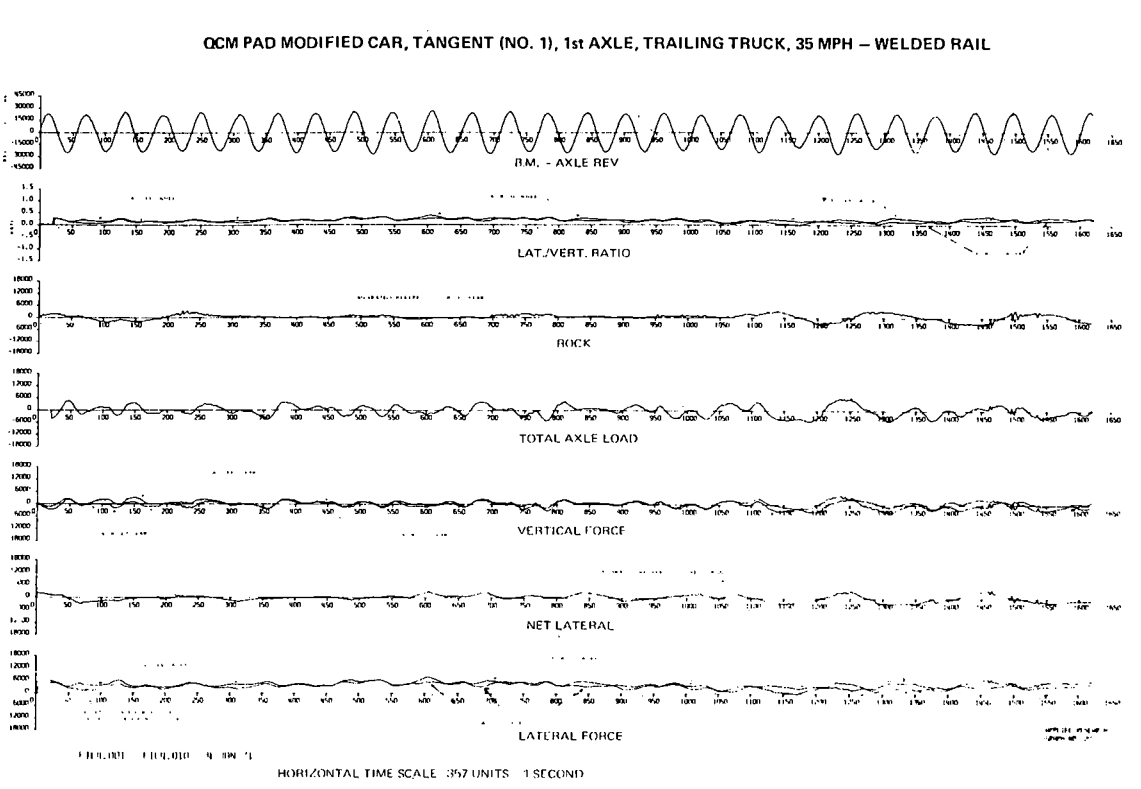


Fig. 4 Computer produced graphs of selected forces on welded tangent track

ute instantaneous force measurements or ratios into various range groupings, to determine the percent of time the forces were within these designated intervals, and to calculate various other statistical measures, such as standard deviations.

The recording of all data onto magnetic tape in analog form provided versatility and flexibility. Strip chart displays were produced as desired. Representative portions of the total data were selected for detailed analysis and converted to 800-bpi digital tape at a 357 sample per second rate and 11-bit resolution on each channel.

Conversational Fortran computer programs were written and a DEC PDP-10 time-shared computer utilized in analysis. The responsive man-computer interaction afforded by this arrangement proved invaluable and averted the problems and pitfalls inherently present in historic transactions between the analyzing scientist, a programmer "middleman," and a batch type computer processor.

The computer application described not only provided an accurate quantified print-out of the instantaneous lateral and vertical forces at each wheel, but also allowed many other types of instantaneous comparisons. Through appropriate combinations and calculations, simultaneous values, referred to as "rock," total axle load ("bounce"), and lateral to vertical force ratios (L/V), were determined for each 0.003 of a second. Numerous graphical representations of each of these quantities as they varied with time were also output from the computer. Two typical examples are illustrated in Figs. 4 and 5, for the case of welded tangent track and jointed curved track, respectively. The vertical bearing forces, total axle load, and "rock" traces are shown in reference to deviations from static.

Table 2 is an example of the result of a further step in processing of the data. In this procedure, each instantaneous value is allocated to a particular "cell" which represents a force level range which envelopes it. The number of times in which individual data points are allocated to each of these cells is counted. In this way, a "frequency distribution" is produced.

DISCUSSION OF RESULTS

A good portion of the findings from this

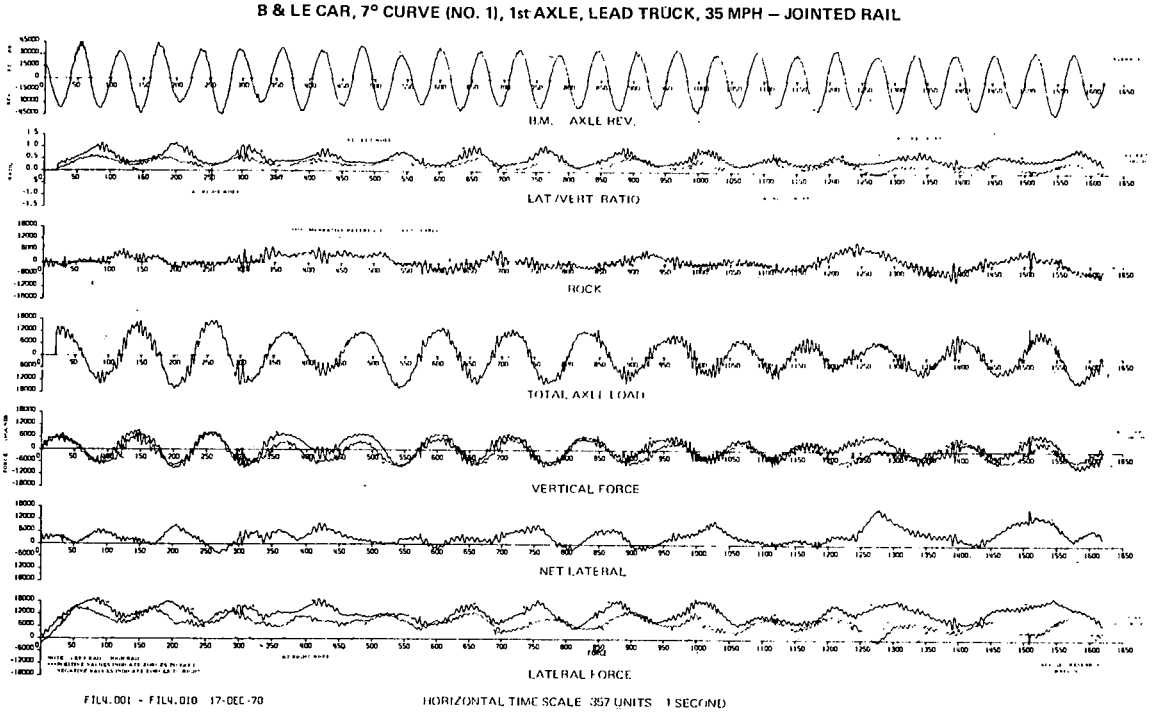


Fig. 5 Computer produced graphs of selected forces on jointed rail in a 7-deg curve

study were expressed in the form of recommendations for alleviating specific problems. Viewed in a more general context, the results supply a surprisingly wide variety of information in several subject areas, many of which have been previously studied.

The more salient conclusions can be discussed by considering three of the car types tested:

1 B&LE plain bearing car — Std. P.B. car
2 QCM roller bearing car — Std. R.B. car
3 QCM roller bearing car with lateral pads — modified R.B. car.

Signatures

Relatively minor modifications in car components did produce different patterns of wheel-rail interaction, yet each remained reproducible (and thus predictable) under identical test conditions. The terms, "signature" and "fingerprint," carry with them qualitative connotations which include: (a) uniqueness, (b) reproducibility, and (c) utility. It turns out that under adequately defined conditions, these are appropriate descriptions for the frequency distributions produced from allocation of measured values into cells (defined ranges) on curves over 3 deg, and apply equally well to distribution of the 357 sample per second values of the lateral forces at each wheel, the net lateral on the axle, the vertical force on each bearing and at each wheel, the lateral to vertical force (L/V) ratio at each wheel, the dynamic load shift from one side of the axle to the other (rock), and the total axle dynamic load (bounce). Sample signatures for several of these measurements, via cumulative frequency graphs, are displayed in Figs. 6 through 9.

Lateral track deflections, as measured from the trackside LVDT's, also satisfied signature requirements. Fig. 10 is an example of representative rail displacement signatures, for selected individual cars, obtained from track instrumentation at a fixed location in a 6-deg curve on jointed track. Fig. 11 has similar information for both travel directions from a 5-deg curve location on welded rail.

The potential implications of the establishment of a well-defined quality control type signature or fingerprint system are far reaching. Objective evaluation of car design changes, track inspection through measurements obtained from a "standard" car, and automated inspection of cars in a moving train from trackside instrumentation are suggested.

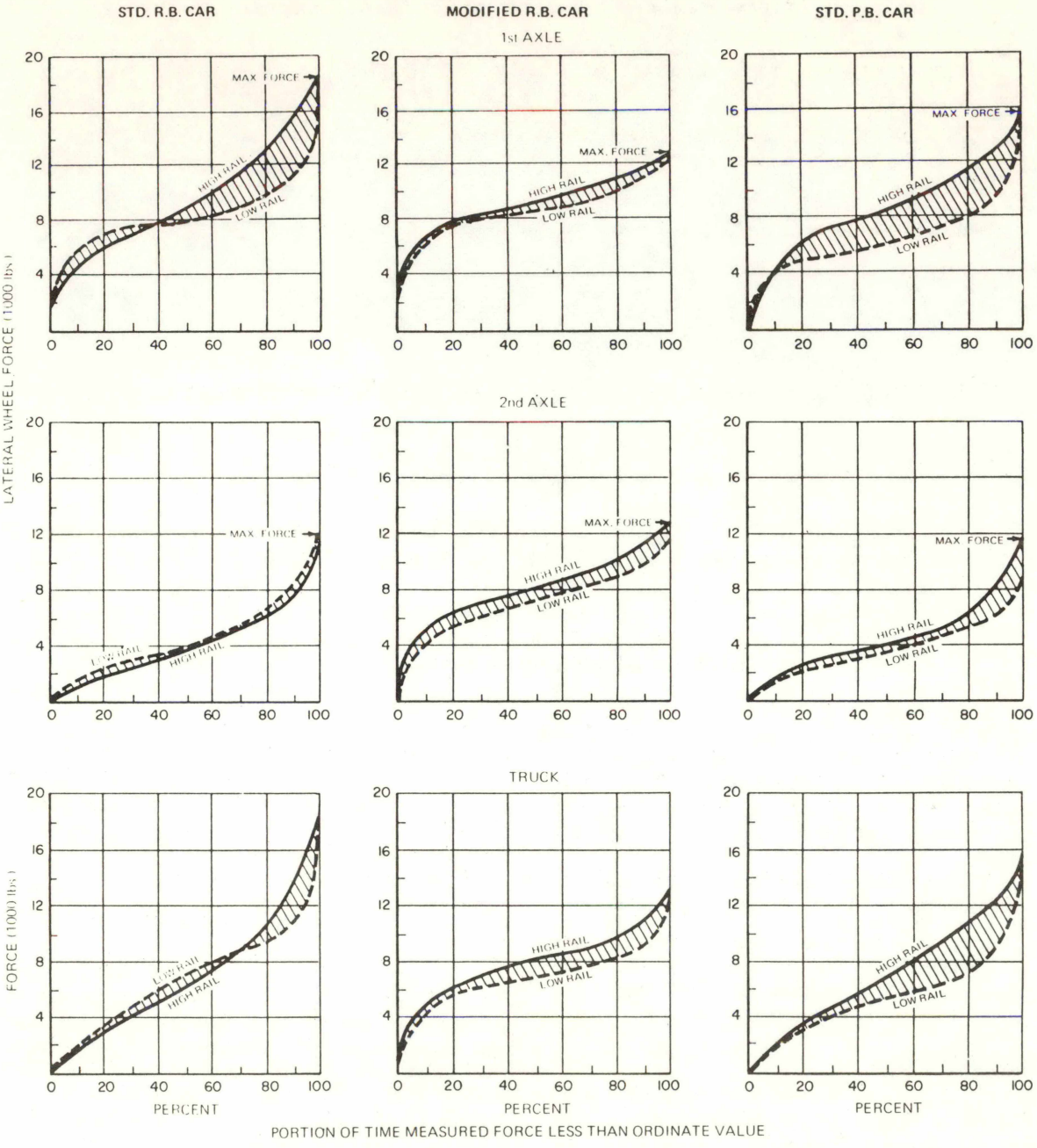


Fig. 6 Lateral wheel force comulative frequency distribution signatures for three different car designs at 35 mph on a welded rail 5-deg curve

Curved Versus Tangent Track

Due to differences in suspension, there are definite differences in car performance in respect to lateral and vertical reactions on curved and tangent track.

Fig. 12 shows typical net lateral axle (L_N) relationships at 35 mph for the force level one standard deviation (1 σ) above the mean. The points plotted are values corresponding to the axle on the truck with the highest 1 σ value.

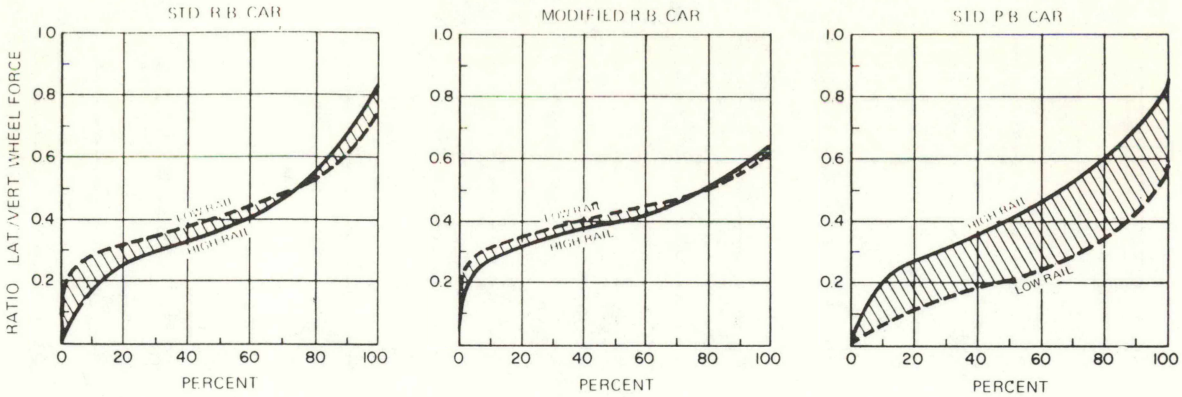


Fig. 7 L/V first axle wheel ratio cumulative frequency distribution signatures for three different car designs at 35 mph on a welded rail 5-deg curve

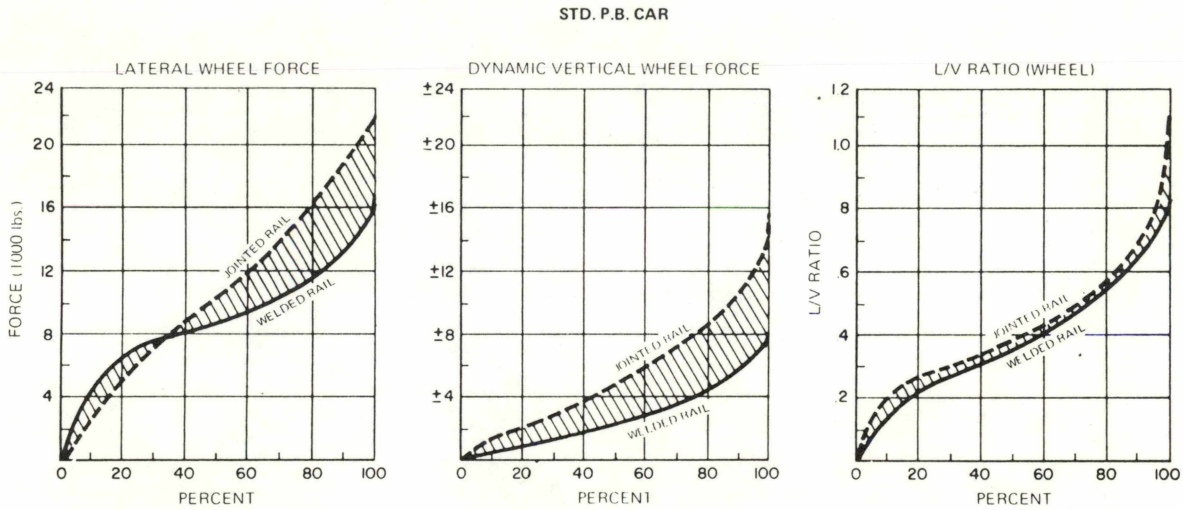


Fig. 8 Selected cumulative frequency distribution signatures for high rail side of first axle of the standard plain bearing car at 35 mph on 5-deg curves of jointed and welded rail

Table 3 Vertical Axle Loading (Thousands of Pounds)

Car	Range of Total Axle Load During Long Term Bounce Cycles		Characteristic Dynamic Peak To Peak "Shocks"	
	Tangent	5° Curve	Tangent	5° Curve
Standard Plain Bearing	44 - 84	56 - 72	5	5
Standard Roller Bearing	58 - 70	59 - 69.	15	5
Modified Roller Bearing	58 - 70	60 - 68	10	3

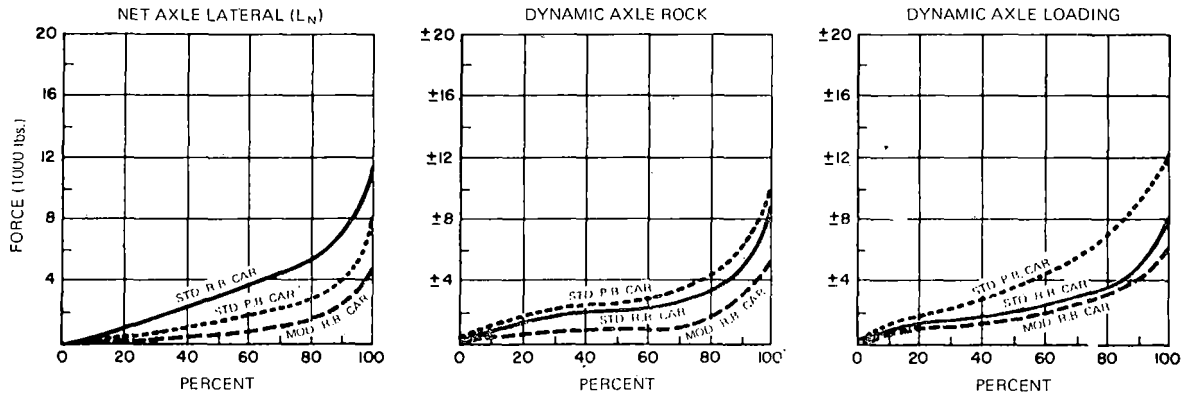


Fig. 9 Comparison of cumulative frequency distribution signatures for lead axles of three different car designs at 35 mph on a welded rail 5-deg curve

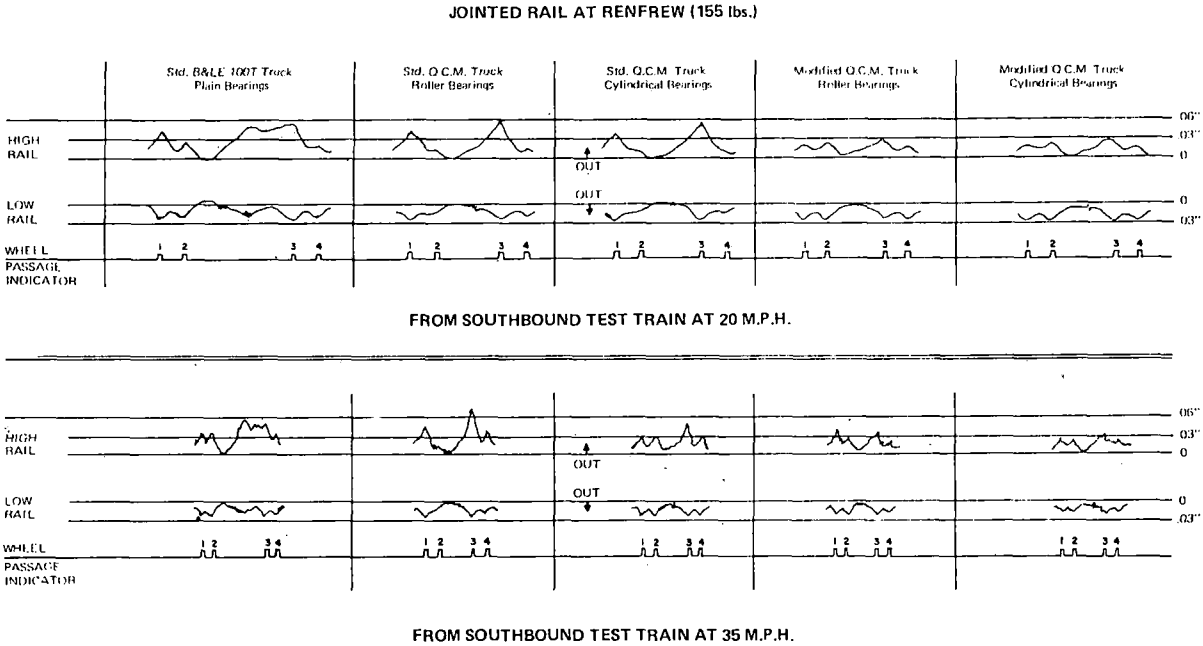


Fig. 10 Representative lateral rail displacement fingerprints for five different car designs at a location in a 6-deg jointed rail curve

The graph in Fig. 13 illustrates the similarly defined differences in fluctuating vertical total axle load force (bounce) for the three car types at a constant 35-mph-speed.

Note the increase in lateral force with curvature and the decrease in vertical bounce for the Standard Plain Bearing car with increasing curvature. (This truck is equipped with an unsnubbed D-2 spring grouping.)

Lateral Force Comparisons

The Standard Roller Bearing car, as used in normal everyday service, is typical of cars equipped with roller bearings allowing little lateral

Table 4 Comparison of Maximum Values for Various Measurements Jointed Versus Welded Rail (Standard Plain Bearing Car on 5-deg Curves at 35 mph)

Measurement	Rail		%
	Jointed	Welded	Reduction
1. Vertical Dynamic Wheel-Rail Force (1000 lbs.)	14	8	43
2. Lateral Wheel-Rail Force (1000 lbs.)	21	16	24
3. Net Lateral Force Per Axle (1000 lbs.)	16	8	50
4. Rock Per Axle (1000 lbs.)	15	8	47
5. L/V Force Ratio per Wheel	1.10	85	23
6. Axle Bending Moment (1000 lb.-ft.)	46	38	17

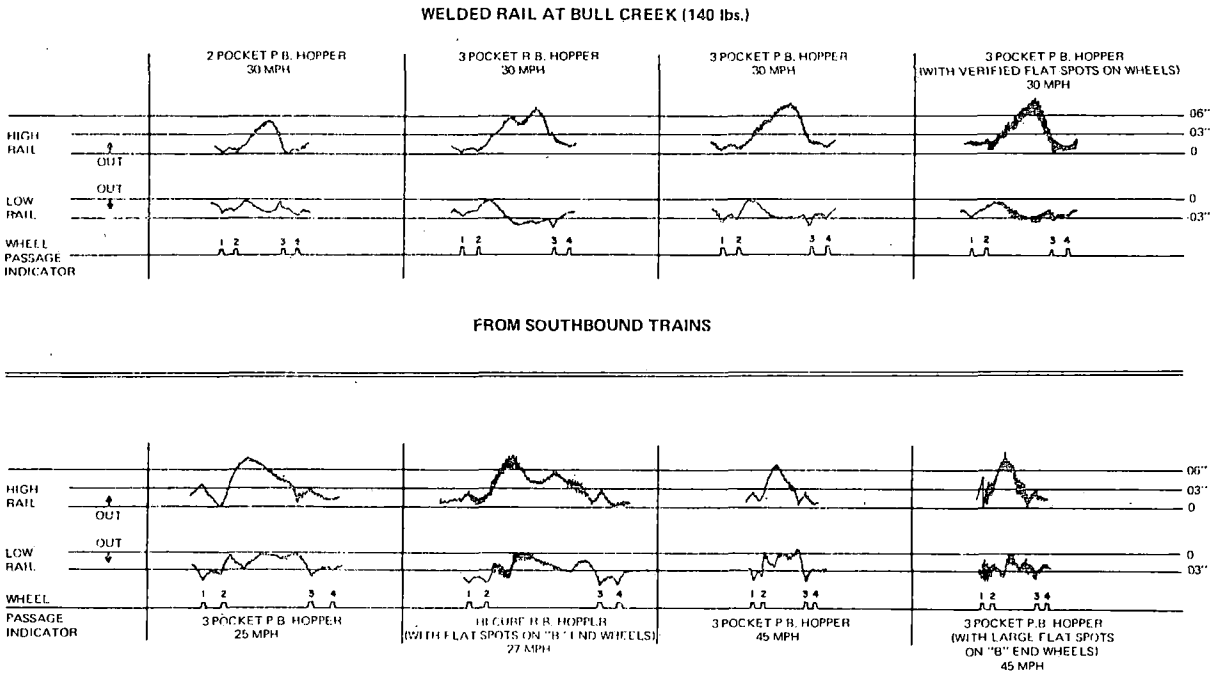


Fig. 11 Selected lateral rail displacement fingerprints from a location on welded rail in a 5-deg curve

movement between the side frames and the axles, and only a small amount of lateral movement between the bolster and the side frames. Such trucks may be termed "rigid" as opposed to more "flexible" plain bearing cars.

On curves, cars with rigid trucks produce a characteristic type of rotative action about the lead, high rail wheel of the truck. This action is substantially diminished when the car is modified to allow lateral movement between the side frames and axles by the insertion of lateral springs (flexible rubber-steel sandwiches called lateral pads).

As Fig. 12 illustrates, lateral force levels are reduced in the modified roller bearing car case and are also less for the plain bearing car in comparison to the standard roller bearing car. In addition, dynamic lateral force frequencies (vibrations) corresponding to the peak-to-peak distance (7 to 12 in.) between corrugations on the

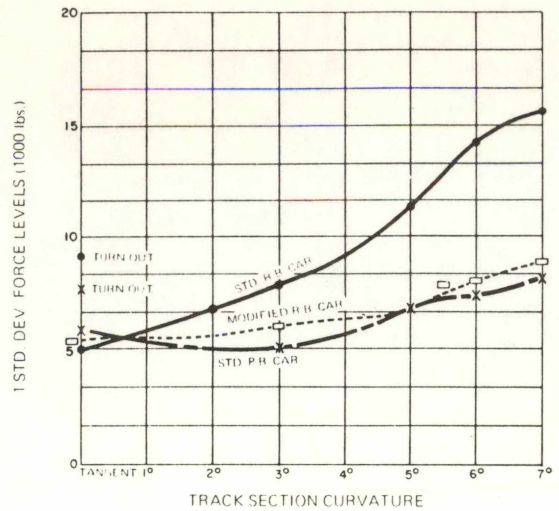


Fig. 12 1 σ level net lateral axle (L_N) forces on representative jointed rail track sections

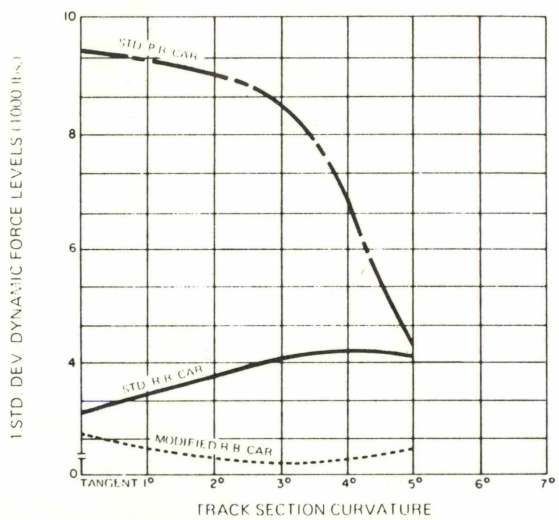


Fig. 13 1 σ level dynamic variations in total axle load ("bounce") on representative welded rail track sections

QCM rail were much more pronounced in the standard roller bearing car measurements than on the other truck designs tested.

Vertical Force Comparisons

Transmittal of the load vertically to the axle bearings and eventually to the wheel-rail contact area is a much more complex occurrence when a car is moving (dynamic force) as compared to when it is standing still (static force). The dynamic pattern of load transmittal as a car traverses a given section of track is determined chiefly by the suspension system of the truck (bolsters, springs, bearings, wedges, snubbers, etc.), and, as Fig. 13 depicts, the magnitude may vary significantly depending upon the characteristics and clearance between these components.

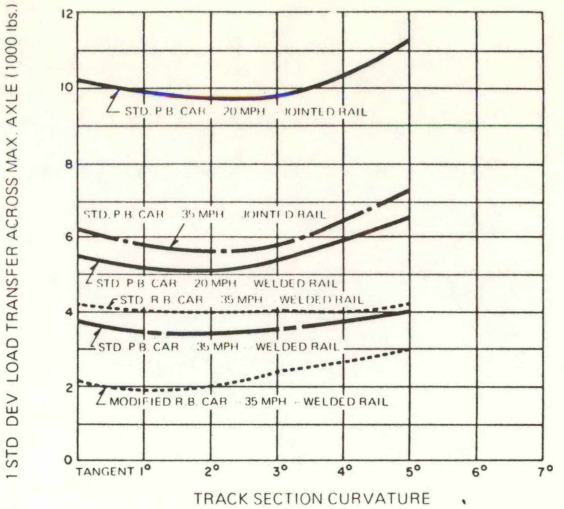


Fig. 14 1 σ level rock comparisons for selected car designs, speeds, and rail types

Rock is a method of viewing the transmittal of vertical force. The dynamic transfer of vertical load from one side of an axle to the other is termed "rock" in this paper. The traces in Figs. 4 and 5 each contain a labeled time-varying graphical interpretation of this quality as defined by:

$$\text{Rock}_R = \bar{B}'_R - \bar{B}'_L \qquad (10)$$

where each B' term is the "dynamic" portion (deviation from static) of the bearing load measurement at a given instant of time. Each B' term will assume a negative value when it is less than the static condition.

Repetitive car rocking was observed on jointed track, but only at widely separated locations on welded rail. The nature of the rocking on jointed rail is dependent upon the car design and the speed. Maximum rocking amplitudes on jointed track consistently corresponded to joint half intervals of 19 1/2 ft during speed ranges of 18 to 25 mph. Above 25 mph, rail joint input produces very short duration "shock" or impact effects of as much as 10,000 to 12,000 lb in 0.003 sec — rather than the longer term resonant reactions at lower speeds. Characteristic car actions associated with curving responses tend to make

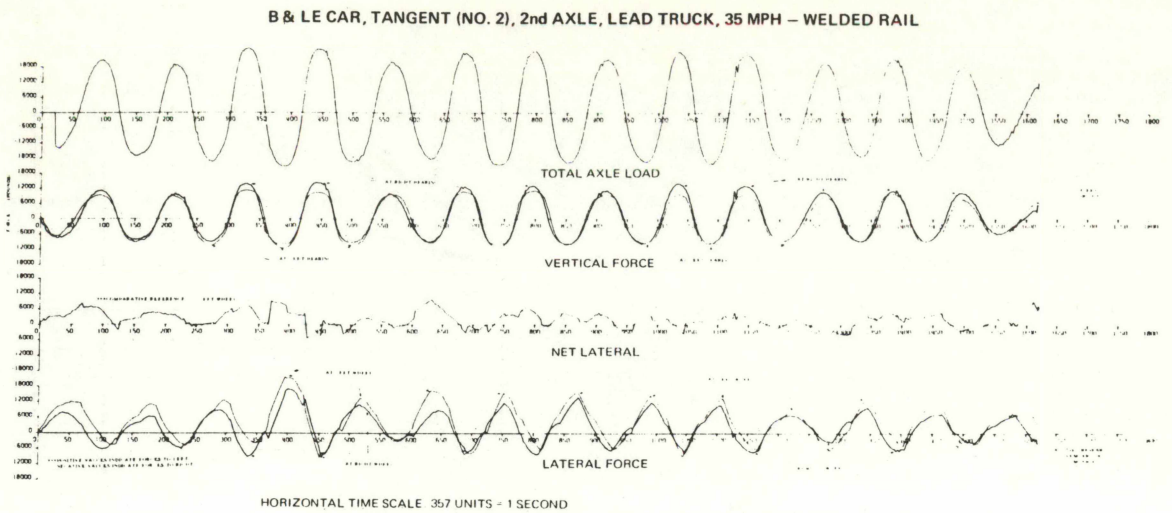


Fig. 15 Characteristic axle loading and lateral force effects of standard plain bearing car in bounce mode on tangent track

joint-excited rock less discernible on curves.

Dynamic lateral-track profile reaction differences can be thought of as changes in track lateral resistance sensed by a car in passage over the track. The joint itself is a prime example of a discontinuity in lateral track resistance with the joint bar, bolts, etc. changing the lateral resistance. Such lateral "input" to a moving car as well as the widely recognized vertical errors at the rail joints, can be a trigger in causing dynamic vertical load shifts ("rocking").

The modified roller bearing car exhibited a higher capability for reducing resonant speed activated rock and of absorbing shocks received at higher speeds in accordance with Fig. 14 which also illustrates overall speed effects.

Bounce is defined as a pattern of dynamic total axle loading characterized by simultaneous and synchronous loading of all bearings of trucks of a car. An axle exhibits prominent bounce characteristics when the bearings on each end load and unload together Time correlated axle loadings on tangent track for the standard plain bearing car are shown in Fig. 15. The values for the total axle load trace shown in this exhibit and in other similar figures represent deviations from static conditions. They were arrived at by:

$$\text{Total dynamic axle load} = \bar{B}'_R + \bar{B}'_L \qquad (11)$$

Certain sections of tangent track caused the exhibited exaggerated bouncing of the plain bearing car during eacn passage.

The standard roller bearing car was equipped with friction snubbing in the spring group, while the standard plain bearing car had an unsnubbed or "free" spring travel system. This helps to account for the very evident differences in behavior when comparing this car to the friction snubbed roller bearing car, such as in Fig. 4. There are definite trade-offs. Friction snubbing generally reduces the amplitude of vertical force fluctuations but apparently initiates high-frequency vertical vibrations (60 to 80 Hz) which are also translated into lateral vibrations corresponding to the wavelength of rail corrugations.

Table 3 is a simple quantified comparison summary for the conditions of welded rail and 35-mph car speeds.

In-depth examination of the dynamic vertical forces in relationship to the simultaneous lateral forces measured in this study proved that the outboard bearings on axles, as present in freight cars, transform pure vertical force input at the bearing into well-defined lateral as well as vertical force elements at the wheel-rail contact area. In fact, bounce-type vertical loading and unloading of the axle will produce a reciprocating lateral reaction at the rail tending to force the rail inward and outward, respectively, as exaggeratively depicted in Fig. 16. (The axle does not bend in the reverse direction as shown in the lower portion of the sketch.)

Outboard Bearing Effect

Railroad freight cars normally have outboard bearings; i.e., the load is applied to the axle outside the wheels. The study called attention to

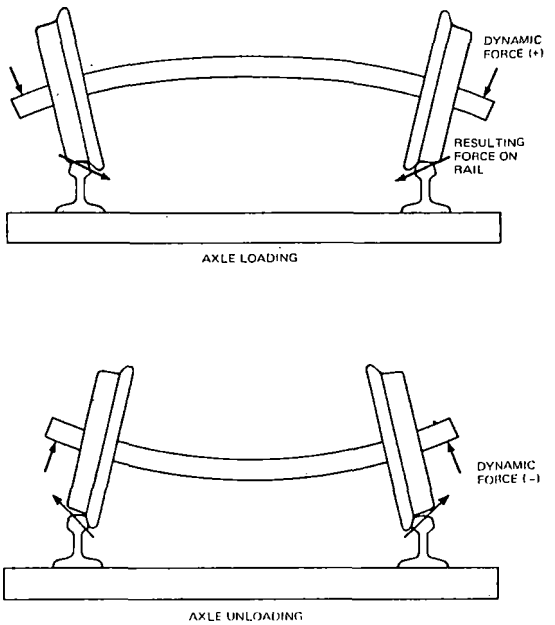


Fig. 16 Grossly exaggerated depiction of fluctuating force on rails due to alternate axle loading and unloading in bounce mode

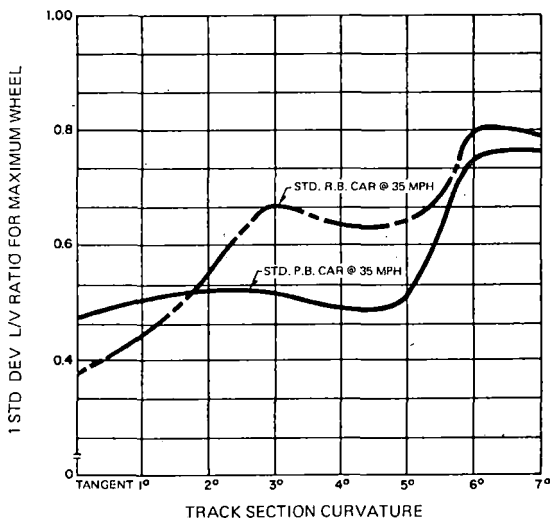


Fig. 17 1 σ comparisons of maximum wheel L/V ratios for standard plain and roller bearing cars on jointed rail

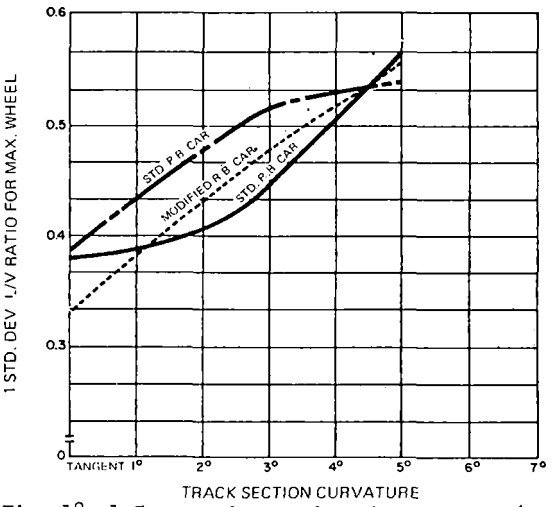


Fig. 18 1 σ comparisons of maximum wheel L/V ratios on welded rail at 35 mph

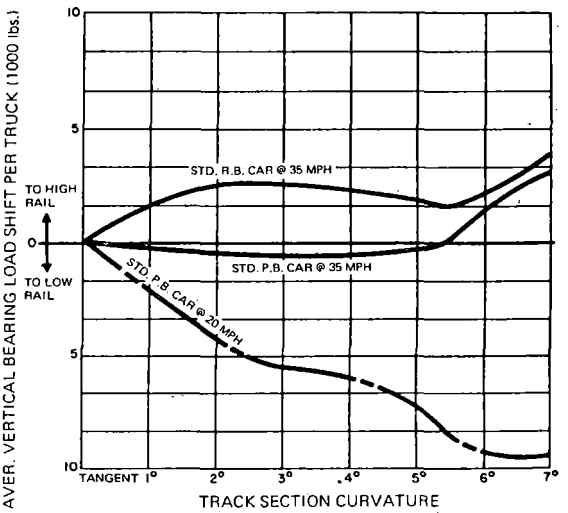


Fig. 19 Comparisons of average vertical bearing load shifts across lead truck on given curves at constant speeds of 20 and 35 mph

the fact that for either outboard or inboard bearings, the "offset" point of loading causes what might be intuitively visualized as a characteristic flexing of the axle as it unloads and loads in a bounce mode. Since the wheels are an integral right angle extension of the axle, the bouncing action produces not only vertical wheel-rail forces, but also significant and corresponding lateral wheel-rail forces.

Both trackside instrumentation and car instrumentation show that reactions resulting in an inward force on the rail are incurred. The amount of inward force is limited by the coefficient of friction between the wheel and the rail, since slipping will occur if the resisting frictional force is exceeded. While frictional coefficients as high as 0.6 were evidently reached in this

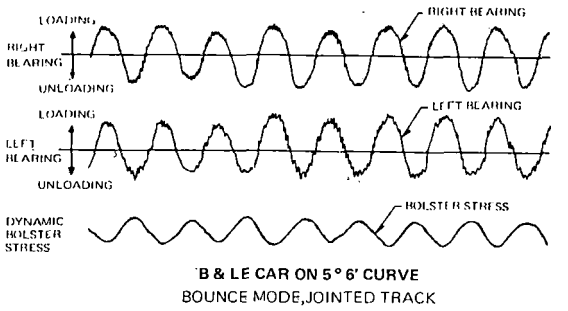


Fig. 20 Dynamic bolster stresses resulting from bounce mode of standard plain bearing car

study, no coefficient values in excess of 0.3 were found when the rail was being forced inward, even though such inward forces, which are limited by the coefficient of friction, infrequently reached as high as 12,000 lb per wheel. The reason for this is that the vertical force on the wheel at these peak lateral force periods was always in excess of 40,000 lb (maximum inward lateral always occurring at very heavy axle loadings). Thus, relatively high wheel-rail lateral forces were reached in these tests within the bounds of commonly accepted values of frictional coefficients.

This relationship of direct variation of lateral force with varying vertical loading has not been recognized to any extent in the industry, which may help to explain why many conflicting reports on lateral force levels and action are found. The importance of this finding, as applied to rail wear, is that vertical force dynamics alone can produce lateral stresses which could, under certain conceivable conditions, cause a type of lateral rail yielding at the wheel-rail contact area.

Wheel-Plate Stresses

Past research papers[3] have stated that wheel plate failures would be more prevalent if forces applied in a direction away from the wheel flange actually occur, because these stresses would be additive to residual stresses that are inherently developed in the wheel heat treatment process. Present wheel design practice assumes wheel forces applied toward or against the flange as being the rule, and these are "subtractive." This study indicates that outward forces on the wheel can occur with a higher degree of regularity, due to prominent car bouncing on certain long tangent track sections, than could be expected from exposure to retarders, etc. — which can also produce wheel stresses away from the flange.

Fatigue failures of this type result from the accumulation of repetitive cycles of a critical force level which may take place over varying lengths of time. Even though industry-wide wheel plate failures are rare, this study suggests that certain specific conditions involving consistent operations of "bounce-prone" cars over "bounce-exciting" track sections could result in a relatively higher frequency of failure. In these low-speed tests, lateral force application cycles with a component away from the wheel flange exceeding 4000 lb were recorded on certain long tangent track sections at the rate of 150 per mile.

Unit Train Hazard

In addition to all other considerations concerning track, operating conditions and car design, there is a clear and definite hazard of incurring cyclic type rail deterioration due chiefly to the passage of identical consists of loaded cars with heavy wheel loads within consistent speed ranges.

The study has very conclusively shown that on curves over 3 deg, a selected point on the track will behave in an identical manner during each passage of a given consist of cars at a given speed in a given direction; i.e., the entire trace of the deflections of the selected point on the track will be the same each time the given consist passes that point at that speed. Also, each car will tend to behave in a characteristic repetitive manner in traveling over a particular section of track on a curve; i.e., it will bounce or rock at the same point, etc. Another point selected on the track at a different location in the curve will have a different overall trace, but will exhibit the same consistency upon repetitive passages of identical consists. This means that if nominal axle loads are near the critical threshold, each dynamic impact will cause an excessive force to occur repeatedly in the same places.

Tangent track does not, to as high a degree, produce this repetitive type action. It appears that the reason for this difference between curves and tangent is the "following" or guiding action of the wheel of the lead axle on the high side of the curve as opposed to a more random "hunting" action on tangent track. Another way of saying this is that the lead wheel on the high side of curves follows or conforms to the lateral profile of the rail, imparting a predetermined pattern of car action much like the role of a stylus on a template. The necessity to conform to any one rail is usually lacking on tangent track, and

[3] Bruner, Levy, Jones, and Wandrisco, "Effect of Design Variation on Service Stresses in Railroad Wheels," ASME Paper No. 67-WA/RR-6, Pittsburgh, Pa., Nov. 1967.

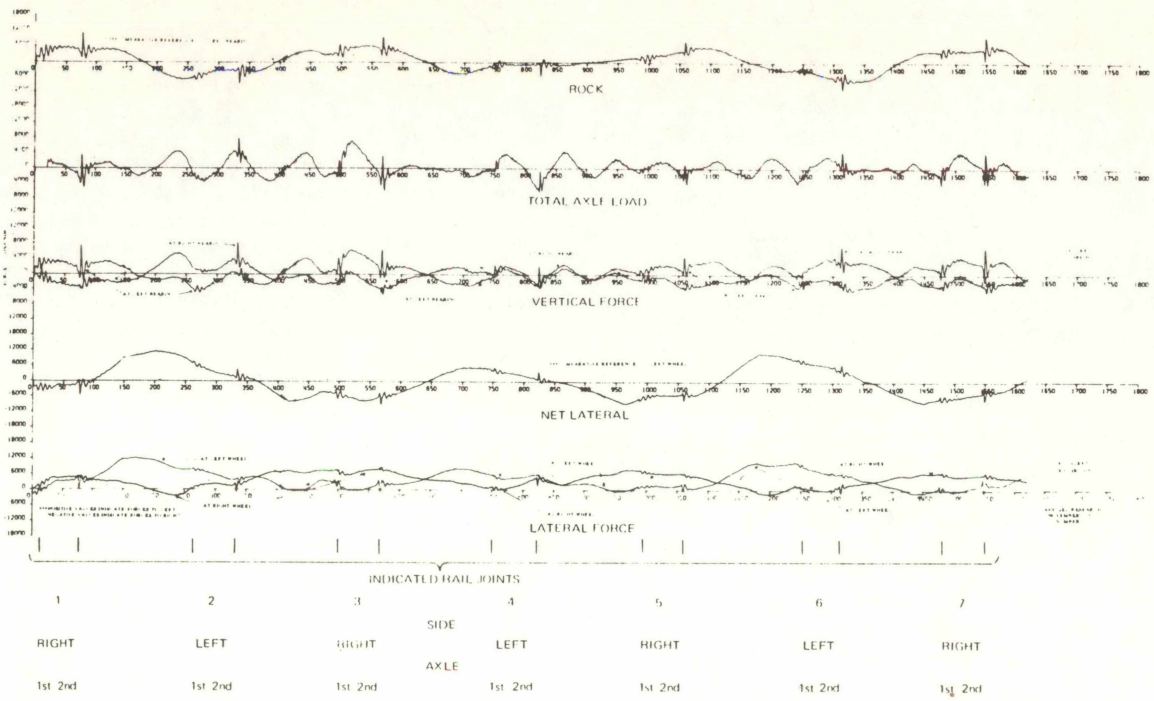


Fig. 21 Detection of rail joints from measured car performance values on tangent track at 20 mph

there is no significant travel distance differential requirement between the opposite wheels of an axle as there is on curves. As a result, the action of trucks in curves, where wheel coning cannot overcome rolling distance differentials, is not only repetitive in nature but, in addition, tends to produce relatively higher dynamic (vibrational) wheel-rail force levels than tangent track.

Regardless of the reason for the difference between tangent and curves, the basis for a definite wear pattern is established on curves; i.e., a high lateral force (with the same repetitive motion) will always occur at the same point on the track each time an identical train consist traverses that section of the curve.

The case in which loaded cars travel in both directions over single track may actually be better than when loaded cars consistently travel in one direction and empty cars in the other. The reason for this is that "uneven" wear is more of a problem than a somewhat greater rate of "uniform" wear. Empty cars do not generate anywhere near as high vertical and lateral forces as loaded cars and, therefore, are less likely to have any noticeable offsetting effect on the wear pattern established by the loaded cars. Conversely, in the case of loaded cars traveling in both directions, the loaded car pattern in one direction might offset a different wear pattern incurred in the opposite direction. Differences in wear patterns by direction occur because of track and operating condition variations; i.e., train action will be considerably different on an ascending versus a descending grade. The study did show differences because of direction, but no thorough analysis was made to determine whether the effects would usually tend to be offsetting — producing a more even overall rail wear pattern.

Consistent heavy wheel loads and consistent higher speeds will accentuate the problem because the force levels increase with speed and load. Cars with less dimensional clearance between truck components in their design and manufacture will also tend to provide undesired consistency in operating behavior in this respect. And, as on the QCM, cars of the same age, having been subjected to the same operations over a period of time, tend to wear in a similar manner and, in turn, will then react to a more consistent pattern on curves.

Paradoxically, though, it is not necessary that each car in the consist be identical with

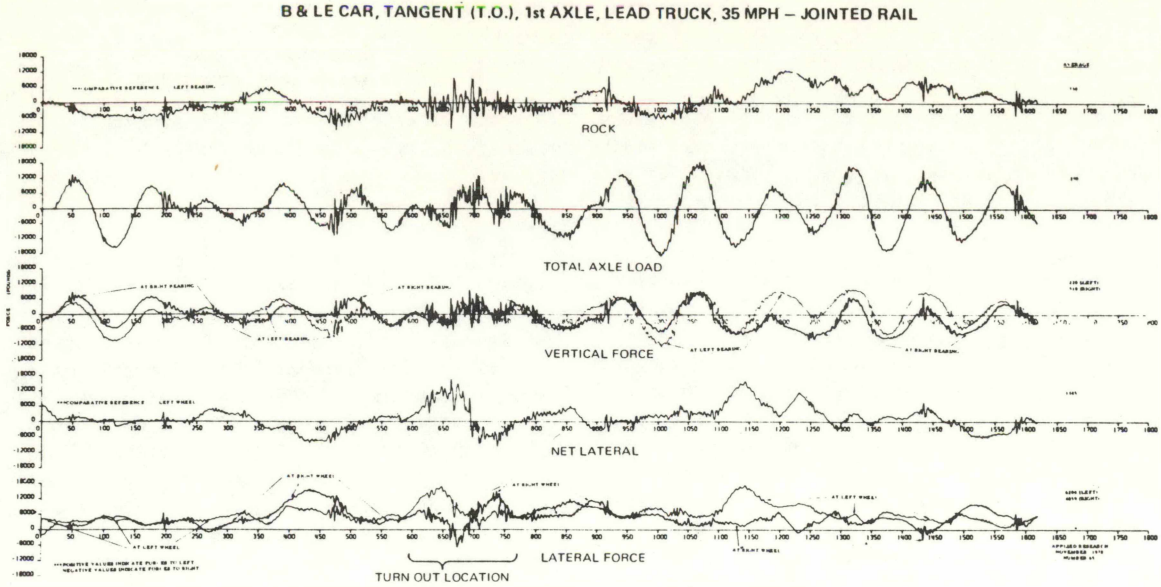


Fig. 22 Relative effects of turnout negotiation by the standard plain bearing car at 35 mph

each of the other cars for a repetitive pattern to be established. Even with different types of cars, the total force and dynamic action sequence for the given consist will be the same for each passage of the train — under the conditions described. Thus, the hazard is present whenever the same cars remain coupled together for successive trips.

There seems to be no practical way of completely eliminating the hazard of patterned rail damage on railroads having a predominance of unit train operations, such as the QCM. Hopefully, car design changes, which reduce the level of wheel-rail forces and modify undesirable repetitive car actions on curves, will significantly prolong the period of acceptable operation before the rail damage becomes intolerable. Specific actions which would: (a) minimize unsprung weights of cars, (b) make vehicle suspension "softer," (c) vary car design characteristics which might influence natural periods of oscillation, (d) reduce static and dynamic axle loading, (e) adjust track parameters to better match operations, or (f) decrease the frequency and level of lateral wheel-rail forces in curves, are examples of appropriate objectives.

Jointed Versus Welded Rail

Marked positive advantages of welded over jointed rail were confirmed. Results indicate that car components, as well as track structure members, should last longer and that derailments should be reduced, upon conversion from jointed to welded rail.

Fig. 8 illustrates comparisons concerning the lateral, vertical, and L/V ratio values for a particular case of the standard plain bearing car at 35 mph.

Other comparative maximum values incurred by the same car on similar 5-deg curves are shown in Table 4.

L/V Ratios

The obvious intent of this ratio is to provide an index comprehending the relationship between the force tending to drive the wheel, axle, truck, or car off the rail versus the force trying to keep the wheel, axle, truck, or car seated on the rail. The instantaneous ratio of the lateral force to the simultaneous vertical force at each wheel was determined for each fraction of a second and displayed during the analysis of the tests. Similar instantaneous ratios comprehending the net lateral (L_N) on each axle to vertical axle force and total net truck lateral to total truck vertical force were also computed. The degree of applicability or usefulness of these ratios is debatable, but they are of value in making relative performance comparisons in conjunction with other information. The short-term magnitude of such ratios is certainly not by itself an absolute indicator for forecasting derailments.

Great differences in L/V were found between curves and tangents, jointed and welded rail,

speed ranges, and the various car designs tested. The frequency distributions constructed met the "signature" test and facilitated quantitative comparisons in all the mentioned areas.

Fig. 17 compares the L/V performance of the standard roller bearing car to the standard plain bearing car on jointed rail, while Fig. 18 compares the three car types on welded rail.

Superelevation of Curves

An analysis of the behavior of three different car designs, when traveling at 35 mph over three separate, but statically similar, 5-deg curves, revealed that in actual practice, the true equilibrium speed is not solely a function of the degree of curvature and the amount of superelevation of the curve — which commonly used formulas state. On the first curve, the two roller bearing cars exhibited lateral and vertical force behavioral measurements which indicated they were traveling at a speed somewhat above true equilibrium. On the second curve, the modified roller bearing car appeared to be at equilibrium speed and the other two car types below. All of the cars seemed to be below equilibrium speed on the third curve. This points out that there is a difference between static, no-load measured superelevation and the elevation sensed by a moving loaded car. Effective superelevation (under dynamic load) of the different curves and the dynamic response of the individual car designs affect equilibrium behavior. Cars with high centers of gravity are more sensitive to superelevation errors in respect to corresponding amounts of deviation from equilibrium behavior.

The evaluation of when a car is or is not at equilibrium speed, in itself, is difficult — even when all of the vertical and lateral force reactions are known, as they were in this analysis. Defined criteria for vertical and lateral force characteristics may be met on the first axle of the truck but not on the second (frequently the case with standard roller bearing car).

Fig. 19 is a graphical example of comparing equilibrium behavior based on average vertical truck load shifts between the high and low rails on selected curves at constant speeds of 20 and 35 mph. Differences between the standard roller bearing and plain bearing cars at 35 mph can be seen.

In practice, many railroad M of W Departments periodically adjust superelevation based upon the track wear pattern evolving under operations, thus deviating from purely calculated elevations. This tends to match the amount of elevation to predominant wheel loads and car speeds. Of course, optimizing track wear by continuing to reduce the superelevation past a certain level could produce an uneconomical risk of high rail derailment, especially for certain cars. For instance, for a car that has a tendency to bounce, this equivalent to increasing the "L" term in the L/V ratio under conditions where the vertical (V) load may at times of axle unloading be drastically reduced.

Track Input

The study quantitatively confirms that rail, roadbed, and general track structure conditions act as stimuli for inducing various reactions in passing cars. Track geometry, such as alignment, vertical profile, lateral profile, gage, cross level, together with other qualities, such as ballast characteristics, etc., can and do produce very noticeable effects in car behavior. Each type of car design has a peculiar degree of sensitivity to certain combinations of track conditions and thus may react very differently from another type of car design. However, each car has a characteristic "carprint" or signature of action which is reproducible upon repetitive passage over a given section of track. When track conditions change, the measured car reactions also change.

Lower lateral wheel-rail forces are produced when the individual wheel and axle sets of the car can independently absorb lateral track irregularities. Cars having either plain bearings or lateral pad equipped axles possess this ability of independent accommodation to a greater degree than cars with common unmodified roller bearing axles.

It is important to realize that statically measured track dimensions do not necessarily yield an accurate index as to what the car will sense under moving (or dynamic) conditions. Historic track maintenance determinants relying heavily upon physical measurements of gage, cross level, alignment, etc., under no-load conditions do not always reliably appraise the acceptability of the track under normal operating conditions. Some factors, such as roadbed resiliency, are not even susceptible to historic methods of measurements. Dynamic track geometry measurements correlated with the type of car performance measurements described in this paper would provide a powerful combination in dealing with the problem of "setting track standards."

This study did show that the frozen roadbed, encountered in operations of the QCM Railroad during the winter season, does decrease track resiliency and that resulting dynamic loads during this time are higher by as much as 30 percent than under unfrozen roadbed conditions. Once track profile irregularities, such as corrugations, have begun, they act to trigger dynamic actions in the car, and the frozen track with a higher spring

constant will consequently produce higher impact forces. Thus, further deterioration of the rail will proceed at a faster pace on railroads with the roadbed frozen for extended periods.

Car Design Implications

Theoretically, captive cars operating constantly over the same track could be designed in accordance with measured stresses and car behavior to achieve lower costs and better performance. For example, railroad cars operating strictly on welded rail and with a small percent of curves over 3 deg and none over 4 deg could be of a more economical design (smaller axles, etc.) than on a similar railroad with higher curvatures. This is true because bending moments, torsion in the axles, and stresses in other car components are greater on curves of higher curvature. Data, such as produced in this study, could be used to assist in designing cars to fit individual situations of this kind.

The study showed that the bolsters on the 100-ton standard plain bearing car tested were being subjected to flexing stresses of up to ±5000 psi to give a total stress of greater than 20,000 psi as the axles loaded and unloaded in the characteristic bounce mode of the car. Such bouncing, interestingly enough, took place to a greater extent on certain sections of tangent track than on curves. The quantitative force and frequency of occurrence measurements obtained can serve as a basis for design modification in the bolsters to reduce fatigue failures.

Fig. 20 is typical of truck bolster stress action in relation to bounce or bearing loading on each end of an axle of the standard plain bearing car tested.

POTENTIALS FOR AUTOMATED QUALITY CONTROL

As an outgrowth of the various techniques developed and used in the study and in conjunction with pertinent findings, a mechanized and objective method of quality control to detect, by exception, departures of individually measured track sections and individually monitored cars from expected limits (signatures) is deemed highly feasible.

Locating and Describing Track Defects

Track conditions do act as stimuli inducing certain behavioral patterns into cars. The response of the car to this "input" is dependent upon its design, but is repetitive in nature; i.e., when the car is subjected to the same stimulus, it will behave similarly. The instrumentation used during these tests adequately measured and pinpointed the location of various track condition inputs.

For instance, individual joints and other physical track discontinuities, such as track turnouts, were accurately located in the tests by the characteristic action which they induced into the instrumented cars. Each of the five different car designs tested reacted in an individualistic manner when passing through a given turnout, and induced reactions can be ranked according to severity. Furthermore, the difference between trailing and facing movements was distinguishable. The standard roller bearing design car generally behaved in a more unstable manner when negotiating turnouts, as compared to the other car designs tested. Figs. 21 and 22 illustrate joint and turnout input effects on the measured forces of the standard plain bearing car.

As previously mentioned, certain locations on tangent track caused the plain bearing car to go into a particularly exaggerated bounce, and this occurred each time the car passed these points. While a cursory inspection of the track was not sufficient to isolate the track condition responsible for inciting this reaction, it is felt that a concentrated effort would generally succeed in uncovering the necessary cause-effect relationships to correlate car reactions to track conditions and that this information could be methodically accumulated.

One of the major hypotheses evolving from this investigation contends that periodic (say monthly) measuring runs with a standard car will be successful in detecting deviations from normal signature trends for a given segment of track (i.e., particular curve) which, when they become of a statistically significant magnitude, can be "flagged." Then a computerized follow-up process could isolate more exactly the location within the track section responsible for the "out of control" reaction and, through a correlation catalog amassed from past experience, suggest the probable cause.

For this purpose, a box car has been remodeled to house equipment for in-motion receiving and processing of data from instruments attached to a nearby car having known (standard) signatures. Equipment on-board the recording car, consisting of an FM recorder, an A to D converter, a mini computer, two magnetic digital tape drives, a CRT display, and a teleprinter, was installed to accomplish real-time analysis and evaluation of data with on-the-spot "exception" reporting, while still permitting more detailed "after-the-fact" analysis.

Detecting and Describing Car Behavioral Defects

Car behavior can be measured to varying degrees from instrumentation located on the car itself or by transducers located at trackside sites.

Instrumentation located on the car possesses the added utility of being able to assess track conditions.

The car-located instrumentation used during these tests measured the reactions of five different car designs over many variations in track conditions. Each of these designs generated an individual signature in response to various track inputs, and the quantified results showed surprising differences between the standard roller bearing car and the same car with the lateral pad modification, which is a relatively minor car component revision. The instrumentation used supplied enough information to evaluate the degree of improvement of one design over the other.

The implications for future utilization and application of such equipment are obvious, and these tests indicate that a "package" to accomplish the job, whenever car design evaluation appears beneficial, is practical and feasible.

Trackside instrumentation can also appraise the differences in reaction of various car designs, but only in respect to a single track location. It has the advantage over car-located instrumentation of being less costly and requires only a one-time installation expense, while car-located instrumentation may necessitate the application of appropriate measuring devices for each car design to be tested. Trackside instrumentation has the disadvantage of being stationary. However, knowing the predictable and repetitive behavior of all cars on curves over 3 deg, it was possible to precisely locate trackside measuring equipment to appraise the performance of the five car designs tested and produce results in substantial agreement with the car-located instrumentation. In fact, the trackside equipment was able to:

1 Detect the presence and direction of each train
2 Detect each axle of each passing vehicle
3 Pick out and distinguish between locomotives, cars, and cabooses
4 Differentiate between the different type of cars (two-pocket hoppers, three-pocket hoppers, roller bearing cars, high cube hoppers, etc.)
5 Determine the speed of each train
6 Detect abnormal behavior in any car and pinpoint the axle on the car producing the abnormal effect
7 Transmit all of the foregoing information via radio to a remote location where it was recorded on magnetic tape.

Figs. 10 and 11 are representative individual car extractions from longer train traces but illustrate some of the aforementioned capabilities.

Recently, the track instrumentation package has been refined and enlarged to simultaneously measure lateral deflections at each wheel-rail contact point of a car. Ways of automatically comparing "carprints" to standards and communicating "exceptions" to the train dispatcher are being tested.

CONCLUSIONS

1 The methods employed for measuring and analyzing lateral and vertical forces at the wheel-rail interface appeared to give generally acceptable and meaningful results, although there is room for refinement and improvement.

2 Results from this study allowed development of plausible hypotheses for explaining the phenomena responsible for specific wheel-rail problems and provided feasible directional guides for relief.

3 Experimentally designed collection of data of this type can raise pertinent questions and supply unusual insights into many other subject areas of the car-rail system.

4 Preliminary findings that various force measurements or relationships meet signature criteria under defined conditions encourage prospects of successful extension of techniques into mechanized track inspection, automatic monitoring and detection of abnormally behaving cars, and more immediate evaluation of the effect of car component changes.

There is a great need for additional and continuing work to secure experimentally controlled factual data which adequately describes the dynamics of the car-rail system under a wide range of conditions. Such "field" measurements, in addition to furnishing direct information, would contribute necessary support for verification of mathematical models, development of computer simulations, and design of test lab facilities.

ACKNOWLEDGMENTS

The authors make special acknowledgment of the competent work performed by Brewer Engineering Laboratories, Inc., in delivering the necessary transducer and measurement skills to fulfill the terms of a very exacting contract. Deep appreciation is extended to the respective managements of the QCM and B&LE railroads who have financially supported our efforts and in many other ways inspired and encouraged us. Finally, we thank the personnel of both railroads who participated in the actual tests and various representatives of other companies and universities who furnished component parts and/or valuable advice.

SEABOARD COAST LINE RAILROAD

— 70 Ton Box Car —

ROAD TEST

For

TRUCK & COMPONENT WEAR INVESTIGATION



REPORT No. 1-73

Reprinted from Seaboard Coast Line Railroad, 500 Water Street, Jacksonville, Florida 32202.

I. INTRODUCTION

In early 1972 the Seaboard Coast Line Railroad discovered accelerated truck and component wear on 150 50-ft., 70-ton, 20,000 series box cars built in 1967 by Pullman-Standard. In particular, severe wear was present on the truck wheels, the truck side frames at the roller bearing adapters, and the bolster center plate rim.

After two meetings with technical representatives from Pullman-Standard on July 12 and August 9, 1972, as well as a preliminary test run witnessed by both Seaboard Coast Line and Pullman-Standard representatives on July 18, specific test requirements and schedules were formulated toward determining the cause of the accelerated truck and component wear.

Following the test schedule agreement, specific assignments were made. During the week of August 21, 1972 all test runs were completed and documented under SCL/P-S R&D Test No. 2960.

II. EXECUTIVE ABSTRACT

From actual observations and close review of all movie film showing the action of the test box car, the accelerated truck and center plate wear was determined to be due to "shimmy" or truck "hunting". This truck "hunting" action originates from a natural tendency of the truck wheels to oscillate laterally between rails at specific freight car operating speeds, depending upon the condition of the wheels.

At some critical speeds, this lateral oscillation becomes a sympathetic harmonic transmitting a force frequency which causes a pendant type motion of the car body.

However, the primary factor related to truck "hunting" was found to be the contour of the wheel treads. Wheels with new AAR tread contour commenced "hunting" at 70 mph on the empty test box car. Wheels with a slightly worn tread contour commenced "hunting" at significantly lower speeds. In fact, in some test runs worn tread wheels began "hunting" at speeds as low as 50 mph.

Other factors believed to have major effects on the critical speed which cause truck "hunting" include:

1) The type of truck journal bearing (roller or friction)
2) The amount of load on the truck
3) The type of truck bolster side bearing
4) The rigidity or squareness of the trucks
5) The wetness of the rail.

Over and above the test runs conducted with the basic test car truck, three devices were tested to determine their effect on truck "hunting". These devices were:

1) National Casting's Constant Contact Side Bearings
2) VanGuard Corporation's Trioid Constant Contact Side Bearings
3) Buckeye Steel Casting's Elasto-Cushion Side Frame.

In each case, the device was volunteered to test by the individual maker and the results have been shown to their representatives or authorized personnel for clearance and approval to appear as part and parcel of this test report.

VII. CONCLUSIONS

A. The speed at which sustained lateral wheel "hunting" occurred on empty car tests seemed to be determined by the amount of wheel tread wear. New wheels went into sustained lateral "hunting" at 70 mph, whereas worn wheels started at 50 mph, regardless of whether the other components on the roller bearing truck were new or worn.

B. The worn wheels on the 50-ton trucks behaved similarly to the worn wheels on the 70-ton trucks. Similarly, the smaller diameter center plate on the 50-ton trucks had little or no effect on truck "hunting".

C. The wheel "hunting" on the roller bearing trucks seems to transmit lateral forces through the truck bolster into the car body through the center plate.

The car body behavior due to lateral truck action had a pendant type motion, as if pivoting about some point nearly 10 feet above the rail. (Contrast to car body rock and roll which has greatest motion at the top of the car).

The car body bolster lateral acceleration value increased with train speed and maximum relative longitudinal movement occurred at the center plate bowl and rim during truck "hunting", indicating accelerated wear to the center plate and related truck components is taking place during this action.

D. The only trucks applied to the empty car which perform well with respect to truck "hunting" at 70 mph were the worn plain journal bearing trucks. In this instance the worn wheels "hunted" at 50 mph and above, but the lateral clearance within the truck side frames was enough to allow the wheels and axle to float between the rails.

E. The critical speed at which time "hunting" took place was raised by loading the car. In the loaded car test, adding only half-load capacity of the car increased the critical speed by approximately 10 mph.

F. The observed truck "hunting" action of the Elasto-Cushion side frame truck appeared to be somewhat different than the other truck conditions tested. There appeared to be a sympathetic, harmonic oscillation set-up in the elastomer pads beginning at 50 mph.

G. The National and the Trioid are both constant contact side bearing devices to dampen truck oscillation by introducing resistance to truck swivel at the truck side bearings. The behavior of the trucks with this type of device acted as if they were heavily dampened and made the truck "hunting" mushy until about 70 mph when their effect disappeared.

All devices tested of this type were in new condition and under full side bearing preloads as established by their manufacturer.

H. Examination of worn wheel profiles (shown in Figures 1 and 1A) clearly shows how little wear is necessary to lower the "hunting" critical speed to well within freight train operating speeds.

Paper No.
68—WA/RR-3

General Aspects of the Lateral Dynamics of Railway Vehicles

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Recent developments in research into the lateral dynamics of railway vehicles are reviewed with particular reference to dynamic stability, dynamic response, and curving. Two engineering aspects of this work are stressed—namely, the advantages of applying linear theory to a comprehensive study of vehicle dynamics, and the choice of basic assumptions required to make such studies realistic for design purposes.

Introduction

THE DESIGN of new types of high-speed railway trains should be based on a realistic and comprehensive analysis of their lateral dynamics, with particular reference to the problems of dynamic stability, guidance, and response to track features. The purpose of this paper is to review recent results of research in this field with particular reference to applications to design. Attention is confined to the lateral dynamics of vehicles involving interaction between wheel and rail; the important problems of vehicle roll dynamics and vertical dynamics will not be discussed.

It has long been recognized that a major limitation to satisfactory high-speed running of railway vehicles has been the hunting oscillation. Many of the measures adopted in conventional railway technology in order to raise the critical speed above the operating speed ignore the basic fact that the tendency to dynamic instability is a consequence of the guidance offered by coned wheels. Unfortunately, the full potentiality of this guidance is not exploited on conventional vehicles, and the main theme of this paper and the research it describes are that railway vehicles ought to be designed so that for small displacements guidance is achieved by the use of coned wheels acting in conjunction with the creep forces.

This form of guidance in effect involves feedback, and the basic problem in the lateral dynamics of railway vehicles then is to resolve the basic conflict between guidance and stability with an adequate margin so that the dynamic response to economically realistic track geometries is satisfactory.

These considerations show that a comprehensive approach to the lateral dynamics of high-speed railway vehicles is necessary, and the various facets of this approach are reviewed in what follows.

Basic Concepts

For small displacements of a railway vehicle from the center of the track, guidance is achieved by the forces acting at the wheel treads in combination with the conicity of the wheels. The basic mechanism of guidance may be described with reference to the behavior of a wheelset. If a wheelset is rolling along the track and is slightly disturbed, there is a difference in rolling radii between each wheel due to the conicity. If pure rolling occurred, the subsequent response would consist of the wheelset tracing out a sinusoidal path as it moves along the track. This is termed the kinematic oscillation, and was first described by Stephenson in 1821. An analysis given by Klingel in 1883 shows that the frequency of the kinematic oscillation is proportional to speed and the square root of the conicity.

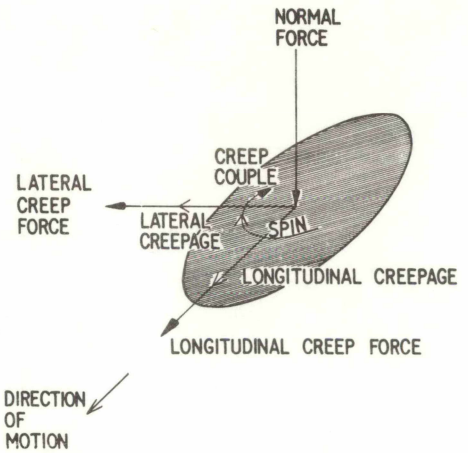


Fig. 1 Deviations from a pure rolling motion can be described in terms of longitudinal creepage, lateral creepage, and spin. These quantities give rise to corresponding creep forces and a couple.

Carter [1][1] pointed out that in reality, pure rolling cannot be maintained in such a motion because of the phenomena of creep. When tangential forces, as shown in Fig. 1, are transmitted across the contact area between wheel and rail, elastic strains occur which, when combined with the basic forward translatory and rolling motion, cause fractional deviations in the motion. These deviations are the creepages and spin, Fig. 1 (Johnson [2]).

For small values of the creepages and spin there is a linear relationship between these quantities and the tangential forces, and the creep coefficients are functions of the elastic constants of the wheel and rail, their curvatures, and the normal reaction. As the creepages and spin increase there is a departure from linearity until ultimately there is gross sliding of the wheel on the rail, so that a typical creep characteristic is as shown in Fig. 2.

A complete numerical solution assuming Coulomb friction has been given by Kalker [3] who has also reviewed theoretical work. Experimental work has been reviewed by Hobbs [4]. An improved understanding of frictional mechanisms operating in the interface is necessary, particularly where the surfaces are contaminated, and the rough surface theory being developed by Paul and Nayak at M.I.T. promises to make a valuable contribution in this direction.

Creep modifies the transient response of a wheelset. For a typical elastically restrained wheelset [5], the transient response

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

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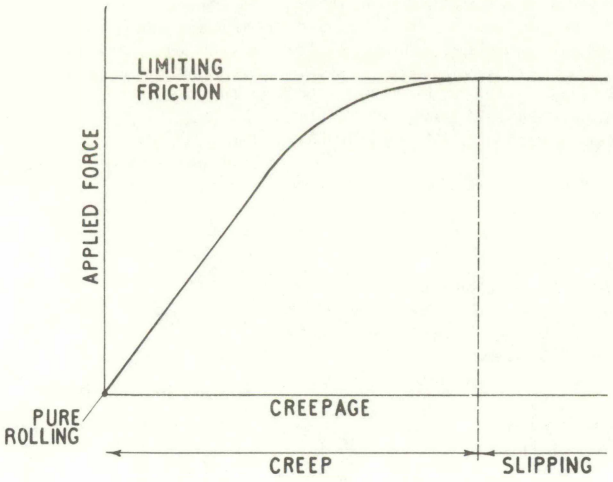
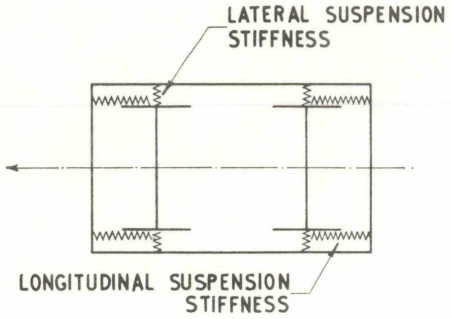


Fig. 2 The relationship between longitudinal creepage and longitudinal creep force, for constant values of lateral creepage and spin



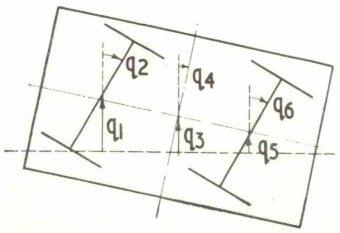


Fig. 3 Plan view of two-axle vehicle or bogie showing lateral and longitudinal suspension stiffness, and the six generalized coordinates necessary to describe the lateral motion

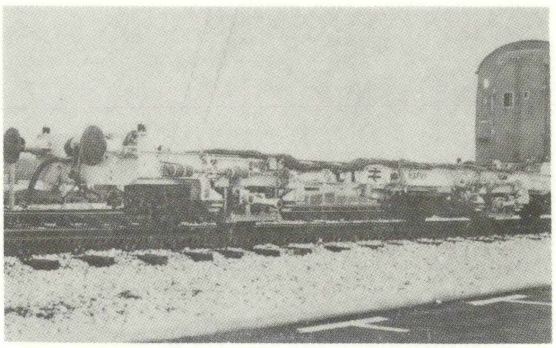


Fig. 4 Variable-parameter two-axle experimental vehicle HSFV-1

occurs at the kinematic frequency but is damped at low speeds. At high speeds the system is dynamically unstable, and so there is a critical speed dependent upon the elastic restoring forces and the inertia of the wheelset. Motions at speeds above the critical speed are limited by the wheel flanges and slipping at the treads and it is this limit cycle that is properly referred to as hunting.

The transient response of a stable wheel set depends on the magnitude of the input (change of cross level or alignment or other external force), the kinematic frequency, and the damping factor. An optimum response requires a high conicity and adequate elastic restraint, but as the critical speed is inversely proportional to the square root of the conicity, the requirements of guidance and stability conflict. This is the fundamental problem in railway vehicle suspension design.

In response to track curvature, an unconstrained wheelset adopts a radial position on the curve, but is displaced laterally so that pure rolling occurs. In response to cant deficiency or excess, a wheelset yaws, and thus generates a reacting lateral force. As mentioned previously, however, elastic restraint is required to provide an adequately damped response. For a two-axle vehicle or bogie, Fig. 3, this restraint is provided by the longitudinal stiffness between wheelsets and frame [6]. On curved track equilibrium is established by further lateral displacements of the wheelsets which cause the vehicle to take up a yawed attitude on the track. These further lateral displacements generate a yawing couple due to the longitudinal creep forces. If the creep forces are greater than the maximum friction available, slipping will occur and the flanges must come into play. The linearized theory of steady-state curving has been developed by Boocock [7]. Again, there is the fundamental conflict between the needs of guidance and stability.

Linear Dynamics

These basic concepts of guidance and stability apply not only to the single wheelset but to vehicles and trains. Comprehensive equations of motion including the degrees of freedom of wheelsets and body, the creep forces, and general geometries of wheel and rail have been derived [8]. The equations of motion consist, essentially, of systems of second-order linear differential equations with large numbers of degrees of freedom, and solutions have been

obtained analytically and by digital and analog computers ([9], for example).

Concurrently with theoretical work, experimental work has been carried out specifically in order to provide a proper basis for comparison with the theory. The experiments have been carried out with models [10], existing conventional railway vehicles [11], and a variable-parameter two-axle experimental vehicle HSFV-1 [12, 13], Fig. 4.

In general, favorable comparisons between theory and experiment have been obtained.

Applications of Linear Dynamics

The major application of the theory outlined here is to the design of guided vehicles which are dynamically stable and which have optimum damping so that the response to economically realistic track geometries is good up to very high speeds.

In addition to the obvious function of the suspension of supporting the vehicle body, the suspension must

(*a*) Provide a good ride for the payload.

(*b*) Minimize stresses and wear in the vehicle and track structure.

(*c*) Control the wheel-rail forces so as to guide the vehicle along the track.

(*d*) Maximize the adhesion available for both traction and braking.

While the performance of a given suspension system can be judged by comparison with human vibration tolerance criteria or derailment criteria, for example, such criteria are generally only useful in situations where the performance is thought to be marginally acceptable. In order to make a radical improvement in performance, a more fundamental approach to the application of theory is desirable.

It is therefore suggested that the aim should be to design a vehicle so that guidance is achieved by the creep forces acting on the wheel treads thus avoiding flange contact in normal running conditions. This aim implies that the amplitudes of the wheelset motions are within the flangeway clearance and, in addition, slipping at the treads should be prevented. Clearly, if slipping occurs, guidance is lost until the flanges come into play. Also, as a further function of the suspension is to maximize adhesion, it is necessary to minimize the demands on creep. Thus a linearized small disturbance analysis is relevant to the stated design problem.

Within the scope of a linearized small disturbance analysis, the design requirements may be stated in the following way.

(i) Dynamic stability, with a suitable margin over the operating speed limits.

(ii) Acceptable response to discrete track features such as curve entries and exits, switches and crossings, and to randomly distributed irregularities.

(iii) Satisfactory steady-state curving performance.

Using the theoretical basis referred to previously, it is possible to analyze vehicle and suspension configurations and optimize the design. Some examples are given next, but it should be noted that many of the parameters are fixed by the operating characteristics of the vehicle (size, weight, and speed) and by the nature of the wheel-rail contact conditions (as discussed in the next section).

For predictions of vehicle response it is necessary to specify the shape of the track in terms of curvature, cant, and variations in cross level and alignment This, together with aerodynamic forces, provides the input or forcing functions for the equations of motion. For a specific stretch of line, the track geometry can be specified by continuous records, but the description of a complete route can only be done on the basis of a statistical sample. For example, a realistic description of track irregularities can only be based on the idea that irregularities are continuously and randomly distributed. Discrete features, such as curvature and cant, can be treated as deterministic inputs. In order to obtain proper measurements of track irregularities, Gilchrist [14] has constructed a track measuring system and some power spectra are now available [15]. This information is being used in studies which relate the continuous, random response of vehicles to the random track irregularities.

One important application of response studies is in the formulation of rational loading cases for structural design of vehicles and track.

A further application of response studies is of fundamental economic importance. By establishing relationships between loads in vehicle and track and irregularities in the track as affected by track usage and maintenance procedures, it should be possible to optimize the economics of the vehicle-track system. This is particularly important in relation to high speeds, and there is an obvious trade-off between track alignment and suspension sophistication.

Wheel-Rail Kinematics

The most important kinematic parameter is the effective conicity which is defined as the rate of change of rolling radius with lateral displacement of the wheelset. For a purely coned wheel the effective conicity is simply equal to the cone angle, but the more general case in which the transverse curvatures of both wheel and rail are taken into account must usually be considered.

This is because for a purely coned wheel the area of contact with the rail is small and the contact stresses are high. Local wear occurs and this produces hollowness of the tire tread. After an initial period of rapid wear, the wear is distributed over the whole tread and no further change in the profile takes place. It is desirable to design vehicle suspensions so as to operate with this fully developed worn profile as the effective conicity is larger than that for a purely coned wheel with 1 in 20 taper. Improved guidance results, though with vehicles of conventional design the critical speed is reduced. A further advantage of the worn profile is that the contact stresses are reduced, and because reprofiling is unnecessary, the work-hardened tread surface is maintained thus increasing wheel life. Tire profiles based on these considerations have been designed by King [16], Fig. 5.

A linearized theory of wheel-rail kinematics has been given in [8], and this applies to both purely coned and profiled wheels. This theory can be applied in general investigations of the possible wheel-rail configurations as indicated in [8] and [9]. Measurements of existing railway wheel-rail combinations indicate that the kinematic characteristics can be very complex. In such cases equivalent linear values of the effective conicity (and other geometric parameters) can be established by hand (King [17, 18]) or by using special measuring devices and a digital computer (Gilchrist [19]). Estimates of the effective conicity may also be obtained by the describing function method or by statistical linearization [20]. These measurements show that large variations of the effective conicity can occur with values ranging from 0.05 to 0.50. A major reason for these variations is irregularity of gage.

One of the next steps in this research is to study analytically the development of worn wheel and rail profiles. This can be represented as a feedback loop, Fig. 6, in which the starting point is a track with defined geometry including the initial railhead profile, and a vehicle with initial wheel tread profile. The vehicle dynamic response to track geometry gives rise to creepage which is related to changes in the wheel and rail profiles by wear equations. Thus the effective conicity is altered and the dynamics of the vehicle changes. Of particular interest in this study is the long term stability of the profiles, particularly for the new suspension concepts discussed in this paper.

In view of the rapid wear of purely coned wheels, and the increase of effective conicity with increasing hollowness, the validity of design studies performed on the assumption of purely conical profiles is questionable, and it is desirable to cover the range of effective conicities just mentioned.

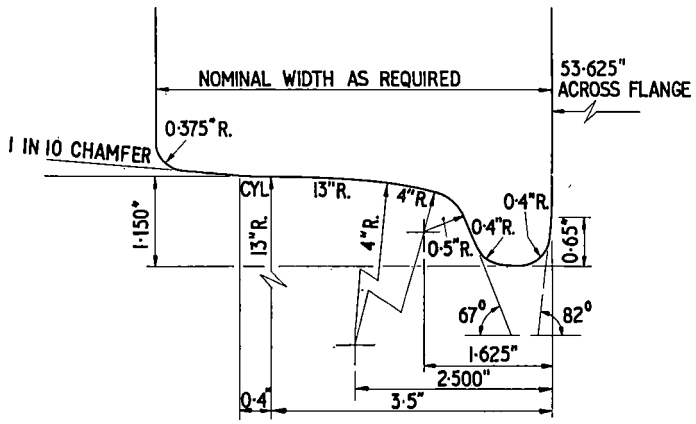


Fig. 5 The RD 4 standard tire profile for use on British Railways

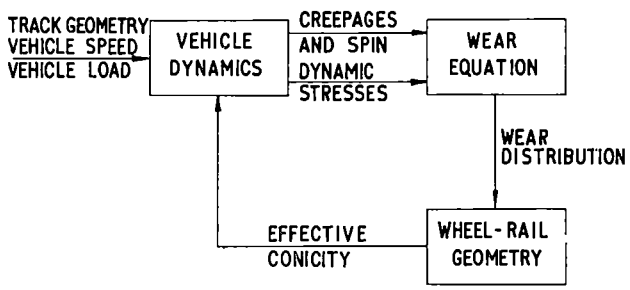


Fig. 6 The wheel tread wear feedback process

Conventional Railway Vehicles

From the earliest days of the railways, the basic form of guidance has been the wheel flange. Coning of the treads was introduced simply so as to prevent the almost constant rubbing of one or the other flange against the rail which occurs with a cylindrical tread. The customary coning of 1 in 20 or 1 in 40 is quite ineffective in curves, and consequently previous to the work described in [7] almost all scientific work on the behavior of railway vehicles in curves ignored the coning of the wheels and assumed flange guidance. Flange guidance leads to flange wear and side-cutting of the rails even with present-day trains, and this problem is likely to become more serious at the higher speeds which are of current interest. Very large dynamic forces can be developed at the entry to curves, and the track then requires more maintenance in order to preserve an acceptable standard of alignment accuracy.

On straight track, a major limitation to high-speed operation has been the hunting oscillation. In the past, the hunting oscillation was regarded as an unavoidable consequence of the use of coned wheels, and the very low critical speeds of the early four wheeled vehicles were regarded as inherent properties of this configuration. The early adoption of the bogie vehicle was probably due, in part, to its higher critical speed. For many years, critical speeds of bogie vehicles have been consistent with present operating speeds, and this has been achieved by the evolution of a design which depends for its stability on rather stiff primary lateral and longitudinal springs and on a rather stiff rotational spring between bogie frame and body. The design procedure has been described by Matsudaira [21] in connection with the trains for the New Tokaido Line.

In order to obtain a sufficiently high critical speed, the overall yaw stiffness is high and wheel treads are maintained to a conicity of 1 in 20 or 1 in 40 by reprofiling. Consequently, flange contact occurs readily on curves.

The susceptibility of a vehicle with stiff suspension to misalignments, and the low damping in the lateral modes of oscillation are both responsible for premature slipping, and this may be a reason for the apparent reduction in the adhesion available at higher speeds.

These comments, which may be quantified in particular cases, suggest that while current practice enables bogie vehicles to be run at high speeds, this is only achieved at a relatively high cost in terms of vehicle and track maintenance.

Suspension Concepts

The previous discussion of conventional vehicles indicates that their lateral dynamics is very complex because, for a significant proportion of the time, their behavior is nonlinear. This degree of complexity is a barrier to rational innovation. As suggested previously, a more fruitful approach lies in designing the vehicle so that, under all reasonable conditions, it operates on the linear portion of the creep characteristic, and continuous guidance is provided using the creep forces acting between wheel and rail thus avoiding contact of the flange with the rail. The powerful methods of linear dynamics may then be applied to the design problem.

In reviewing the application of linear theory, it is useful to discuss the influence of the lateral and longitudinal suspension stiffness between the wheelsets and the frame in the case of a two-axle vehicle or isolated bogie, Fig. 3.

The effects of varying both suspension stiffnesses together are illustrated in Fig. 7. In this diagram the regions of dynamic instability, unsatisfactory response, and nonlinear curving are indicated. The clear area indicates the region in which the

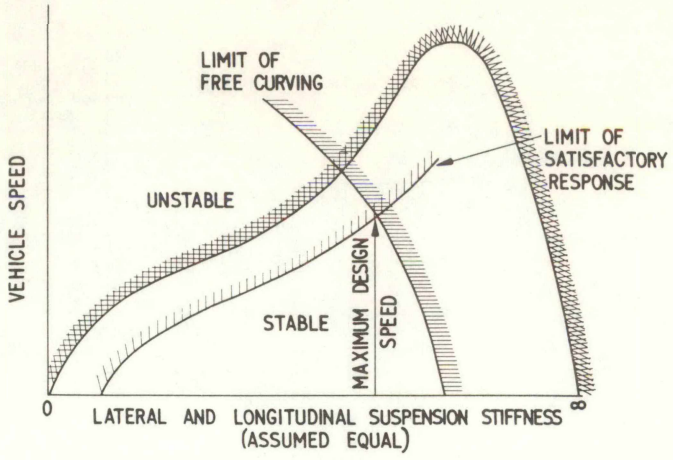


Fig. 7 Lateral dynamics of a two-axle vehicle in terms of lateral and longitudinal stiffness and vehicle speed, in the case of zero suspension damping

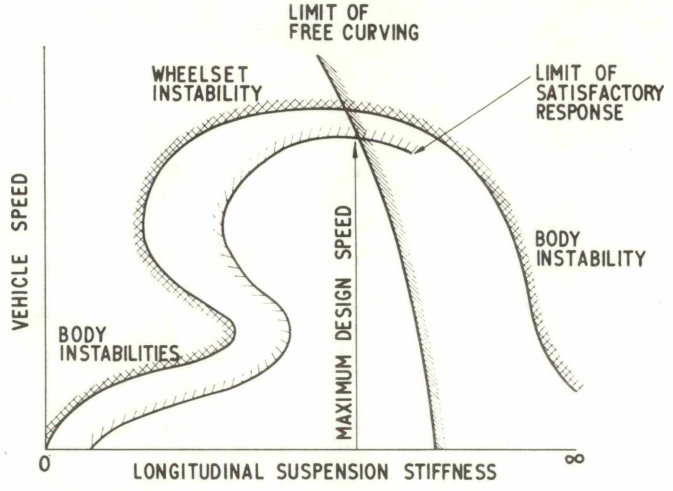


Fig. 8 Lateral dynamics of a two-axle vehicle in terms of longitudinal suspension stiffness and vehicle speed, in the case of optimum values of lateral suspension damping and stiffness

performance of the vehicle would be considered satisfactory on the basis already discussed.

For small values of stiffness, dynamic instability involves motion at the kinematic frequency of the wheelset. The lowest critical speed occurs when the kinematic frequency is rather close to a natural frequency of the body on the suspension. This mode of instability may be termed body instability as the basic mechanism is that the modes of oscillation of the body on the suspension are excited by the wheelset oscillation, and the resulting body motions in turn sustain the wheelset oscillation, thus closing the feedback loop. Determination of critical speeds in this case involves the analysis of a system with six degrees of freedom [10], Fig. 3, except when the wheelbase is very long in which case the system may be reduced to three degrees of freedom [22].

For infinite values of the suspension stiffnesses, instability involves motion at the kinematic frequency of a rigid framed vehicle. The system may be considered to have only two degrees of freedom and the stability has been investigated by Van Bommel [23] and Stassen [20]. It is clear from Fig. 7 that the concept of a rigid framed vehicle is completely unrealistic because the extreme values of stiffness required to even approximate the behavior of a rigid framed vehicle cannot be achieved in practical engineering structures. The effect of small suspension flexibility has been discussed in [24].

For a range of intermediate values of suspension stiffnesses, instability occurs at a frequency which is intermediate in value between that of a single axle and that of a rigid two-axle vehicle and the critical speed reaches a maximum. This maximum value can be well above present operating speeds, and this result suggests that a method for designing vehicles for stable running at high speeds is to select appropriately high suspension stiffnesses. Unfortunately, as Fig. 7 indicates, this is in conflict with the requirement for flange-free curving, and a satisfactory design can only be achieved for a low operating speed or a very low ruling track curvature.

Model experiments on a vehicle with stiff suspension were described in [10]; full-scale experiments on a special two-axle bogie are being carried out in conjunction with O.R.E. Committee B.52 of U.I.C.

Fig. 8 shows the effect of varying the longitudinal suspension stiffness only, for appropriately chosen values of the lateral suspension damping and stiffness. If the longitudinal suspension stiffness is zero the critical speed is very low because the wheel set is not restrained. As the longitudinal suspension stiffness is increased, the critical speed increases but is still comparatively low and is associated with the body instability. Provided that a criterion given in [25] is satisfied, for a range of values of longitudinal suspension stiffness, the body instabilities can be eliminated completely, and the operating speed is then limited only by the onset of the wheelset instability. The wheelset instability involves motions of the wheelset at the kinematic frequency with the body or frame moving steadily forward, and the critical speed can be accurately obtained from an analysis of a system with only two degrees of freedom [5, 8]. Very large values of longitudinal stiffness introduce a further form of body instability which is analyzed in [25], and which again occurs at rather low speeds. However, performance for large values of longitudinal suspension stiffness is limited by curving as indicated in Fig. 8.

This concept was first described in [26] and more complete analyses are given in [22, 25]. Detailed calculations show that it is possible to design high-speed vehicles meeting the suggested requirements for stability, curving, and response for use on existing railway tracks using this concept.

The validity of this concept has been demonstrated, with the variable-parameter research vehicle HSFV-1 shown in Fig. 4. This vehicle has been used in an extensive program of laboratory and track experiments on both stability [12], response, and curving [13].

The two design concepts just described should be considered as representative of a broad spectrum of more complex configurations. For example, in the case of a bogie vehicle as there is a secondary suspension in addition to the primary suspension, there is the possibility of body instabilities corresponding to both the bogie frame and the body, as well as the wheelset instability. As the wheelset mass is of the same order as the bogie frame mass, the flexible, damped suspension concept just described is not usually optimum and it seems that a bogie vehicle must have a relatively stiff lateral and longitudinal primary suspension (in accordance with much of current practice). The secondary suspension may be:

(*a*) Stiff in both lateral and rotational senses.
(*b*) Stiff in rotation and flexible and damped laterally.

For moderate speeds, it is no longer necessary that the rotational stiffness be high and then it is possible to design a freely pivoting bogie with consequent improvement to the curving performance. For very high speeds, the rotational stiffness must be high and this degrades the curving performance. It therefore appears that to achieve stable running at very high speeds and flange free curving on existing main-line track curvatures a two-axle vehicle with flexible, damped suspension is superior to a bogie vehicle.

However, such generalizations must be viewed with caution because conclusions on the optimum suspension design must depend, for example, on the range of effective conicities and track geometries adopted, and this where economic considerations exert a major influence.

Conclusion

Rapid progress in the conceptual or parametric design of high-speed railway vehicles is being made and a major reason for this is the adoption of a philosophy of design which is consistent with linear theory. Consideration of nonlinearities is obviously necessary in certain limiting conditions, but need not be allowed to dominate the design for normal high-speed running conditions.

A distinction may be made between existing conventional railway vehicles with stiff suspensions and nonlinear flange guidance and linear vehicles with flexible, damped suspension and linear, tread guidance. The latter class is represented by the experimental vehicle HSFV-1. The two classes of vehicle in reality represent extremes of a continuous spectrum and future developments will demonstrate the utility of the distinction.

Most of the foregoing work has been concerned with the design of vehicles to run, at very high speeds, on existing tracks. This is extremely attractive in economic terms, but for extremely high speeds over, say, 200 mph it may be worthwhile investigating new track forms. Such investigations can be carried out on the basis of the linear theory, and until system studies of this kind have been completed it is unwise to draw conclusions about the maximum speeds possible with railway vehicles. However, neither extrapolations based on existing practice nor facile studies based on oversimplified dynamics are likely to be reliable guides to possible developments in railway technology.

Acknowledgment

The author is grateful to S. F. Smith, Director of Research, British Railways, for permission to publish this paper.

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9 Wickens, A. H., "Recent Developments in the Lateral Dynamics of High Speed Railway Vehicles," *International Railway Congress Association Bulletin*, Vol. 44, No. 12, Dec 1967.

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11 Gilchrist, A. O., et al., "The Riding of Two Particular Designs of Four-Wheeled Railway Vehicle, "*Proceedings of the Institution of Mechanical Engineers, London*, Vol. 180, Part 3F, 1966, pp. 99–113.

12 Hobbs, A. E. W., "Experiments on the Lateral Stability of the Experimental Vehicle HSFV-1," B.R. Research Department Report DYN/53, 1967.

13 Boocock, D., "The Steady State Performance of Experimental Four-Wheeled Vehicle HSFV-1 on Curved Track," B.R. Research Department Report DYN/51, 1967.

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18 King, B. L., "An Evaluation of the Contact Conditions Between a Pair of Worn Wheels and Worn Rails in Straight Track," B.R. Research Department Report DYN/37, 1966.

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Vehicles With Special Reference to Bogie Design for the New Tokaido Line," *Proceedings of the Institution of Mechanical Engineers, London*, Vol. 180, Part 3F, 1966, pp. 58–66.

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23 Van Bommel, P., "Application de la Théorie des Vibrations Nonlinéares sur le Problème du Mouvement de Lacet d'un Véhicule de Chemin de Fer," Doctoral thesis, Delft Technological University, 1964.

24 Wickens, A. H., "Static and Dynamic Stability of a Two-Axle Railway Vehicle Having a Stiff Suspension," to be published.

25 Wickens, A. H., "Static and Dynamic Stability of a Two-Axle Railway Vehicle With a Flexible Suspension," to be published.

Printed in U. S. A.

Computer Analysis of Truck Performance

Brann, R. P., "Some Aspects Of The Hunting Of A Railway Axle," Journal of Sound and Vibration, Vol. 4, No. 1, 1966, Academic Press Incorporated, Berkeley Square House, Berkeley Square, London SW1, England, pp. 18-32.

Equations of motion are derived to describe the hunting mode of a railway axle running at constant velocity along straight track. It is assumed that the wheel and rail-head profiles take some arbitrary shape. This shape gives rise to nonlinearities in the equations. The equations are first linearized, and approximate expressions are derived for the frequency of the oscillation and conditions of stability. Asymptotic stability for all initial conditions of the nonlinear system is then considered in the manner of Aiserman, and the equations are examined for stable limit-cycle by applying the first approximations of Krylov and Bogoliubov. It is shown that, when running at low velocities, the axle will execute limit-cycle oscillations even though the wheel's flanges do not contact the rails. Small increases in velocity, however, quickly result in flange contact.

RRIS: 037211

TDOP: 05-001

British Railways Board, "Predict High Speed Performance Of The Four-Wheel Bogies Of The Linear Induction Motor Research Vehicle," FRA Contract CN-DOT-FR-30261, July 1972, Federal Railroad Administration, Department of Transportation, 400 7th Street, S.W., Washington, D.C. 20590.

The purpose of this contract is to predict analytically the high speed performance of the four-wheel bogies of the Linear Induction Motor Research Vehicle, guiding selection of design parameters and later supporting test data interpretations.

RRIS: 036618

TDOP: 05-002

Bussman, D. R., "Random Vibrations Of A Multi-Wheeled Vehicle," ASME, 67-VIBR-60, 345 East 47th Street, New York, N.Y. 10017.

The vehicle considered in this paper is a three-dimensional N-wheeled vehicle. The speed of the vehicle is assumed constant, and it is also assumed that the wheels maintain ground contact at all times. The suspension system is assumed to be a simple spring-viscous damper system connected in parallel. The spring and damper constants can be varied from wheel to wheel, or can be held constant. Power spectral densities are derived for the vertical acceleration of the mass center, and power spectral density is also determined for the vertical acceleration of any point not at the mass center of the vehicle. The input for the aforementioned equations is the power spectral density of the traversed terrain.

RRIS: 047425

TDOP: 05-003

Carman, R. W., "Railway Freight Car Suspension - The Effect Upon The Car Body Structure," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 58-63.

The effects of harmonic roll on the operating performance of a 100-ton aluminum gondola car for hauling coal are presented. An analog computer simulation of the gondola car operating on 39-foot jointed rail was made to study forces that develop in centerplates, side frames, wheels, and truck bolsters.

TDOP: 05-004

Clark, J. W. and E. H. Law, "Investigation of the Truck Hunting Instability Problem Of High-Speed Trains," ASME, 67-TRAN-17, 345 East 47th Street, New York, N.Y. 10017.

The problem of truck hunting instability and the design implications for high-speed trains were investigated. Linearized theoretical analyses were made to identify the most important design parameters for speeds of 125 to 200 mph. Truck wheelbase, truck mass, wheel coning ratio, and springs to oppose yaw and transverse motions of the truck relative to the carbody all have large effects on the critical speed for secondary hunting instability. It appears probable that lightweight passenger cars could be designed for stable operation at speeds considerably in excess of 200 mph.

TDOP: 05-005

Cooperrider, N. K., "The Hunting Behavior Of Conventional Railway Trucks," ASME Journal of Engineering for Industry, May 1972, American Society of Mechanical Engineers, 345 East 47th Street, New York, N.Y. 10017.

This article represents a study of truck hunting taking into account the non-linear effects such as the intervention of the flanges and the behavior of the adhesion. The author shows the equations covering the system in the case of two truck models as well as the system of external forces applied, based on the interactions between the rail and the wheel tread and the reactions at the flanges. The systems of equations were resolved by means of a computer. The author shows the conclusions reached concerning the hunting frequencies, riding stability, and the contacts between the flanges and the rail. These conclusions are different from those reached as a result of a study based on linear hypotheses. Reference is made in the bibliography to a number of basic studies of the subject.

RRIS: 046979

TDOP: 05-006

DePater, A. D., The Stability Of Running Of A Railway Vehicle, Report No. 306, April 1965, Technological University, Delft, The Netherlands.

This report is drawn from a series of lecture notes concerning the nature of vehicle-track interaction and deals with the behavior of a bogie running on tangent track. Some of the elements considered in the report are:

- Equations of motion for a railway vehicle
- Methods for solution of nonlinear problems of mechanics
- Description of symmetrical vehicle with two degrees of freedom
- Equations dealing with the stability of the vehicle
- Equations dealing with the behavior of a bogie equipped with cylindrical tyres
- Application of the Krylov-Bogoliubov method

TDOP: 05-007

Diboll, W. B., Jr. and H. S. Bieniecki, "Suspension Dynamics By Computer Simulation," ASME, 68-RR-4, 345 East 47th Street, New York, N.Y. 10017.

This paper reports on an analytical study of the effect of changing the design parameters of a two-mass, six-degree-of-freedom, suspension system. Rail cars with coil and air springs were analyzed by analog and digital computer. Spring stiffness, spring spacing, damping rates, height of center of gravity, and total mass were varied. The effects on frequency and response were determined.

TDOP: 05-008

Elmaraghy, W. H., M. A. Dokainish, and J. N. Siddall, "Minimax Optimization Of Railway Vehicle Suspensions," ASME, 74-WA/RT-3, 345 East 47th Street, New York, N.Y. 10017.

The minimax response of a complex dynamic system, such as a railroad vehicle, can be obtained by choosing certain (optimum) values of the stiffness and damping elements in the system. The railway vehicle is mathematically modeled as a linear, stable, strictly dissipative multidegree-of-freedom

dynamic system. The system is excited at more than one point by synchronous harmonic forces. A minimax principle reduces the problem to that of a finite dimensional optimal design problem. Nonlinear mathematical programming techniques are used to minimize the nonlinear objective function representing the maximum resonant response at a point of the system and, subjected to linear or nonlinear constraints, over a certain frequency range. The frequency range may be finite or infinite. Dynamic response of the system before and after optimization is shown, and three-dimensional plots for the constrained and unconstrained objective function versus the two most important design parameters are illustrated.

TDOP: 05-009

International Railway Congress Association, "The Transversal Stability When In Motion Of Railway Vehicles On The Straight," Rail International, November 1959, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 1058-73.

The theory of car rocking was investigated by means of an analog computer. Wheel-rail dynamics were first established for an isolated axle with differential equations. The theory was then expanded to a two-axle locomotive bogie with the axles having transverse play, and an ordinary bogie vehicle with sliding surfaces between the body and the bogies. Yet to be established at the time of writing was the relationship between the actual performance and the theoretical predictions.

RRIS: 037688

TDOP: 05-010

International Union of Railways, Problems Of Interaction Of Vehicles And Track - Measurements Made With Rail Vehicles During 1961 To Assess The Clearance Gauge Allowing For Rolling Movements Of The Vehicles, Interim Report No. 5, October 1962, Office for Research and Experiments, Utrecht, Netherlands.

The purpose of these measurements was to examine the behavior of bogies relative to their position on the track and to assess the lateral and rolling motions of the moving vehicles caused by forward traveling speed, curve radii, and superelevation in curved track. The values so determined were to be taken as a basis for setting up a kinematic vehicle gauge. One of the most important results of the experiments is the linear dependence which was confirmed for all vehicle types between the angles of rolling or between the transverse displacements of the sprung masses and the excess transverse accelerations, respectively. This seems to indicate that the magnitude of the quasistatic angles of rolling can be precalculated, provided the mathematical layout of the problem is sufficiently accurate. This would, however, entail complex considerations, because it would be necessary for all force effects of the connecting members between the individual masses to be incorporated in the process with reasonable accuracy.

RRIS: 040419
TDOP: 05-011

Keith-Hitchens, J. G., "Throwover Chart For Rolling Stock," Railway Gazette, Vol. 113, December 30, 1960, Temple Press Limited, 161-166 Fleet Street, London EC4, England, p 767.

A throwover chart, which is a simple and straightforward method of obtaining the throwover of a vehicle at its several points up to 100 ft over buffers and with a 4-ft bogie wheelbase on any curve, is described and illustrated.

RRIS: 037986
TDOP: 05-012

Koffman, J. L., "Bogie Pitching," Railway Gazette, Vol. 112, No. 15, April 8, 1960, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 418-421.

In this paper, mathematical studies dealing with the effects of bogie pitching on body oscillations are reported. The factors considered are:

- Spring stiffness
- Inertia
- Frame pitching
- Wheel spinning

RRIS: 039314

TDOP: 05-013

Koffman, J. L., "The Riding Properties Of Bogie Vehicles--3," Railway Gazette, Vol. 111, No. 20, December 25, 1959, 161-166 Fleet Street, London EC4, England, pp. 398-601.

This article states that vehicle development has led to the introduction of railcars and locomotives with unsymmetrical body-weight distributions resulting in the use of bogies of differing designs at each end. The author explores theoretical predictions of railcar and locomotive riding performance as an aid to the rationalization of bogie design.

RRIS: 037309

TDOP: 05-014

Koffman, J. L., "Some Aspects Of Bogie Pitching," Railway Gazette, Vol. 122, July 15, 1966, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 574-576.

This article discusses the use of dampers between the axlebox and bogie frame that would be equally effective in compression and expansion because pitching takes place symmetrically and regularly about the lateral axis. Proprietary hydraulic dampers are theoretically the most suitable devices to deal with these oscillations, provided the correct characteristic has been determined, produced, and maintained. Body and bogie primary suspension

oscillations with and without auxiliary friction dampers are shown for a passenger coach traveling at 70 mph. The beneficial action of the friction damper in reducing the magnitude of the fore and aft accelerations is apparent.

RRIS: 040113

TDOP: 05-015

Koyanagi, S., "The Motion Of A Guided Railway Car With Independently Rotating Wheels - Calculations Based On The Shin Kansen Car Model," Railway Technical Research Institute Quarterly Report, Vol. 14, No. 3, September 1973, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 186-171.

The speed limit of railway vehicles is generally held to be about 350 km/h, owing to the instability caused by hunting motion. But, if an independently rotating wheel track guided by a guide rail (IRWTG) is adopted, a railway vehicle applicable beyond that speed limit is realizable. This paper describes the principle of an IRWTG and the results of numerical examinations.

RRIS: 050876

TDOP: 05-016

Labuschagne, T. J. and H. Scheffel, "Some Aspects Of The Interaction Between Railway Vehicle And Track," Transactions of the South African Institution of Civil Engineers, Vol. 11, October 1969, Johannesburg, South Africa.

This paper briefly describes some aspects of an experimental research project on the South African Railways into the conditions affecting the riding quality of railway vehicles. Parameters that influence vertical riding quality are given in connection with the theory of forced vibrations. Results recorded

during riding quality tests were found to be in good agreement with the theory. A method for simulating vertical oscillations of railway vehicles on a digital computer is outlined.

TDOP: 05-017

Law, E. H., "Nonlinear Wheelset Dynamic Response To Random Lateral Rail Irregularities," ASME, 73-WA/RT-3, 345 East 47th Street, New York, N.Y. 10017.

Nonlinear equations of motion for a rail vehicle wheelset having profiled wheels and wheel flange contact with flexible rails are presented in this paper. The effects of spin creep and gyroscopic terms are included. The rails are considered to have random lateral irregularities which are described by prescribed power spectra. The equations of motion are integrated numerically, and the effects on the dynamic response of quantities such as speed, track roughness, wheel wear, flange clearance, and lateral stiffness of the rails are investigated.

TDOP: 05-018

Law, E. H. and R. S. Brand, "Analysis Of The Nonlinear Dynamics Of A Railway Vehicle Wheelset," ASME, 73-Aut-F, 345 East 47th Street, New York, N.Y. 10017.

The nonlinear equations of motion for a railway vehicle wheelset having curved wheel profiles and wheel-flange/rail contact are presented. The dependence of axle roll and vertical displacement on lateral displacement and yaw is formulated by two holonomic constraint equations. The method of Krylov-Bogoliubov is used to derive expressions for the amplitudes of stationary oscillations. Conditions are then derived for the stability characteristics of the stationary oscillations. The expressions for the amplitude and the stability conditions are shown to have a simple geometrical interpretation which facilitates the evaluation of the effects of design parameters

on the motion. It is shown that flange clearance and the nonlinear variation of axle roll with lateral displacement significantly influences the motion of the wheelset. Stationary oscillations may occur at forward speeds both below and above the critical speed at which a linear analysis predicts the onset of instability.

RRIS: 048239

TDOP: 05-019

Liepins, A. A., "Digital Computer Simulation Of Railroad Freight Car Rocking," ASME, 68-RR-3, 345 East 47th Street, New York, N.Y. 10017.

A mathematical model for the simulation of railroad freight car rocking is presented. The equations of the model are developed into a digital computer program. The model response is validated by two series of tests involving a 100-ton covered hopper car on ASF Ride-Control trucks. The model is considered reliable for engineering predictions.

RRIS: 047428

TDOP: 05-020

Mano, K., S. Arai, and K. Yokose, "Stability Region Of The Nonlinear Hunting Vibration Of Railway Vehicle Truck," Railway Technical Research Institute Quarterly Report, Vol. 9, No. 2, June 1968, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 109-110.

Applying the describing function method to the estimation of nonlinear hunting vibration of a railway truck, the stability region of that truck can be estimated easily by the computer. The results obtained by this method coincide well with those obtained by an analytical method.

RRIS: 033171

TDOP: 05-021

Manos, W. P., "The Dynamics Of Rail Car Suspension Systems," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 28-31.

In this presentation, equations of motion are derived for the harmonic roll phenomenon for railroad freight cars. The resulting mathematical models are modified to determine the effects of snubbing, spring travel, length of rail, etc. These parameters are evaluated with respect to car performance and other car characteristics.

TDOP: 05-022

Manos, W. P. and J. C. Shang, "Dynamic Analysis Of Rolling Freight Cars," ASME, 65-WA/RR-8, 345 East 47th Street, New York, N.Y. 10017.

The dynamic equations of motion describing car rocking are presented. These equations are then simplified and used to determine carbody roll frequency and its dependence on roll amplitude. As a result of the analysis, a graphical technique was developed which shows the rail disturbance as a function of truck center distance for any desired rail length, wheel base, track profile, and rail stagger. In addition, a brief account of the experimental history is presented.

RRIS: 047422

TDOP: 05-023

Matsudaira, T., S. Arai, and K. Yokose, "Combined Effect Of Frictional And Elastic Moments Against Truck Turning Upon Hunting Of Truck," Railway Technical Research Institute Quarterly Report, Vol. 7, No. 3, 1966, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan.

The truck hunting is analyzed theoretically, and a numerical calculation for an actual example is made to provide data on truck design. By a numerical example, the combined effects of friction and rigidity against truck turning upon the hunting speed was investigated and was checked by

analog computation. The spring action against truck turning plays the leading role in increasing the critical hunting speed. The frictional resistance plays an important supporting role in making the spring action effective over the range of a large amplitude, though the friction itself does not actively prevent truck hunting.

RRIS: 040126

TDOP: 05-024

May, J. T., "Metroliner Dynamic Behavior Investigation," Paper No. C 72 928-9-IA, Institute of Electrical and Electronics Engineers, Inc., 345 East 47th Street, New York, N.Y. 10017.

The Metroliner Cars used in regular rail passenger service on the Northeast Corridor are the subject of a program sponsored by the Federal Railroad Administration, U.S. Department of Transportation, to improve the ride quality performance of the cars. The technique of mathematical modeling of the car was utilized in this program to predict and postulate methods of obtaining improved performance. The model was generated, verified, and then various parameters were altered to obtain optimum vehicle response. The altered parameters were then used to specify various car component modifications and to generate a specification for a new truck.

TDOP: 05-025

Meacham, H. C. and D. R. Ahlbeck, "A Computer Study Of Dynamic Loads Caused By Vehicle-Track Interaction," ASME, 69-RR-1, 345 East 47th Street, New York, N.Y. 10017.

Computer analyses of vehicles and track are producing enlightening results regarding actual dynamic loads and the manner in which various parameters of vehicle and track structure affect these loads. Using this information, it is possible to more intelligently decide how to alleviate the high wheel-rail stresses caused by today's unique traffic and track conditions. Possible solutions range from better track maintenance to different wheel-rail geometries to changes in stiffness and damping of trucks and in the track structure itself.

TDOP: 05-026

Melpar Incorporated, "Dynamic Railcar Simulation Program," Contract CN-DOT-C-111-66, Falls Church, Virginia.

A generalized digital simulation has been programmed in the basic FORTRAN language for calculating the motions and forces during operation of a multi-membered railcar. The railcar is driven at selected speeds along a pair of rails represented by recorded numerical measurements. All massive components of the railcar are treated as general mechanical members with six degrees of freedom, coupled to each other by an arbitrary set of linear elements or a programmed set of nonlinear functions having given spring rates, damping constants, etc. The model includes simulation of truck hunting phenomena with cylindrical or tapered wheel treads and simulation of the compliance properties of the rail roadbed.

RRIS: 039207

TDOP: 05-027

Newland, D. E., "Steering Characteristics Of Bogies," Railway Gazette, Vol. 124, October 4, 1968, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 745-750.

This article, which is a description and analysis of the fundamental steering properties of four-wheel bogies, considers the limitations of bogie vehicles for high-speed operation and compares their performance with estimates for the APT, a 250 km/h advanced passenger train. Because bogies act to attenuate the effect of road bed irregularities, there are strong reasons to favor bogie vehicles whenever passenger comfort is a major consideration. A possible disadvantage is that bogies designed to be dynamically stable at high speeds may need a primary suspension that is too stiff for good steering properties. It is concluded from mathematical studies that a dynamically optimized bogie still provides satisfactory guidance, comparable with a four wheel vehicle with a very soft suspension, and bogies should not, therefore, be regarded as unsuitable for high-speed operations.

RRIS: 037464

TDOP: 05-028

Norfolk and Western Railways, Methods Of Loading Single Trailer On A Long Flat Car, November 11, 1968, 8 North Jefferson Street, Roanoke, Virginia 24017.

Computations of the stability of a TTX 89-ft 4-in. flat car with one trailer situated on the A end or rear hitch ascertained that in this situation the car is unstable. Accordingly, the recommended loading position of a single trailer is shown to be on the B end, while the loading of a single trailer on the A end is shown as the prohibited method.

RRIS: 033295

TDOP: 05-029

Siddall, J. N., M. A. Dokainish, and W. Elmaraghy, "On The Effect Of Track Irregularities On The Dynamic Response Of Railway Vehicles," ASME, 73-WA/RA-1, 345 East 47th Street, New York, N.Y. 10017.

The steady state response for models of a six-axle locomotive running on a sinusoidally irregular track has been investigated. Two mathematical models have been set up, a full model for the "stationary" vehicle in which creep between wheels and rails was neglected, and a full model for the "moving" vehicle in which creep forces, gravity stiffness effects, and wheel tread profiles were considered. The use of the generalized method of complex algebra to obtain the steady state response of the railway vehicle components to varying input frequencies was used.

The results given in this paper are for the case of sinusoidal lateral track irregularities only, but the method is general enough to allow also for vertical track irregularities. For the "stationary" vehicle, the input frequency is increased from zero to 3 cycles per second. For the "moving" vehicle, the input frequency is a function of the track wave length and the vehicle forward speed, and is given in terms of the vehicle speed. The frequency response curves are computer plotted in each case. For the "moving" vehicle, responses for the cases of both new and worn wheels are obtained. The natural frequencies for the full model are also calculated.

The results obtained show the effect of the creep forces and the condition of the wheels on the steady state response. It is recommended that slip and corresponding creep forces, wheel tread and rail profiles, and the gravity stiffness effect be included in the steady state response analysis of railway vehicles to track irregularities.

The analysis given may be used to check the response of any proposed design for a railway vehicle to economically attractive track irregularities. It may also be used to adjust geometry, spring rates, and damping characteristics in order to maximize operating speeds while providing optimum damping for the truck and body motions.

This paper illustrates and describes the mathematical models used and gives the generalized form for the differential equations of motion and the methods of solution. The equations of motion for the wheelsets, including the creep forces and the wheel tread profiles, are derived in detail.

TDOP: 05-030

Ullman, K. B., "Developing A Performance Specification To Improve Ride Quality," Proceedings of Conference on Track/Train Dynamics Interaction, December 1971, Association of American Railroads, Research and Test Department, Chicago, Illinois, pp. 330-339.

This report discusses the Metroliner Ride Improvement Program that was undertaken by a joint Department of Transportation/Rail and Supply Industry/ Railroad Task Force. The program used road, lab, and component testing to arrive at a dynamic model which was then used to develop a functional truck design specification.

TDOP: 05-031

Uschkalov, V. F., "An Investigation Of The Vibration Of A Railway Car Moving Along Tracks Deformed By Random Irregularities," (translation), Vestnik Vniigt, No. 3, 1975, Moscow, USSR, pp. 20-23.

This article examines the stationary condition of forced vibration of a railroad car moving at a constant speed along irregular track. The object of research is the standard quadraxial gondola on a biaxial truck. As a computational model, the gondola's elastic body is presented with concentration of the mass on the nodes. The vibration of the gondola is examined on a vertical plane. The bending of the frame elements when there is transverse vibration is ignored.

TDOP: 05-032

Westinghouse Research Laboratories, "Engineering Design Study Of Active Ride Stabilizer For The Department Of Transportation's High-Speed Test Cars," by W. O. Osbon and T. H. Putnam, FT-Contract, CN-DOT-3-0267, June 1969, Pittsburgh, Pennsylvania.

This report describes an engineering study concerning the application of an active suspension to one of the U. S. Department of Transportation's high-speed test cars. The objective of the study was to establish quantitatively the ride improvement that could be expected from the stabilizer as well as to determine power requirements, vehicle modifications, and the basic equipment design parameters. Quantitative assessment of expected ride improvement was carried out through computer simulation of the vehicle with the stabilization equipment in conjunction with simulated subgrade disturbances. These results are discussed in detail with regard to computer records for the stabilized and unstabilized vehicle. To equip a test car with the proposed active stabilization system involves modifications to the car suspension. These modifications are listed, and detailed descriptions are given.

RRIS: 039199

TDOP: 05-033

Wickens, A. H., "The Dynamic Stability Of Railway Vehicle Wheelsets And Bogies Having Profiled Wheels," International Journal of Solids and Structures, Vol. 1, 1965, Pergamon Press Incorporated, Maxwell House, Fairview Park, Elmsford, N.Y. 10523, pp. 319-341.

The dynamic instability of railway vehicle bogies and wheelsets is caused by the combined action of the conicity of the wheels and the creep forces acting between the wheels and rails. The instability is investigated for the case where the wheels are profiled rather than purely conical. Equations of motion are formulated, and stability criteria obtained that indicate the effect of varying the parameters of the system. The nature of the motion at the critical speed is investigated, and the mode of energy conversion between the forward motion of the vehicle and the lateral motion of the bogie or wheelset is explained.

RRIS: 040123

TDOP: 05-034

Wickens, A. H., "General Aspects Of The Lateral Dynamics Of Railway Vehicles," ASME, 68-WA/RR-3, 345 East 47th Street, New York, N.Y. 10017.

Recent developments in research into the lateral dynamics of railway vehicles are reviewed in this article. Particular reference is made to dynamic stability, dynamic response, and curving. Two engineering aspects of this work are stressed--namely, the advantages of applying linear theory to a comprehensive study of vehicle dynamics, and the choice of basic assumptions required to make such studies realistic for design purposes.

RRIS: 047432

TDOP: 05-035

Wickens, A. H., "Recent Developments In The Lateral Dynamics Of High Speed Railway Vehicles," Bulletin of the International Railway Congress Association, (paper presented at the "High Speeds" Symposium, Vienna, 1968).

Recent developments in the theoretical and experimental aspects of the lateral dynamics of railway vehicles are reviewed in this paper, with particular reference to the problems of dynamic stability, guidance, and response to track features.

The influence of track and wheel geometry, suspension characteristics, and overall vehicle configuration on dynamic behavior is discussed in relation to linear theory. The implications of the linear theory for vehicle design are discussed both in relation to improved forms of existing railways and various possible new forms of guided, wheeled systems.

TDOP: 05-036

Yokose, K., "Combined Effect Of Frictional And Elastic Moment Against Truck Turning On Running Stability Of A Truck With Elastically Suspended Axles," JSME Bulletin, Vol. 13, No. 61, July 1970, Japan Society of Mechanical Engineers, 1-24 Akasaka, 4-Chome, Minato-ku, Tokyo 107, Japan, pp. 888-896.

The Frank-Koenig method is used to search for a criterion of stability for polynomials with complex coefficients. It is made clear that the turning stiffness of the truck, the friction of the side bearers, the supporting stiffness of the axles, the slope of wheel treads, and the creep coefficients between wheel and rail affect the truck hunting. According to these results, the optimum value of the longitudinal supporting stiffness of the axle and the optimum value of the lateral stiffness which gives the maximum critical hunting speed are connected to the magnitude of the turning amplitude of the hunting truck.

RRIS: 046924

TDOP: 05-037

Yokose, K., "Combined Effect Of Frictional And Elastic Moment Against Truck Turning On Running Stability Of Elastically Suspended Axles," Railway Technical Research Institute Quarterly Report, Vol. 9, No. 4, December 1968, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 220-223.

In this paper, the author analyzes truck hunting where springs and friction are connected in series against truck turning. A describing function method is applied to the analysis of the truck hunting, and the Frank-Koenig method is used to determine the criterion of stability for polynomials with complex numbers. It is made clear that the turning stiffness of the truck, the friction of the side bearer, the supporting stiffness of the axle, the slope of the wheel tread, and the creep coefficient between wheel and rail affect the truck hunting.

RRIS: 033187

TDOP: 05-038

Yokose, K., "Effect Of Non-Linear Characteristic On Hunting Of Car - Effect Of Side Play Between Axle And Bearing Metal," Railway Technical Research Institute, Vol. 13, No. 2, June 1972, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 107-111.

This paper describes an equation of motion for a two-axle car covering side play between the axle and the bearing metal. An observation on the value of side play is also made. Nonlinearity of the elasticity and the side play was treated by the describing function method. Then the characteristic equation of the two-axle car was derived, and the sinusoidal hunting velocity was calculated. A one-fifth sized model of a two-axle car was used in order to experimentally verify the theoretical treatment of hunting. The experimental results and the theoretical calculation coincided qualitatively. Moreover,

when the damping of the supporting device for the carbody was selected properly, the above results also coincided quantitatively. It was proved that the side play not only effectively prevented the body hunting, but also lessened the side thrust experimentally.

RRIS: 041123

TDOP: 05-039

Yokose, K., "On Hunting Of High Speed Railway Vehicles," Japan Society of Mechanical Engineers Bulletin, Vol. 14, No. 73, July 1971, Japan Society of Mechanical Engineers, 1-24 Akasaka, 4 Chome, Minato-ku, Tokyo 107, Japan, p 713.

To clarify the hunting phenomenon of high speed railway vehicles, general equations of motion were derived for bogies on the new Tokaido Line, and the roots of characteristic equations were obtained by digital computer. The supporting stiffness of the axle with respect to the truck frame was found to be capable of eliminating the body hunting when a proper supporting stiffness is chosen. When the creep coefficient decreases, body hunting in the yaw mode occurs. The analytical results closely agreed qualitatively with the experimental results obtained using a one-fifth model car at the critical speed for truck hunting.

RRIS: 037416

TDOP: 05-040

Young, J. W. and D. N. Wormley, "Optimization Of Linear Vehicle Suspensions Subjected To Simultaneous Guideway And External Force Disturbances," ASME, 73-Aut-O, 345 East 47th Street, New York, N.Y. 10017.

The optimum linear suspension for a vehicle traveling over a guideway with random irregularities and subjected to random external force disturbances is derived. The optimization is based upon a performance index which

weighs mean square vehicle acceleration (a measure of passenger comfort) and mean square relative displacement between the vehicle and guideway (a measure of required suspension dynamic excursion). The form of the optimum suspension is not restricted to a specific mechanical, magnetic, or fluid suspension configuration. Performance curves and numerical design examples are presented to illustrate the influence of both guideway and external force disturbances upon optimum suspension performance capability.

RRIS: 056841

TDOP: 05-041

Field Analysis of Truck Performance

American Steel Foundries, The ASF Six-Wheel Ride Control Truck Under The "Whopper Hopper," Publication JHL/T-103-5-65, Chicago, Illinois.

American Steel Foundries' (ASF's) six-wheel ride control truck was tested under an Atlantic Coast Line large hopper car of 135-ton capacity. The article discusses the performance of the truck, which features articulated side frames. The tests in question were conducted during December of 1964 with a view toward:

- Determining that the six-wheel truck would operate safely in interchange service
- Comparing the six-wheel truck's ride quality to that of a four-wheel, 2-1/2-in. travel, 90- to 100-ton truck
- Checking the ASF design parameters

Thirty-eight channels were used to record information. Data reported included:

- Bolster swivel
- Roll frequency
- Side frame motion
- Vertical and lateral bolster movement

TDOP: 05-042

American Steel Foundries, Car Rocking Tests Of Standard Truck Arrangement, Test Engineering Department, Product Engineering Division, Chicago, Illinois.

This report concerns a series of tests undertaken by American Steel Foundries to:

- Directly compare the roll performance of various truck arrangements and modifications currently available to the railroad industry
- Determine which modifications were most effective in reducing car rocking
- Acquire basic data concerning the effect of car rocking on truck components

TDOP: 05-043

American Steel Foundries, Road Testing Freight Car Trucks, 1943, Chicago, Illinois.

A descriptive publication on the 1940-1942 equipment and instrumentation used in service to obtain truck performance data under various conditions of loading. The instrumentation included accelerometers, three-way recorders, lateral recorders, spring deflection recorders, and indicating and recording speedometers. The publication includes a variety of photographs illustrating test trains and procedures.

TDOP: 05-044

Association of American Railroads, C&O - B&O Ride Control Tests For OHSGT, by D. E. Monselle, June 10, 11, 12, 1969, AAR Research Center, 3140 South Federal Street, Chicago, Illinois 60616.

This report concerns a test program prepared and performed by the C&O - B&O Railroad under contract to the Office of High Speed Ground Transportation (OHSGT), U. S. Department of Transportation (DOT). The purpose of the program was to develop basic data on the vibration environment within present day rail cars created by differences in track geometry and truck suspension systems. The collected data was to be used to:

- Develop performance specifications for high-speed freight car suspension systems
- Develop needed information covering the effect of track inputs on the vibration environment in rail cars
- Aid in the development of accurate rail car dynamics computer simulation programs
- Aid in the development and check-out of track measurement instrumentation

This document represents a trip report by the author covering three of the four days of testing conducted on the sixth and final week of the program. Descriptions of the test consist, instrumentation, facilities, and trucks tested are included.

TDOP: 05-045

Association of American Railroads, "Effect Of Spring Travel, Height Of Center Of Gravity, And Speed On Freight Car Clearance Requirements On Curved And Tangent Track," Report of the Joint Committee on Relation Between Track and Equipment of the Engineering and Mechanical Divisions, Association of American Railroads, In Collaboration with AREA Committee 28 - Clearances, AREA Bulletin, Vol. 5A, 1958 Proceedings, American Railway Engineering Association, 59 East Van Buren Street, Chicago, Illinois 60605, pp. 305-361.

The American Railway Engineering Association proceedings for 1958 includes information on clearance requirements on curved and tangent track to accommodate car movements resulting from then new designs of trucks. Both static and dynamic tests are reported. The interrelationship between truck changes and resultant effects on carbody motion are of special interest.

The tests were made with two fully loaded 70-ton cars with nominal center of gravity heights of 70, 85, and 100 in. statically, and under a range of speeds from 5 to 65 mph over tangent and curved track. Records were taken on both cars of body roll, vertical and lateral accelerations, and speed.

RRIS: 052380

TDOP: 05-046

Association of American Railroads, First Progress Report Lateral Action Of Common Designs Of Freight Car Trucks Having Axles With And Without End Collars, Report No. F 4000, December 28, 1950, AAR Research Center, 3140 South Federal Street, Chicago, Illinois 60616.

The objective of certain instrumentation in these tests was to obtain fundamental information on the lateral action of different common designs of freight car trucks having standard and modified parts in the journal box

assemblies, and having axles with and without end collars. The tests were proposed to determine:

- The motions of truck parts and the relation between these motions and lateral carbody accelerations
- The origins of lateral disturbances to the carbody
- The relative importance of these origins
- The manner of transmission of disturbances from the rail through the truck to the carbody

A mechanical recorder was attached to one truck to obtain records of the relative lateral movements of truck parts (axle, bearing, wedge, frame) at both ends of one axle, and records of the vertical and lateral displacements of both side frames relative to the bolster. The conclusions are as follows:

- The most important origin of lateral disturbance to the carbody is the nosing of the wheel-axle assemblies and truck, due to coning of the wheel treads.
- The effects of axle end collars on truck action are to increase the activity of truck parts, but to limit the forces transmitted through the truck.
- Increase of lateral clearance, which does not reduce the beneficial effects of axle end collars, should improve the lateral riding quality of conventional trucks with snubbers.
- Curved track produces a quieting influence on truck action and improves lateral riding quality.
- Definite improvement in lateral riding quality should result from the use of a device which would cause the carbody to seek a centered position between lateral clearances.

RRIS: 033736

TDOP: 05-047

Association of American Railroads, Investigation Of Lateral Forces Developed By Diesel Locomotives On The Spiral Of A Two Degree Curve On The CB&Q Railroad, Publication No. 55451, December 9, 1969, AAR Research Center, 3140 South Federal Street, Chicago, Illinois 60616.

On February 18, 1969, the CB&Q Railroad experienced a derailment of a freight train at Crete, Nebraska. Three large tank cars loaded with anhydrous ammonia were standing on a parallel siding, and one car was fractured resulting in eight deaths. Indications were that the high rail was turned over on the spiral approaching the curve by one of the G E Class U28C diesel locomotives pulling the train. Accordingly, the railroad requested an investigation by the AAR to measure lateral forces on a spiral under conditions similar to those existing at Crete. The ensuing tests showed no lateral forces applied to the outer rail nearly large enough to cause rail overturning. It could only be concluded that some abnormal condition of the track equipment or train action existed at the time of the accident at Crete.

TDOP: 05-048

Association of American Railroads, Lateral Action Of Freight Car Trucks, Report No. f3800, June 1951, AAR Research Center, 3140 South Federal Street, Chicago, Illinois 60616.

This work, although old, represents excellent reference material. Three types of trucks were tested:

- The ASF A-3 Ride Control Truck
- The AAR Selfaligning Spring Plankless Truck
- The AAR Spring Plank Truck

Among the conclusions were the following:

- The most important origin of lateral disturbance to the carbody is the nosing of the wheel-axle assemblies and truck due to coning of the wheel treads.

- The effects of axle end collars on truck action are to increase the number of small lateral shocks to the carbody but to decrease the magnitude of potentially larger shocks.
- Increase of lateral clearance, which does not reduce the beneficial effects of axle end collars, should improve the lateral riding quality of conventional trucks with snubbers.
- The spring plank has a beneficial effect on riding quality at low train speeds, but a detrimental effect at high train speeds.
- Curved track reduces lateral activity of truck parts and improves lateral riding quality.
- Definite improvement in lateral riding quality should result from the use of a device which would cause the carbody to seek a centered position between lateral clearances.

RRIS: 040244

TDOP: 05-049

Association of American Railroads, Rocking Car Derailment Testing During 1966, by D. E. Monselle, AAR Progress Report, June 1967, 3140 South Federal Street, Chicago, Illinois 60616.

This report summarizes the results of a test program conducted on the Louisville and Nashville Railroad during the year 1966. The program was conducted to further investigate the rocking and derailment tendencies of the new, large 100-ton hopper cars operating on staggered jointed track. The purpose of the testing was to study the influence of track condition, train operation, and truck characteristics on the rocking and derailment tendencies. Track conditions investigated included extreme curvature, unbalanced elevation, irregularity in alignment, low joints, and rail joint stagger. Train operating conditions included slack run-in and run-out and the different effects of train pull and buff operation. Truck characteristics

considered included increased spring travel, increased gib clearance, supplementary snubbing, centerplate friction, centerplate rock, and lateral restraint of the truck bolsters. Some of the special devices tested were a roller bearing centerplate, a cylindrical centerplate with elastic side bearings, a carbody-to-truck side frame, a double-acting hydraulic damper, and a modified truck with truck bolster to spring plank lateral restraint.

Excessive out-of-cross level and extreme superelevation were the most influential track variables tested. None of the train operation variables made any significant contribution to the harmonic roll phenomenon.

In terms of truck variables, the double-acting hydraulic dampers and increased lateral restraint on the truck bolsters both proved to be effective roll-mitigating measures. Results for many of the other variables were inconclusive. Increased centerplate friction seemed to encourage derailment tendencies. The relative rolling motion between carbody and trucks as allowed by centerplate rock proved to be an important factor in wheel lift and derailment.

TDOP: 05-050

Association of American Railroads, Studies To Eliminate Harmonic Roll Of Heavy Cars, Progress Report AAR-MR-440, April 1965, 3140 South Federal Street, Chicago, Illinois 60616.

This document combines a theoretical discussion of harmonic roll with a report on then-current field testing activities with heavy cars. It is reported that means have been studied to eliminate harmonic roll, and that these studies indicate that by using lower spring rates, extending the center of the spring nests, and applying swivel centerplates and preloaded laminated side bearings, the condition can be attenuated.

Experimental truck arrangements are under development that will allow the truck to follow the irregularities of track, and will prevent the initiation of the forcing frequency into the body of the car at such a phasing as to cause harmonics in the car rolling. Also, snubbing and energy absorption devices have been found that, when applied in conjunction with other modifications, lessen wheel lift.

RRIS: 040352

TDOP: 05-051

Association of American Railroads, Tests Of Trucks And Truck Springs Designed To Produce Easier Riding And Reduce Harmonic Action, AAR Progress Report, March 3, 1934, 59 East Van Buren Street, Chicago, Illinois 60616.

The tests made in this investigation showed that freight cars equipped with 1933 Standard A.R.A. or old Standard A.R.A. coil springs developed vertical oscillation at critical speeds. This oscillation was more pronounced at half load than at full load. It was determined that this bouncing could be reduced or practically eliminated by the use of various stabilizing devices in conventional trucks, or by special trucks, when trued wheels were used. It was thought that the use of these devices would effect savings by the reduction of lading and equipment damage in cars that operated at speeds above 40 miles per hour.

RRIS: 040246

TDOP: 05-052

Baguley, R. W., "Vehicle Testing For Higher Speeds," Railway Engineering Journal, Vol. 2, No. 3, May 1973, Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England, pp. 19-31.

This paper reviews the work undertaken by the Testing Performance Section at the Derby Technical Center in Great Britain, with particular reference to higher speeds. Attention is focused on riding, interaction between vehicle and track, stressing, braking, and aerodynamic effects.

RRIS: 047894

TDOP: 05-053

Bertram, L. W., "Car Roll And Wheel Lift Tests On Simulated Service Track: Test Conditions And Results At PRR Altoona Test Site," Technical Proceedings - 1966 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, N.Y. 14043, pp. 23-26.

This presentation deals with harmonic roll tests conducted with 100-ton hopper cars by the Pennsylvania Railroad Company. During the testing over a specially modified track section, it was ascertained that the Stucki hydraulic snubber, installed between the top of the truck bolster and the side frames, did eliminate wheel lift and reduce roll tendency. Further work with this device is proposed.

TDOP: 05-054

Brodeur, R. H., "A Program For Evaluating The Ride Quality Of The Present Standard AAR Freight Car Truck Under An 89 Ft. Flat Car," 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. 357-381.

The basic 70-ton, 3-11/16-in. spring travel truck was studied by Trailer Train Company by operating long container- and trailer-type cars over a high-speed, mainline track. This study was undertaken because of maintenance problems that developed in truck centerplates, adapters, wheels, container support pedestals, and trailer hitch components.

Eighty test runs had been completed as of this report. No analysis had yet been accomplished. The report includes photographs of the data acquisition system and of some data samples.

TDOP: 05-055

Caldwell, W. N., "Ride And Tracking Vibration Problems Of TOFC-COFC Cars In High-Speed Service," 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. 154-200.

In this presentation, the author reports on a large-scale research program undertaken by Canadian National Railways with regard to Trailer On Flat Car-Container On Flat Car (TOFC-COFC) cars. The presentation begins by defining the two vibrational problem areas in COFC service:

- Vertical ride of long (85-ft) COFC cars carrying up to four, 20-ft containers
- Lateral ride of COFC cars equipped with the standard three-piece frame, roller bearing truck

Among the variables tested with regard to the lateral stability problem were:

- Low conicity wheels (1/40 taper). These will lead to stability up to 75 mph, but will also lead to increased maintenance costs since their service life appears limited to 100,000 miles.
- Lateral clearance roller bearings. These controlled hunting up to 92 mph (highest speed tested), but after a year's service developed premature failure of grease seals in some cases.
- Tie rods to increase track frame rigidity. Some ride improvement was noted with these devices, but performance was not comparable to a COFC car operating with plain journal bearings.

- Constant contact side bearings. Constant contact side bearings displayed the following operating disadvantages:
  - They require a higher degree of dimensional accuracy in adjusting the side bearing preload.
  - They degrade the low-speed tracking capability of the trucks because of a tendency to grab on curved track.
- Bearing rocker suspension device. A mechanical device was developed by CN's Operating Department. The device connected the roller bearings to the side frames through a rocker arrangement. This device resulted in stable running to 75 mph, but failed mechanically. A revised version was still under study as of this writing.

TDOP: 05-056

Canadian National Railways, Derailment Of 100 Ton Hopper Car CN 373041, by J. F. Scott, Report No. 24/T-207, 1968, Technical Research Centre, St. Laurent, Quebec.

This memorandum describes the mechanism by which a 100-ton covered hopper car derailed and overturned during harmonic-roll testing on curved track in the Montreal Yard. The investigation ascertained that the leading inner wheel on the trailing truck lifted clear of the guard rail on a curve approach and caused the derailment.

TDOP: 05-057

Canadian National Railways, Roll Motion Of 100-Ton Hopper Cars, by J. F. Scott, J. Johnson, and A. Charenko, September 1969, Montreal, Quebec.

This report comprises a summary of Canadian National Railway's harmonic roll tests with 100-ton covered hopper cars. From 1967 to 1969, approximately 60 test series were run with five different hopper cars on two controlled test tracks. These tests evaluated the various car suspensions available.

On the basis of test results, suspension systems were grouped in performance categories. Only four tested suspension systems controlled wheel lift on the CN curved track. These four suspensions have been placed in Performance Category A. Devices placed in Category B controlled wheel lift on tangent but not on curved track. Category C was comprised of devices which improved ride without preventing wheel lift on tangent track. Devices in Category D did relatively little to control roll oscillation.

At the time of this writing, CN felt it was in a position to confidently select suitable suspension systems for new 100-ton car orders. However, since car weights and operating speeds continue to increase, the report stressed that continuing research is needed.

The four tested suspension systems rated in Category A were:

- Four Model 1100 MDA hydraulic dampers with D-5 main springs and single side springs
- Four Model 1100 MDA hydraulic dampers with D-3 main springs and single side springs
- Unity No-Sway Side Bearings with D-5 main springs and double side coils
- Eight AR-15 Armstrong Beverley hydraulic dampers on hard setting with D-5 main springs and single side springs (the Armstrong dampers are unsuitable for practical application)

TDOP: 05-058

Canadian National Railways, Study Of The Vibration At Low Speeds Of 100-Ton Hopper Cars, July 1967, Technical Research Centre, Montreal 3, Quebec.

This report concerns a series of road tests of a 100-ton hopper car NAHX loaded with potash. The tests were undertaken by CN in their Turbot Yard, Montreal. The object of the test program was to evaluate the effectiveness of various modifications to the track and to the car in controlling the build-up of roll motion.

Throughout the program, two test sections were used. The first was kept at a 3/4-in. cross level elevation as a basis for comparison, the second was altered to investigate the influence of track at the source of vibration excitation.

Studies of track influence found that track surface certainly constitutes a valid influence on harmonic roll, but track is difficult to maintain within desirable cross-level variation without a great deal of expense. However, consideration should be given to increasing the amount of maintenance at track locations likely to give trouble, particularly sidings. This will reduce the likelihood of a roll resonance condition. The possibility of changing to a longer distance between joints can be a help, but potential application is limited.

Studies of the influence of suspension modifications involved a total of 10 modifications. Springs with 3-11/16-in. travel with double springs under the friction wedges and the No Sway Side Bearing Stabilizer (Unity Railway Supply Company) appeared to be the two modifications having the greatest attenuating effects on wheel lift.

TDOP: 05-059

Canadian National Railways, Vibration Of 100-Ton Hopper Cars Analysis Of Results, by K. Henderson, A. Charenko, and J. Johnson, October 1967, Montreal 3, Quebec.

This report is the second to be issued concerning the vibration at low speeds of 100-ton hopper cars. The first report provided background information, introduced the CN program of test work, and summarized observations and experimental results. This report constitutes an analysis of results.

Graphs illustrating the distribution of stabilizing and overturning moments have been prepared for all significant modifications tried and for various track profiles. The comparisons were made at moment ratios of 0.5, 1.0,

1.25 and at the highest moment ratio observed for the various modifications to the car suspension and track profile. The numerical value of the highest ratio observed showed that two modifications produced considerable improvement. They are: the use of 3-11/16-inch travel springs with double coils under the friction wedges, and the use of the Unity No Sway Friction Side Bearings. However, even for these two modifications, the largest moment ratio observed was greater than one. This indicates that they did not offer a sufficient degree of improvement. The No Sway Friction Side Bearing showed the greatest amount of improvement. It should be stated that there is very good correlation between the ranking of the suspension modifications according to the highest moment ratio measured and the performance as observed visually. Wheel lift was not observed with the two modifications mentioned above, but because the moment ratio was observed to be slightly greater than one, the car must have been on the verge of wheel lift.

TDOP: 05-060

Canadian Pacific Railway Co., Analysis Of Riding Qualities Of 86-Foot Container Car CP 520536, Report No. S-405-69, October 1969, Department of Research, Montreal, Quebec.

This report discusses the results obtained in a ride environment test of a newly fabricated container car, Car No. CP 520536. Acceleration measurements were taken in the vertical-lateral plane.

The car, loaded with one empty container, was instrumented with the assistance of the National Research Council at their Uplands Railway Laboratory in Ottawa. It was run on test from Ottawa to Toronto, and from Toronto to Montreal. In Montreal, the wheels, which were essentially new, were replaced with a worn set, and the car was run to Toronto and back. After completion of the test, the wheels were replaced with the original ones, and the car was returned to revenue service.

It was found that with worn wheels, and to a lesser extent with new wheels, a lateral harmonic motion was induced in the trucks by the rail joints at speeds from 50 to 65 mph. This lateral oscillation of the trucks was carried through the carbody into the container. In the worst case recorded, with worn wheels at 63 mph, the lateral acceleration levels on the container floor exceeded 1 g for over seven percent of the running time. Vertical accelerations were also responses to the rail joints, and random peak accelerations of up to 4 g were recorded on the container floor.

TDOP: 05-061

Canadian Pacific Railway Co., Tests Of Trainmaster Locomotive Trucks, File No. 5012, 1958, Montreal 3, Canada.

These tests concerned variations in lateral on the axles of trainmaster locomotive trucks. Wheel loads and lateral forces were measured,and certain design changes were implemented as a result of the tests.

TDOP: 05-062

Chesapeake and Ohio Railway Company, 1956 Freight Truck Tests On Northern Region, Research Report No. 37, February 21, 1957, (Chessie System), 2 North Charles Street, Baltimore, Maryland 21201.

The purpose of these tests was to check the damping features and riding characteristics of various designs of trucks and package groups. The program consisted of three series of tests run concurrently. The first series involved tests of trucks that the railroad had in service since 1948 on Series 36000 gondola cars. This test was to determine the wear characteristics of trucks in service. The second series tested eight, 50-ton, 2-1/2-in. spring deflection packages for determination of their riding qualities. The third series was to determine the proper springing for the road's 70-ton hopper cars.

The results of Series 1 tests, when compared with tests conducted in 1948 on new trucks by the railroad, indicated that there was a general trend toward deterioration of ride with age (wear) due to decrease of available damping force. The report suggested that maintenance practices for ride control trucks should be established on the basis of ride deterioration rather than complete failure of snubber parts.

On the basis of Series 2 tests, a comparison was made among ASF, Buckeye, National, and Barber friction snubbing systems. From the results of Test Series 3, it was felt that the 3-1/16-in. spring group of seven outer and six inner coils would give proper springing for the road's 70-ton hopper and covered hopper cars.

TDOP: 05-063

Chesapeake and Ohio Railway Company - Baltimore and Ohio Railroad Company, Study To Lessen Rocking Action On Cars Moving Over Staggered Rail Joints, File: CC-220.4, April 29, 1969, Operating Headquarters Building, Huntington, West Virginia.

This document describes the progress being made in ongoing harmonic roll tests being conducted at this writing by the C&O - B&O. Test procedures, track locations, instrumentation, and types of data outputs are detailed.

TDOP: 05-064

Federal Railroad Administration, Investigation Of Boxcar Vibrations, by R. W. Luebke, Test Report No. FRA-RT-70-26, August 1970, 400 Seventh Street, S.W., Washington, D. C. 20591.

The vibration environment within a 50-foot, 70-ton boxcar and its running gear was measured by accelerometers and recorded on magnetic tape. The program included evaluations of load, speed, track irregularities, flat wheels, friction damping, variable rate springs, spring travel, and truck design on the vibration environment within the carbody. It was concluded that an increase in load and spring travel reduced the vibration levels in the carbody. All new truck designs tested produced reductions in the carbody vibration levels. Friction damping levels presently used in freight car trucks were found to be nearly optimum. Flat wheels produced a substantial increase in truck vibrations and a smaller increase in carbody vibrations.

RRIS: 039417
TDOP: 05-065

Federation des Industriels Ferroviaires, "Details Of Some Recent Test Work On The S.N.C.F. 1. Railway Dynamics Section (R.D.S.) 2. Brake Test Section (B.T.S.) 3. Rolling Stock Sound And Heat Proofing Section 4. Vitry Tests Station," French Railway Techniques, No. 1, 1969, 92 Rue Bonaparte, 75 Paris 6E, France, pp. 60-70.

This report concerns test work by the various sections of the S.N.C.F. including tests of stability of a gas turbine unit at high speeds, and a study of the unit on straight and curved track. Tests of noise levels in coaches with various bogies and of a spring test bed for determining a vehicle's spring rate are also reported.

RRIS: 033367
TDOP: 05-066

Federation des Industriels Ferroviaires, "Report On Some Recent Tests By The Division Des Essais De Material Of The S.N.C.F. 1. Railway Dynamics Section (S.D.F.) 2. Brake Testing Section (S.E.F.) 3. Vitry-Sur-Seine Testing Station (S.E.V.)," French Railway Techniques, No. 3, 1970, 92 Rue Bonaparte, 75 Parie 6E, France, pp. 105-115.

This article represents a report of French rail technology including tests concerning:

- Braking and effects on vehicle stability
- Stability of an experimental gas turbine vehicle
- Locomotives with rubber block suspension

The second section deals with testing of braking systems of passenger and freight units. The last section reports testing of a modified suspension system, an air suspension system, and an Eddy Current brake.

RRIS: 033374

TDOP: 05-067

General American Transportation Corporation, A Survey Of Shock And Vibration Environments In The Four Major Modes Of Transportation, by R. W. Schlock and W. E. Paulson, (Shock and Vibration Information Center, Naval Research Laboratory, Washington, D. C.).

This report comprises a review of all available information and test data describing shock and vibration environments in four major modes of transportation. This information has been collated, analyzed, and combined to provide acceleration versus frequency envelopes for each of the four transportation modes.

Data descriptive of the railroad shock and vibration environment have been categorized by over-the-road operation and coupling. Thus, a distinction is made between operational shock and shock resulting from train dynamics. A tabular display depicting the acceleration versus frequency envelope for the railroad is included in the report.

TDOP: 05-068

General Motors Corporation, Analysis Of DD-35 Locomotive & Truck In Operation On New York Central R.R. Tribes Hill 2 Degree 58' Curve, by L. F. Koci, Report #118, July 15, 1964, Electro-Motive Division, Chicago, Illinois.

This report is a study of the conditions associated with operating the DD-35 locomotive over the 2-degree, 58-minute curve involved in a derailment at Tribes Hill on February 21, 1964. The expected lateral loading considered likely to occur on the Tribes Hill Curve under the most extreme conditions would give net lateral wheel load of 15,900 pounds with instantaneous peaks to approximately 25,000 pounds. This extreme value of 15,900 pounds represents only 37 percent of the associated vertical wheel load. Analysis and test results accumulated from numerous tests, including curve negotiation of many different locomotive models and measurements of wheel stresses, axle stresses, and lateral loads, indicate that the instantaneous peak values acting over and above the base loads are not significant in rail turnover since lateral loadings representing 90-100 percent of vertical load can be applied to the rail at a single wheel location without any indication of rail turnover or wheel climbing.

RRIS: 040191

TDOP: 05-069

General Motors Corporation, SD Truck Performance On The Baltimore & Ohio Railroad And Related Wheel Wear On The Cumberland Division, by H. A. Marta, Test Report #128, December 5, 1966, Electro-Motive Division, Chicago, Illinois.

This test program was designed to determine the causes of excessive rail wear and associated wheel flange wear and to solve derailments involving SD-35 locomotive units that occurred during operation of the first SD-35 units on coal trains between Grafton and Terra Alta, West Virginia. The

test program measured the actual wheel-to-rail lateral loads and the associated truck motions under all possible operating conditions. Also, wheel wear tests were run on standard wrought steel and new cast steel wheels as a first step in an effort to find a better wear resistance wheel. Test results are reported, and recommendations are made.

RRIS: 040190

TDOP: 05-070

Guins, S. G., "Dynamics Of Shimmy In Passenger Car Trucks," ASME, 50-SA-21, 345 East 47th Street, New York, N.Y. 10017.

This study was concerned with only one phase of passenger truck dynamics, and therefore the solutions developed won't necessarily be valid when one encounters severe lateral oscillations in a passenger car. Tests were conducted to investigate the causes and the solutions of "shimmy" motion. It was found that this type of truck motion can be not only controlled but prevented if a proper angular damping is introduced between the truck frame and the carbody.

RRIS: 040384

TDOP: 05-071

Hengel, M. F. and R. J. Chinn, "The Effects Of Increased Work Being Performed At The Truck/Carbody Connection In Rigid Underframe Cars," Technical Proceedings - 1972 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 21-34.

Illustrated presentations are given showing problems in freight car centerplates as experienced by two different railroads. Factors influencing the wear observed are discussed. Component damage is shown in illustrations as related to broken body centerplates, broken centerplate wear rings, and broken riveted, bolted, or welded attachments.

TDOP: 05-072

International Union Of Railways, Tests On The Track On The Riding Stability And The Guiding Quality Of Vehicles By Means Of A Special Vehicle. Characteristics Of The Experimental Vehicle, Interim Report No. 1, Question B52, June 1963, Office for Research and Experiments, Utrecht, Netherlands.

In order to experimentally study the riding stability and hunting movements of vehicles, it was decided to consider a vehicle of the simplest possible design, i.e., a two-axled isolated bogie. Each of the factors entering into the dynamic phenomena connected with its running can be separately varied. This experimental bogie is placed under the middle section of the body of an ordinary bogie coach. The bogie and the experimental coach are described in detail and also the methods adopted for measuring the interesting magnitudes.

RRIS: 033201

TDOP: 05-073

International Union of Railways, Tests On The Track On The Riding Stability And The Guiding Quality Of Vehicles By Means Of A Special Vehicle - Results Of The First Track Tests, Interim Report #12, Question B52, June 1963, Office for Research and Experiments, Utrecht, Netherlands.

The report gives an account of the results of the tests made with an experimental bogie. The first part of the report supplies data relating to four series of tests during which the various parameters of the bogie (lateral play between axle-box and axle and axle load) and also the riding speed were successively varied. These tests have made it possible to establish conclusions relating to the wave length of the hunting movement, the amplitude of the transverse movements of the bogie, and the maximum transverse forces occurring between bogie and axles. The second part of the report supplies data relating to the tests during which the wheelbase of the test bogie was varied. The data obtained permitted some conclusions relating

to the wave length of the hunting movement, the transverse displacement of the bogie frame, the maximum angle of rotation of the bogie, and the transverse forces. The third part of the report contains an account of the results obtained during tests which studied the same magnitudes as the previous tests with the wheel tyres of the test bogie having been machined in accordance with the wear profile "Muller No. 2." All the tests were made in one tangent section, which was in an excellent state of repair and had a relatively constant gauge.

RRIS: 033732

TDOP: 05-074

Japanese National Railways, "Dynamics Of High Speed Rolling Stock," by T. Matsudaira, Railway Technical Research Institute Quarterly Report, October 1963, Kunitachi, Box 9, Tokyo, Japan, pp. 21-27.

Tests on passenger ride comfort on curved track and under braking conditions were carried out in March 1962 to supplement the data obtained by the previous tests. Tests were planned minutely with the theory of sensory inspection and experimental design, and the analysis of the subjects' evaluation of ride comfort was made with a statistical technique. The main difference between these tests and the previous ones lies in the method of evaluation of ride comfort. The evaluation was marked separately for degree of sensation against acceleration, for degree of mood or comfort, and for judging whether the ride was permissible or not in terms of sensation or mood.

RRIS: 033737

TDOP: 05-075

Jennings, J. R., "Relation Of Wheel Tread Wear And Brake Shoe Wear," ASME, 61-WA-217, 345 East 47th Street, New York, N.Y. 10017.

Brake shoe requirements on freight cars equipped with conventional brake rigging are not completely equitable at all shoe locations. Studies indicate that replacements of mated shoes on the same brake beam can vary from 8 percent on the inside beams to 13 percent on the outside beams. Four diagonal locations are found to have a shoe requirement approximating 55 percent, and the four mating positions 45 percent of total replacements. The unequal shoe wear affects wheel wear adversely insofar as realizing uniformity of service life on the wheels in the four positions as well as on mated wheels at the respective locations. The pattern of wheel wear disclosed is not conducive to economy of car maintenance nor to the best standard attainable in car utilization.

TDOP: 05-076

Kiefer, P. W., A. M. Miers, and L. D. Hays, "Protection Of Lading And Equipment," ASME, 48-A-76, 345 East 47th Street, New York, N.Y. 10017.

This article describes instrumentation and testing techniques which led to the adoption by the New York Central System of boxcars with improved trucks, brake rigging, and end-of-car cushioning devices. Detailed descriptions of vertical and lateral accelerometers used in comparative tests of truck performance are given. Graphic representations of freight carbody lateral and vertical accelerations at various speeds and with several equipment arrangements are shown. The following trucks are evaluated:

- The conventional spring plankless truck
- Standard Car Truck Co. Type S-I-L
- Standard Car Truck Co. Type S-I-L-P
- Symington-Gould Type XL
- Chrysler Design, Symington-Gould-Type FR-5D

TDOP: 05-077

Koch, B. R., High-Speed Research Train Instrumentation, PRC 68-2, March 1968, 7700 Arlington Boulevard, Falls Church, Virginia 22046.

Track geometry can now be determined at speeds in excess of 150 mph. Cross level, gauge, profile, alignment, warp, and rate of change of gauge can be measured at high speeds by using noncontact sensors.

TDOP: 05-078

Koci, L. F. and H. A. Marta, "Lateral Loading Between Locomotive Truck Wheels And Rail Due To Curve Negotiation," ASME, 65-WA/RR-4, 345 East 47th Street, New York, N.Y. 10017.

Curve-negotiation mechanics and forces resulting from locomotive trucks negotiating curves are well recognized. Recent theoretical analysis has made possible analytic explanation of such curving phenomena. However, meaningful and reasonable prediction of forces resulting from service conditions has been limited owing to the lack of experimental data pertaining to the friction-creep relationship between railway wheel and rail. An instrumented wheel-axle assembly was developed and used on 2-, 3-, and 4-axle trucks to study the effect of creep and the transverse load reactions resulting between wheel and rail. Instrumentation was used to measure these forces and the reactions between axles and truck frame under operating conditions. Test results confirm predicted phenomena and indicate the effect of creep on resulting loads. This paper includes a brief and general review of curve-negotiation mechanics and presents the test results and their relation to the theoretical analysis.

TDOP: 05-079

Koffman, J. L., "Body Rolling As Influenced By Bogie Suspension--1," Railway Gazette, Vol. 113, September 16, 1960, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 330-331.

Design variables affecting the angle of roll of bogie vehicles are discussed in this paper. A standard Russian coach bogie with divided bolster swing links is shown. Tests carried out with modern coaches running through curves at a steady speed resulting in an unbalanced lateral force equivalent to 0.1 g show that the angular displacement varies between 0.019 and 0.032 radians. With some modern locomotives with bogies using laminated bolster springs and operating at speeds of up to 90 mph, the angle of roll on straight track was found not to exceed about 0.005 radians. Roll stiffness of coaches, railcars, and diesel and electric locomotive bogie suspensions for British Railways stock are tabulated.

RRIS: 037696

TDOP: 05-080

Koffman, J. L., "The Frame Displacement Of The Three-Piece Diamond Type Bogies," Glasers Annalen ZEV, Vol. 98, No. 2, February 1974, Siemens (Georg) Verlagsbuchhandlung, Luetzowstrasse 6, 1 Berlin 30, West Germany, p 56.

The use of the three-piece, diamond-type bogies in the 120 km/h-freight-liner container trains of British Railways has caused occasional damage to track crossings, and has also led to wheel howling. The need for suppression of bogie hunting made it necessary to raise the bogies' rotational resistance. Tests have shown that when running through curves, as well as on straight track, the side frames can lozenge to an extent causing the wheel flanges to strike the edge of the crossings or to run on tangent track as if the bogie were in a curve. To eliminate this, future preference will be given to the use of single frame bogie designs.

RRIS: 054610

TDOP: 05-081

Koffman, J. L., "Tests Of British Railways Air-Sprung Wagon Bogies," Railway Gazette, Vol. 123, August 4, 1967, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 573-576.

This report concerns the testing of air suspensions for freight trucks. Air springs, with their ability to maintain a virtually constant height and a stiffness proportional to the load, should ensure a high equivalent static deflection regardless of load. The results of rather limited tests presented here show favorable riding qualities.

RRIS: 037824

TDOP: 05-082

Koffman, J. L., "The Torsionally Stiff Bogie Wagon," Railway Gazette, Vol. 123, August 18, 1967, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 629-632.

This paper begins by stating that torsional stiffness is an important factor bearing on the ability of vehicles to run over twisted track. Results of British Railways tests of tank cars with tare of 26 to 28 tons and 41.75-ft. truck centers are described. The design limitations for wheel load are calculated. Possible suspension changes to eliminate derailment tendencies are discussed.

RRIS: 040118

TDOP: 05-083

Luebke, R. E., "Performance Testing Of Freight Car Trucks," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 52-57.

This report states that the basic requirement for performance testing of freight car trucks is the need to reduce lading damage caused by vibration. To do this, accurate data are needed that will define the railroad environment in terms usable by a packaging engineer. The need for objective performance specifications is also stressed.

The author also reports on a joint series of tests undertaken by the C&O/B&O and FRA to evaluate the effects of load, speed, track irregularities, flat wheels, snubbing force levels, primary spring rates, variable rate springs, and overall truck design on the vibration environment in a 50-ft, 70-ton box car.

TDOP: 05-084

Mauzin, A., "The Behavior Of The SNCF Stock At High Speeds," French Railway Techniques, No. 2, 1966, Federation des Industriels Ferroviaires, 92 Rue Bonaparte, 75 Paris 6E, France, pp. 93-96.

An evaluation of S.N.C.F. rolling stock running at speeds of 200 to 250 kilometers per hour, using three different types of equipment, all designed for 150 kilometers per hour, showed no abnormal findings as to transverse forces exerted on the tracks by the equipment or transverse accelerations in the body of the last coach. One locomotive developed a lateral pressure on the track of 4.9 tons, or an L/V ratio of 0.70 at 250 kilometers per hour; another, 3.1 tons for an L/V ratio of 0.52; while the third developed 4.1 tons for an L/V ratio of 0.59. The latter two tests were at 200 kilometers per hour on a 3000-meter curve. The transverse acceleration in the last coach always remained under 0.08 g, which corresponded to the best obtainable in comfort with existing equipment.

These outstanding performances were due to the fact that the period of truck side-motion is very much under the natural period of the body, which puts it far from any hazard of resonance.

TDOP: 05-085

Mauzin, A., "A Contribution To The Study Of Transversal Comfort Of Carriages," French Railway Techniques, No. 1, 1966, Federation des Industriels Ferroviaires, 92 Rue Bonaparte, 75 Paris 6E, France, pp. 24-38.

To improve test techniques on the S. N. C. F., and to provide for advances in design, investigations have been implemented by the use of a motorized rail car, on which any number of factors relating to suspension could be varied, as desired. In addition, to supplement these online tests of new designs of trucks and vehicle suspensions, a test bed was built for measuring the periods and damping of the body rolling and side-to-side motions. Another test bed for measuring the frictional torque of the body rotation, and a special coach for measuring the bogie side-to-side motion were also constructed.

The significance and use of the results derived from the test beds were:

- The body-bogie assembly of a railway vehicle is a mechanical system comprising a drive motion (the truck) and a driven motion (the body).
- Such mechanical systems are elementary. They set up resonance phenomena, which at times can become dangerous. With the conditions known, the remedy can be applied either by modifying the value of a period of vibration or by introducing friction.
- To obtain the best possible riding comfort, body motion should be the least practicable; hence, the periods proper to the body should be longer than those of the truck side-motion.

TDOP: 05-086

Melrose, M. F., "Truck Performance," Technical Proceedings - 1967 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043.

This presentation is a general treatment of truck performance problems with special emphasis on rock and roll. Various truck components in various states of wear are illustrated. The importance of standards for comparison in the evolution of truck performance is stressed.

TDOP: 05-087

Missouri Pacific and ACF, Preliminary Results Of Missouri Pacific/ACF Truck Hunting Study, 1973, 210 North 13th Street, St. Louis, Missouri 63103.

This document contains a number of progress reports on a truck hunting study undertaken jointly by ACF and Missouri Pacific. Instrumentation for the tests consisted of accelerometers located on the test truck bolsters and side frames and a lateral accelerometer attached to each carbody bolster. Displacement gages of the "bending beam" type were installed to measure truck bolster rotation at each end of the car, and a special beam gage indicated when the truck side frames were out of tram. In addition, linear potentiometers measured vertical and lateral truck bolster motion with respect to the car bolster. Two video cameras were positioned to view the A-end truck.

Equipment used in conjunction with the tests included:

- Three types of constant contact side bearings (ASF Simplex, Ride Control, and Unity)
- Dynamically balanced wheel and axle sets
- Worn centerplates
- New centerplates
- Wheels with hollow treads
- Wheels with treads machined to a 1/40 taper

Observations concerning the relative effectiveness of some of this equipment in controlling hunting are given. Wheel profiles tested are illustrated.

TDOP: 05-088

Murphy-Richter Publishing Company, "Truck 'Hunting' Investigation," Progressive Railroading, Vol. 16, No. 3, May 1973, 9 South Clinton Street, Chicago, Illinois 60606.

Truck and component wear problems prompted the Seaboard Coast Line to conduct a series of carefully controlled, highly instrumented truck hunting road tests on a 70-ton box car built in 1967. Thirteen test runs were made at speeds of up to 70 mph over a 20-mile stretch of tangent track, which had 132-lb welded rail laid approximately two years before. The tests made it evident that accelerated truck and centerplate component wear can be attributed to shimmy or truck hunting, which originates from a natural tendency of the truck wheels to oscillate laterally between rails after a critical operating speed is reached.

RRIS: 050557

TDOP: 05-089

Nefzger, A., "Performance Results Of High Speed Running Investigations Carried Out On German Federal Railways," Glasers Annalen ZEV, Vol. 93, No. 11, November 1969, Siemens (Georg) Verlagsbuchhandlung, Luetzowstrasse 6, 1 Berlin 30, West Germany, pp. 337-346.

This paper concerns theoretical and experimental investigations to solve problems of wheel flange wear at speeds of 200 km/h and higher. The progress of the developmental work is described in terms of test results. The problems of wheel tread stressing are discussed. The tests carried out on wheelsets flexibly attached to the frame are considered as a step toward optimization of the overall design.

RRIS: 046921

TDOP: 05-090

Peterson, L. A., "Care And Feeding Of Modern Unit Trains," Proceedings of Conference on Track/Train Dynamics Interaction, December 1971, Association of American Railroads, Research and Test Department, American Railroads Building, 1920 L Street, N.W., Washington, D.C. 20036, pp. 84-118.

This report discusses problems encountered in the operation of unit trains and investigations to pinpoint problem sources, and concludes with a plan for further work in this area. Special hazards to the track posed by unit trains include:

- High static wheel loads from the special car designs
- Danger of a mistake in a unit train car design being repeated in many units
- The repetitious running of identical consists over the same track for extended periods of time

TDOP: 05-091

Peterson, L. A., W. H. Freeman, and J. M. Wandrisco, "Measurement And Analysis Of Wheel-Rail Forces," ASME, 71-WA/RT-4, 345 East 47th Street, New York, N.Y. 10017.

This paper describes a method used to continuously measure, record, and analyze the lateral and vertical forces between wheels and rails of several types of railroad freight cars under a variety of car and track conditions. The method, using analog-to-digital conversion and computerized data handling, has produced results relating to a multitude of car and track behavior subject areas. Especially important is the definition, development, and verification of performance "signatures" that are generated in a unique and characteristic manner by each car in negotiating a given curve. The finding of such "signatures" to be completely reproducible and yet sensitive enough to change with relatively minor track or car component variations, i.e., modifications, supports the belief that these techniques can be applied beyond pure experimental scopes into routine practices such as:

- Trackside inspection of cars in passing trains
- Mechanized track inspection
- Truck design evaluation

TDOP: 05-092

Pocklington, A. R., "Effects Of Lateral Forces When Propelling Round Sharp Curves," Railway Gazette, Vol. 121, December 3, 1965, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 942-945.

Tests were made to investigate the effects of lateral forces arising from the propulsive effort of a train. The train consist was 12 coaches of multiple-unit stock (weight: 450 tons) with the tractive force provided by two electro-diesel locomotives used as electric locomotives. The purpose of these tests was to determine whether the speed restrictions on certain turnouts would have to be reduced for propelled trains, since turnout curves are not normally fitted with check rails, and the curved track in single and double slips cannot be fitted with check rails. The coach adjacent to the locomotives was instrumented to record distance moved, speed, axlebox lateral forces, bolster lateral displacement, and tractive effort at the auto-coupler. Load measuring baseplates were placed under the outer rail of the curve at six adjacent sleepers to record lateral forces and vertical loads.

RRIS: 040098

TDOP: 05-093

Pocklington, A. R., "Propulsion Of Trains At Speed," Railway Gazette, Vol. 121, April 16, 1965, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 317-329.

Safety aspects on unrestricted main-line track were investigated by instrumenting axleboxes to measure lateral force on hauled vs. propelled trains. It was found that it is the combination of sharp curves and high propulsive efforts, both demanding low speeds, which requires close examination.

RRIS: 040091

TDOP: 05-094

Pocklington, A. R., "Propulsion Of Trains On Sharp Curves," Railway Gazette, Vol. 121, November 19, 1965, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 909-912.

A model to simulate the forces resulting in overturning and flange-climbing with propelled trains on sharp curves led to a series of tests arranged on a sharp curve at Dover. An axlebox lateral-force gear was used to measure the elements of the flange force present at the axleboxes, and load measuring baseplates were fixed between the outer rail and the sleepers to measure the vertical and lateral loads imposed by the wheels. The lateral loads are equal to the flange forces less the lateral tread friction at the outer rail. The results of these tests showed that the increased lateral force when propelling considerably exceeded the value attributable to the propulsive effort and the angle between the coaches.

RRIS: 040096

TDOP: 05-095

Railway Technical Research Institute, Japanese National Railways, Report On Dynamic Tests Of AAR Standard Truck Built By Sumitomo Metal Industries, Ltd., December 1970, Southern Pacific Transportation Company, One Market Street, San Francisco, California 94105.

In 1970, the Railway Technical Research Institute of the Japanese National Railways (JNR) in collaboration with Sumitomo Metal Industries, Ltd. performed dynamic bench tests on the three-piece, friction-snubbed freight car truck for the Southern Pacific Transportation Company. Upon the conclusion of testing activities, the following truck modifications were recommended:

- The centerplate resistance should be controlled through the use of liners or lubricants.
- The roller bearing motion in the direction of car travel contributes to the phenomenon of hunting and should be controlled through the use of shims mounted in the pedestal openings fore and aft of the adapters.

- To aid in maintaining tram, an intertie between the side frames should be used in conjunction with elastomeric adapter pads in place of the standard roller bearing adapters.
- Supplemental control of the lateral bolster motion with respect to the side frames should be provided through the use of two hydraulic dampers. Each of these units should be mounted on diagonal sides of the truck bolster in the interest of separating the control given by vertical snubbing forces from that given by lateral snubbing forces.
- Constant contact side bearings should be used.

TDOP: 05-096

Reynolds, D. J., "Hunting In Freight Cars (A Brief Description Of Cause And Cures, With Some Results Of Road Tests Of Various Modifications)," ASME, 74-RT-2, 345 East 47th Street, New York, N.Y. 10017.

This article contains a description of the forms of hunting observed in freight cars with three-piece trucks, using linear mathematics to give an engineering understanding of the oscillation and the forces causing it. The factors in the expressions giving the onset speeds show what can be done to raise or remove these limits. Supporting test results are quoted, and the practical and economic limits appropriate to the freight car are emphasized. The initial need to establish the incidence and cost of hunting is mentioned.

RRIS: 054014

TDOP: 05-097

Sakai, S., "Increase Of Train Speed On Curves," Railway Technical Research Institute, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan.

For a train to reach the destination in shorter time, speeds on curves and gradients must be increased. It would involve enormous investment to eliminate curves and gradients through rerouting the track in a mountainous country like Japan. Thus, the alternative would be to selectively invest in high-speed trains. With this in mind, Japanese National Railways has in-

troduced a bogie (TR 96) that is an experimental model built to explore the possibilities of increasing speed on curves. The data collected using this truck will offer the basis upon which a new high-speed train will be developed for actual service.

RRIS: 033856

TDOP: 05-098

Schmidt, J. J., "Influence Of Track Irregularities On Freight Car Vibrations," 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. 125-151.

This report covers a broad study undertaken jointly by the Chesapeake and Ohio Railway/Baltimore and Ohio Railroad and the Federal Railroad Administration of the U.S. Department of Transportation. The purpose of the study was to define a broad spectrum of the vibration environment of freight cars and their running gear. Test facilities included a prepared test track section, a test train including DOT research cars T-2 and T-4, and a subject 50-ft, 70-ton box car.

Based on the results of the study, the following conclusions were drawn:

- The vibration environment within a freight carbody is comprised of vibrational energy at all frequencies between 1.5 and 520 Hz. This vibrational energy, however, is concentrated in the forcing frequency range of 1 to 10 Hz and in the 2 wave bending mode of the carbody, which, in this case, was centered at 80 Hz. Other car designs would yield different carbody bending mode frequencies.
- The vibration energy within the freight car truck is concentrated at the natural frequencies of the wheel-on-rail and of the track structure, and at the wheel forcing frequency for other than round wheels.

TDOP: 05-099

Scott, H. L. and G. E. Reed, "Car Roll And Wheel Lift, Problem Characteristics," Technical Proceedings - 1965 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, N.Y. 14043, pp. 30-38.

Mr. Scott's portion of the program defines the problem of wheel lift and harmonic roll with heavy cars. His presentation points out that traditional solutions involve the track rather than the truck. Mr. Reed reviews various studies and solutions that have dealt with the problem and suggests that a universal solution will be dependent upon the development of a universal technique for evaluating the phenomenon.

TDOP: 05-100

Seaboard Coast Line Railroad, Measurements Of Lateral Rail Movement Of No. 10 Turnouts Negotiated At Various Speeds By 4-Wheel Truck And 6-Wheel Truck Diesel Locomotives, by B. K. Conrad, Test No. 164, January 1964, 500 Water Street, Jacksonville, Florida 32202.

These tests were made to determine the relative lateral movement of rail structure in a No. 10 turnout being negotiated at various speeds by an EMD diesel electric locomotive having four-wheel trucks and weighing 257,000 pounds per unit, and by a General Electric diesel electric locomotive having six-wheel trucks and weighing 360,000 pounds per unit. In practically all instances, the maximum lateral movement of both rails under the G. E. locomotive was greater than that of those under the EMD locomotive. Bar graphs show comparisions of maximum lateral movement of both rails caused by EMD and G.E. locomotives negotiating the No. 10 turnouts on 100-pound, 115-pound, and 131-pound rail at the various speeds tested. The data indicate that the turnout track movement under the heavier locomotive was considerably greater at 20 miles per hour than at the lower speeds tested.

RRIS: 040323

TDOP: 05-101

Seaboard Coast Line Railroad, Road Test For Truck And Component Wear Investigation, Report No. 173, 500 Water Street, Jacksonville, Florida 32202.

This report deals with a series of tests undertaken to determine the causes of accelerated truck and component wear on a series of 70-ton, 50-ft, box cars. One of the cars experiencing the rapid wear syndrome was chosen for testing. "Floor" windows of clear Plexiglass were installed in the B-end of the car for the purpose of direct observation and in order to allow the test crew to take motion pictures of the truck action. Instrumentation consisted of eight accelerometers and nine position transducers wired to the recording instruments mounted in the caboose. Thirteen test runs were made with various wheel, truck, and centerplate conditions in conjunction with an empty car.

Results of the tests indicated that the accelerated wear problem stemmed largely from truck hunting. The primary factor related to truck hunting was found to be the contour of the wheel treads. Wheels with new AAR tread contours began hunting at 70 mph, while wheels with a slightly worn contour hunted at speeds as low as 50 mph.

TDOP: 05-102

Shilkoff, M. J., "How Fast Can Trains Go?," SAE Journal of Automotive Engineering, Vol. 75, No. 7, July 1967, Society of Automotive Engineers, 485 Lexington Avenue, New York, N.Y. 10017, pp. 74-92.

Two high-speed trains are described: the French Aerotrain, which rides on an air cushion on an inverted-T-shaped track; and a modified standard rail car fitted with jet engines by the New York Central, which was tested on standard track. General characteristics are presented for the Aerotrain and data on dimensions, weight, and performance are listed. Discussion on the NYC train includes: train design, jet conversion, and jet engine characteristics; computer simulation of acceleration and deceleration;

and track and vehicle instrumentation for measuring axle and wheel stress, vertical and lateral acceleration, displacements, speed, temperature, and crosslevel and running statistics for the train.

RRIS: 033740

TDOP: 05-103

Simmons-Boardman Publishing Corporation, "Tests To Determine Truck Action," Railway Age, Vol. 85, No. 17, October 27, 1928, P. O. Box 350, Bristol, Connecticut 06010, pp. 798-800.

The purpose of this study by T. H. Symington and Son, Inc. was to measure the static effects of warped track surface in causing derailments under varying truck conditions. The tests were confined to the determination of the amount of depression or superelevation in the outside rails of curves required to cause derailment of a boxcar of rigid body construction, which had a total weight of 82,000 lb. These determinations were made in terms of track warp over a distance equal to that between the truck centers. The study also determined the turning resistance of trucks under various conditions. The results of the tests indicate that the centerplate resistance is not the major factor in the resistance to turning, which must be overcome by rail pressure at the flange, under the truck conditions prevailing when such turning is normally required. The flange pressure required to turn the truck on level track ranges approximately from 600 to 1,000 lb with centerplates greased. Forces greater than these are the result of side-bearing friction caused by the tilting of the centerplate as the side bearings on one side of the truck are brought into contact. The results do not indicate that the presence of graphite or grease on the center bearings had any consistent influence on total truck resistance under track conditions that bring the side bearings into action.

RRIS: 040157

TDOP: 05-104

Simmons-Boardman Publishing Corporation, "Truck Hunting Accelerates Wear," Railway Locomotives and Cars, Vol. 147, No. 4, May 1973, P. O. Box 350, Bristol, Connecticut 06010.

A test project undertaken jointly last year by Seaboard Coast Line and Pullman-Standard to determine the causes of premature wear in the truck areas of 150 five-year-old SCL 70-ton box cars has found truck hunting to be a major source of wearing action within the truck. Hunting, described as the natural tendency of the truck wheels to oscillate laterally between the rails at specific operating speeds, appears to be more prevalent with empty, roller bearing-equipped cars. The condition of the wheel treads, the tests revealed, will determine, more than anything else, the speed at which the hunting action will begin. There are other factors, however, that will also determine the critical speed at which hunting is initiated. Among these are:

- The type of journal bearing--roller or solid
- Load on the truck
- Type of side bearing
- Truck rigidity or squareness
- Degree of moisture on the rail

The joint SCL/P-S project was conducted in 1972 on a 20-mile stretch of SCL welded rail mainline between Waycross and Offerman, Georgia. Aside from one of the 70-ton box cars that had been experiencing the premature wear problem, equipment used in the test included a locomotive, an office car, and a special caboose that doubled as the instrument car. To better observe and photograph the box car's hunting action, flooring at the B end in the bolster area was removed and replaced with clear Lexan "windows."

RRIS: 046064

TDOP: 05-105

Southern Pacific Transportation Company, Harmonic Oscillation Tests Of Copper Concentrate Cars Tucson Division, February-March 1971, 1 Market Street, San Francisco, California 94105.

This report concerns a series of tests conducted by Southern Pacific Transportation Company to monitor the rock and roll characteristics of cars in copper concentrate service. Tests were conducted over uniformly shimmed test track at Stockham siding containing both 33- and 39-foot rail sections.

It was established on Stockham siding that the critical speed for the loaded copper concentrate car was 13 mph and 23 mph for the empty car to create resonant rock-and-roll. A greater degree of carbody sway (5 degrees 6 minutes) occurred with the empty base car than occurred with the loaded base car (4 degrees). With the inclusion of track cross level, amounting to 1 degree 30 minutes between joints, a roll angle of 6 degrees 36 minutes was indicated with an empty car and 3/4-inch low joints on 33-foot rail sections. This is above the desirable limit of six degrees.

Road tests substantiated that the roll conditions were greater on 33-foot rail sections and could be aggravated by operations approaching the critical speeds. The tests also pointed out that acceleration through critical speed range could create large roll angles.

Two stabilizing devices were tested both at Stockham siding and in over-the-road tests. A spring seat device shifted loaded critical speed to near the empty critical speed, but had such consistent roll action in operation that it is felt that rapid car deterioration (centerplate, bolster, springs) would occur, so the device could not be recommended. An elastic side bearing showed merit in effectively reducing the carbody sway on the loaded car over the entire speed range. The loaded H-100-4 Hopper car included in the test consist at Stockham showed a critical speed of 16 mph on the 33-foot rail sections, but a lesser tendency for rock-and-roll.

TDOP: 05-106

Southern Pacific Transportation Company, Union Pacific, and Pacific Fruit Express, Dynamic Tests Of Freight Car Trucks Under Mechanical Refrigerator Cars - 70-Ton Capacity, January 1971, Roseville, California.

The objective of these tests was a comparison of the ride characteristics of various designs of premium-type trucks currently being marketed with the ride characteristics of the AAR standard friction-snubbed truck.

The trucks tested were as follows:

- The AAR standard 70-ton (used as base case), narrow pedestal roller bearing truck with a Barber S-2 friction snubbing device, 3-11/16-in. travel AAR D-5 truck springs with 28-20 groupings, and Stucki single roller side bearings with new 2-wear CM-33 wheels on D-11 raised wheel-seat axles
- The XL-70 truck, a roller bearing version of the XL truck design by Symington Wayne Transportation Division of Dresser Corporation with an interlocking bolster-side frame arrangement with independently sprung journal housings
- The Rockwell Express truck, consisting of shallow pan-type side frames interconnected with ball-jointed transoms holding the side frames square while still allowing walking motion over bumps with the bolster located between a fixed phenolic wear pad and a spring loaded phenolic wear pad
- The National Castings Division of Midland Ross Corporation's truck, which employs the side frames as swing hangers by modification of the side frame pedestals to permit swinging of the side frames on the adapters and elimination of the conventional side frame spring seat and use of a rocker spring seat to permit swinging of the side frames relative to the spring seat
- The General Steel Industries' Chevron freight truck with a rubber Chevron spring arrangement replacing the present coil springs, snubbing device, and column guides

Tests were conducted on relatively level track between Roseville and Marysville, where the trackage presented a good variety of track conditions. Tests were conducted with both new and worn wheels on all trucks except the Chevron truck, which was tested only with new wheels.

Tests were made at speeds of 65, 55, and 45 mph, with measurements taken of lateral and vertical force acceleration inputs to truck side frames and carbody floor, of wheel lateral and vertical forces on the rail, and of truck swivel motion.

The following conclusions were reached:

- Vertical forces received by the carbody floor were not adequately snubbed or controlled by any of the five trucks tested. Maximum vertical force reaction approached the 1 g level, which is considered unsatisfactory.
- Lateral force levels at the carbody floor were at a low amplitude with all of the trucks tested with new wheels with the exception of the Chevron truck. Lateral carbody force reactions to rail joints and other track irregularities increased materially at higher speeds and when worn wheels were substituted for new wheels, especially with the AAR standard truck and the XL-70 Symington Wayne truck. The spring plank swing motion arrangement on the National Swing Motion truck showed excellent control of carbody lateral force levels, while the transom-linked side frames of the Rockwell truck showed acceptable control of these force levels.
- Vertical and lateral force levels were lowered when speed was reduced, but they did not reach acceptable levels until below 50 mph.
- Wheel to rail lateral to vertical force ratios were never high enough for concern. Vertical forces between wheel and rail varied more with trucks with more harmonic bounce (Rockwell and Chevron).
- Further efforts to improve truck snubbing arrangements need to be made.

RRIS: 039696

TDOP: 05-107

Stassen, H. G., "Random Lateral Motions Of Railway Vehicles," International Union of Railways, No. 28, January 1969, Office for Research and Experiments, Utrecht, Netherlands, pp. 16-19.

This study was concerned with the random motions made by a truck of experimental design. The behavior was studied on a test track, and was compared with empirically predicted behavior. Results of the program demonstrated a close agreement between observed and predicted performance.

RRIS: 039416

TDOP: 05-108

Suckow, F. G., "Particulars Of Car Impact And Road Test Demonstrations," Technical Proceedings - 1964 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, N.Y. 14043, pp. 49-54.

The Symington car impact test plant is briefly described in this presentation. The results of a road test of the XL-70 truck conducted during this conference are included. The portion of the test from Depew to Batavia showed good results with a considerable amount of high-speed running. The XL trucks performed well, while considerable periods of truck swiveling with accompanying car nosing were noted in a car equipped with ASF trucks.

RRIS: 039522

TDOP: 05-109

Torns, A. H., "Wheel Hunting And Irregular Rail Wear," Railway Gazette, Vol. 83, December 21, 1945, Temple Press Limited, 161-166 Fleet Street, London EC4, England, p 636.

In combination with the hunting of rolling stock wheels, there is the constant side-to-side movement of individual pairs of wheels. The standard taper of coning of wheel-treads is at 1 in 20, so the tread of a new tyre may be at right-angles to the center-line of the rails. Experiments have been tried in varying the angle of taper of the treads and the radius of the

railhead, but the wear of tyres and rails complicates the problem, and hitherto has made it difficult to come to any final conclusion as to how the hunting phenomenon may be avoided. Persistent bogie hunting can cause bogie stock to ride uncomfortably. Of even greater importance is the wear-and-tear of rolling stock and resultant damage to rails.

RRIS: 039480
TDOP: 05-110

Union Pacific Railroad, Lateral Stability Tests Of Cabooses With General Steel Casting Outside Swing Hanger Trucks And LFM Caboose Express Truck, by P. E. Flebbe, 1416 Dodge Street, Omaha, Nebraska 68102.

These static tests were conducted by the Union Pacific Railroad to compare the lateral stability of General Steel Casting trucks with outside swing hangers with the LFM Caboose Express truck. No skewing or unstableness of either truck was noted at compression drawbar forces up to 300,000 lb on tangent and 13-degree curved track.

RRIS: 04001
TDOP: 05-111

Union Pacific Railroad, Tests Of F-70-23 Modified And Unmodified Baggage Flat Cars, by P. E. Flebbe, Files 400-310-3/413-011, 1416 Dodge Street, Omaha, Nebraska 68102.

These tests were made to determine the modifications that would be necessary to improve the stability of flat cars, Class F-70-23, redesigned for container service. Preliminary squeeze tests with unmodified trucks resulted in these cars skewing and derailing, indicating that the original 2-1/2-in. bolster-to-truck-frame clearance on each side should be reduced to a maximum of 1/2 in. on each side. Tests made with instrumented wheels and truck bolsters on both unmodified and modified trucks under these flat cars (with 2-1/2-in. bolster to truck frame clearance and

with clearance reduced to 3/8 in.), showed the following: on trucks with unmodified clearance, lateral pressure of wheel to rail reached 10,000 lb with 200,000 lb squeeze, resulting in derailment of test car. L/V ratio exceeded 0.68. Further tests on unmodified trucks with 450,000 lb squeeze developed lateral pressure of wheel to rail up to 15,000 lb, resulting in rail turnover and derailment. L/V ratio exceeded 0.68 (AAR figure for limiting value of this ratio). On trucks with modified clearance, lateral pressure of wheel against rail reached a maximum of only 2,000 lb, for an L/V ratio of 0.22. In all these tests, all unmodified cars assumed a skewed position, while all modified cars with restricted bolster to truck frame clearance remained in line with axis of track. Final tests with the modified flat car were made under drawbar pull of 250,000 lb, with a maximum lateral force of wheel against rail of only 4,000 lb. It was determined that the limitation of the truck bolster to truck frame clearance did not adversely affect the curving ability of the modified car, and rendered the car stable under all tracking conditions.

RRIS: 039695

TDOP: 05-112

Union Pacific Railroad, Tests Of Head End Passenger Equipment Permanently Assigned To Freight Service, by P. E. Flebbe, Files 340-103-1, 413-011, 400-310-3, August 28, 29, September 5, 15, 1969, 1416 Dodge Street, Omaha, Nebraska 68102.

Tests were made on three types of baggage cars that have been permanently assigned to freight service, equipped with trucks as described below:

- Six wheel trucks with outside swing hangers
- Six wheel trucks with inside swing hangers
- Four wheel trucks with outside swing hangers

These tests were made in buff or squeeze to determine the stability of these cars before and after modification to reduce lateral truck frame to truck bolster clearance to plus or minus 3/8 inch. In the first case, the modified six-wheel truck showed a reduction in lateral wheel force to draw-bar force ratio of 79 percent. In the second case, the modified six-wheel truck showed a reduction in this ratio of 73 percent, and in the third case, the reduction was 54 percent for the modified four-wheel truck. The modified six-wheel trucks produced no skewing. The cars with four-wheel trucks did assume a skewed position, but the L/V ratio did not exceed 0.66. All of the cars with modified trucks had good stability under all test conditions.

RRIS: 039694

TDOP: 05-113

Union Pacific Railroad, Truck Hunting, Tracking, And Dynamic Stability Tests Of Stock And Ore Cars Equipped With Various Anti-Hunting Trucks And Bolster Snubbing Devices, Report No. TCR-19, July 1973, Office of Research and Standards, 1416 Dodge Street, Omaha, Nebraska 68102.

Various preloaded side bearings, a centerplate extension pad, and several truck designs were tested under an empty triple-deck stock car and an empty ore car. Fifteen test runs were made. Each one included high speed runs on level, tangent continuous-welded rail, six-degree left- and right-hand curves, and 20-mph travel over selected low rail joints. Devices were evaluated in terms of their effectiveness in controlling lateral truck hunting at speeds near 75 mph without a serious adverse effect on the car's curve negotiability and rock and roll stability. A hunting severity index was defined to evaluate and compare the hunting tendencies of the cars equipped with the various devices. One truck and one bolster restraining device were approved.

TDOP: 05-114

Union Pacific Railroad and Southern Pacific Transportation Company, High Speed Tracking Test Of Empty Bulkhead Flat Cars Between Las Vegas - Yermo, May 1972, 1416 Dodge Street, Omaha, Nebraska 68102.

The joint tests of the Union Pacific and Southern Pacific reported herein were conducted for the purpose of developing an understanding of the hunting action of trucks on bulkhead flat cars operated at maximum authorized speeds on both railroads. A further purpose was to investigate hardware concepts for application to the trucks in an attempt to add stability to the empty bulkhead flat cars during the truck hunting period. For this purpose, existing hardware was chosen to demonstrate concepts of control.

The data herein are preliminary in character. Inasmuch as limited service conditions were studied, more extensive data on a wider range of operating parameters and endurance and life testing, including an appraisal of maintenance costs, should be developed to provide a basis for firm conclusions with regard to the systems employed.

Study conclusions included:

- The empty bulkhead flat cars tested showed lateral truck hunting occurred at speeds as low as 45 mph, and sustained hunting was observed at speeds as low as 50 mph.
- No hunting was observed at speeds up to 77 mph with the National Castings Division Swing Motion Truck.
- Truck unstable lateral hunting speeds were increased to 60 mph with the Standard Car Truck Centerplate extension "C-PEP" device and the Vanguard Corporation urethane preloaded Trioid side bearing.
- When bolster gib clearances were restored to pre-1968 standard, and excessively worn roller bearing adapters were replaced with new adapters, carbody roll was decreased, and there was an increase in threshold speed of lateral hunting to 50 mph.

- A new bulkhead flat car, which had never been under load, was equipped with worn wheels and showed lateral truck hunting tendencies at 50 mph.
- A 1966 bulkhead flat car equipped with new wheels showed no hunting to 66 mph.
- It was shown that both hunting and the damping of hunting action in a given car spread somewhat to immediately adjacent cars.

TDOP: 05-115

Wickens, A. H., "The Dynamics Of Railway Vehicles On Straight Track: Fundamental Considerations Of Lateral Stability," Proceedings Of The Institution of Mechanical Engineers, Vol. 180, 1 Birdcage Walk, London SW1, England, pp. 29-44.

Existing and recently developed theories of the lateral motion of railway vehicles are discussed in relation to experimental work on both models and full-scale vehicles. It is shown that a realistic theory, taking into account flexibility between the wheelsets and the frame in the longitudinal, lateral, and vertical directions, and the influence of wheel and rail profiles, yields values for critical speeds which are consistent with experimental results. The influence of various parameters on stability is discussed. Examples of suitable choices of suspension and other parameters for high-speed operation are given, and experiments carried out in order to verify these concepts are described.

RRIS: 040124

TDOP: 05-116

Wiebe, D., "The Effects Of The Lateral Instability Of High Center Of Gravity Freight Cars," Anthology of Rail Vehicle Dynamics, 1971, ASME, 345 East 47th Street, New York, N.Y. 10017, pp. 253-267.

High center of gravity freight cars experience extreme weight shift from side to side as a result of lateral resonance on track with cross-level differences from alternately staggered rail joints. Dynamic measurements from tests made on track with controlled 3/4-in. cross-level difference changes illustrate the force and motion magnitudes resulting from resonant and near-resonant operating speeds. These magnitudes include: side bearing loads of 138,000 lb and spring group loads of 100,000 lb, accompanied by centerplate separation and wheel lift.

The rotational energy input to the carbody can be approximated for a given motion cycle and is proportional to the product of the amplitudes of the track profile and carbody motion. The high lateral forces on the truck at the side bearing and centerplate make the truck unstable and cause wheel lift on one side. This lateral force at a given end of the car is proportional to the corresponding vertical side bearing load.

RRIS: 047433

TDOP: 05-117

Laboratory Analysis of Truck Performance

Byrne, R. and W. E. Thomford, "Summary Of Japanese Reports On Railroad Vehicle Trucks," Technical Proceedings - 1973 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 28-30.

The article discusses Japanese National Railway research programs that contributed to the successful design and construction of passenger car trucks for new Tokaido Line equipment. A summary of Japanese reports by Dr. M. Kunieda, Mr. K. Yokose, and Dr. N. Matsui, relating to topics such as: hunting, curve negotiation, derailment quotient, and wheel-rail interfaces is given in the paper. Mention is also made of a Japanese study of the standard AAR truck, which was funded by Southern Pacific Transportation Company.

TDOP: 05-118

Canadian Pacific Railway, Mass, Inertial, And Frame Stiffness Properties Of A 100-Ton Freight Car Truck, by D. D. Taylor and R. F. Gonsalves, Report No. S-408-70, March 1970, Department of Research, Montreal, Quebec, Canada.

This report presents some of the physical properties of a 100-ton freight car truck. The truck tested was a Symington-Wayne Barber Stabilized Type S-2-A-O, No. COU DBO 9149. Items reported on include:

- The masses of the various components of the truck
- The moments of inertia of these components
- Some of the stiffness characteristics of the truck frame under applied loads in the horizontal plane

The project was carried out as part of the CP Rail contribution to a research program initiated by the N.R.C. Subcommittee on Train Dynamics and Lading Damage. The program's purpose was to improve the lateral ride and guidance

dynamics of rail vehicles at high operating speeds. The data were required for input into a truck simulation computer program. It was also felt that a tabulation of truck properties would be useful for other research work. The test program was carried out at the Uplands Railways Laboratory of the National Research Council of Canada in Ottawa by the staffs of the Canadian Pacific Research Department and of the N.R.C. Department of Mechanical Engineering.

A majority of the data are presented in tabular form. In the case of the stiffness characteristics of the truck frame, the data are presented in raw form only, and any required stiffness parameters can be developed by the reader.

TDOP: 05-119

Cappel, K. L., "Simulation Of Vehicle/Track Interaction," Technical Proceedings - 1971 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 58-63.

The main purpose of this presentation is to describe the extent to which it is possible to reproduce wheel-rail interactions in the laboratory. Various laboratory installations throughout the world are illustrated, and the Department of Transportation's wheel-rail simulator is described. The capabilities and limits of "roller rig" simulator such as the DOT's are discussed.

TDOP: 05-120

Cappel, K. L., "Wheel/Rail Dynamic Simulator - Its Capabilities And Utilization," Proceedings of Conference on Track/Train Dynamics Interaction, December 15 and 16, 1971, American Railroads Building, 1920 L Street, N.W., Washington, D. C. 20036, pp. 282-300.

In this presentation, the wheel/rail simulator to be installed in the Department of Transportation's Pueblo Rail Lab is described and compared to other simulators in Great Britain and Japan. Emphasis is placed on the flexibility of the DOT design, which will be able to simulate a wide variety of track conditions.

TDOP: 05-121

Dilg, W. C., "The Riding Properties Of Bogie Vehicles--2," Engineering Interchange for Railroad Advancement, Vol. 111, December 11, 1959, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 538-540.

The author explores various theoretical aspects of railway vehicle riding performance as an aid to the rationalization of bogie design. It is shown that low ride index values can be retained by minimizing either the natural frequency or the amplitude or both of the oscillation. The importance of damping with respect to amplitude ratio is shown, and it is pointed out that dampers are actually undesirable at speeds in excess of 1.41 of the resonance speed. It is also pointed out that damping is of relatively little importance with short swing links. In order to determine the ride index of a rail vehicle, the magnitude of excitation amplitudes, the natural frequency, and the damping factor of the system must be known.

RRIS: 037225

TDOP: 05-122

Henderson, K. A. and J. Johnson, "A Criterion For The Control Of 100-Ton Hopper Car Roll Motion," (paper presented March 27, 1968, and also appears in Anthology of Rail-Vehicle Dynamics, 1971) ASME, 345 East 47th Street, New York, N.Y. 10017.

The chief purpose of this paper is to introduce a technique of analyzing the roll motion of heavy cars. The moment ratio is presented as being a realistic yardstick which can easily be calculated from measurements

which are reasonably simple to make. The moment ratio, as a criterion, can be used for the following purposes:

- The highest value of the moment ratio can be used as a basis for ranking the effectiveness of various suspension modifications. The highest value of moment ratio can also be used to judge the sufficiency of any modification.
- The speed range over which wheel lift can occur can be determined by examining the relation between train speed and moment ratio. The critical speed is the range during which the moment ratio is greater than one.
- The number of consecutive track irregularities needed to cause wheel lift can be found by analyzing the build-up of moment ratio throughout the test track. This can be used as a measure of the likelihood of wheel lift occurring in service.
- Examination of the composition of the stabilizing and overturning moments used to calculate moment ratio can provide information concerning how an improvement was achieved.

RRIS: 047434

TDOP: 05-123

Japanese National Railways, "Dynamics Of High Speed Rolling Stock," by T. Matsudaira, Railway Technical Research Institute, November 1962, Kunitachi, Box 9, Tokyo, Japan, pp. 13-22.

This article is devoted to the dynamics of high-speed rolling stock. Areas of discussion include testing of allowable accelerations in a curved track, limits of wheel side thrust, effects of load upon wheel side thrust, truck hunting, and reports concerning bogie assemblies especially designed for high-speed operation in conjunction with air springs and rubber draft gear.

RRIS: 033738

TDOP: 05-124

Japanese National Railways, "Running Safety," (unpublished report) February 23, 1972, Kunitachi, Box 9, Tokyo, Japan.

This unpublished JNR document deals with a theoretical approach to the running safety of rail vehicles. Operating limits are stated, and the methods of computation are illustrated. The limits considered are:

- Allowable limit for the decrease in static wheel load
- Allowable limit for the decrease in dynamic wheel load
- Allowable limit for lateral pressure of rolling stock
- Allowable limit for the ratio of lateral pressure and wheel load
- Allowable limit for wheel-rail colliding speed
- Allowable limit for horizontal vibration accelerations of the carbody
- Allowable limit for running speed

TDOP: 05-125

Johnson, M. R., "The Analysis Of Truck Forces On Curved Track," (unpublished) April 17, 1957, Battelle Memorial Institute, BCL-240, 505 King Avenue, Columbus, Ohio 43201.

This report contains an analysis of the static, steady-state forces on a four-wheel truck moving on curved track. The truck is used in the general sense to describe any four-wheel rigid wheelbase of conventional truck size or of a size as large as experimental four-wheel cars. This work should represent one phase of a program to attempt a rationalization of various truck phenomena. Appended to this report is an 8-part series of published articles entitled, "The Mechanics of a Locomotive on Curved Track." These articles appeared in The Railway Engineer, 1934-1935.

RRIS: 033741

TDOP: 05-126

Koci, L. F., Locomotive Truck Design And Effect On Rail, report submitted to Electro-Motive Division and General Motors Institute, July 15, 1959, LaGrange, Illinois 60525.

This paper describes analytical and experimental investigations of the nature of the action between the wheel and rail when a locomotive negotiates a curve. The three sections of the report treat lateral forces due to curve negotiation, dynamic vertical forces, and dynamic lateral forces. Calculation of curve negotiation forces and data obtained from tests run with a model truck and track layout are presented in the appendix.

TDOP: 05-127

Koffman, J. L., "Carriage And Railcar Bogies: Their Design And Development--V," Railway Gazette, Vol. 115, September 29, 1961, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 363-366.

The determination of bolster-spring oscillation amplitudes, swing-link displacements, and their phasing using oscillographic instrumentation can be useful in assessing ride qualities. Vibrograph wedge tests permit determination of damping factors. The frequency of lateral/swaying oscillations can be determined by placing the vehicle on a transfer table and imparting an impulse by suddenly stopping the moving table. Similarly, body-nosing can be excited by placing the vehicle on a turntable with the center of gravity of the vehicle body being in line with the center of rotation. A sudden stop of the turntable will then excite body-nosing oscillations at their natural frequency.

RRIS: 040084

TDOP: 05-128

Koffman, J. L., "The Riding Properties Of Bogie Vehicles--1," Railway Gazette, Vol. 111, November 27, 1959, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 483-487.

This article discusses the problems of values assigned to ride quality as indicators of the effectiveness of a suspension system of a rail vehicle. In order to find an index of such performance, a mechanical device such as the Askania Universal Vibrograph is recommended. The charts that are produced by the Vibrograph can be evaluated, and the application of the evaluations to vehicle design is discussed.

The instrument may be used to measure vehicle body oscillations. Vertical accelerations and resonance are included in the measurements. The article concludes with a reminder that the factors recorded by the vibrograph should be used in combination with the factors of noise and vibration before they can be considered meaningful design parameters.

RRIS: 037224

TDOP: 05-129

Koffman, J. L., "Rotational Resistance Of Modern Bogies," Rail Engineering International, Vol. 3, No. 7, September 1973, Broadfields (Technical Publishers) Limited, Little Leighs, Chelmsford, Essex CM3 1PF, England.

Evaluations of conditions relating to tyre flange wear and derailment have been published, but little has been reported concerning vehicles which incorporate secondary suspension. These vehicles are therefore dealt with in this article. Included among these are Metalastik rubber assemblies and the Flexicoil assemblies. The construction of the Z-curve is also explained.

RRIS: 048226

TDOP: 05-130

Koffman, J. L, "Vertical Oscillations Of Bogie Vehicles," Railway Gazette, Vol. 113, July 15, 1960, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 73-80.

This article considers the oscillations of bogie vehicles and the factors that must be known in order to calculate the frequency of these oscillations. As a result of the calculations, the preference for "good" riding qualities results in the selection of helical springs for primary and secondary springs. For locomotive springing, leaf springs are preferred. After a discussion of amplitude limits and pitching and the relationships of tractive forces to static deflections, the application of rubber and air springs is treated.

RRIS: 037223

TDOP: 05-131

Langer, B. F. and J. P. Shamberger, "Lateral Oscillations Of Rail Vehicles," Transactions, ASME, 1935, 345 East 47th Street, New York, N.Y. 10017.

The authors point out in this paper that lateral oscillations, which either do not occur or are negligible at slow train speeds, are of vital importance at the high train speeds now necessary. Lateral forces are responsible for a large portion of all railroad derailments. Therefore, the prevention of lateral derailments requires a knowledge of both the conditions which cause high lateral forces and of the allowable operating limits in relation to those forces.

At the time of this writing, studies of lateral motion had been restricted to forced oscillations, the life of which depended upon some periodic force such as cylindrical action or track irregularities. The present report deals with a type of oscillation which, after a minor disturbance, may build up to dangerous proportions and sustain itself indefinitely on perfect, tangent track. This oscillation is known as "nosing." Illustrations of models used in research and of oscillations recorded using those models are included in the report.

TDOP: 05-132

Lawson, K. L., "Status Report On Pueblo Railab," Technical Proceedings - 1971 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 16-19.

This presentation reports on the progress as of 1971 on the construction and implementation of the dynamic simulation facilities at Pueblo, Colorado.

TDOP: 05-133

Matsudaira, T., "Hunting Problem Of High-Speed Railway Vehicles With Special Reference To Bogie Design For The New Tokaido Line," Proceedings of the Institution of Mechanical Engineers, Vol. 180, Part 3F, Institution of Mechanical Engineers, 1 Birdcage Walk, London SW1, England, pp. 58-66.

This paper describes a preliminary experiment investigating truck hunting by means of a model vehicle, and tests with an experimental bogie and a prototype bogie at a rolling stock testing plant. A considerable amount of various calculations concerning hunting, carried out during the design stage and during running tests of the prototype bogies on the test track section of the new Tokaido Line, are included. The report also gives a number of major test results with some discussion.

In designing the prototype bogie, special attention was paid to the following in order to prevent hunting:

- Mode of axle-box support and its stiffness
- Combination of frictional and elastic restoring force against bogie rotation

According to the running test at 246 km/h this bogie exhibited very high running stability.

RRIS: 037209

TDOP: 05-134

Matsudaira, T., N. Matsui, S. Arai, and K. Yokose, "Problems On Hunting Of Railway Vehicle On Test Stand," ASME, 68-WA/RR-2, 345 East 47th Street, New York, N.Y. 10017.

The stand test in which a vehicle is set on a roller rig to simulate its running state is available as one of the effective means for experimental study on vehicle hunting. Here several questions arise: whether or not a scale model test can substitute for an actual vehicle test; whether or not a bogie with a dummy load on a test stand can represent a whole car; and whether or not a stand test produces a result similar to a field test. The present paper summarizes the results of the studies concerning these questions conducted at the Railway Technical Research Institute of the Japanese National Railways.

TDOP: 05-135

Scales, B. T., "Behaviour Of Bogies On Curves," Railway Engineering Journal, July 1972, Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London, SW1, England, pp. 19-24.

This paper is concerned with the results from a study undertaken by the author in collaboration with the Bessemer and Lake Erie Railroad. The study investigated the behavior of bogies on curved track as part of an overall program to reduce derailments and excessive rail wear on curves.

The research described deals with a standard, three-piece bogie. The paper states that, on a large radius curve, free curving can be obtained from the wheel coning. Bogie behavior on large radius curves is described. Behavior on small radius curves is analyzed and explained in detail. The advantages of a radial bogie are given, and a prototype radial bogie is described.

RRIS: 046838

TDOP: 05-136

Temple Press Limited, "Oscillograph Recording Equipment On The Southern Region," Railway Gazette, Vol. 106, February 22, 1957, 161-166 Fleet Street, London EC4, England, pp. 219-221.

The amplifier unit described in this article is basically a six-channel, self-balancing, push-pull amplifier that will accept symmetrical and asymmetrical inputs. The outputs of the six amplifier channels are fed to six cathode ray tubes. The indications on the cathode ray tubes may be recorded simultaneously on photographic paper. The equipment has been used on a variety of projects. The recording equipment has been used for the measurement of the rail bending stresses set up in bull-head and flat-bottom rails due to the passage of vehicles. Tests involving the measurement of stresses set up in an L.T.E. bogie frame gave useful information regarding stresses under static and dynamic conditions. The equipment has also been used in tests on electrical equipment for multiple unit trains.

RRIS: 037625

TDOP: 05-137

Voorhees, J. E., "Car Test Facility," Technical Proceedings - 1968 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 45-51.

This presentation concerns Batelle's plans for the construction of a railroad dynamics laboratory. A key feature of this laboratory will be a dynamic simulator. Research and development programs aided by the dynamic simulator would include:

- The development of advanced suspension systems
- The establishment of dynamic performance specifications for cars
- The establishment of maintenance specifications for track
- The establishment of authentic fatigue tests for individual truck components, based upon detailed observation and measurement of their behavior

- The accumulation of a body of knowledge of vehicle dynamics to aid the design of improved freight cars
- The refinement of computer simulation techniques to aid the design and prediction of car dynamics performance

TDOP: 05-138

Wickens, A. H., "Practical Research Into Vehicle Dynamics," Railway Gazette, Vol. 123, June 16, 1967, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 467-469.

Stability theory is evaluated by running an experimental car, HSFV-1, on a roller rig and on track. The car underframe and drawgear are standard, and the suspension is designed for an unladen weight of 10 tons and a loaded weight of 35 tons. The wheel profiles conform to the RD6 profile. The first tests consisted of vertical and lateral damper tests; vertical and longitudinal spring stiffness tests; accurate turning and measurement of wheel and roller rig profiles; measurement of masses, moments of inertia, and centers of mass; and rebuilding of vehicle for correction of dimensions. The next step was to measure the natural modes and frequencies of the vehicle, again with wheels fixed, in the frequency range 0 to 10 c/s. The longitudinal springs were modified for stability tests. The vehicle was completely stable up to a speed of about 100 mph, when wheelset instability occurred. The purpose of initial track tests was to confirm that the lateral stability of the vehicle on the track was satisfactory and consistent with the roller rig tests and to measure the response of the vehicle to track irregularities. The results and techniques established by the work on HSFV-1 are applicable to all railway vehicles.

RRIS: 040117

TDOP: 05-139

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