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U.S. Transit Track Restraining Rail

Volume I: Study of Requirements and Practices

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December 1981
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16. Abstract
This report is sponsored by the Office of Rail and Construction Technology of the Urban Mass Transportation Administration (UMTA), and is a part of a program to assist transit properties in the development of track. This report covers a study of restraining rails in transit track, which is part of the current research program of UMTA and was initiated: 1) to assist in the analysis, design, and maintenance and operation of transit track; 2) to compile guidelines for the use of restraining rails; and 3) to devise concepts for possible improvements.

This report, Volume I, describes current practices in the use of restraining rails and provides data from rail lubrication tests. It evaluates the benefits of alternative practices, presents concepts for advanced designs, discusses simplified analysis of the costs and benefits of restraining rail installations, recommends the design and fabrication of modifications and new concepts, and recommends tests to obtain additional information for improvements in track adjustment and practices in order to reduce rail wear.

Volume II of this report contains recommended criteria and practices for the use of restraining rails in transit track and suggestions for the evaluation of requirements. It includes descriptions and limits for measurable conditions, such as guard distance, frangeway width, and descriptions of conditions that are not normally measurable. As a research product, it does not repeat the description of the investigation and other details in Volume II.

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PREFACE

This report was prepared by ENSCO, Inc. under Contract No. DOT-TSC-1771, managed by the Transportation Systems Center, Cambridge, Massachusetts. The contract is part of a program to assist Transit Properties to improve track, which is sponsored by the Office of Rail and Construction Technology, Office of Technology Development and Deployment, Urban Mass Transportation Administration of the U.S. Department of Transportation. The objective of the contract is to develop information to assist the design, installation maintenance and operation of transit track of increased integrity and safety, with emphasis on the cost effective use of restraining rail.

The report describes current practices in the use of restraining rail and provides data from rail lubrication tests. It evaluates the benefits of alternative practices, presents concepts for advanced designs, discusses simplified analysis of the costs and benefits of restraining rail installations, recommends the design and fabrication of modifications and new concepts and recommends tests to obtain additional information for improvements in track adjustment and practices in order to reduce rail wear. Guidelines for the use of restraining rail, compiled under the project, are presented in a companion report, entitled "U.S. Transit Track Restraining Rail, Volume II Guidelines, Report No. UMTA-MA-

Useful data in the report have resulted from the generous efforts of many transit engineers and others. Throughout the study, very effective assistance and guidance were provided by Messrs. Gerald Saulnier and Michael Wiklund of the Transportation Systems Center, the technical monitors. Valuable help and useful suggestions were provided by Mr. C. O. Buhlman of the American Public Transit Association (APTA) and by members of the Project Liaison Board: Messrs Raj. T. Bharadwaja, MTAB, Homer Chen and T. J. O'Donnell, WMATA, J. F. Delaney, PATH, M. A. Fateh, UMTA, C. S. Green, NYCTA, V. P.

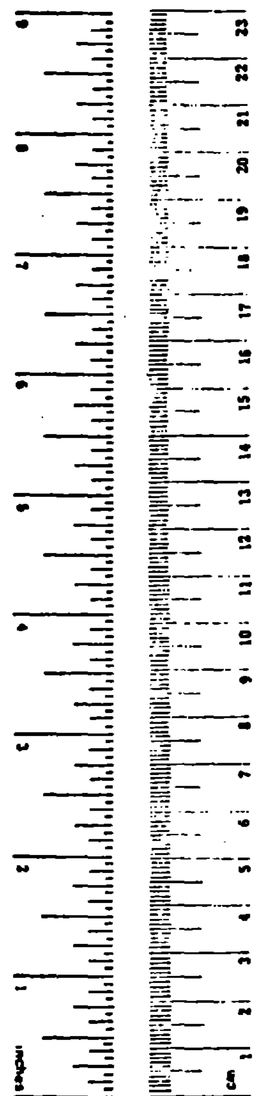
Mahon, BART, C. T. McGinley, MARTA, C. L. Standord, SEPTA, and P. E. Swanson, CTA. As Chairman of the Track Subcommittee of APTA, Mr. E. A. Tillman, associate vice president of Daniel, Mann, Johnson & Mendenhall, was most generous with his knowledge and time. A very large and accurate amount of information was provided on design, maintenance and lubrication by Mr. E. E. MacPhail, manager plant, Mr. R. I. Kingston and other engineers of the Toronto Transit Commission.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

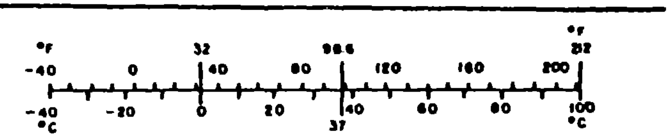
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. SUMMARY

1.1 PROBLEM

Restraining rail is installed close to the gage side of the low running rail in transit track curves to reduce the wear of the high rail and guard against derailments. It does this by contacting the backs of the wheels and guiding them through the curves. The design, installation and maintenance of restraining rail follow practices that were developed for the guards used at frogs and switch points where the conditions of use are generally more severe than in curves.

There is general consensus that restraining rail is desirable in curves to about a 500-foot radius, and in longer radius curves that show excessive wear or tendencies of car wheels to climb the high rail. Opinions vary as to the value of restraining rail and the cost effectiveness of practices followed in its installation and maintenance. Information collected by members of the Track Construction and Maintenance Subcommittee of the Ways and Structures Committee of the American Public Transit Association (APTA), indicated that opportunities could be found for improvements in restraining rail practices. As a result, APTA recommended that the Urban Mass Transportation Administration (UMTA) include a study of restraining rail in its research and development programs.

1.2 PROJECT SCOPE

The general objective of the project, as stated by the Transportation Systems Center (TSC), was to develop information to assist the design, analysis, maintenance and operation of transit track of increased integrity and safety. Specifically, it was to provide criteria for the cost-effective use of restraining rails in reducing wear of rail and track components in curves.

and turnouts, as well as enhancing safety. It included development of guidelines, analysis of significant factors and development of concepts for improvements in design and maintenance.

1.3 PROJECT WORK

The literature was studied, starting with a review of data from a comprehensive questionnaire initiated and compiled by members of the APTA Subcommittee. Practices in the installation of restraining rails, their maintenance and lubrication, and the operation of cars in curve track were observed and discussed with transit engineers; and the conditions of components were examined in relation to their functions and maintenance practices.

Great variations were seen in local factors that affect cost, such as wage rates, availability of skills and shipping distances. Accordingly a cost base was chosen which varies with the local factors. This is the cost of replacing a standardized length of the high rail -- the rail which a restraining rail is installed to protect. Alternatives in the maintenance and replacement of curve track were analyzed, and their costs were estimated as percentages of the cost base.

Environmental effects were observed and discussed, and the only significant ones found were noise and the drip of excess lubricant from elevated structures. Limited measurements made during tests of lubrication effects indicated an average peak-noise reduction of 20 dBA when cars were guided through 150-foot radius curves by a lubricated restraining rail, versus an unlubricated high rail.

Examination of components showed that the stresses developed were well within their capacities, except where looseness, corrosion and wear had occurred over a long time. Potential opportunities for reducing costs were apparent in some of the components. These include the heavy steel chocks used where restraining rail is bolted to the low rail, threaded connectors, new Tee rail used for restraining rails, the sheared flange of a Tee rail used as a horizontal restraining rail, and existing fastener designs used for new requirements.

The effects of lubrication were tested at the Transportation Test Center (TTC), where roll out distances and corresponding time intervals were measured for a transit car coasting (drift mode) over curve track. The tests included 12 track conditions varying from no restraining rail and an unlubricated high rail, to a heavily lubricated restraining rail. The data obtained were widely scattered, indicating wide variations in friction and lateral forces, even at low speed. Similar indications have been seen in the variations in rail wear found in some curves. Overall test averages showed a reduction of 67% in drift resistance obtained by installing and lubricating a restraining rail which kept wheels from flanging on the high rail. This indicates that a lubricated restraining rail would have a useful life three times that of a dry high rail. In addition, the useful life of a restraining rail is extended by the relatively large area over which the wear is distributed.

Two design concepts were developed. One, for slab track with widely spaced fasteners, consists of a structural channel with a wear-resistant rubbing strip fastened to an adjustable brace. The second, for ballasted track, consists of a flat bar of wear-resistant steel on an adjustable brace which can be installed on existing ties, and which would transfer lateral forces to the base of the running rail. In each concept, folded, welded plates would be used instead of castings; lateral resiliency could be provided by an elastic pad or spring clips; and the number of threaded connectors would be minimized.

The guidelines developed from the study are printed as a separate report to facilitate their use.

1.4 CONCLUSIONS

Restraining rails are desirable in mainline curves to a 500-foot radius and in longer radius curves that show excessive wear or tendencies of car wheels to climb the high rail. The value of a restraining rail depends largely on its effects in reducing the wear of the high rail.

Tests of lubrication effects in 150-foot radius curves and consideration of the relatively large area of a restraining rail contacted by a wheel, indicate that its useful life should be three to four times that of an unlubricated high rail used alone.

The cost of fasteners for vertical restraining rails may be reduced and the useful life of the bolts increased, by using light spacers that will not transfer vertical loads between the rails, instead of large chocks.

Worn restraining rail is suitable for the loads encountered in smooth curves. It could be relaid to reduce costs in cases where the flange does not have to be sheared.

Redesign of the facing end of a restraining rail installation would reduce damage in the unlikely event that an approaching car were to derail.

Information is not available to predict the irregular dynamic behavior of cars in curves; and some effective actions to reduce wear and energy requirements (such as setting the optimum height for the restraining rail) cannot be determined. Needed information could be obtained from instrumented tests and studies of the dynamic characteristics of transit cars.

Implementation of the Guidelines will be assisted by the development of low-cost automatic inspection equipment, which is being undertaken in support of track maintenance.

The concepts developed under the project provide opportunities for new cost-effective designs.

1.5 RECOMMENDATIONS

Detailed tests should be conducted to obtain additional information for improving the Guidelines (such as the optimum height of a restraining rail) and for improving practices to reduce rail wear in curves.

The suggested modification to the connectors for vertical restraining rail should be designed, fabricated and tested; and other modifications to components with potential for improving cost effectiveness should be investigated.

Designs should be developed for the facing end of a restraining rail to reduce potential damage in the remote chance that cars derail while approaching a restraining rail.

The design concepts presented in this report should be investigated, final designs completed for the two most favorable, and prototypes fabricated and tested in laboratories and in service.

2. BACKGROUND

2.1 USE OF RESTRAINING RAILS

Transit cars are guided by the rails on which they ride. In a curve, the high (outside) rail forces the cars to change direction by its reaction against the leading wheel of each truck. This effect is aided by the superelevation of the outside rail and the inward cant of the rail acting against the tread of the wheel. The wheel tread is often tapered so that it can negotiate curves of long radius at low speed without the wheel flange coming into contact with the high rail. In sharp curves, however, unless a restraining rail is installed, most of the force to turn the cars comes from the gage side of the high rail reacting against the wheel flanges. This results in rapid wear of the high rail.

Rail wear has caused great expense to many Transit Properties that have sharp curves where necessary to keep tracks within available urban rights-of-way and avoid obstacles such as building foundations, bridge approaches and tunnels. In areas where very sharp curves are subject to frequent traffic, the high rail has had to be replaced as often as every nine months. In such cases, everything feasible is done to reduce the rate of rail wear, including the installation of restraining rails.

A restraining rail is a guardrail installed close to the gage side of the low running rail in a curve, where it provides a flangeway and contacts the backs of the wheels to help guide the car around the curve. The functions of restraining rails are similar to those of the short guards at switch points and frogs used in both railroad and transit track -- to reduce wear and increase protection against derailment. Other benefits, such as reducing flange squeal and reducing the wear of wheel flanges, are sometimes attributed to restraining rail.

Restraining rails have been used in transit tracks, and in light rail or street car tracks, since the construction of the earliest systems. In the late nineteenth century, they were commonly used in railroad tracks also, since the locomotives then in service were likely to derail on sharp curves because of their long wheelbases. Later, when movable trucks were introduced and wheelbases were shortened, the railroads gradually eliminated restraining rails. Most transit systems, however, have sharp curves and other conditions for which restraining rails are advantageous.

Transit cars have several characteristics that give them a greater tendency towards wheel climb than railroad cars have. When unpowered railroad cars negotiate curves, the drawbar pull actually has a lateral component towards the lower side of the curve*. This is not true of self-propelled transit cars. The powered transit car wheels tend to scuff the rails and exert an additional downward force on the high rails in curves, and the reaction to this force is thought to increase any tendency to climb the rail.

Another transit car characteristic is the relatively large mass of a truck and traction motor assembly. Its high inertia requires a larger lateral force than the truck of a railroad car does to turn it in a curve.

For transit systems, the restraining rails usually have been designed with a Tee rail section similar to that used for the running rails. The types and sizes of bolts and other fastener components are then selected that have given good service in turnouts. This has been a reasonable and safe practice, as the load environment in a turnout is generally more severe than in

*One exception is when a train is put into buff by heavy braking of lead locomotives.

a smooth curve. With few exceptions, the components used have been adequate for the maximum stresses that have developed. Exceptions were found where the components were fastened rigidly to structural concrete, and large stress concentrations resulted in cracks and broken bolts. Other failures of components were seen where tracks had deteriorated over many years, and the fasteners were loose, worn and corroded.

The Tee-shaped restraining rails may be installed in either the vertical position, as shown in Figure 1, or the horizontal position, as shown in Figures 2 and 3. The horizontal position provides greater stiffness to resist the lateral force of car wheels on the rails. This is valuable where the rail fasteners are relatively far apart. The horizontal restraining rail is usually lighter weight than the running rails in the track.

In some installations, the restraining rail is fastened tightly in the track without provision for adjustment as shown in Figure 1. In these installations, the rate of wear of the restraining rail controls the rate of wear of the high rail. When the surface of the low rail also wears slowly at about the same rate, all three rails can be replaced eventually in a single, efficient operation. In the meantime, the tight fastenings require little or no maintenance.

As shown in Figure 2, some installations have adjustable fasteners supporting the restraining rail. In these cases, the restraining rail can be moved towards the low rail at intervals after wear has occurred, to prevent car wheels from flanging on the high rail.

In most transit systems, restraining rails are lubricated to reduce friction and the resulting wear and noise. Lubrication is also thought to smooth the movement of the wheels through

Fig. 1

Vertical Restraining Rail

In Ballasted Track

Extra bolts and spacers are used at the joint in the restraining rail to fasten it to the running rail.

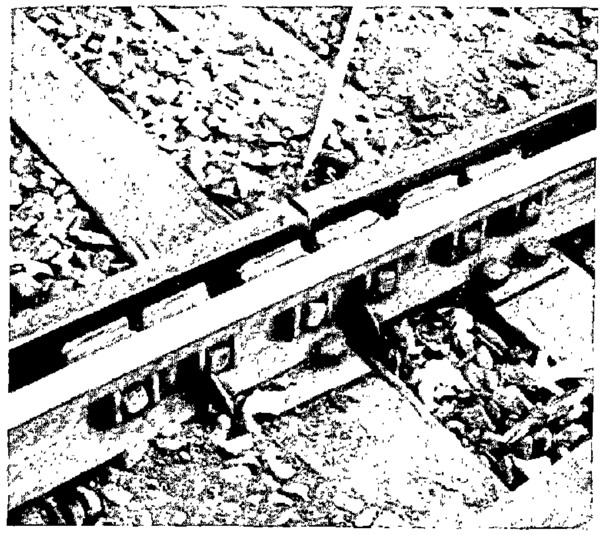


Fig. 2

Horizontal Restraining

Rail in Ballasted Track

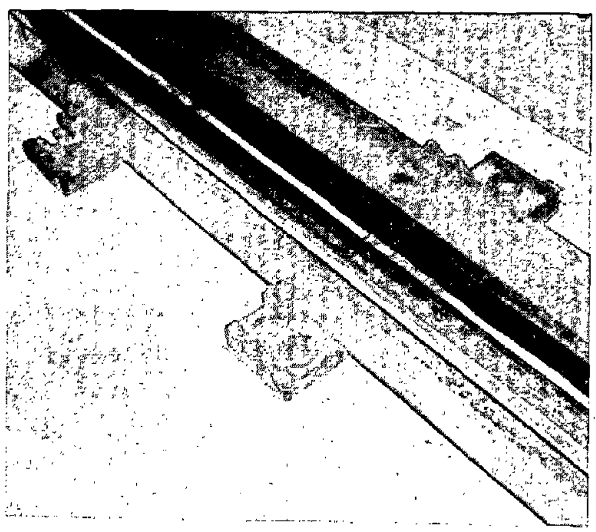
The braces that support the restraining rail are installed at alternate ties, a spacing of 48 inches. The restraining rail can be adjusted by installing shims between it and the braces.

Fig. 3

Horizontal Restraining

Rail in Concrete Slab

A restraining rail fastener, at the upper right, is installed at alternate fasteners for the running rail. This gives a spacing of 60 inches center to center.



curves, so as to improve ride quality slightly. However it adds to the work of routine track maintenance.

Restraining rails are generally lubricated by automatic devices installed in the track and activated by pressure from the car wheels. Manual lubrication is also common, particularly as a temporary measure when an automatic lubricator is awaiting repair or where tracks are used infrequently.

Criteria for the design, installation and maintenance of restraining rails vary among the Transit Properties, based upon their experience with the characteristics of the curves in their tracks and the dynamic responses of their particular cars when operated through the curves. At present, restraining rails are installed in sharp curves to upper limits of radius ranging from 470 to 1500 feet at different Transit Properties.

Along with its benefits, restraining rail has been noted to have several disadvantages. These include: the initial costs of the extra rail and fasteners, interference with tamping of ballasted track, additional routine track maintenance and additional work when worn rails have to be replaced. With these disadvantages in mind, the designers of transit systems built recently have successfully minimized requirements for restraining rails by avoiding small-radius curves wherever feasible.

2.2 AMERICAN PUBLIC TRANSIT ASSOCIATION ACTIONS

The American Public Transit Association (APTA) and antecedent organizations have been very effective in sponsoring, producing, or assisting in the collection of data for publications aimed partly at improving track systems and standardizing where economical. Lately the initiation of these efforts has been the responsibility of the Track Construction and Maintenance Subcommittee of the Ways and Structures Committee of APTA. The

APTA publications[1,2,3,4] provide much useful information, and reference 1 has specific information on flangeway widths and gage widening required in curves, but little of the information in it or other track publications applies to the restraining rails used in rapid (heavy rail) transit systems. This gap in information, and the wide variety of practices and problems seen in the use of restraining rails, indicated to many Subcommittee members that opportunities for improvement could be uncovered by systematic investigation.

As a first step, data were collected and compiled on the current criteria and guidelines used for restraining rails by the various Transit Properties[5]. The information obtained was to be used in a determination by the Subcommittee as to the feasibility of developing general standards and guidelines or the need for further investigation.

The compiled information showed a very wide spread in the design and maintenance of restraining rail and in the estimates of costs. After review of the replies to the questionnaire, the Subcommittee concluded that detailed study of restraining rails would be necessary in order to determine what standards would be desirable. As background information, it was noted that while

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[1] Engineering Manual, American Transit Engineering Association, Rev. 1931.

[2] Subway Environmental Design Handbook - Volume I, Principles and Applications, Associated Engineers Report UMTA-DC-06-0010-74-1, 1975.

[3] Rail Rapid Transit Systems, Technical Reference Report #1, American Public Transit Association, June 1977.

[4] Guidelines for Design of Rapid Transit Facilities, American Public Transit Association, January 1979.

[5] Smith, Roy T., and Wagner, Frank, "Questionnaire on Restraining Rails", American Public Transit Association, May 1976.

restraining rail offers benefits, it also introduces undesirable effects such as:

Increased noise from wheel squeal.
Increased initial construction costs.
Increased maintenance costs.
Problems in changing out worn rails.
Interference with proper ballast tamping.
Development of sharp wheel flanges in some cases.

In September 1977, the Subcommittee submitted to APTA a recommendation for a study of restraining rails accompanied by an outline of the scope of work required to develop guidelines. This was discussed with representatives of the Urban Mass Transportation (UMTA) and the Transportation Test Center (TTC), and it resulted in UMTA sponsoring the current project.

A Liaison Board was formed by APTA shortly after work was initiated on the current project, with the functions of reviewing plans and progress, providing information and making appropriate recommendations. Lubrication was among the items discussed by the Board at its first meeting. The consensus was that lubrication involved more complex problems than had been envisioned in the planned work, and that a more thorough investigation was appropriate, including tests of the effects of lubrication. The Board recommended that criteria and specifications for proper lubrication of restraining rail be developed. As a result, the project was amended to include additional study and tests of lubrication effects at the Screech Loop of the Transportation Test Center.

2.3 RESEARCH NEEDS

Transit track problems and opportunities for significant improvements have been recognized in a series of research studies under the UMTA Urban Rail Construction Technology Program. These studies are intended to increase the performance, reliability and safety of urban rapid transit track systems through improved track performance criteria, more reliable track components and more cost-effective maintenance techniques. The current restraining rail project was initiated as one of the studies. Another, just nearing completion, is an investigation of the suitability of slab track to meet some of the U.S. transit requirements for track at grade.

In discussions of research needs at a Transit Track Systems Seminar sponsored by UMTA and TSC in April 1979, two items on restraining rail were among those proposed by members of a Design and Construction Workshop:

> Study of the use or non-use (effectiveness) of restraining rail.
>
> Study of restraining rails -- how to install them in vertical and horizontal positions; and how to write better specifications with appropriate tolerances.

It was agreed that these studies were desirable. However as they covered work already included in the project under consideration by UMTA, they were not included in the recommendations on track research needs[6].

2.4 PROJECT SCOPE

The Restraining Rail Study was undertaken by the Transportation Systems Center (TSC) in support of the UMTA Office of Rail and Construction Technology, Office of Technology Development and Deployment. The general purpose stated was to develop information to assist the design, analysis, maintenance and operation of transit track of increased integrity and safety. Specifically, the project was planned to provide design criteria for the cost-effective use of restraining rails in reducing wear of rail and other track components in curves and turnouts, as well as enhance safety.

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[6]Cunney, E. G., Boyd, P. L., and Woods, J. A., U.S. Transit Track Assessment and Research Needs, ENSCO, Inc. Report UMTA-MA-06-0100-79-16, December 1979.

The project included consideration of ballasted track, direct fixation track and track installed on open-deck structures. It required the development of guidelines for the optimized use, design and installation of cost-effective restraining rails. To support this requirement, it covered the study and analysis of all significant factors affecting restraining rails, plus the development of concepts for improvements in design and maintenance.

3. DATA COLLECTION

3.1 LITERATURE SEARCH

Only a few reports or papers were found with direct information on restraining rail. These were mostly APTA products discussed in Section 2.2, and surveys[7] made by engineers preparatory to the design of new transit systems. The literature search consisted of a survey of track bibliographies, research records, research bulletins, indices of engineering and scientific journals, transportation trade magazines, railroad and transit manuals, engineering handbooks and other publications containing track-related information. Since there were so few direct references available, the literature search was continued at a low level throughout most of the project in an effort to find useful information in publications on railroad track, tribology and lubrication.

Information on current restraining rail installations was found in the APTA Questionnaire on Restraining Rails[5], and in plans, criteria, maintenance standards and other documents obtained from the Transit Properties. In the general literature of the transportation industries and research, information was found on rail wear, the effects of lubrication on sliding friction and peripheral items; very little of the information, however, is applicable to restraining rail. The dearth of applicable information apparently results from the discontinuance of restraining rails by railroads and by European transit systems. It has been almost a hundred years since American railroads, which then had long-wheelbase locomotives, made extensive use of restraining rails.

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[7] Dunn, R. H., et.al., Trackwork Study Volume 1, Trackwork Practices of North American Rapid Transit Systems, DeLeuw Cather and Co. Report WMATA-DCCO-TWS-1, Nov. 1967.

[5] Smith and Wagner, p. 11.

Useful information on rail wear and its proximate causes, which can be related to restraining rail, was found in a paper by King[8], and information from the Facility for Accelerated Service Testing at TTC[9,10]. A clear and concise introduction to wear processes in general and some effects of lubricants has been provided by Colangelo and Heiser[11]. Detailed information on sliding friction, wear and lubrication effects, is available in the classic volume by Rabinowicz[12]; and, on lubrication in the Handbook of Lubrication[13]. Other publications with information relevant to the project are referred to where appropriate in the text.

3.2 INFORMATION FROM TRANSIT PROPERTIES AND MANUFACTURERS

Concurrent with the literature search, visits (coordinated through UMTA and APTA) were made to nine of the U.S. Transit Properties and the Toronto Transit Commission. Properties using restraining rail in curves as well as in turnouts were visited in order to discuss benefits and problems in their restraining rails; obtain plans and criteria; observe train

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[8] King, F. E., "Rail Wear and Corrguation Problems Related to Unit Train Operations: Causes and Remedial Actions", Paper, 12th Annual Railroad Engineering Conference Proceedings, FRA Report ORCD-76-243, October 1976.

[9] "Preliminary Evaluation of Locomotive Wheel Flange Lubrication", Technical Note FAST/TTC/TN 79-18, July 1979.

[10] Evaluation of Rail Wear at the Facility for Accelerated Service Testing", Technical Note FAST/TTC/TN 80-04, March 1980.

[11] Colangelo, V. J. and Heiser, F. A., Analysis of Metallurgical Failures, John Wiley & Sons, 1974.

[12] Rabinowicz, Ernest, Friction and Wear of Materials, John Wiley & Sons, 1965.

[13] Standard Handbook of Lubrication Engineering, O'Connor, J. J., Editor-in-Chief, McGraw-Hill, 1968.

operations over curves with and without restraining rails; and to examine rail conditions closely, and measure samples of rail wear in some 87 curves selected as representative by the transit engineers.

In most cases, where traffic and other conditions permitted, measurements of wear were made in the spiral and body of the curve with a small pantograph device that provided an approximate profile of the railhead cross section. Crosslevel, gage and flangeway width were measured at the same locations with a combination level-gage. Information on the major curve characteristics was obtained from the transit engineers. This included length, radius, type of rail, time in service, estimated useful life, fastener data and other details. In many cases, the information was verified from track charts and records provided at the time of a visit or in later correspondence. In a few cases, records were not available, and the data were provided from memory and observations of conditions. Relevant information on car characteristics was obtained mostly in later correspondence with the transit engineers and from an APTA compilation[14].

The data obtained were sorted, and those that appeared most significant were tabulated, as shown in Appendix A, in an effort to identify patterns and dominant factors affecting rail wear. Unfortunately the effects of many important factors (such as curve radius, weight of car, traffic density and operating speeds and modes) appear so intermingled as to preclude the development of valid weighting. Many of the data are suspect because they were obtained from memories or estimates. Many of

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[14] Roster of North American Rapid Transit Cars, 1974-1976, American Public Transit Association Report No. UMTA-DC-06-02121-79-1, Jan. 1977.

the traffic speed data seem doubtful, as the accelerations and braking felt in many cars running through curves indicated that they were not always operated smoothly at planned speeds. In some cases, remote control systems did not seem to provide smooth changes in car velocities.

Data on the useful life of rails were weakened by the sensible practice of replacing a worn rail at a time when the work is convenient and seems most cost effective. This is reasonable, because a worn rail does not suddenly become hazardous or useless, and it may be continued in use until weather and other conditions favor the replacement work.

Costs and environmental factors were discussed during the visits to the properties, and additional information was obtained by telephone and correspondence. Most of the information on costs was incomplete and related so closely to local practices and conditions that it could not be generalized. At two properties, however, sufficient detailed records were available to provide a sound base from which to project the costs of restraining rails versus replacement of high rails in curves.

Lubrication practices and costs were discussed in detail, and lubrication was observed closely in the tracks visited. Additional information was obtained from the manufacturers of lubricants and lubricating devices. Manufacturers were queried also for information on components used in transit track, and good data were found in their catalogues and comments on practices and costs.

3.3 LUBRICATION TESTS

Tests of the effects of lubrication on the drift resistance of a transit car were conducted at the Transportation Test Center (TTC) in accordance with a test plan approved by TTC and TSC. The State-of-the-Art-Transit-Car was provided and operated by TTC for the tests. TTC personnel collected data on rail forces, noise and weather; while ENSCO personnel collected data on car travel distance and speed, and limited data on noise.

Prior to the tests, track geometry was surveyed in the four test zones indicated in Figure 4 in order to find any deviations from smooth conditions that could affect test results. The coefficient of sliding friction was measured on the rails, with and without lubrication, with a device that was pulled to

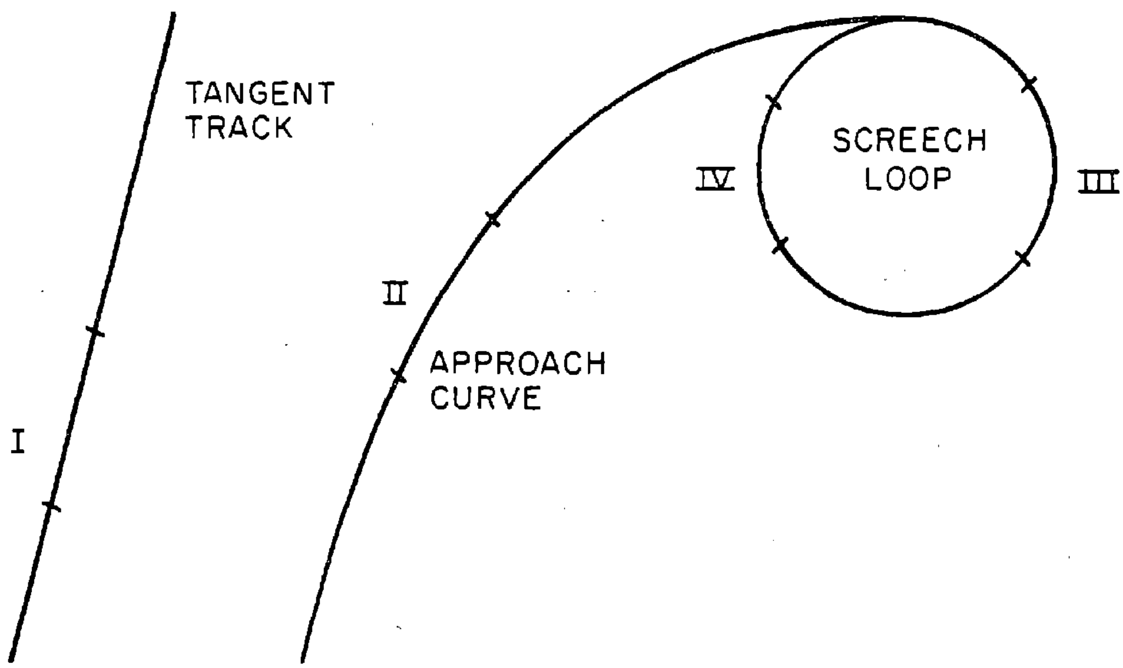


Fig. 4
Test Zones

slide along the rail. Water spray nozzles were installed on the car to wet the rail with water for certain tests, and a small rail-lubricating device was used to obtain a uniform difference between light lubrication for certain tests and heavy lubrication for others.

Strain gage arrays and recording equipment were installed at two locations in Test Zones II and IV for measurement by TTC personnel of forces on rails, concurrent with the drift resistance tests.

Each of the test zones was divided into sections, as indicated in Figure 5 for Zone II, so that data could be obtained from low speed runs in which the car would stop within a test zone.

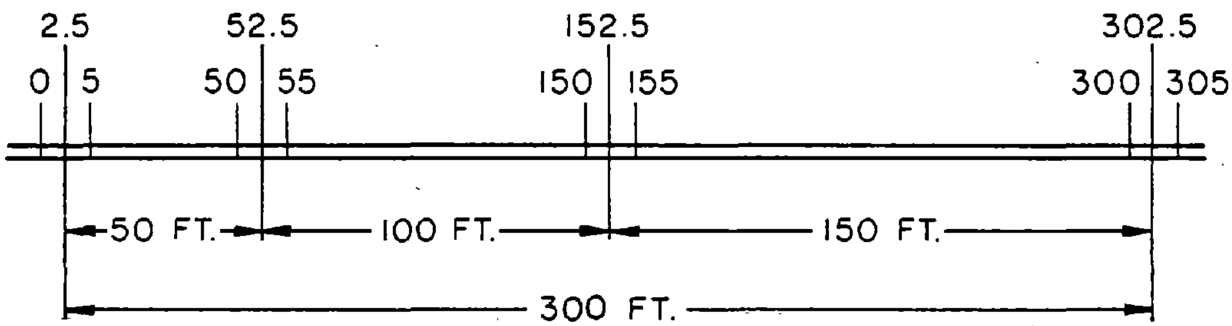


Fig. 5
Diagram of Test Zone II, Curve Track.

The beginning and end of each section of the test zones were marked with speed traps. Each trap consisted of a pair of short reflective tapes installed on the field side of the rail and separated by a distance equivalent to a track centerline distance of 5.000 feet.

To meet the requirements of the restraining rail study, a system to measure mean speed of the test vehicle was developed by ENSCO. This speed measurement system consisted of a precision timer; passive, track-mounted, optical reflectors; and a microcomputer provided by ENSCO. The precision timer was used to collect data based on signals generated by the movement of the car over the optical reflectors. These data, which provide elapsed time, were sent to the microcomputer and used as the basis for the computation of mean vehicle speed. The microcomputer used was a modified Level I, TRS-80; the necessary computations were performed by a program written in BASIC.

The mean speed measurement system had two displays. The first was a six digit display located on the precision timer, which provided elapsed time in seconds. The second display was in the microcomputer and provided mean vehicle speed in miles per hour. From this display, the operator recorded the mean vehicle speed computed between the two adjacent reflective tapes in each of the speed traps.

The tests are summarized in Table 1. The three tests, indicated in a test zone for each test condition, have three different entry speeds into the zone: approximately 15, 10 and 5 miles per hour. The maximum entry speed was limited to 15 miles to assure safe operation through the small, level turnout between Test Zone II and the Screech Loop.

The State-of-the-Art car that was used in the tests is shown in Figure 6 on the 150-foot radius Screech Loop at TTC. Figures 7, 8 and 9 show car wheels in relation to the restraining rail, while Figure 10 shows a car wheel on the high rail of the Screech Loop. These figures also show the nozzles through

TABLE 1. DRIFT RESISTANCE TESTS

Test Series Condition	No.	Total Number of Tests In Zone Tangent	Transition	1/2" Elev	1-1/2" Elev
Dry Rail No Restraining Rail	1	3	3	3	3
Water Spray No Restraining Rail	2	3	0	3	3
Lubricant No. 1, Lt. No Restraining Rail	3	0	3	3	3
Lubricant No. 1, Lt. w/water spray No Restraining Rail	4	0	0	3	3
Lubricant No. 1, Hvy. No Restraining Rail	5	0	3	3	3
Dry Rail w/Restraining Rail	6	0	0	3	3
Water Spray w/Restraining Rail	7	0	0	3	3
Lubricant No. 1, Lt. w/Restraining Rail	8	0	0	3	3
Lubricant No. 1, Lt. Water Spray w/Restraining Rail	9	0	0	3	3
Lubricant No. 1, Hvy w/Restraining Rail	10	0	0	3	3
Lubricant No. 2, Lt. w/Restraining Rail	11	0	0	3	3
Lubricant No. 2, Hvy. w/Restraining Rail	12	0	0	3	3

which water and detergent were sprayed on the rails for test conditions 2, 4, 7 and 9. For comparison, a heavily lubricated restraining rail in a subway track is shown in Figure 11.

The tests on tangent track in Test Zone I were run in two directions. This was done in order to detect the effects of track grade and indications of other factors, as well as the normal drift resistance that could be expected for the car when operated on a level, tangent track. This drift resistance for tangent track provides the baseline representing all of the resistance not attributable to track curvature.

Fig. 6

Test Car

The State-of-the-Art Test Car is shown on the Screech Loop at TTC. It is headed in the counter clockwise direction in which all the tests in curve track were run.

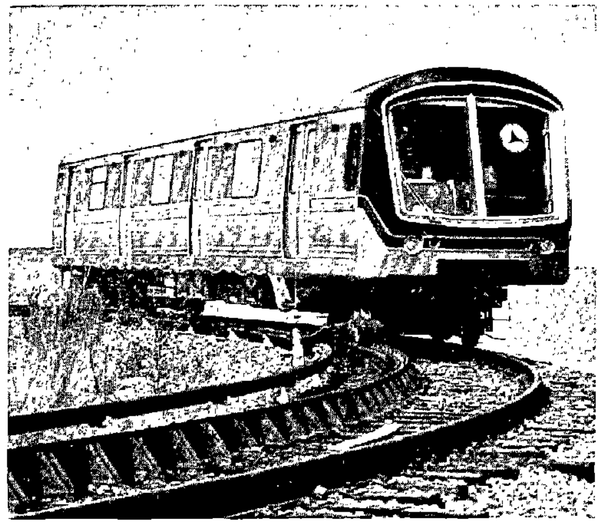


Fig. 7

Wheel in Flange Contact With the Restraining Rail

The car wheel is shown in static contact with the restraining rail before the tests were run. The nozzle is part of the applicator system for water sprays.

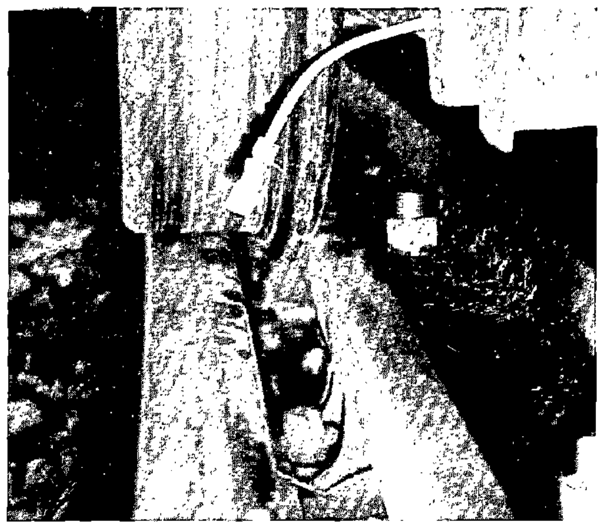


Fig. 8

Restraining Rail Braces

The rail braces shown in this picture are designed for adjustment by loosening the bolts, shifting the rail to correct position, inserting shims between the rail base and the brace, and tightening bolts.

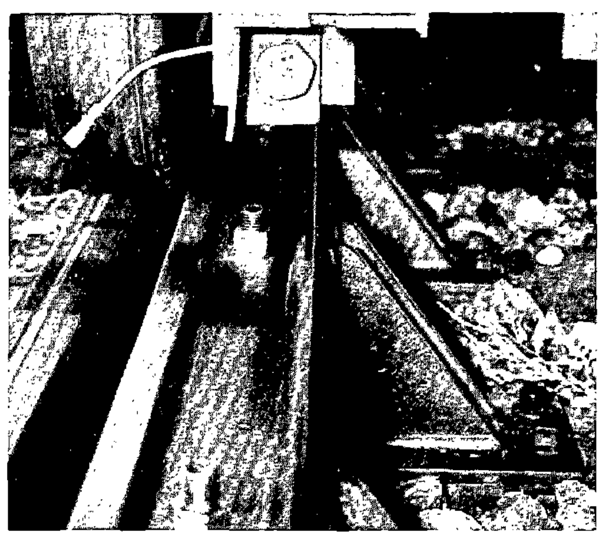


Fig. 9

Restraining Rail Joint

A bolted joint is shown in the restraining rail in this picture. The running rail in the foreground has welded joints.

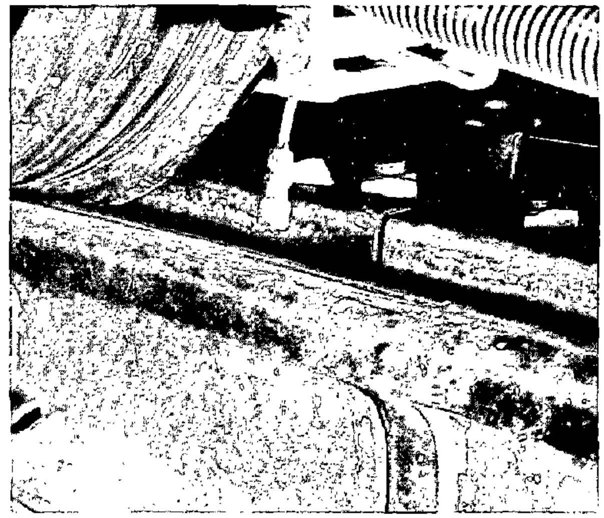


Fig. 10

Wheel on High Rail

A wheel is shown on the high rail with its flange well clear of the gage side of the rail. It is prevented from flanging by the restraining rail contact at the opposite wheel.

Fig. 11

Restraining Rail in

Subway Track

Both rails in this track have bolted joints which are staggered. The lubricated restraining rail has worn so that the near edge is no longer rounded.

Noise was measured with a hand-held sound level meter in Test Zones III and IV at the highest operating speed under each of the 12 test conditions. The meter was held in the same locations at the start of each test zone, at approximately the same distance from the track, same attitude and angle to the track for each measurement. Environmental factors such as wind velocity and direction varied, however, as the measurements had to be made at different times on different days.

Tests were begun on 22 September. There were two scheduled interruptions of testing, one for steam cleaning of the rails after tests in which lubricants were used, and one for installation of the restraining rail in Test Zone III and IV. All tests were completed on 4 November.

The drift resistance data from the tests were reduced by converting measured distances and times into entry speeds, average speeds, time intervals and average drift resistance forces. These were tabulated for the various test conditions under each of the four test zones as shown in Appendix B. Noise data are listed in Table 5, Section 5.2.

4. CURRENT PRACTICES

4.1 INSTALLATION

The APTA Questionnaire[5] of 1976 provides a comprehensive summary of current practices. Very little change has occurred since this survey of restraining rail was completed, except in the design of new systems where the use of restraining rails is avoided (other than in turnouts and crossings) by the design of long-radius curves. A few more automatic lubricators have been installed to replace manual lubrication, and more are planned. Restraining rails have been installed in several additional curves where the high rails showed very high rates of wear without them, and in some where the wear pattern indicated a tendency of wheels to climb the high rail.

The upper limits of curve radius for the installation of restraining rails range from 470 to 1500 feet at different Transit Properties; and, where the rate of wear justifies it, to 2600 feet at one Property. Restraining rails are generally considered desirable in sharp curves, because they limit the movement of cars towards the outside of curves, so that the tendency of powered wheels to climb the high rails is resisted. Requirements vary among the properties, however, based on their observations of variations in rail wear and other factors. Economy is a major consideration. Use of a restraining rail is generally justified, if the value of the savings obtained by reducing the wear of the high rail exceed the costs of the restraining rail.

Standard Tee rails are used for restraining rails. They are installed in either the vertical or horizontal position according to design practices that have been standardized by individual

[5]Smith and Wagner, p. 11

Transit Properties based on local experience. Tee rails are readily available, including worn rails that have been removed from track and which would be suitable for restraining rail service. Regular carbon steel rails have been used as well as control-cooled rails, heat-treated rails and premium alloy steel rails, with little difference reported in the rates of wear. The heat-treated rails are, in effect, spring steel and are very difficult to bend. Tee rails are generally bent to the curvature of the track by the supplier or the property for curves up to a 500-foot radius, when the rails are used in the horizontal position, and to a 300-foot radius when they are used in the vertical position. On long-radius curves, straight rails are usually held in the curve position by the fasteners. Bolted joints are used in most cases, and rails that are bent in track have a tendency to straighten near a joint, bending the joint bars and causing an alignment irregularity.

Components for the installation of the Tee rails as restraining rails are readily available. They appear fully adequate for the service since they are the same components used by the Transit Properties and railroads to fasten guards in turnouts where the service is generally more severe than in smooth curves. Components that were seen to be damaged appeared to have flaws in them that had produced stress concentrations, or to have been left loose or poorly supported by ties, so that they had deteriorated over long periods of hard use.

The restraining rail is commonly installed in a vertical position in ballasted track where it is bolted to the running rail as shown in Figures 1 and 2. Large, adjustable C clamps are used in a few of the older installations, as shown in Figure 12, but these have become very expensive. Special braces that mount on tie plates, such as those shown in Figure 13 and others somewhat similar, are used in many cases. Such braces do not require drilling the rails for through bolts and are thought to simplify the work of replacing worn rails. In all cases, the vertical restraining rails facilitate connections to standard guards at turnouts and crossings as shown in Figures 13 and 14.

Horizontal restraining rails are used in slab track, as shown in Figures 3 and 15. Here, their lateral stiffness in the horizontal position and the strength of the fasteners to the slab permit long intervals between fasteners, so that few are required. They are also used in many cases with wood ties. Wherever used, the horizontal restraining rails require special fastenings to join them to the standard vertical guards in special trackwork, such as the heavy castings shown in Figures 16 and 17.

Where clearances permit, restraining rails are installed from 0.25 to 0.50 inch higher than the low rail. This height differential increases as the surface of the low rail wears under traffic. The added height of the restraining rail is considered to provide increased resistance to wheel climb. It also provides additional area of the rail to contact the wheels and resist frictional wear. Observation of the wear areas of rails in active track indicated that restraining rails, installed at heights above the low rails, had wear areas 50% or more bigger than the wear areas of the high rails in similar track.

Adjustable fasteners for restraining rails (as shown in Figures 2 and 12) are favored by many transit engineers, as the rails can be moved to compensate for wear and thus keep car wheels from flanging on the high rails. This is advantageous in cases where the flange of the high rail is not lubricated, since the migration of lubricant to the running surface of the rail would interfere seriously with braking and traction. However, it has been noted that newly installed restraining rail often does not contact wheels until the high rail has worn to some degree.

Fig. 12

Heavy Duty Clamps Holding Restraining Rail

The adjustable wedge between the large C clamp (yoke) and the rail, holds both rails firmly against adjustable spacer blocks. The wedge is secured with a vertical pin.

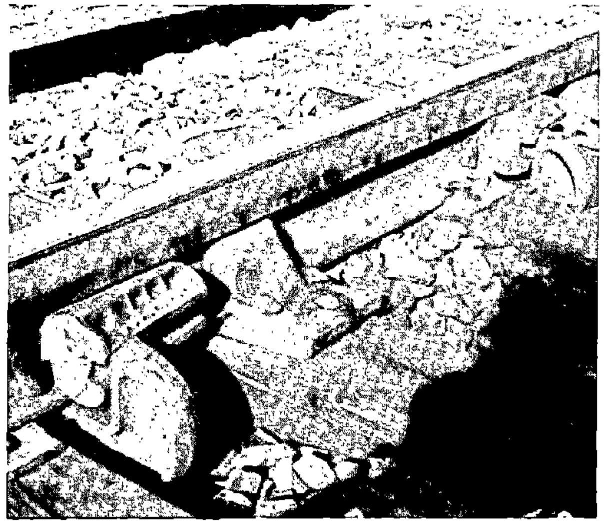


Fig. 13

Braces Mounted on Large Tie Plates

Less costly than the heavy duty clamp, the braces provide a simpler installation but are not adjustable.

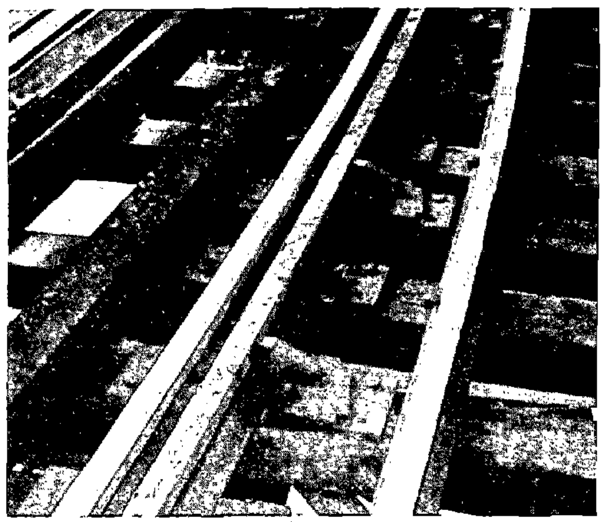


Fig. 14

Vertical Restraining Rail Adjoining Special Trackwork

The restraining rails are bolted to the running rails where spacer blocks can be seen between them. The connections to the wing rails of the frog in the center are simple plates and bolts.

Fig. 15

Horizontal Restraining Rail on Braces in Concrete Slab Track

The rail is fastened to the brace by a vertical bolt through its web. Adjustments are made by loosening the bolts and inserting shims between the near side (base) of the rail and the base.

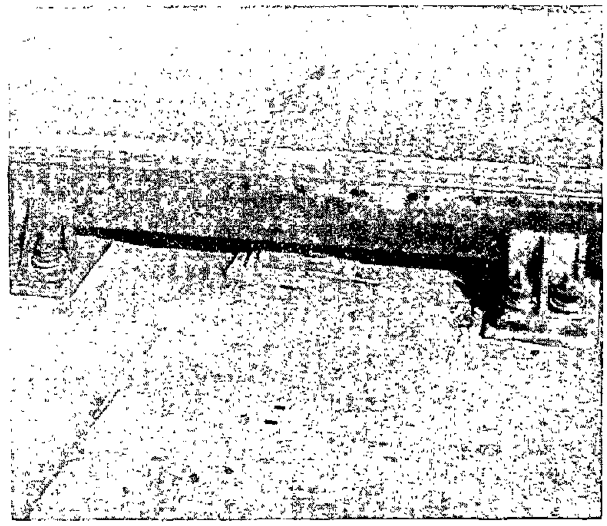


Fig. 16

Restraining Rail Connection

The connection of a horizontal restraining rail to a vertical guardrail in a turnout requires large, special fittings.

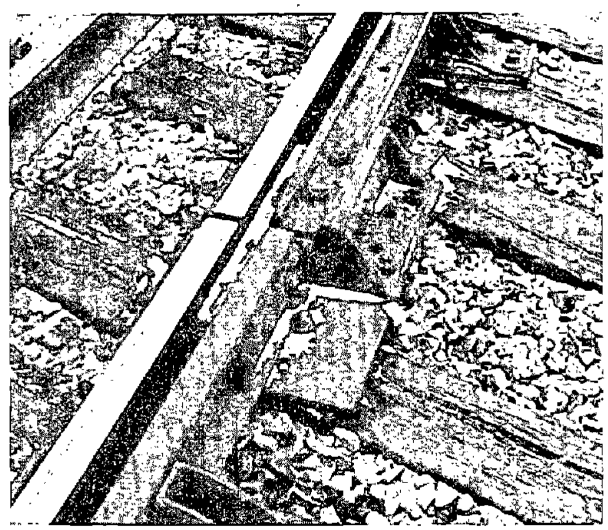
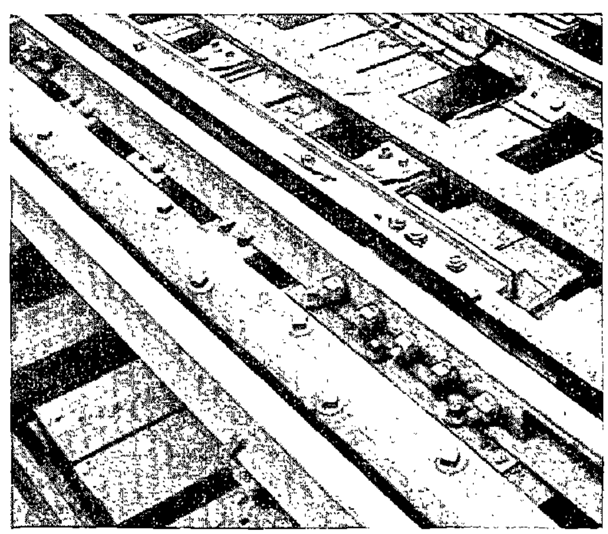


Fig. 17

Restraining Rail Connection in Elevated Track

The connection of a horizontal restraining rail to a vertical guard on an open-deck structure uses large fittings similar to those shown above for ballasted track.



4.2 MAINTENANCE

Preventive maintenance consists of regular inspection of the rails and fittings by a track walker who tightens loose nuts and replaces broken lock washers and bolts as well. Other worn or broken components are replaced and adjustments are made by track maintenance crews. In some cases, threads are lubricated and nuts are tightened to specified torques.

Rail wear is observed and measured. As the wear approaches levels set by the Property, restraining rail with adjustable fasteners is moved towards the low rail to compensate for wear and keep wheels from flanging on the high rail. Rail replacement is planned as the wear approaches limits set by the Property. Rails are sometimes replaced long before the wear on them makes replacement necessary. In a few cases, the maintenance is behind the workload, a condition attributed to shortages of manpower and skills; and loose bolts (Figure 18) and poor support under the rails (Figure 19) have contributed to rapid wear and damage of the fastenings. The bolts that hold the restraining rail and low rail together have often broken under such conditions. One Property has eliminated this problem by using 1.25-inch diameter, high strength bolts and improving rail support conditions. Temporary measures, such as bracing the rails against a tunnel wall where fasteners have shifted (Figure 20) are used until standard repairs can be made.

The bolted connections of vertical restraining rails to running rails, and many of the braces used, do not permit adjustments of the rail. When these fasteners are used, the rate of wear of the restraining rail governs the rate of wear of the high rail, so that the two require replacement about the same time. Where surface wear occurs on the low rail at about the same rate as the gage wear on the high rail, a practice has developed of

Fig. 18

Bolted Restraining Rail

Loose nuts as seen near the center of the picture permit the bolts and spacers to move under load. This accelerates wear and increases stress.

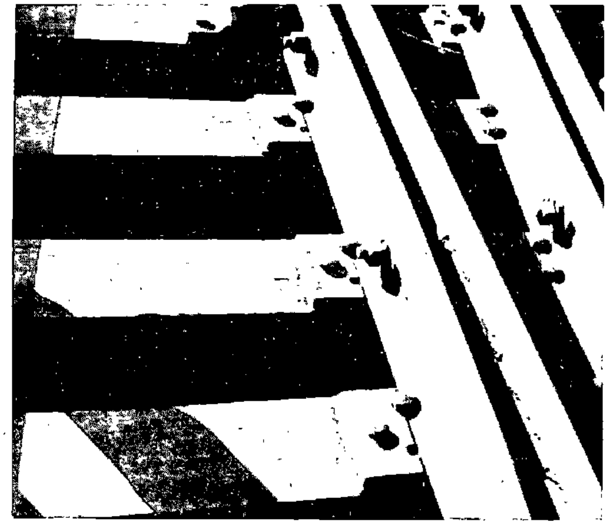


Fig. 19

Mud Pumping

Weak support under the rails results in large forces being transferred through the fasteners, rapid wear and damage.

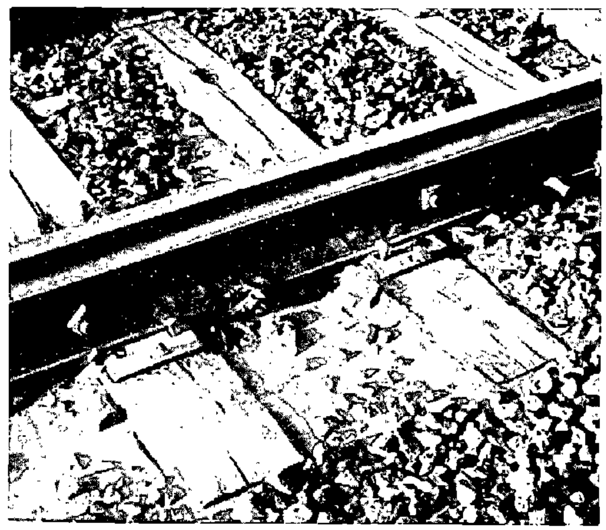


Fig. 20

Temporary Repair

The rail assembly is braced against the tunnel wall in an area where fasteners had loosened, to prevent the stiff restraining rail from straightening part of the curve.

shearing, bending, drilling and bolting the restraining and low rails together in a shop; and handling them as an assembly in work on track. Replacing the three rails at the same time is favored, because it increases the efficiency of track maintenance and reduces the frequency of on-track maintenance work.

Less surface wear is tolerated on a rail with a restraining rail fastened to it than would be permitted for a low rail on a curve without a restraining rail. The low rail fastened to a restraining rail is seldom permitted to wear down to the point that wheel flanges begin to hit the spacer blocks bolted between the rails. Such impacts produce a very unpleasant rumble in the cars and increase the wear of wheels and bearings as well as fastener components.

Lubrication is a major part of the work and cost of maintaining restraining rails, and usually it is the most effective effort made to reduce noise and wear. Because of its importance, it is discussed separately in the next subsection.

4.3 LUBRICATION

Transit engineers favor the lubrication of all restraining rails in order to reduce both noise and wear. Lubrication of the restraining rail is acceptable in systems where surface contamination of the running rails is considered a serious problem (usually systems where the transit cars have disk brakes). The lubricant applied to a restraining rail is considered unlikely to migrate to the surface of an adjacent running rail, as it contacts only the side of the wheel flange away from the tread.

In transit systems where surface contamination has not caused serious problems, such as cars developing excessive numbers of flat wheels or sliding past passenger boarding locations, the gage side of the high rail also is lubricated. This lubrication is generally limited to curves where high rates of wear and high noise levels are found when the rail is left dry. In cases where surface contamination is not a problem, the high rails are lubricated wherever restraining rails are installed.

In systems where surface contamination would be troublesome, transit engineers often apply very small amounts of lubricant to the gage side of high rails that are particularly noisy. The frequency and amount of lubricant are related by experience to the ambient humidity which often has large effects on noise generation.

Interference with electrical or signal circuits was mentioned in only two cases. In one, use of a lubricant containing molybdenum disulphide was thought to cause problems with the electrical circuits, and its use has been discontinued. In the other case, water spray on noisy curves in yard track has interfered with signal circuits and is also thought to accelerate the rate of deterioration of embankment under the ballasted track; but the advantages of the water spray are considered to outweigh its disadvantages. Manual lubrication with a grease is substituted only in cold weather when the water spray cannot be used.

High pressure lubricants of the types recommended by manufacturers and used by railroads for flange lubrication are used on most transit track. Preferences are based on experience, local availability and costs. Most of the lubricants used have a calcium base, although some with lithium bases are used. Several have graphite added to the oil, and a few have high pressure (HP) or extreme pressure (EP) additives.

Summer and winter grades are used by most properties; others use all-weather grades of lubricants. These lubricants were found to produce too much smoke in the subways of one of the Transit Properties where a nonsmoking, no-silicone lubricant is now used. While it costs more, a smaller quantity is needed, and overall costs seem to be lower.

Automatic lubricators are installed in the tracks just ahead of a curve (where feasible) that requires lubrication. They are activated by passing wheel flanges. A lubricator may serve one curve or several short curves that are close together. The carriage of lubricant along the rails by car wheels varies greatly. Where the lubricant is wanted, it seems to be carried reliably only a few hundred yards at most. Where drip from elevated track is a problem, the lubricant is sometimes carried miles from the lubricator.

Figure 21 shows an old model mechanical lubricator. With the newer pressure-type lubricator, a drum of lubricant is connected to the equipment; this avoids the work of filling a lubricant reservoir and possible contamination of the lubricant in the process. Several transit engineers have indicated that the pressure-type lubricators require less maintenance and can be adjusted more precisely than the mechanical type, to control the amount of lubricant. Adjustments are made when large seasonal temperature changes occur and, in some cases, when the weather becomes very dry or very humid. Metering a suitable amount of lubricant is a problem because of the variable frequency of traffic and the short and variable lengths of trains, as well as changes in temperature and humidity. Figures 22 and 23 show examples of light and heavy lubrication of restraining rails.

High pressure lubricants are favored for manual application, as well as automatic lubrication. In manual lubrication, a higher viscosity is usually preferred, and lubricants with graphite added are often used. Application of the grease to the rail with a stick is considered to give better control of quantity than the use of a brush does.

Fig. 21

Mechanical Lubricator

This is an old model installed near the facing end of a curve to lubricate wheel flanges that contact the restraining rail and those that contact the high rail.

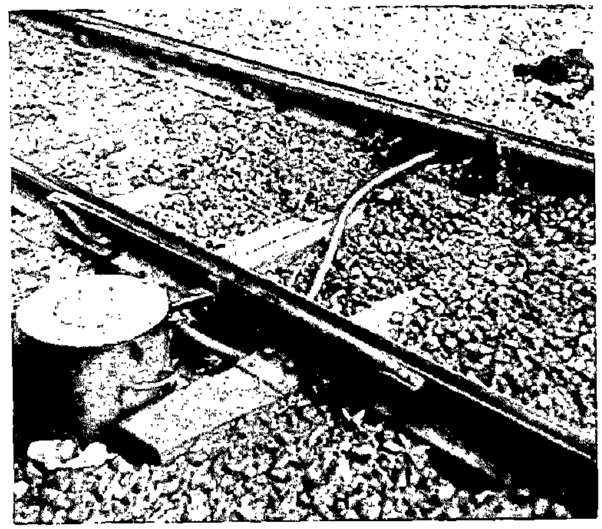


Fig. 22

Lubricated Rail

The horizontal restraining rail shown in this picture is lubricated lightly by a modern mechanical type lubricator.

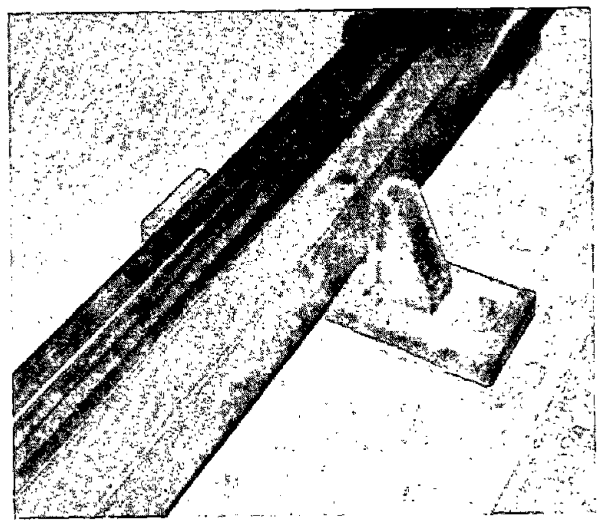


Fig. 23

Heavily Lubricated Rail

The restraining rail is lubricated by an older machine that does not give good quantity control. The adjustable brace is unusual with vertical restraining rail.

A resin-bonded graphite lubricant, such as used on trolley wires, was tried in one case but considered less effective than grease. Car-mounted lubricators are not favored, because servicing and maintaining the large number of units required would add large logistics problems. Experience has shown that equipment added to cars is troublesome. One Transit Property had tried lubricator grease sticks (mounted on trucks and spring loaded to lubricate wheel flanges) but found they were not satisfactory and has discontinued their use.

The opinions of transit engineers varied as to the benefits from lubrication in reducing the rate of wear of restraining rails. Some could see very little benefit in the curves of their systems. The largest estimated benefit was an increase of one-third in the useful life of the rails. Most transit engineers were uncertain and indicated that controlled tests would be necessary to quantify the results of lubrication. None had conducted controlled wear tests, and they were unlikely to have opportunities to do so because of the urgency of current work and large backlogs of track maintenance.

Information in varying amounts was obtained from 16 of the manufacturers of lubricants and lubricators who were contacted. On the whole, they are very positive in stating the advantages of lubrication of running rails. The wide use of rail lubricants by railroad and transit systems was mentioned, and tests were said to have proven the advantages to railroads. The Pennsylvania was said to have extended the life of the low rail on the Horseshoe Curve by a factor of six by lubrication. Claims for extending the life of the high rail were smaller.

Those manufacturers who add graphite (12% or more) to their curve lubricants consider it very important because of the high pressures on the small contact areas between wheels and rails.

They state that the graphite does not squeeze out easily from between the contact surfaces, and the oil acts mainly as a carrier for the graphite.

Benefits claimed by manufacturers from the use of automatic rail lubricators on railroad curves are:

> Greatly prolonged curve rail life - Postpones by years the time when flange wear reaches allowable limit.
>
> Increased low rail life - Decreases head flow by nearly equalizing wheel slippage on both rails.
>
> Wheel screeching eliminated on curves and through switches.
>
> Reducing aligning cost, since trains move more smoothly.
>
> Reduced fuel consumption.
>
> Increased train speeds.
>
> Lower regaging costs because of reduction in flange wear.
>
> Less need for services of helper engines.
>
> Increased tonnage ratings, where curvature governs.
>
> Decreased derailment hazards by reducing the tendency of wheels to climb the rail.
>
> Quadrupled switch point life.
>
> Greatly reduced wheel flange wear.

The AREA Manual[15] supports lubrication by stating that it can extend the life of rail on curves, decrease costs of tires, result in longer service of ties, and decrease train resistance thus permitting less fuel consumption. Hay[16] noted that a reduction of approximately 50% in curve resistance was obtained in tests of lubrication over 168 miles of track with many curves.

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[15] AREA Manual for Railway Engineering, American Railway Engineering Association, 1962 revision.

[16] Hay, W. W., Railroad Engineering Volume 1, John Wiley & Sons, 1953.

4.4 OPERATIONS

Traffic frequency, density, length of trains, maximum wheel loads and other car characteristics vary greatly among the Transit Properties. Most noticeable were the variations in accelerations and braking in curves even on the same property and even where traffic speed was stated to be computer controlled.

More rail wear than expected on curves of medium radius was seen where accelerations and braking were felt frequently in the cars. Less wear than expected was found where speed control seemed very effective and cars passed through sharp curves at a constant or slightly increasing speed without change in the traction mode.

5. ECONOMIC AND ENVIRONMENTAL CONSIDERATIONS

5.1 COSTS AND BENEFITS

5.1.1 NEED FOR COST ANALYSIS

Costs are important considerations in maintenance work. Safety and operations come first but, underlying them, are many maintenance decisions that should be made on the basis of cost effectiveness.

Almost daily, the transit engineer and track supervisor are faced with decisions--whether to increase the maintenance work on this section of track and save on the future costs of major repairs, or to let the maintenance slide a little and plan large repairs/replacements in the next few years. Such decisions cannot be made from a checklist or the use of a simple formula.

Excellent techniques have been developed for evaluating major project costs under the headings of value engineering[17] and life-cycle costing[18]. Unfortunately, these techniques appear time consuming in relation to day-to-day decisions on small maintenance jobs; so a simplified approach seems more appropriate.

Original costs, sunk costs, and depreciated costs are of interest, but their consideration introduces complications that can be avoided by using replacement costs. Current replacement costs are independent of the age and condition of the assets and thus provide a good base for comparison of the available alternatives in maintenance and replacement.

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[17] Dell'Isola, A. J., Value Engineering in the Construction Industry, Construction Publishing Co. 1974.

[18] Brown, R. J., and Yanuck, R. R., Life Cycle Costing, Fairmont Press, Inc., 1980.

5.1.2 COST AND VALUE FACTORS

A restraining rail adds to the cost of investment in the track and the cost of maintenance, since the rail and its components usually require inspection, adjustment, lubrication and eventual replacement. In some cases, a restraining rail is considered highly desirable, or even necessary, for its contribution to system safety by increasing track resistance to derailments in curves where car wheels are seen to have strong tendencies to climb the high rail. In other cases, the major benefit of a restraining rail is to reduce the frequency and cost of transposing and/or replacing the high rail during the life cycle of the track. In the latter cases, the value of a restraining rail depends on several definite factors:

- The frequency at which the high rail has to be transposed or replaced in the absence of a restraining rail.
- Costs of replacing the high rail over the life cycle of the track.
- Costs of the restraining rail over the life cycle of the track. These costs include the original investment in the restraining rail, fasteners, and lubricator; annual maintenance costs; and the costs of replacements when needed during the life cycle.
- The discount rate. This determines the present values of future costs and the future savings that are to be obtained by increasing the intervals between replacements of high rail.

The costs of installing and maintaining restraining rails vary greatly among the transit properties because of large differences in factors such as availability of materials, shipping costs, wage rates and levels of available skills. Costs also vary from site to site on a single property because of the access conditions, traffic frequency and other factors. In addition, costs change with inflation and changes in the economy that are not uniform across the country. As a result, the economic considerations are very complex.

In order to reduce this complexity and avoid considerations that are not generally applicable to all transit properties, a variable cost base was developed. Several possible cost bases were considered, but it was recognized that an appropriate base for the evaluation of restraining rails would be the replacement cost of the high rail that a restraining rail protects.

The economic function of a restraining rail is to extend the useful life of the high rail and thus reduce the recurrent cost of its replacement. Restraining rail costs tend to have a uniform relation to high rail costs over time at a specific property for comparable work sites. This relation can be expected to remain unchanged, if changes are introduced into local accounting practices; and it should not be affected appreciably when wide-area cost factors change, such as the ratio of energy costs to labor costs.

The base can be standardized to the cost of replacing a standard 300-foot length of restraining rail at a good, accessible site under good working conditions. Variations in job size, location and working conditions can be accounted for by cost factors that are developed by comparing specific jobs to the standard job. Thus high rail cost can provide a variable base for the evaluation of restraining rails, with the variations occurring among locations and the relation remaining constant at a specific location. Should the cost relation be changed as a result of major changes in designs or maintenance techniques, the effects of the changes will be well known and easily accounted for by the transit engineers who make them.

5.1.3 VARIABLE COST BASE

To simplify and standardize the variable cost base, it is assumed that the high rail is replaced on fasteners in good condition, or restored, on a good roadbed. Only the costs of the rail and its actual installation are included in the base. All other costs such as adding ballast, tamping, replacing broken bolts and worn pads and replacing joint bars are considered extras and are not included in the simplified base. Such work would be required in any event, with or without the use of a restraining rail.

Fortunately good cost data were available in two important cases --horizontal restraining rail with adjustable fasteners and vertical restraining rail fastened to the low rail with high strength bolts. Both work sites were in subways with good access, and the work had been done without interruptions by traffic. The running rails were welded in both tracks, and the joints in the restraining rails were bolted.

The costs were verified from the records of other trackwork costs on the same transit systems; and the costs of some components and work items (such as track bolts and shearing of rail flanges) were checked with data from manufacturers and from other transit systems. Material costs from 1978 were updated to 1980 costs by a factor of 1.22 and 1979 costs were updated to 1980 by a factor of 1.12. Labor costs were updated to 1980 by a factor of 1.15 for 1978 costs and a factor of 1.08 for 1979 costs. Information on the cost of opening and fastening bolted joints in running rails (versus cutting and welding) were available from several sources.

As a first step, the 1980 costs of replacing a 300-foot length of high rail were separated from other costs. Labor costs included the immediate overhead costs of vacation, sick leave, cluded the immediate overhead costs of vacation, approximately 60% of direct pay. Costs did not include the general overhead of the transit organization. Estimated 1980 base costs are shown for the rail on wood ties at 24-inch spacing. The cost items include the following:

Cut and bend the new rail (shop)	140+890 =	$1030
Labor; set up lights		80
Operate transportation and materials handling equipment		120
Cut old rail		60
Remove and load old rail and scrap		170
Load and deliver new rail		230
Weld new rail		550
Install and adjust new rail		870
Materials; 300 feet of heat treated rail		2540
9 thermite welds		490
Equipment and fuel		810
		$6950

The comparable 1980 base cost for replacing a 300-foot length of 100-pound heat-treated, continuous welded high rail on direct fixation fasteners at 30-inch spacing was found to be $6780.

Costs were then estimated for the following:

Transposing the high and low rails.

Replacing both high and low rail at the same time (10% gain in labor efficiency).

Installing a restraining rail at the time of track construction.

Installing a restraining rail when the high rail is replaced.

Replacing a restraining rail separately.

Replacing a restraining rail at the same time as the high rail is replaced.

Replacing all three rails at one time.

In each case, the costs were compared to the base costs for replacing the high rail alone. An example is replacement of 300 feet of vertical restraining rail, on wood ties, in 1980:

Purchase the rail drilled for joint bolts and with one flange sheared. Cut and bend it in the shop. Drill it to fit holes in web of low rail, and install it on track.

Labor, Cut and bend in shop	30+890 =	$ 920
Set up lights		80
Operate transportation and materials handling equipment		120
Remove and load old rail		300
Load and deliver new rail		230
Clamp, drill and bolt to low rail (151 fasteners)		3320
Install joint bars		70
Install electrical bonds		70
Material, rail with flange sheared and web punched for joint bolts		3750
Equipment and fuel		790
Total (140% of base costs)		$9650

The costs of the alternatives in installing and replacing rails of standard 300-foot lengths were estimated for both vertical and horizontal restraining rail installations. The costs were then calculated as percentages of the base costs as shown in Table 2. Except for the second alternative, the low cost ratios in Table 2 are associated with a horizontal restraining rail with widely separated braces on a concrete invert, with the rail flange not sheared; and the high cost ratios are associated with a vertical restraining rail with its flange sheared, on double tie plates and wood ties, bolted to the low rail between ties. The cost ratios for other types of installations tend to fall in between these two extremes.

The cost ratios in Table 2 cover one-time installation or replacement only, exclusive of subsequent maintenance/replacement throughout the life cycle of the track.

TABLE 2. RELATIVE COSTS OF ALTERNATIVES IN THE INSTALLATION AND REPLACEMENT OF RAIL IN CURVES

Alternatives	Ratio to Base Cost Low	High
Transpose high and low rail	45%	50%
Replace both high and low rail at the same time	160%	175%
Install a restraining rail during construction of a track	110%	195%
Install a restraining rail in an existing track at the time the high rail is replaced	130%	200%
Replace a restraining rail only	85%	140%
Replace a restraining rail when the running rails are replaced	75%	105%
Replace high rail only (base cost)	100%	100%

5.1.4 ANALYSIS

Determination of the most advantageous action in the maintenance or replacement of rail in curve track requires careful investigation and systematic analysis of the information collected. The usual steps in the process are:

- Identify the major problem and subproblems.
- Measure rail wear and determine useful lives of rails.
- Identify possible changes in rates of wear and other factors that may result from future traffic levels and vehicle replacements.
- Determine what acceptable actions could be taken to correct or reduce the problem, including actions that may not require rail replacement or additional maintenance.

Problems include such conditions as excessive rail wear, high noise levels, uncomfortable ride, and indications of tendencies for wheels to climb the high rail. Subproblems may include inability to obtain new rail in a reasonable length of time, wear and damage of fasteners, and deteriorating roadbed. Corrective actions may include installing a lubricator, adjusting and controlling the traffic operating mode to reduce the rate of wear, increasing wear limits, improving maintenance inspection, and procedures to keep the restraining rail well-lubricated and adjusted to prevent wear of the high rail. In any specific case, some of the possible actions may not be feasible because of shortages of funds or other conditions.

If the potential corrective actions include new equipment or replacements or a change in maintenance procedures, a cost analysis is indicated. As a prerequisite to a cost analysis, cost records have to be sorted, evaluated and updated; and other actions have to be completed as outlined in Figure 24. The development of estimated costs for standard units of rail replacement and maintenance, and cost factors for job size and location, will make it easier to estimate the costs of acceptable alternatives for all future jobs.

The remaining steps in the analysis process are:

- Estimate the costs of the preferred, acceptable actions over the life cycle of the track, including the costs of future maintenance and replacements.
- Evaluate the benefits of the preferred, acceptable actions.
- Compare costs and benefits to determine the most cost-effective action.

Simple procedures should be used in estimating future costs and savings, and figuring their present values for comparisons. Complex, highly detailed procedures are not worthwhile because of the uncertainty of future conditions and the future value of today's money. Thus present year costs can be used in estimating future work, and the present interest rate can be used to discount the future costs to their present values.

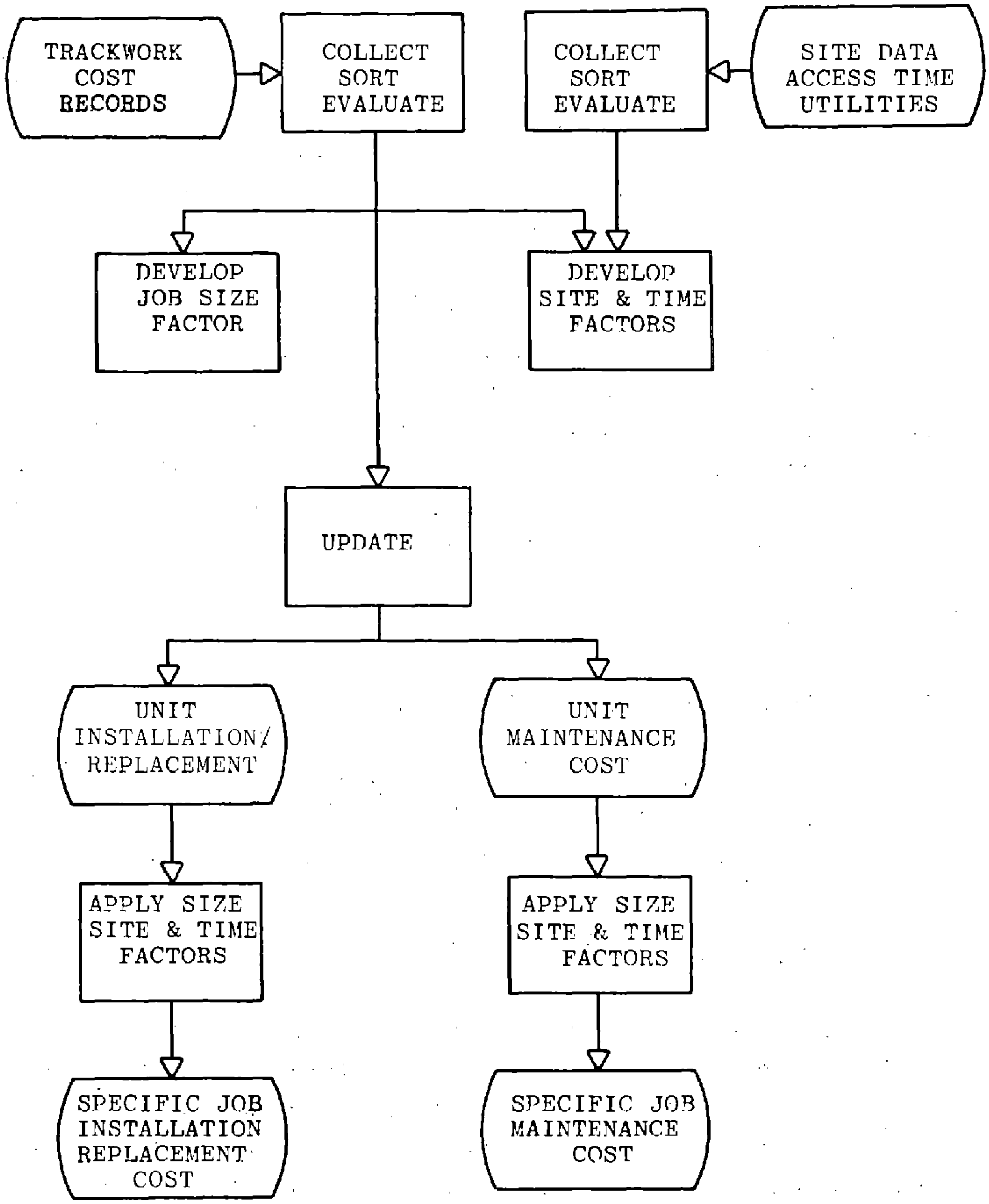

Fig. 24

Functional Activity Sequence in Estimating Costs of Installation, Replacement and Maintenance of Rails in Curves.

The equations used to figure present value can be found in textbooks covering the mathematics of finance:

$$PV = C(1+i)^{-y}$$

where C is a future cost or savings in present (base year) dollars
i is the present interest rate, and
y is the year in which the cost or savings will occur;

and $$PV = A\left[\frac{1-(1+i)^{-y}}{i}\right]$$

where A is the annual recurring cost or savings, and
y is the number of years the cost or savings will occur.

The equations above indicate the very large effects of interest rates on the value of future savings in maintenance or replacement costs.

Consider an investment in wear-resistant alloy steel rail to extend the interval to the next rail replacement from 10 to 20 years, and save \$2000 per year, years 11 through 20. At a discount rate of 5%, the present value of the savings would be:

$$PV = 2000\left[\frac{1-(1+0.05)^{-20}}{0.05}\right] - 2000\left[\frac{1-(1+0.05)^{-10}}{0.05}\right]$$

$$\doteq \$9480$$

At 15%, the present value of the future savings would be only \$2480.

A simplified example of a cost analysis is presented in Appendix C. It uses the base costs discussed in Section 5.1.3 and the relative cost alternatives listed in Table 2. In any specific case, costs should be developed from local cost records as outlined in Figure 24. Good cost data are important since errors are compounded when they are updated and multiplied by many factors during the analysis.

The evaluation of benefits other than savings in rail maintenance/replacement is difficult. Possible benefits and disadvantages should have been considered carefully before a decision was made on the acceptable alternative actions. If some actions seem more beneficial than others, efforts should be made to estimate their relative values, so that they can be weighed along with estimated savings in rail maintenance/replacement costs. The effort alone will often help dampen unreasonable preferences that tend to arise.

5.2 ENVIRONMENTAL FACTORS

Comments have been heard to the effect that restraining rail often increases the noise of wheel squeal. Very loud noises are heard on many tight curves where restraining rails are in use, and it seems likely that poorly lubricated restraining rails do increase noise levels when wheels rub them as well as the adjacent high rails. On the other hand, if a restraining rail keeps the wheels from contacting the high rail, and especially if it is well-lubricated, so as to reduce the effective coefficient of friction, it should reduce the noise level.

The force at the wheel-rail contact may greatly affect the amount of sound power produced. When a wheel flanges on a high rail, the force normal to the rail is larger than the lateral force. As indicated in Figure 25a, it is equal to the lateral force divided by the cosine of the angle θ between it and the lateral force. The vertical component of this force comes from the wheel load. When a wheel flanges on a restraining rail as indicated in Figure 25b, the force normal to a restraining rail that develops is generally close to the line of the lateral force and not much larger than it.

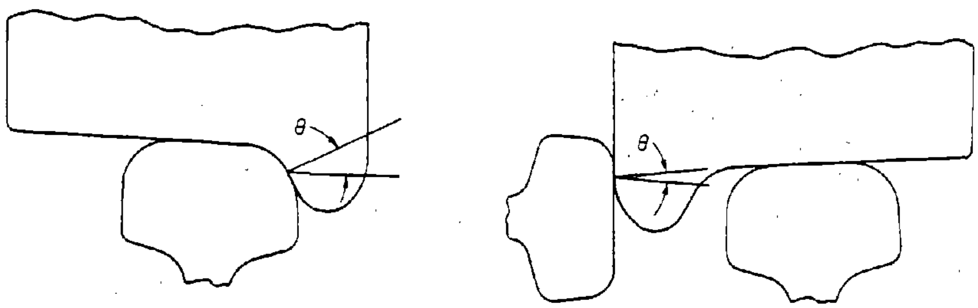

Fig. 25
Flanging in Curve Track.

Observations and measurements of peak noise during regular traffic operations at the Transit Properties indicated only that noise is often a serious problem. The background conditions that affect noise propagation varied greatly from place to place, and the loudest noises were heard at very sharp curves with restraining rails.

Measurements of noise levels, taken with a hand-held sound level meter during tests of lubrication effects on 150-foot radius curves at TTC, show substantial noise reductions with restraining rails installed. Peak noise levels, recorded at the highest test operating speeds at the beginning of each test zone are listed in Table 3.

The average peak noise reduction indicated when cars were guided through the curves by a lubricated restraining rail versus a dry high rail was 20 dBA; low readings marked with asterisks in Table 3 were omitted from the average.

TABLE 3. NOISE MEASUREMENTS IN SCREECH LOOP

Zone	Car Speed	R in Place	Lubrication H	Lubrication R	Water Spray	T °F	Wind Speed	Wind Dir.	Peak dBA
III	21.3					78	5	300	108
IV	20.5					82	5	300	109
III	21.3	+				65	4	160	79*
IV	19.6	+				65	4	160	80*
III	22.7				+	78	5	300	105
IV	21.2				+	84	5	310	106
III	21.2	+			+	70	4	160	78*
IV	20.0	+			+	70	4	160	80*
III	21.1		1-Lt			103	5	330	90
IV	23.0		1-Lt			103	5	340	96
III	22.3	+		1-Lt		82	8	360	78*
IV	21.5	+		1-Lt		89	8	360	84
III	22.0		1-Lt		+	103	5	330	87
IV	21.0		1-Lt		+	103	5	340	99
III	18.8	+		1-Lt	+	82	8	360	72*
IV	21.7	+		1-Lt	+	93	8	360	79*
III	21.7		1-Hy			60	7	290	92
IV	21.5		1-Hy			60	7	290	93
III	21.5	+		1-Hy		80	5	25	85
IV	20.1	+		1-Hy		81	5	25	91
III	20.6	+		2-Lt		78	8	360	90
IV	20.5	+		2-Lt		82	8	360	87
III	20.7	+		2-Hy		86	8	360	86
IV	19.4	+		2-Hy		98	8	360	83

Notes: R: Restraining rail
H: High rail
+: yes;
Dir: Direction, clockwise from North in degrees
T: Rail temperature in degrees Fahrenheit
Lt: Light application (Number in front indicates lubricant type)
Hy: Heavy application (Number in front indicates lubricant type)
*Exceptionally low readings

The effects of lubrication and a water spray on peak noise levels have been tested on 115-foot radius curves in the subway of the Port Authority Trans-Hudson Corporation[19]. Both the high rails

[19] Reduction of Wheel Wear and Wheel Noise, Phase I Exploratory Research, Port Authority Trans-Hudson Corp. Report, UMTA Project IT-09-0058, TS No. A541, Oct. 78.

and the restraining rails in these curves are normally lubricated automatically. In the tests, lubrication of the flange of the high rail only versus dry rails indicated peak wayside noise reductions under traffic of 4 to 16 dBA. With a water spray and no lubrication of the high rail but with residual grease films on the rails, peak reductions of 12 to 15 dBA were recorded.

Restraining rails have some small environmental influences that may be grouped under hazards and pollution. The installation and maintenance of a restraining rail requires some extra work on track that necessarily involves small hazards. However the net effect of a restraining rail is a slight improvement in overall safety.

The friction of car wheels against a restraining rail produces wear particles that help to degrade the local environment. However indications are that the total wear occurs at a lower rate when a well-lubricated restraining rail is in use than when a high rail alone guides car wheels through a curve.

The most noticeable pollutant found with restraining rails is waste lubricant. This is also found where a restraining rail is not installed and the high rail is lubricated, but more seems to be used where restraining rails are installed. In a subway or at grade, the lubricant is a nuisance to the track mechanics, not to the general public. None of the tracks run through a water supply catchment area. The drip of lubricant from elevated tracks, however, may inconvenience many people. To avoid this, drip shields are, and should be, installed at strategic places, particularly in and near stations where many people walk under the elevated structures.

6. RESTRAINING RAIL STRUCTURES

6.1 FORCES ON RESTRAINING RAILS

6.1.1 GENERAL

When a transit car runs through a curve, the track forces the car to follow the curve, accelerating it towards the center of curvature. In a superelevated smooth curve, two types of lateral body forces are exerted on the distributed mass of a moving transit car: the inertial force caused by acceleration towards the center of curvature and the lateral component of gravitational force developed by superelevation of the track. The lateral forces (measured in a plane parallel to the surfaces of the rails) are in dynamic balance with lateral reaction forces at the wheel-rail contact points developed through friction and the flange reactions.

The resultant of the lateral forces at the wheel-rail contact points must be equal and opposite to the resultant of the body forces. The total lateral force is not shared equally by all the wheels because of the curving dynamics of the truck and wheel conicity. In fact, lateral reaction forces from the left and right rails may have equal and opposite components which add to the loads on the individual rails but have no effect on either the vehicle or the track panel. Any lateral components in the couplers, transmitting forces from adjacent cars, must also be added to the overall dynamic balance of forces.

The vehicle body forces in the longitudinal direction consist of the inertial forces resulting from acceleration or deceleration*. They must be in dynamic balance with the sum of all contact forces

*The gravitational component may become significant if the track has a steep grade.

at the wheels developed by traction, braking, rolling resistance, other resistance, and any longitudinal forces transmitted through couplers.

Distribution of the longitudinal forces over the wheels is unequal, as in the case of the lateral forces. It is affected by truck curving dynamics, wheel conicity, and the fact that two wheels are rigidly mounted on a single, solid axle which causes longitudinal slip in curves.

The interactions of wheels and rails consist of complex, dynamic phenomena that are always changing in response to motions of the car, conditions of the wheels and rails, and each other. These dynamic phenomena are affected greatly by changes in the speed and operating mode of the car, track irregularities and other factors. Nevertheless, while the total situation is obscure, many of the phenomena that comprise it can be examined separately under assumed steady state conditions. Although the information obtained is only approximate, it provides insights into the range of the effects of the phenomena, and indicates possibilities for reducing effects that cause wear of wheels and rail.

Data have become available recently from measurements of lateral forces in curves of transit tracks. Under extreme conditions of tight gage and unworn, cylindrical wheel profile, the average lateral force at the lead wheel approached the magnitude of the vertical wheel load[20,21]. Unusual operating conditions may further increase this lateral force, since excessive speed and rail

[20] Phillips, C., et. al., Measurement of Wheel/Rail Forces at the Washington Metropolitan Transit Authority Volume I: Analysis Report. Interim Report. TSC Report No. UMTA-MA06-0025-80-6, July 1980.

[21] Ahlbeck, D. R., Harrison, H. D., and Tuten, J. M., Measurement of Wheel/Rail Forces at the Washington Metropolitan Transit Authority, Volume II: Test Report. Interim Report. BMI Report No. UMTA-MA06-0025-80-7, July 1980.

irregularities may occur some day in any curve. The additive effects of these conditions could increase the lateral force as much as 50%, to 1.5 times the wheel load. This large peak force would occur at the lead wheel, and a considerable part of it would be used in overcoming lateral creep forces at the rail surface. The unbalanced effect on the car would not be serious, but the rail and fasteners should be designed to withstand the worst-case peak force.

6.1.2 FLANGE FRICTION

Flange friction is the immediate cause of wear of wheel flanges, restraining rails and the gage side of the high rails in curves. It also increases energy requirements, since it contributes primarily to resistance rather than tractive force.

When a leading wheel flanges, it usually contacts the rail at a point forward of the axle and the tread contact patch as indicated in Figure 26.

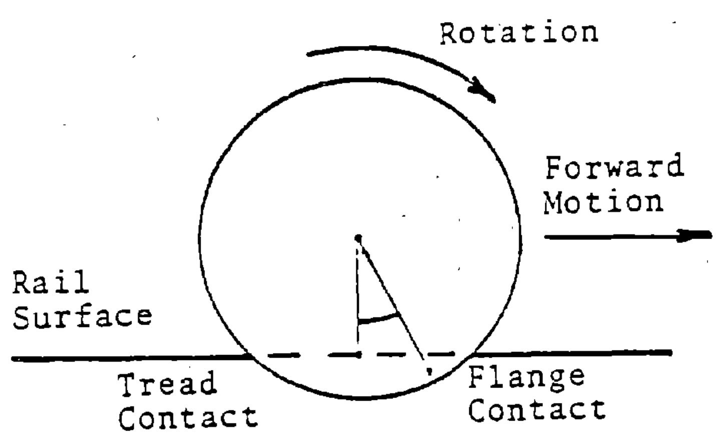

Fig. 26

Wheel-rail Contacts.

The net motion of the flange at its contact with the rail has downward and backward components; and its backward component is small, since the flange contact is close to the level of the tread contact where the instantaneous motion is zero.

Since the flange contact patch covers an area on the wheel, the motion of different points in the contact patch on the wheel have slightly different directions and velocities. As the wheel turns and moves forward, some of these points slide past each contact point on the rail, which experiences rapid variations in the direction of the frictional force and the sliding velocity. This action may account for some of the extreme effects of abrasive and adhesive wear found in sharp curves, where the rail wears very rapidly and has a rough, scuffed appearance.

The frictional forces developed by the sliding motions of the flange contact against the rail are proportional to the lateral flange force and do not have much relation to the sliding velocity. The forces at the start of sliding, however, may be very high, indicating what has been called the coefficient of static friction. Changes to and from static friction can occur rapidly in wheel contacts with a restraining rail, as the flange contact points pass the level of the tread contact.

Rabinowicz[12] notes that the measured coefficient of friction in laboratory tests was actually higher at low speeds than at higher speeds for steel sliding on steel, except under conditions of perfect lubrication by a soap. Accordingly, large frictional forces can be expected even at very low speeds when the lateral flange forces are large.

6.1.3 SURFACE FRICTION AND RELATED EFFECTS

Surface friction at the contact patch between the wheel tread and the rail is very important in curves, because of its large effects on the flange forces, as well as the wear it causes. The surface friction is caused by movement between wheel tread and rail at the contact patch, which was omitted from the simplified discussion in the preceding subsection. The largest movements between wheel tread and rail surface are creep and

[12] Rabinowicz, p. 16

movements between wheel tread and rail surface are creep and slip. Creep is the lateral, sliding motion that occurs as the wheels are forced by the curved rails to move sideways while, at the same time, they are forced to roll along lines defined by the track structure. Slip is the longitudinal sliding motion that occurs when the angular velocity of the wheel times the radius is not equal to the longitudinal speed of the wheel. This discussion is limited to the simple case of cylindrical wheels which are forced to creep from a straight line to the curve of the rail and slip the full difference between their curved paths on the rails.

6.1.3.1 Creep

The creep distance that the wheels are forced to slide sideways depends on the radius of the curve and the distance the car travels along the curve. In a very short distance, a curve does not diverge measurably from its tangent; so a cylindrical wheel may be considered to creep in very small increments to the tangent from the straight line along which it rolls as indicated by chord AD in Figure 27.

Each very small increment of creep is proportional to the mid-chord distance from chord to tangent at m in Figure 2. This is equal to L tan α; where L is half the chord length, and α, the angle between chord and tangent (angle of attack) is equal to half the central angle subtended by the chord.

Each small increment of creep is then:

$$\Delta C = \Delta L \tan \alpha$$

and

$$\Sigma \Delta C = C = L \tan \alpha$$

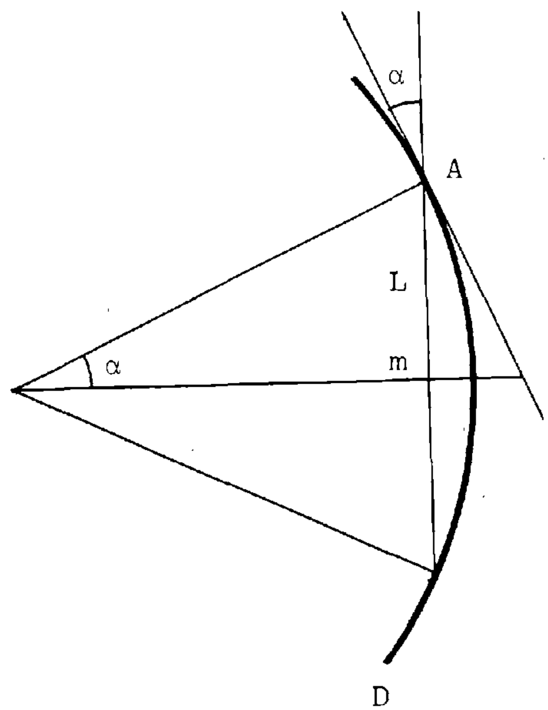

Fig. 27

Curve Geometry.

Since the angular relations do not change appreciably as a car moves uniformly around a smooth curve, the total creep distance can be summed as above for any travel distance.

King [8] indicates that the creep force does not continue to increase as the angle of attack increases, because the effective lateral coefficient of friction reaches the limit of wheel-rail adhesion at an angle of attack of about 1 degree. The rates of wear of rail and wheels tend to increase, however, because the contact area between wheel flange and rail moves farther forward of the axle as the angle of attack increases. This increases the velocity of the downward component of the motion of the wheel against the rail (scuffing) and tends to decrease the size of the contact area which increases the unit pressure. Both of these processes tend to increase the rate of adhesive wear.

As noted by King [8], the lateral component of tread creep forces required to slide the two wheels on an axle laterally in a curve is:

$$F_{creep} = 2\mu_e N$$

where μ_e is the lateral tread coefficient of adhesion and N is the wheel load normal to the rail.

Typical values reported by King [8], that have been found for the coefficient of wheel-rail adhesion which limits the resultant of the slip and creep forces, range from 0.15 to 0.35.

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[8] King, p. 16

6.1.3.2 Wheel Slip

Slip affects the lateral coefficient of adhesion, μ_e, because the full frictional resistance acts along the line of the resultant of the forces that cause the wheel to creep laterally and slip longitudinally. Slip itself is affected by braking forces which resist the rolling motion of the wheels and by tractive forces applied to the wheels.

The axles of a powered truck are coupled to a traction motor which exerts a braking-type drag on the wheels when the car is operated in the drift mode, and a tractive force in the traction mode. In the drift mode, the drag increases any tendency of a wheel to slip ahead and resists any tendency to slip backwards. Hence we would expect the low wheels of the truck to slip ahead to accommodate the difference in the wheel paths when a powered truck is operated in the drift mode, and no slip to occur at the high wheels.

Slip occurs with a car in a braking mode similar to the slip in the drift mode, with the braking force adding to the internal resistance of the traction motor. In extreme cases, all wheels may slide forward.

In the traction mode, we would expect the opposite effects on the wheels of a powered truck, backward slip at the high wheels and no slip at the low wheels.

While no theory is yet available to describe completely the exact effects of slip, it appears that frequent changes in the operating mode transfer the effects of slip from the creep forces at one rail to those at the other, and may result in large fluctuations in the flange forces. This is consistent with the excessive wear that has been found in some curves where changes in operating mode occur frequently.

Observation of wheel actions in a few sharp curves indicated that slip may occur in short, irregular increments rather than continuously. This action may result in erratic fluctuations in the levels of the creep forces and, consequently, the flange forces.

6.1.3.3 <u>Other Effects</u>

Other movements at the contact patch consist of spin, as the contact patch rotates about its center while the wheel moves through the curve, and differential movements within the contact patch. These differential motions result from elastic and plastic deformations of the materials and the distances of points in the patch from the instantaneous center of the roll axis. Fortunately the effects of these motions seem relatively small in relation to the effects of lateral creep and longitudinal slip.

6.1.4 SUPERELEVATION

Superelevation of the outer rail of curved track causes gravity to provide part of the force that accelerates a car towards the center of curvature, so that it follows the curve. The car reacts to the forced acceleration with a "centrifugal force" of:

$$F = \frac{V^2}{R} \qquad (1)$$

where M is the mass of the car in slugs

V is the speed of the car in feet per second

R is the radius of curvature in feet

and V^2/R is the centripetal acceleration in feet per second per second.

Superelevation gives the gravitational force on the car a lateral component as indicated by H in Figure 28. Then:

$$H = \frac{WE}{B} \qquad (2)$$

where W is the weight of the car, in pounds

E is the superelevation in inches

B is the distance between wheel contacts on the rails, in inches.

For a balanced condition: H = F

Then: $$\frac{WE}{B} = \frac{MV^2}{R} = \frac{WV^2}{32.2R}$$

$$E = \frac{BV^2}{32.2R} \qquad (3)$$

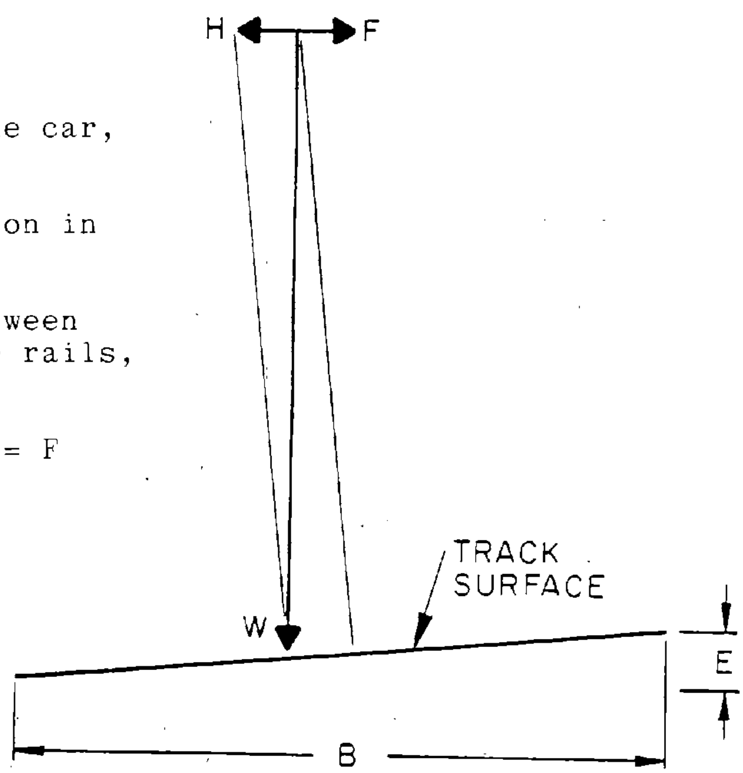


Fig. 28
Forces Affected by Superelevation.

For a balanced condition, with R = 150 feet, B = 60 inches, and converting V to miles per hour:

$$E = \frac{60V^2}{32.2 \text{x} 150}\left(\frac{88}{60}\right)^2 = 0.027V$$

Then, for examples:

for	V = 15,	E = 6.1	and for	E = 2.0,	V = 10.5
	10,	2.7		1.0	6.1

6.1.5 WHEEL TAPER (CONICITY)

Wheel taper assists superelevation by causing gravity to provide a small, additional lateral force. When tapered wheels move laterally towards the high rail, the high wheel is raised slightly and the low wheel is lowered by a similar amount.

Consider a wheelset with a 1:20 taper that moves laterally one inch from a centered position, so that a wheel flanges. Then the flanging wheel is raised 0.05 inch and the opposite wheel is lowered 0.05 inch, with a combined effect similar to a superelevation of 0.10 inch. For an axle load W of 60,000 pounds and a distance between wheel-rail contact patches of 60 inches, the restoring force on the axle would be, from equation (2):

$$H = \frac{60{,}000 \times 0.10}{60} = 100 \text{ lbs.}$$

Wheel taper usually improves curving by reducing the amount of longitudinal slip that occurs while the two wheels travel on paths of different radii. It does this by increasing the effective rolling diameter of the high wheel when it moves laterally towards the high rail and reducing the effective rolling diameter of the low wheel.

6.2 STRESSES

When a wheel flanges on a restraining rail, the lateral force causes the railhead to deflect as indicated in Figure 28.

In this example, the flange of the restraining rail is sheared, and both the restraining rail and running rail are supported on double tie plates with wood block ties at 24-inch spacing on a concrete slab. Both rails are 100-pound ASCE sections, and they are fastened together with 1.25-inch Ø bolts and spacer chocks installed halfway between the ties. The restraining rail has worn the guard face to the extent that its transverse plane moment of inertia is reduced to 1.35 in^4.

For the condition shown in Figure 29, the wheel is directly above the center of the tie plate and is flanging with a 10 k lateral force. Under this condition, the lateral force at A tends to rotate the restraining rail about point E on the tie plate. This effect is resisted mainly by the bolts and chocks 12 inches away from A, and force and moment are transferred to the low rail which is held down by the vertical wheel load. When

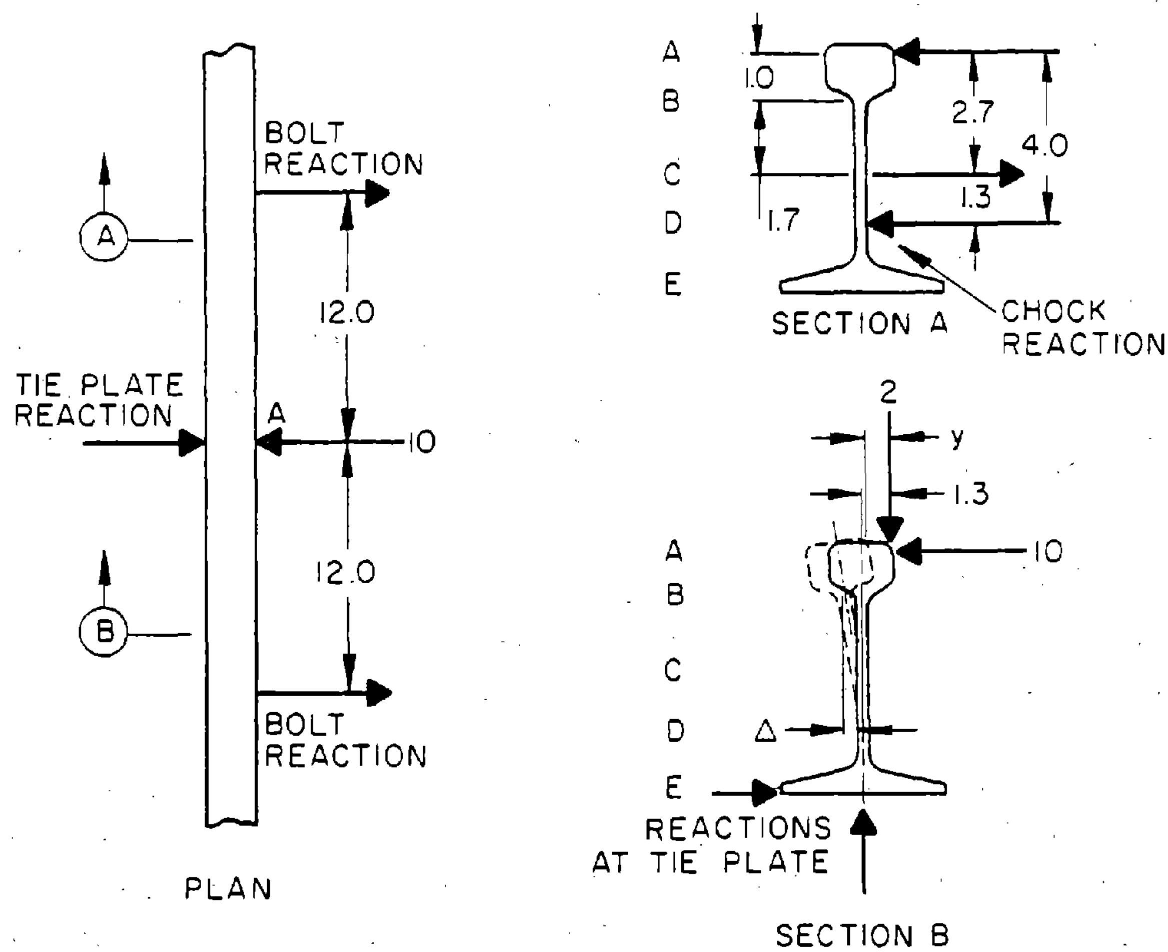


Fig. 29

Diagram of Vertical Restraining Rail Under Load.

the bolt is tight, the stresses are relatively low. High stresses develop when the bolt is loose, and the bolt is subject to shear and moment between the chock and the webs of the rails.

Approximations of the stresses and deflections in this complex structure can be obtained by considering the railhead as a beam and the rest of the structure as an elastic foundation. Reasonable values for the stiffness of the foundation can be found by considering its parts to act as a group of springs in series, so that the several deflections are sequential and cumulative. Equivalent stiffness coefficients are combined to an overall stiffness modulus for the equivalent foundation. The deflection of the railhead can then be determined from continuous beam theory as discussed by Hetenyi[22].

At the point of the force application, A, the deflection is:

$$y = \frac{P\lambda}{2K} \quad \text{and the moment is } M = \frac{P}{4\lambda}$$

where P is the applied force in kips

K is the stiffness of the equivalent foundation in kips/in./in.

y is the deflection in inches

λ is the factor obtained from the stiffness of beam and foundation by the relation,

$$\lambda = \sqrt[4]{\frac{K}{4EI}}$$

where E is the modulus of elasticity of the railhead, 29,000 kips/in.^{2}

I is the lateral moment of inertia about the vertical centerline of a plane transverse section through the railhead, 1.35 in.^{4}

[22] Hetenyi, J., Beam on Elastic Foundation, Scientific Series Vol. 16, Univ. of Michigan Press, 1942.

The moment developed by the 10 k force P shown in Figure 29 is reduced by the opposite moment of the vertical force component (say 2 k) about a point above the centerline of the rail.

This latter moment is:

$$1.30(0.20P) = 0.26P$$

In effect, it reduces the force available to bend the web and stretch the bolts to 0.74P.

Immediate support of the railhead is provided by the web of the rail which is constrained by the bolts and bends between the base of the railhead and the centerline of the bolts, a distance of 1.7 inches. Taking a one-inch slice of the web to find its unit stiffness, with an average thickness of approximately 0.65 inch, its transverse plane moment of inertia is:

$$I_w = \frac{bh^3}{12} = \frac{1.0(0.65)^3}{12} = 0.023 \text{ in.}^4$$

where b is the width of the slice and
h is the depth of the slice (thickness of the web).

Treating the web slice as a cantilever beam fixed at the level of the bolt centerline, a unit lateral deflection at A, in Figure 29 , would produce a deflection at the end of the beam, B:

$$\Delta = \frac{1.7}{2.7} \text{ in.}$$

Applying an imaginary force P_b at B, from the equation for a fixed-end cantilever beam:

$$\Delta = \frac{P_b \ell^3}{3EI_w}$$

where ℓ is the 1.7-inch length from the bolt center to B.

The force P at A necessary to produce a force P_b at B is:

$$\frac{1.7}{2.7}(0.74P) = P_b$$

Then,

$$\Delta = \frac{1.7(0.74P)(1.7)^3}{2.7(3)29000(0.023)}$$

From which, P = 550 k.

This effect extends along the rail an average of about 6 inches each side of a bolt. The value of the modulus, K, over the 24-inch distance between bolts is then:

$$K = \frac{12}{24} P = 275 \text{ k/in.}^2$$

If the bolt is tightened to specified torque, the tension produced by the lateral force P will be less than the residual stresses in bolt and chock; so the bolt will not elongate but will merely transmit force to the web of the low rail. This will flex the web and cause the low rail to deflect towards the restraining rail; the low rail then transfers force to the wood block ties and, through them, to the bolts and concrete invert. Each of these effects is evaluated as above, and the resulting stiffness values are converted to one value for the whole equivalent foundation:

$$K \simeq 50 \text{ k/in.}$$

From the equation for a continuous beam on an elastic foundation with K = 50 and I = 1.35,

$$\lambda = \sqrt[4]{\frac{50}{4(29000)1.35}} = 0.134 \text{ in.}^{-1}$$

$$y = \frac{10(0.134)}{2(50)} = 0.013 \text{ in. maximum at the point of load}$$

$$M = \frac{10}{4(0.134)} = 18.7 \text{ in-kips maximum}$$

Then the maximum stress in the railhead is:

$$f = \frac{Mc}{I} = \frac{18.7(2.175)}{1.35(2)} = 14.4 \text{ ksi}$$

Large deflections would occur with the track supported on ballast. Since the track would be loaded, the lateral modulus would be much larger than that measured on unloaded track (0.6 to 1.0 $k/in.^2$) say 2 $k/in.^2$. Then calculated as above, the maximum deflection is 0.15 inch and the maximum stress is 33.6 ksi.

With a wheel at a point above a bolt, the restraining rail is subject to the lateral flange force, plus the small vertical component; while the running rail deflects under the wheel load and transfers part of it to the restraining rail through the bolt and chock. If the bolt is loose, very little of the load is transferred; the chock rotates, and large shear and moment forces are imposed on the bolt between the chock and the webs of the trails. This situation is illustrated in exaggerated form in Figure 30. Rapid reversals of the stresses occur when the wheel moves past the next tie.

The large transfer of stresses between the rails may be avoided, if they are connected only above the ties where they are supported by double tie plates, or if (instead of a chock) a tube separator is used, which would permit differential vertical movement of the rails.

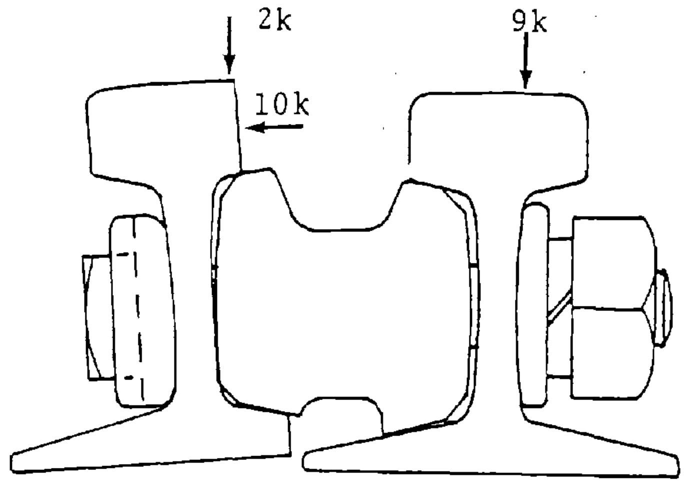


Fig. 30

Deflections at Loose Bolt and Chock with a Wheel Above the Centerline of the Bolt.

A horizontal restraining rail is not connected to the adjacent low rail, so there is no transfer of flange forces between the rails. Stresses caused by flange forces are low because of the lateral strength of the Tee rail when installed on its side. Large stresses, however, could develop in the rail from vertical wheel loads, if rail wear caused a step to develop in the worn face of the rail as discussed in Section 6.4. In such a case, a vertical wheel load of about 9 k plus a 2 k vertical component of the 10 k flange force would be imposed against the "side" of the rail.

Consider an 80-pound ASCE section restraining rail, in the horizontal position with supports at 48 inches, and worn to the extent that the moment of inertia of the railhead is reduced to 1.4 in^4. The railhead deflects downward and rotates under the 11 k vertical force; and the 10 k flange force deflects the rail a small distance laterally and reduces the rotation of the railhead. If the wheel flange were solidly seated on the step, the vertical deflection would reach 0.15 inch, and the maximum stress in the railhead would reach 60 ksi. Fortunately both the wheel flange and step are rounded, and the flange tends to slip sideways off the step. It is doubtful then that the deflection ever exceeds 0.10 inch or the stress ever exceeds 30 ksi.

6.3 WEAR AND DAMAGE MODES

A restraining rail no longer performs its primary functions; when it is worn, loose, or out of position to the extent that it:

- Does not increase track resistance to derailment
- Does not keep wheels from flanging on an unlubricated high rail
- Does not reduce the rate of wear of the high rail.

6.3.1 RAIL WEAR

Restraining rails generally wear at angles relatively close to the vertical (from the plane of the track surface) when compared to the wear of high rails. This is shown by the sharp guard corners visible on the worn restraining rails in Figures 31 and 32, rather than the rounded wear found on the gage side of running rails as shown in Figure 33. Indications of step wear can be seen in Figures 34 and 35. The step wear is considered more serious than flat wear, since long flanges tend to ride on the step illustrated in Figure 36. The lift of the wheel increases the vertical force on the worn restraining rail and may increase the frictional drag on cars, but it also transfers some of the axle load to the high rail; so step wear in a restraining rail should have less effect on safety than it would in a high rail.

Excessive side wear of a restraining rail transfers flange force and wear to the high rail. It also widens the flangeway, so that the angle of attack and the rate of wear tend to increase.

Fig. 31

Worn Restraining Rail

The lubricated restraining rail to the left in this picture has worn almost vertically and has a sharp corner at the guard face.

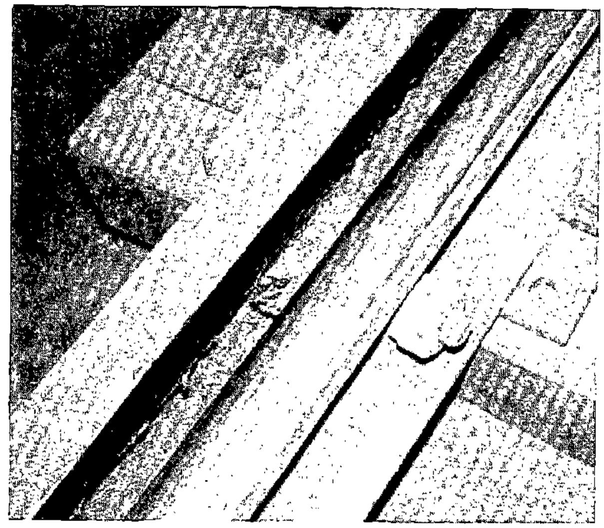


Fig. 32

Worn Restraining Rail

The lubricated restraining rail in this picture has also worn so that the upper corner of the guard face is relatively sharp.

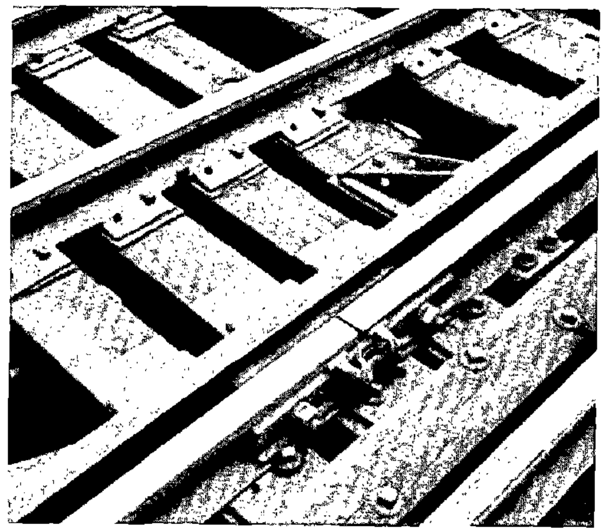


Fig. 33

Worn Restraining Rail

The worn high rail in this picture has a rounded corner at the gage side. The rail is on a high embankment and has an inner guardrail to the right of it in this picture.

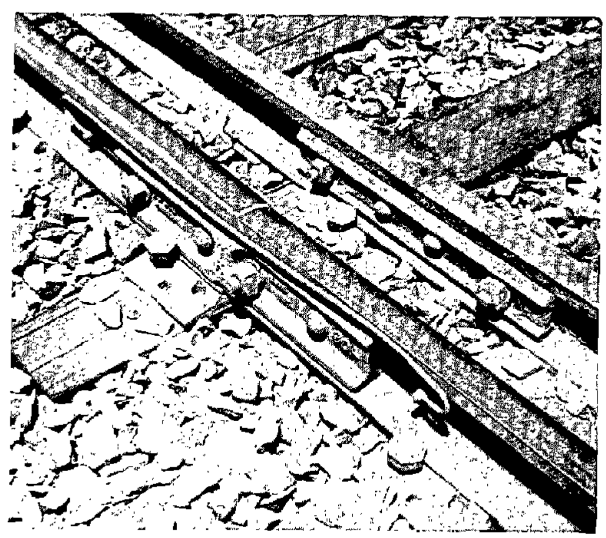


Fig. 34

Step Wear of High Rail

Wheel flanges have worn a reverse curve in the gage side of this rail.

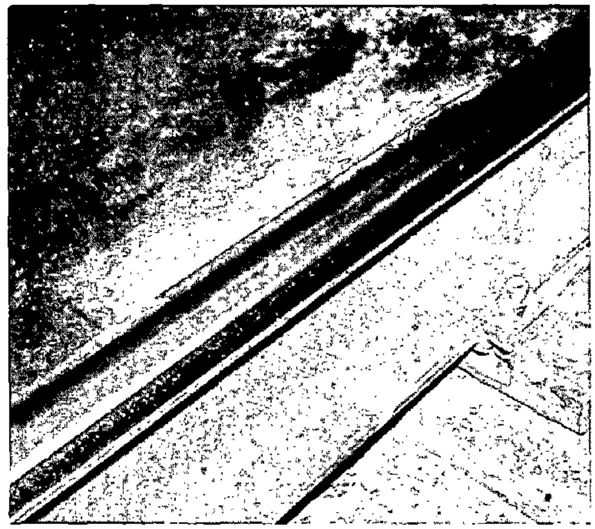


Fig. 35

Step Wear of Restraining Rail

Light is reflected from the horizontal surface of the worn guard face of the restraining rail shown in this picture.

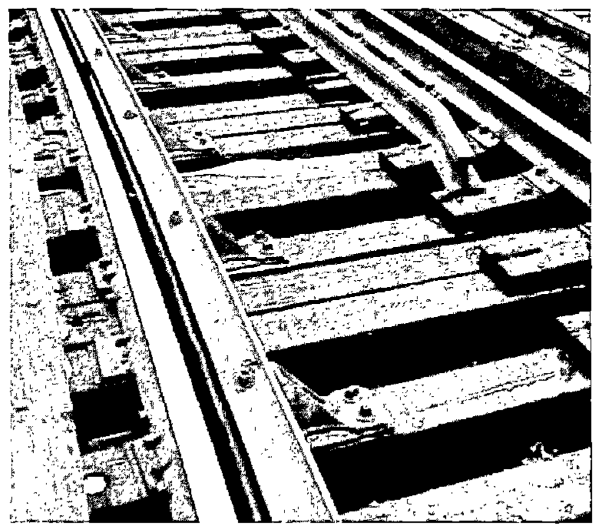
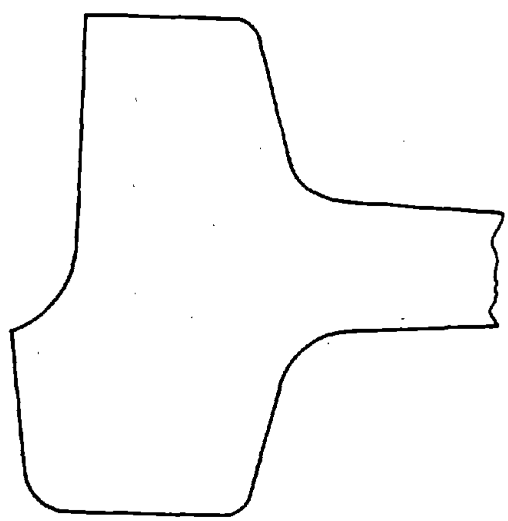


Fig. 36

Cross Section of Worn Restraining Rail

This worn 80-pound ASCE rail is similar to the rail shown in Figure 31.



Restraining rails are replaced long before the loss of material by wear reduces their strength to the point where permanent deformation could occur, let alone structural failure. Accordingly side wear becomes significant only when it moves the position of the guard face towards the high rail to the extent that the restraining rail no longer performs its functions.

Excessive surface wear of the low rail can result in wheel flanges riding on the spacer chocks in the flangeway between the low rail and a vertical restraining rail. The impacts tend to loosen and overstress the bolts and accelerate wear of car bearings as well as degrade ride quality.

6.3.2 WEAR AND DAMAGE OF COMPONENTS

Damage to components may be expected where a track has deteriorated over many years, and the fasteners are loose, worn or corroded.

In vertical restraining rail installations, the degradation of vertical support under the low rail at wood ties results in the transfer of large vertical forces to the restraining rail each time a wheel passes a fastening bolt. Similar transfer of forces to a horizontal restraining rail occurs when wheel flanges ride on the steps that may develop from rail wear as illustrated in Figure 36.

When bolts are loose and spacer chocks are worn, high stress concentration, short stress cycles, notch effects and stress corrosion may all contribute to the eventual fracture of the bolts. The loosening of C clamps, wear of non-adjustable braces, loosening of bolts at fasteners and spikes in ties all permit movements that accelerate wear and increase the stresses in adjacent components.

The side wear of track bolts or spikes permits the movement of tie plates under the lateral force on the restraining rail and the uplift effects of rolling load between trucks. Cases were observed where the tie plates had been shifted permanently more than half an inch towards the high rail and the low rail was seated on the outside shoulders of the tie plates. This lowered the effective height of the restraining rail in addition to transferring lateral force and wear to the high rail.

Brace castings that support horizontal restraining rails have cracked in a few cases. The cracks developed at transitions between the base and the vertical section. Cracks were avoided in other castings from the same lot by installing thick base plates. Cracks are usually avoided by providing large-radius transition curves between the base and vertical sections of castings.

6.4 STRUCTURAL CONSTRAINTS

The restraining rail structure should provide adequate lateral and vertical resistance to wheel forces, while remaining firmly in the position of a smooth space curve. In addition, it should not introduce hazards or increase the overall track maintenance costs, and it should not add to alignment irregularities or noise.

In addition to fitting into the track structure and being assembled from readily available components, a restraining rail has to meet many structural requirements:

- It should be strong enough and fastened firmly enough to resist large lateral and vertical forces while protecting the high rail from wear.

- It should be positioned in a smooth space curve, so it will guide the cars smoothly around the curve, maintaining good ride quality and keeping noise levels low.

- The restraining rail should be stiff enough to span the distances between supports with only small deflections that will not excite harmonic vibrations in the cars.

- In a ballasted track, the restraining rail should transfer approximately equal loads to each of the ties, so that the ballast will tend to consolidate uniformly, the track modulus and compliance will be uniform, and excessive track deterioration at some locations will be avoided.

- In a track with direct fixation to a concrete invert or other strong structure, there is no need for the restraining rail forces to be approximately equal at each fastener. The distance between supports can be related to the stiffness of the restraining rail and not held to the spacing of fasteners for the running rails.

- The restraining rail and its components should not project above the surface of a track so far as to cause hazard or damage to a car.

- The end of a restraining rail installation should not obstruct car wheels, in order to avoid damage in the unlikely case that a car derails while approaching the curve.

- The entire lateral flange force plus a vertical force component resulting from friction must be resisted through the restraining rail fasteners, since the vertical wheel load is on the adjacent low rail and cannot contribute directly to lateral resistance by the restraining rail.

- The restraining rail structure should not add to the overall cost of operating and maintaining the track.

- Stiff restraining rail should be bent smoothly to the design track curvature, so that its spring action under vibrating loads will not warp the track into a series of tangents and irregularities.

7. LUBRCIATION

7.1 LUBRICATION CONDITIONS

The lubrication of curve track in railroad and transit systems often reduces wear significantly, but the results vary. Probable causes of part of the variation are that the wear surfaces are not perfectly smooth, and the large forces tend to squeeze the lubricant from between them. Thus some metal-to-metal contact may be expected even with heavy lubrication.

When a car wheel flanges against a rail, it produces a very severe type of sliding friction. As discussed by Rabinowicz[12] and Czichos[23], the real area of contact is very small compared to the apparent contact patch. It consists of tiny micro contacts, or junctions, where only the high points of the two surfaces are actually pressed against each other as indicated in Figure 37. In this figure, the small junctions in dotted outline represent the real area of contact, while the apparent contact patch is the larger area enclosed by the dashed line.

The total real area of contact is so small that even a moderate flange force produces very high pressures at the tiny contact junctions, and these pressures may far exceed the compressive and shear strengths of the materials. To aggrevate the situation, several high points on the wheel surface may slide across a single, small contact area on the rail, while the wheel moves along the rail, each high point with a slightly different direction of motion and a different sliding velocity as noted in Section 6.1.2. The micro contact areas tend to become work hardened and brittle before they are worn down, and hard particles are often produced by high temperatures and chemical

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[12] Rabinowicz, p. 16

[23] Czichos, H., Tribology. A Systems Approach to the Science and Technology of Friction, Lubrication and Wear, Elsevier Scientific Publishing Co. 1978.

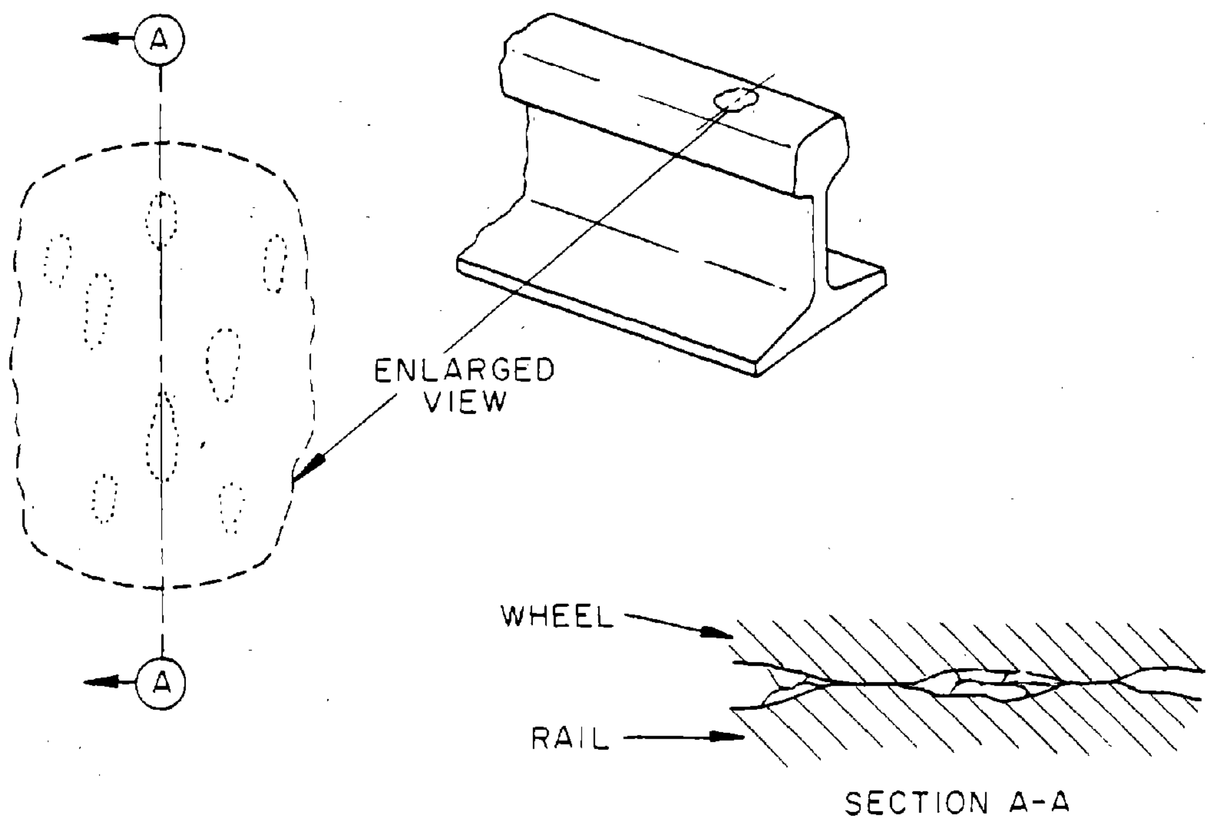

Fig. 37
Apparent and Real Contact Areas.

reactions at the surfaces of the materials and by contaminants on the rail. These conditions produce both abrasive and adhesive wear plus other severe effects.

In the abrasive wear, hardened micro contacts of one of the sliding surfaces actually gouge the contact areas of the other surface and shear off material, much as a grinding stone cuts materials from a work piece. Hard, loose particles become embedded in a surface and then gouge the other surface or alternately slide and gouge both surfaces.

In adhesive wear, some of the micro contacts are thought to bond together briefly with high atom-to-atom forces, before they are torn apart as the wheel rolls on. Rabinowicz[24] has shown that the very tiny particles found in adhesive wear debris are produced by fatigue weakening them until they break away.

Clayton[25] has found that the particular type of debris found where curve rails have worn rapidly can be produced by a deformation and fracture mechanism that can be reproduced in a laboratory. Whatever the exact wear mechanism involved, its results are severe and costly; so lubricants are widely used in attempts to reduce the rate of wear.

The type of lubrication usually achieved between wheel flange and rail is termed boundary lubrication to distinguish it from hydrodynamic lubrication in which the lubricant supports the load and keeps the metal surfaces completely apart. Hydrodynamic lubrication is achieved in pressurized bearings where the film of lubricant actually prevents contact of the bearing surfaces and eliminates the more severe wear processes. In boundary lubrication, the film of lubricant is penetrated by the micro contact areas which carry most of the load at their junctions. Friction and wear behavior of the system are then determined by physical-chemical interactions at the interface[23].

Rabinowicz[12] notes that the speed of the slidng motion may have large effects on the rate of wear. The flash temperature at the

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[24] Rabinowicz, E., "The Dependence of the Adhesive Wear Coefficient on the Surface Energy of Adhesion", Wear of Materials, 1977, Proceedings of the International Conference on Wear of Materials April 77, Am Society of Mechanical Engs. 1977.

[25] Clayton, P., "Lateral Wear of Rail on Curves", Tribology 1978, Materials Performance and Conservation, I Mech E. Conference Publications 1978-6, Mechanical Engineering Publication Ltd.

[23] Czichos, p. 77

[12] Rabinowicz, p. 16

real contact areas may pass through a transition range with a great increase in friction and wear. If a chemically active or EP lubricant is used, an increase in temperature resulting from speed can increase chemical reactions which increases corrosive wear while decreasing adhesive wear. An increase in speed may promote a transition from boundary lubrication towards hydrodynamic lubrication in which most of the micro contact surfaces are separated by a film of lubricant. At moderate speeds, the film of lubricant on the wheel surface will usually have an ability to heal itself after it is damaged in sliding against the rail, which it would not have if the time interval before the next contact were very short.

The observations mentioned above indicate that great variations can be expected in the results obtained with flange lubrication. As early as 1938, the AREA Committee V, Track, concluded that lubrication extended rail life and tie life on curves, reduced wheel wear and decreased train resistance[26]. Later AREA tests reported[13] the use of lubricators in a 7° 30' curve reduced the abraded metal collected from the wear of rail and wheels of four diesel units to about 10% of that collected when lubricators were not used. Increase in the life of the rail up to 700% by lubrication were claimed in one case on the Italian State Railways[27].

Tests with automatic flange oilers in the mid-sixties by the Japanese National Railway[28] showed reductions in the wear found on dry rails by factors varying from 2.9 to 7.3. The same

[26] "Report on Curve Oilers", Roadmaster Association Meeting, May 1980.

[13] Handbook, p. 16

[27] "Wheel and Rail Lubrication", Technical Note, Railway Gazette April 1952.

[28] Fujinawa, I., "Flange and Rail Lubrication", Railway Gazette Dec. 1967.

report compared the effects of manual and automatic lubrication on rail wear. For the curves and traffic investigated, the wear was almost three times as much with manual lubrication as with automatic lubrication. Manual lubrication of railroad curves is uncommon except where short curves are close to switches and other equipment that requires manual lubrication or where track is used infrequently. The long curves and long, infrequent trains on most railroads favor automatic lubrication.

On transit tracks, the high frequency of short trains makes lubrication control very difficult. Lubricators have to be checked and adjusted frequently. Where temperatures vary greatly from summer to winter, two grades of lubricant have to be used. Control is even more difficult with manual lubrication where large amounts of lubricant may be applied at infrequent intervals because of the high unit cost of the work. This generally results in less grease on the contact surfaces and more wasted grease.

7.2 LUBRICATION TEST RESULTS

The effects of lubrication were investigated in tests of drift resistance of a car on the transit test track at TTC. The drift resistance forces, calculated from time and distance measurements, are tabulated in Appendix B. The data have a wide scatter, which can be attributed to many factors:

- Variable influence of speed on the effects of lubrication as discussed in the preceding Section.
- Irregularities in curvature, crosslevel and profile.
- Mill scale and contaminants on the new restraining rail.
- Migration of lubricant to the rail surface.
- Variation in temperature, humidity and dust on the rails.
- Rotational resistance of the truck suspension.

In many of the test runs, the car passed through the speed balanced by superelevation*. Since wheel taper would provide a restoring force within a miles per hour each side of this speed, some free curving might be expected with a resulting drop in drift resistance. This effect, however, could not be distinguished in the test data or scatter diagrams plotted from the data. The effect may have been suppressed by rotational resistance of the trucks or other factors listed in the preceding paragraph. All of the factors listed were examined in relation to test zone locations and scatter diagrams of test results; but the effects were not separable within the data collection and analysis effort conducted under the project.

Drift resistance forces were averaged in Table 4 to reduce the scatter due to random effects and to bring out the trends produced by controlled test parameters. The force figures in parenthesis added to several of the tables in Appendix B were not included in the averages, or trend line calculations; four other figures were also omitted because they are far from other data in the same group and appear to be spurious. These latter are: the last figure (1290) under condition 1 in Table 7, the last figure (1710) under condition 3 in Table 8, the first figure (980) under condition 9 in Table 9, and the last figure (3370) under condition 5 in Table 10.

Data in Table 4 have subtracted from them the approximate drift resistance of the car on tangent track (taken as 1500 pounds from the tests in Zone 1) in order to show the approximate drift resistance resulting from curvature. The mean values shown are merely representative, since the number of test runs and the speeds of the runs were not the same in each set of tests, and drift resistance tends to vary with speed.

*From Equation (3) in Section 6.1.4, the balanced speeds would be 9.7 mph in Zone I, 6.4 mph in Zone III, and 11.1 in Zone IV.

TABLE 4. MEAN DRIFT RESISTANCE FORCES RESULTING FROM CURVATURE (averaged over test speeds between 5 and 15 mph)

	Forces in Pounds		
Test Condition	Zone II	Zone III	Zone IV
1. Dry rails; no restraining rail	1470	2410	1970
2. Water spray, no restraining rail		1350	1620
3. Light Application of lubricant #1 to high rail; no restraining rail	1150	710	1390
4. Same as 3, with water spray added		670	740
5. Heavy application of lubricant #1 to high rail; no restraining rail	1050	910	740
6. Dry rails with restraining rail		930	1130
7. Water spray on rails with restraining rail		1040	960
8. Light applicant of lubricant #1 to restraining rail		770	1210
9. Same as 8 plus water spray		360	590
10. Heavy application of lubricant #1 to restraining rail		1000	860
11. Light applicant of lubricant #2 to restraining rail		980	800
12. Heavy application of lubricant #2 to restraining rail		740	920

NOTE: Drift resistance forces were reduced by 1500 pounds to remove the approximate resistance of the car on level tangent track with trucks centered between dry rails.

Curvature and superelevation; Zone II, 340-ft. R, 0.5 in.; Zone III, 150-ft. R, 0.5 in; Zone IV, 150-ft. R, 1.5 in.

Vehicle: Two two-axle trucks; gross weight 95,500 lbs.

The mean drift resistancesces shown for a dry restraining rail are low. When the drift resistance forces are compared for the dry high rail (Condition 1) and the dry restraining rail (Condition 6) the average reductions are seen to be 61% in Zone III and 43% in Zone IV. These are about the same as the average reductions obtained with lubrication of the high rail or the restraining rail (when installed) under test conditions 3, 5, 8, 10, 11 and 12.

The test car was operated in Test Zones III and IV before the tests in order to wear in the new restraining rail until a continuous line of bright steel could be seen on the rail. In spite of this precaution, the wheels may have been in contact with mill scale on both sides of the narrow strip of bright steel most of the time during the tests. The brittle scale would act as a dry lubricant, since it would be at the top of the micro contact areas and would be easily sheared from the substrate material. Also the flakes of scale hold moisture a little better than a clean steel surface does; and, when combined with slight amounts of residual oil or dust, this would add to the lubricating effect.

Among the other factors listed in the first paragraph of this Section, track irregularities can have large effects even at low speeds. The lateral forces resulting from an irregularity and superelevation may momentarily overcome the lateral creep forces and the inertia of the car, so as to free the lead wheel from the high rail or restraining rail, and reduce the drift resistance force for a short time. Conversely, similar forces may push a trailing wheel against the unlubricated low rail and greatly increase drift resistance for short periods. Except for variations in force data, such effects could not be detected without additional sophisticated test equipment. In addition, as noted in Section 6.1.2, high force levels associated with static friction may develop momentarily in wheel contacts with a restraining rail, if the wheel contact points move through the level of the tread contact.

Scatter diagrams that were plotted from the test data were found to be somewhat confusing and misleading, and they have been omitted from the report. The calculated resistance forces for each test run are listed in Appendix B. Trend lines were plotted from these data, using regression analysis formula for a least-squares fit. Trend lines for the results obtained in the four test zones are shown in Figures 38 through 43; each trend line is numbered with its related test condition as stated in Table 4.

Figures 38 through 43 show that while a linear relationship between drift resistance and speed apparently exists in the results from many of the test sets, there is no consistency from set to set in this relationship, and the slope of the regression line ranges from negative to positive. There is no evidence that the indicated trend would continue beyond the speed limit of a data set even where the relationship appears to be strong. Since laboratory investigations and tests have indicated that the coefficient of sliding friction (under boundary lubrication conditions) does not have a uniform relation to speed, no effort was made to plot curves for the data trends.

The forces measured with the two types of lubricant, and light and heavy applications, do not show consistent differences. Type 1 is a lithium base grease with EP additives that are not listed in its specification. These additives usually include molybdenum disulphide which is considered to rival graphite as a solid lubricant. Both form lubricating surface films on metals[13]. Type 2 lubricant is a calcium base grease with 11.5% graphite added. In the light applications, a bead of grease approximately 0.1 inch in diameter was applied to the rail. This lubricated the rail at a rate of approximately 2.0 ounces of grease per 100

[13]Handbook, p. 16

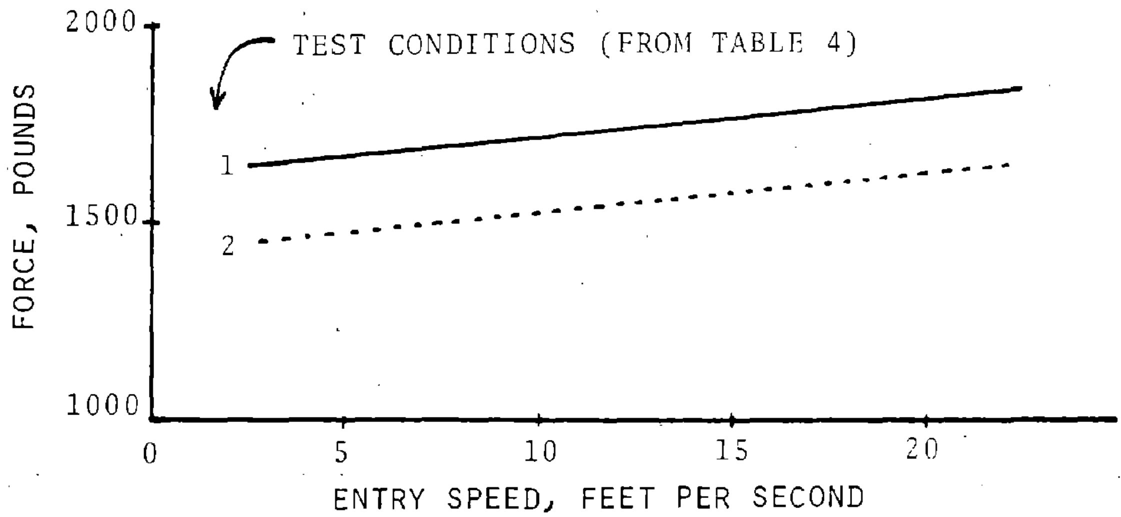


Fig. 38

Drift Resistance, Test Zone I, Tangent Track.

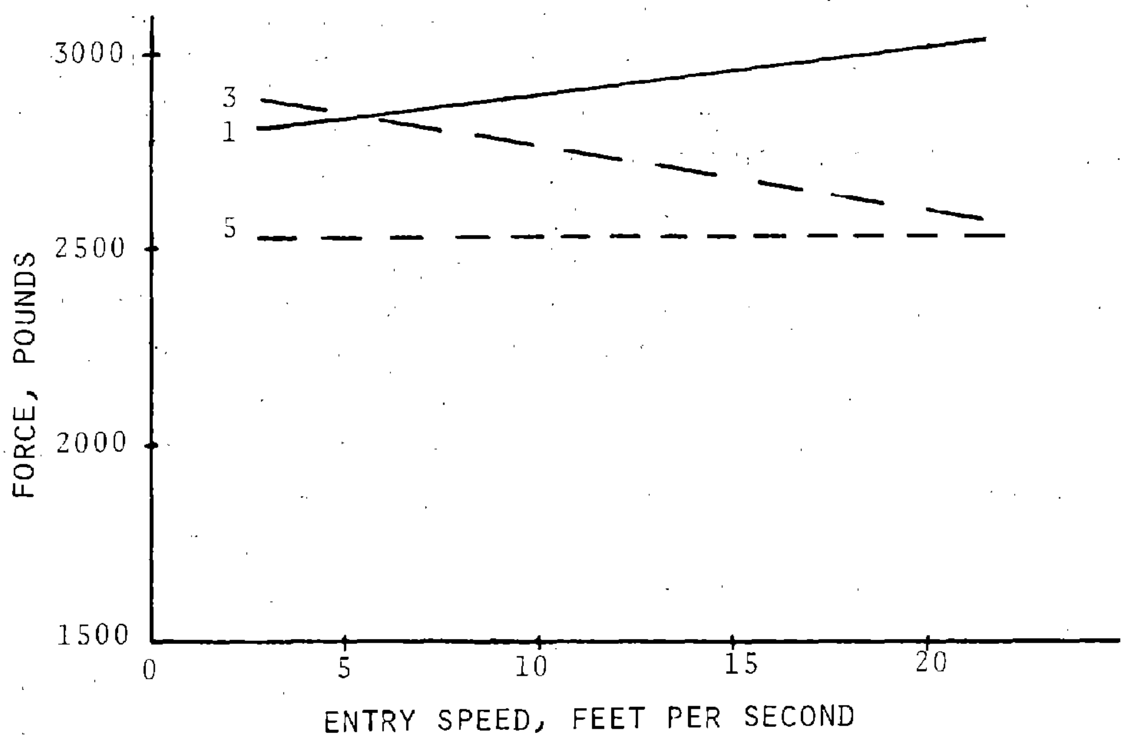


Fig. 39

Drift Resistance, Test Zone II.

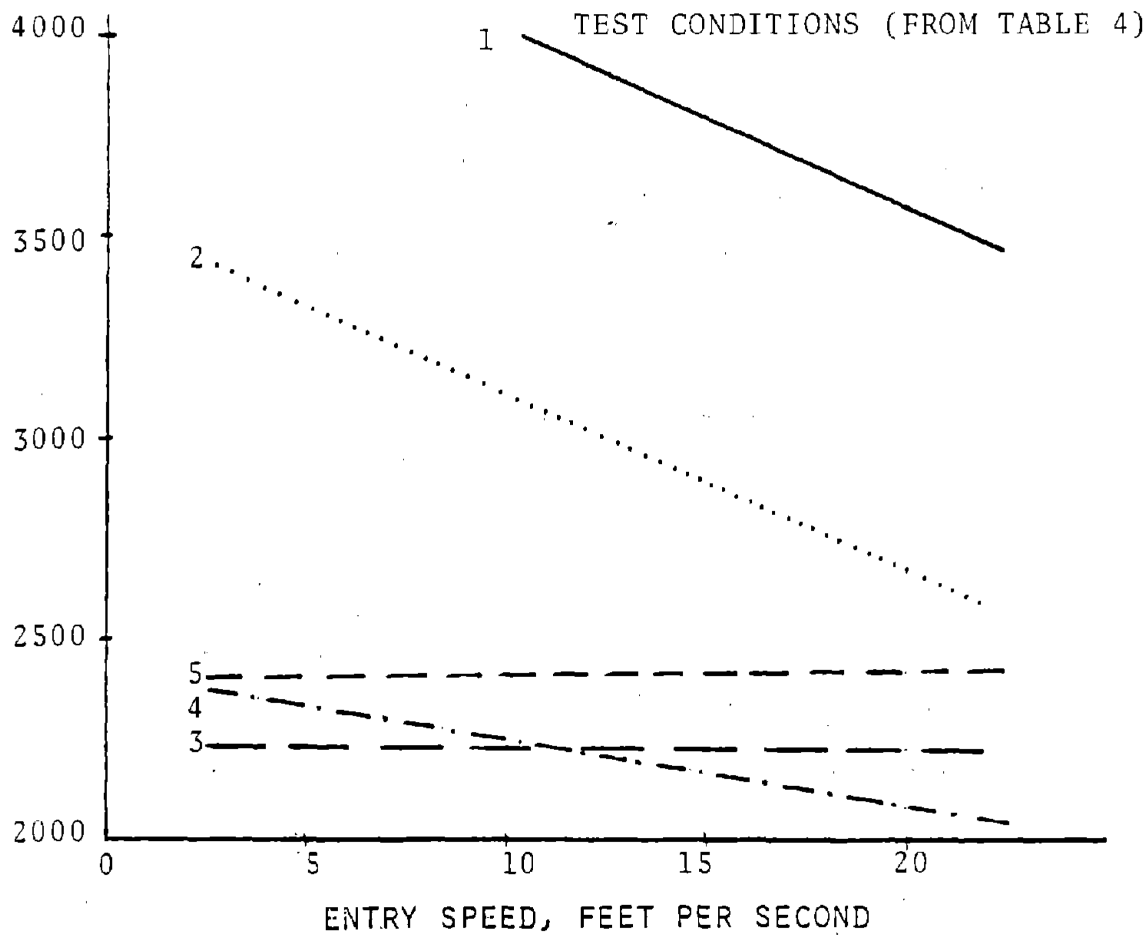


Fig. 40
Drift Resistance, Test Zone III, Without Restraining Rail.

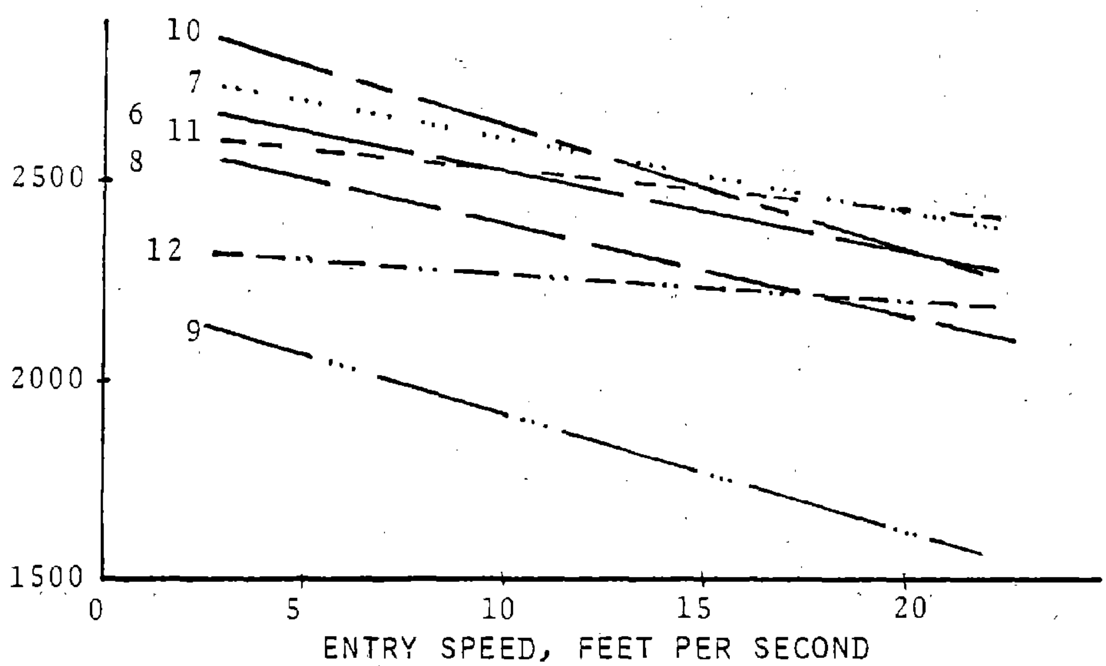


Fig. 41
Drift Resistance, Test Zone III, With Restraining Rail.

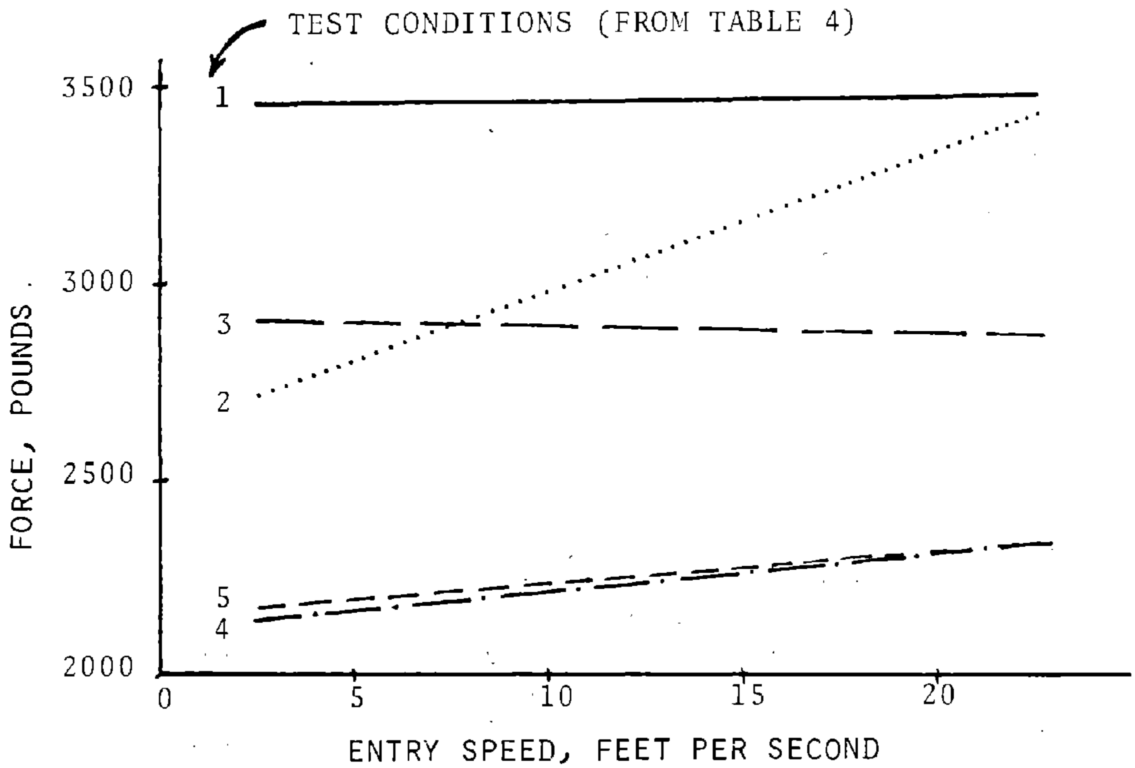


Fig. 42.

Drift Resistance, Test Zone IV, Without Restraining Rail.

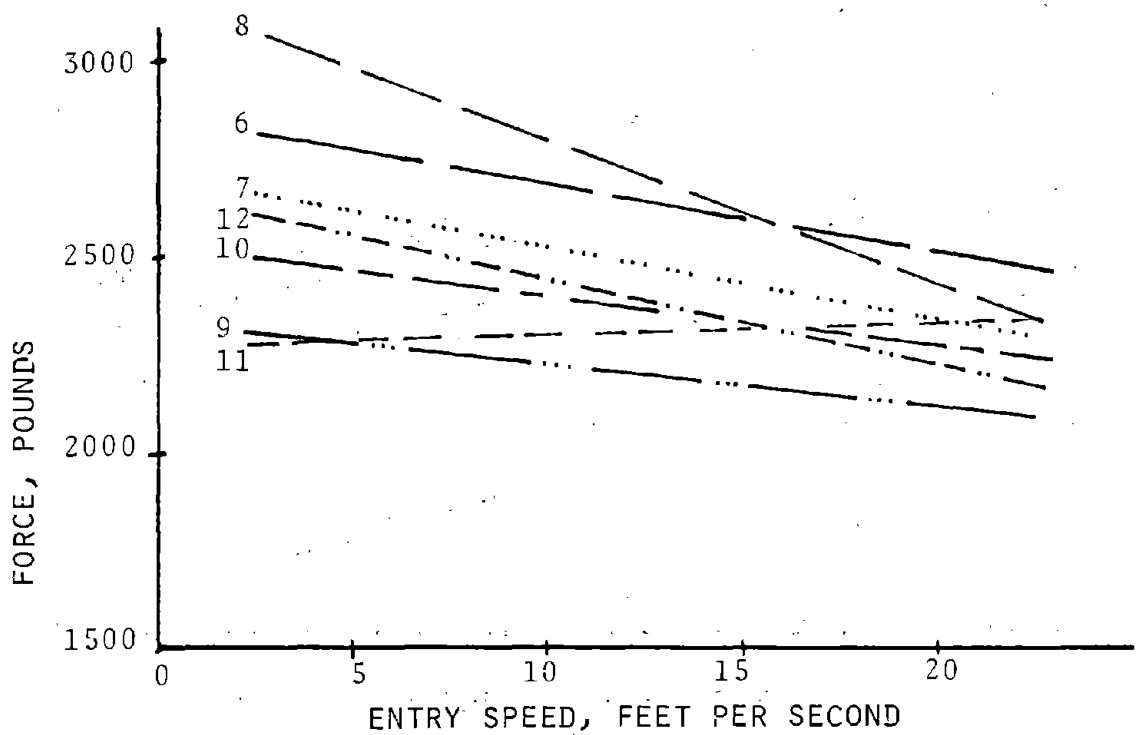


Fig. 43

Drift Resistance, Test Zone IV, With Restraining Rail.

feet. A larger bead was applied for the heavy applications to raise the amount of grease on the rail to approximately 8.0 ounces per 100 feet.

In spite of the wide scatter of data, it is clear that lubrication reduced both the drift resistance and noise level. Comparing the forces under Test Zone III in Table 4, the reduction from the level for a dry high rail ranges from 59% to 69% with a lubricated restraining rail and to 71% with a lubricated high rail. For Test Zone IV, the reduction ranges from 39% to 59% with a lubricated restraining rail, and from 29% to 62% with a lubricated high rail. A wide variation is seen in the forces obtained with water spray conditions (with and without lubrication) when compared to the dry high rail condition.

Lubrication of the high rail in Test Zone II, a 340-ft. radius curve with 1.5-inch superelevation reduced that part of the drift resistance force resulting from curvature to an average of 1100 pounds, a reduction of 25% from 1470 pounds. This is considerably less than the average reduction of 66% obtained in Zone III and 46% in Zone IV with lubrication of the high rail. It indicates that with the operating mode and low speeds used, lubrications would extend the life of the high rail in Zone II only about 30%, while it would extend the life of the high rail about 200% in Zone III and 100% in Zone IV.

The lubricated restraining rail in Zone III could be expected to have almost three times the life of the dry high rail, if the wear occurs over equivalent areas of the two rails. Observation of the rails worn in service, however, clearly indicates that the wear is distributed over a larger area of a restraining rail than a high rail. If on the average, the wear area is 1.5 times the wear area of the high rail, then the life of the lubricated restraining rail would be 4.2 times the life of the dry high rail. In Zone IV, the lubricated restraining rail could be expected to last three times as long as the dry high rail.

A restraining rail is a noise generator added to the track system unless it is dampened to suppress vibrations in the audible range, or lubricated to reduce the slip-stick type of friction that produces noise-generating vibrations.

After a restraining rail is installed to keep wheels from flanging on the high rail, the noise from flange friction will be eliminated; but the high rail will continue to generate noise from surface friction. The low rail will do the same and, in addition, will continue to produce noise from gage side friction whenever wheels flange on it. Unless a restraining rail is lubricated, more noise is likely to be heard on a curve after it is installed than before. Restraining rails are generally lubricated to reduce both noise and wear.

Nothing much is done about the noise and wear resulting from wheels flanging on the unlubricated low rails. In some cases, the rail could be lubricated or damping the rail or wheels may be helpful. Flanging on the low rail may be avoidable with changes in the operating modes and speeds that cause specific cars in a transit system to flange on the low rails of specific curves; but such changes are not determinable from available information on car characteristics and track geometry.

8. DESIGN CONCEPTS

8.1 GENERAL CONSIDERATIONS

Restraining rails and fasteners were examined critically in order to find potential opportunities for improvement and to arrive at concepts that can improve the performance of restraining rails and/or reduce their costs. Often the procurement and installation costs of structures and components can be reduced by simplifying designs to use less material, less complicated shapes and fittings, off-the-shelf structural members, and bent plates or weldments in place of castings. Costs may also be reduced by allowing members to flex and distribute load, and thus permit the use of smaller components.

Maintenance costs can often be reduced by using wear-resistant materials, lubrication, low-friction coatings, replaceable wearing strips, fittings that are easy to adjust and replace, and arrangements that allow flexing to improve the distribution of loads and thus reduce wear and stress concentrations. Controlled flexibility is especially desirable where a curve has irregularities in it that otherwise cause high lateral accelerations and result in high peak forces.

Noise can be reduced by lubrication, special surface coatings, changes in the size and shape of contact surfaces, reducing track irregularities, damping of rails and wheels to reduce noise generation, and treatment of adjacent surfaces to reduce the reflection of noise.

In general, acceptable design concepts should have one or several of the following features:

- Simplified shape or arrangement.
- Lower weight than that of structures now in use.

- Wear-resistant strip.
- Use of available components where feasible.
- Ease of adjustment.
- Ease of replacement in whole or part.
- Require few and simple maintenance tasks.
- Require minimum maintenance parts.
- Require no special tools.

Potential opportunities for savings in design and installation were examined in the use of standard structural shapes for restraining rails rather than heavy Tee rail sections, and in the development of adjustable braces for vertical restraining rail installations. Adjustable braces would permit most of the wear to be taken by the restraining rail, and facilitate its replacement without disturbing the running rails. Also the use of a wear-resistant rubbing strip, welded to a structural steel restraining rail, should extend its useful life by a large factor.

High manganese steel (10-13%) was considered the most favorable material for rubbing strips after evaluation of the data on the wear-resistant properties of many steel alloys that are reported in the literature. It has shown superior performance in frogs, guardrails at switch points, and (when it was available) in curve rails. Curve rails of high manganese steel observed in NYCTA elevated track in Brooklyn in 1979 were said to have been installed in 1919. Although rail sections of high manganese steel are no longer available, this wear-resistant steel is available in bars, plates and castings.

8.2 CONCEPTS

Two preliminary concepts, shown in Figures 44 and 45, were developed for new designs of restraining rails, using structural

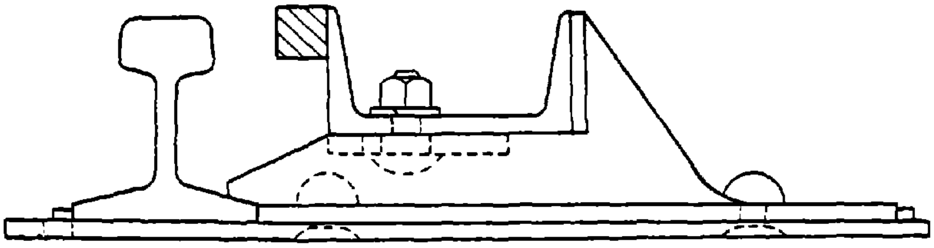


Fig. 44

Combo Restraining Rail on Conventional Brace Plate.

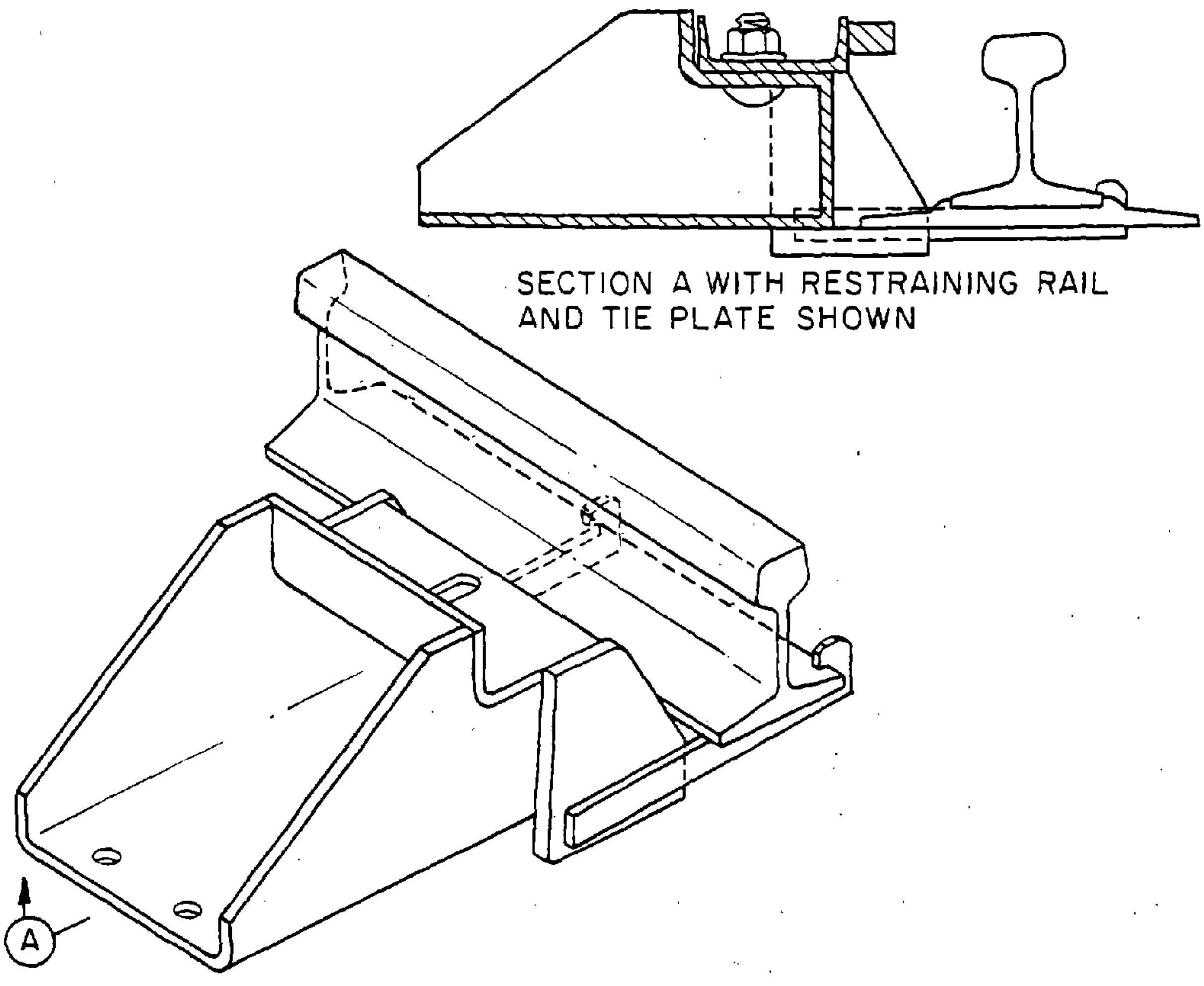


Fig. 45

Retrofit Brace for Restraining Rail.

shapes and wear-resistant steel rubbing strips to lengthen the useful life of the rails.

The concept shown in Figure 44 would substitute for a Tee rail in the horizontal position with widely spaced braces. Bending this composite rail for a short-radius curve would be difficult (as in the case of a Tee rail with part of one flange sheared off) because the section is not symmetrical about a vertical axis. The brace and plate assembly is similar to some that are in use at present on direct fixation track. It may be possible to simplify it by cutting it from one plate and folding up the two sides. The small plate to which the restraining rail is bolted would be welded to the sides. The back plate might be formed by cutting and folding two halves of it from the large plate and welding them where they meet.

The wear-resistant steel bar (1.5 x 1.75) would be skip welded to a structural steel channel (MC 7 x 22.7) for use with widely spaced fasteners. The rubbing strip would be set at an elevation where its lower edge would be higher than the lowest position of a wheel flange so that step wear would not develop.

Figure 45 shows a concept for a lightweight restraining rail for use with closely spaced fasteners. It would substitute for a Tee rail in the vertical position and would not require drilling and bolting to the low rail. The base and sides of the brace are planned to clear the tie plate; so the brace could be placed on a tie, tipped and slid forward to place the flange hooks under and beyond the low rail, then pulled back to engage the flange hooks, and fastened to the tie with screw spikes. The lateral force from a car wheel would be transferred to the low rail through the flange hooks, while the vertical wheel load is on the rail; so the lateral force imposed on the spikes would be small.

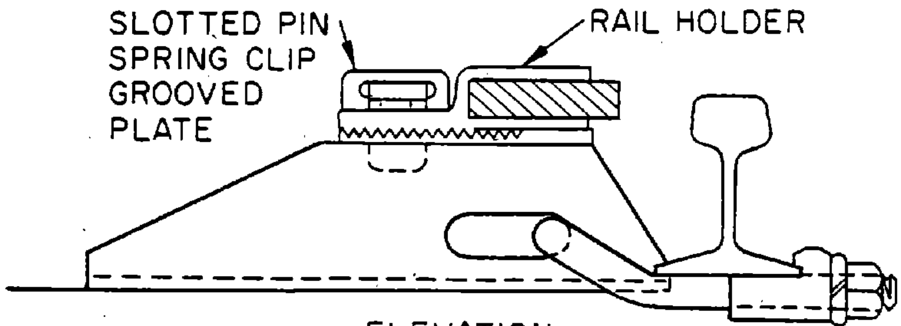
The preliminary concepts presented above require simplification to reduce material and fabrication costs and the work that would be involved in installing and adjusting the restraining rails.

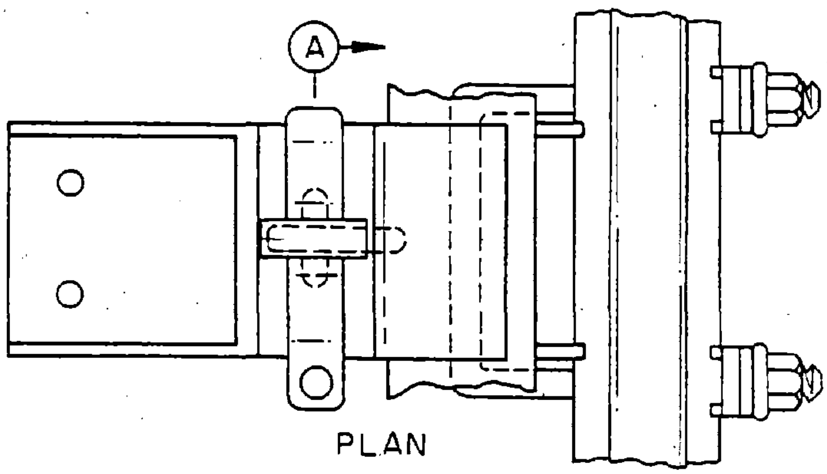
Consideration was given to the following: reducing the number of pieces to be fabricated, eliminating the need for drilling holes at exact locations in the rail, eliminating threaded fasteners, reducing the amount of welding, incorporating readily available parts that are already in use at transit properties, and providing some elasticity in the fastener. This last would reduce peak loads on the restraining rail, so that a lighter weight rail would have adequate strength.

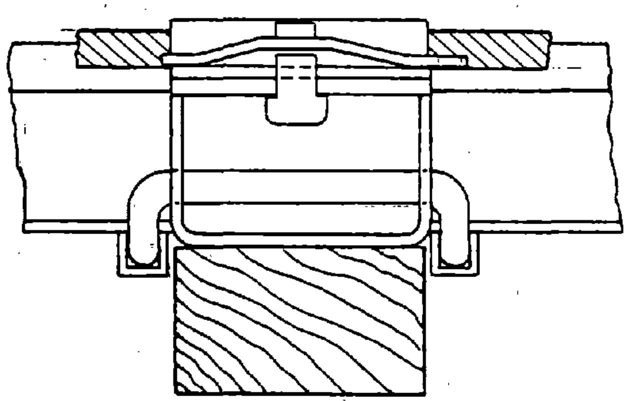
The design concept was modified to permit the use of a flat steel bar at lower estimated cost in cases where fasteners are closely spaced, as in ballasted track. This is feasible with the introduction of fastener components that eliminate needs for welding and drilling the bar. Figure 46 shows a concept for a flat restraining rail that is bonded with structural cement into a short, grooved holder that is seated on the grooved steel top plate of a brace. The rail holder and top plate are grooved so that the restraining rail can be adjusted laterally to compensate for wear, without the use of shims. Grooved components with bolts have been used successfully on the Alaska Railroad[29] to fasten running rails and facilitate lateral adjustments. For the restraining rail, it appears that a spring clip and pin can be used instead of a threaded fastener to keep the rail holder firmly in the grooves.

As an alternative to the hooks shown in Figure 45, the brace in Figure 46 is shown clamped to the base of the low rail by a bent

[29] Weber, J. W., Evaluation of Improved Track Structural Components Under Sub-Artic Conditions, Report of the Portland Cement Association Research and Development Construction Technology Laboratories to the Federal Railroad Administration, Report No. FRA/ORD - 79/01 Jan 1979.







NOTE: THE TIE PLATE UNDER THE LOW RAIL IS OMITTED FROM THE SKETCH FOR CLARITY.

Fig. 46

Wear-Resistant Restraining Rail with Retrofit Brace.

rod with clips and nuts of a type regularly used with gage bars, so that it can be used in retrofit installations. In each case, the brace and fasteners are clear of the tie plate. Properly designed, a fastener of this type could perform the functions of a pair of rail anchors while holding the brace securely.

Discussions of the value of compliance in the restraining rail supports, in order to distribute peak forces in both space and time, resulted in the concept shown in Figure 47. This modification adds a slider which is bonded to the restraining rail and which can move in the rail holder against an elastic pad.

The elastic pad would be bonded to the vertical surfaces of the rail holder and the slider at the factory. This subassembly would be placed on the brace in track to mark its fit to the rail, and the rail and slider would then be bonded together with structural cement before the holder is fastened to the brace. Lateral and longitudinal movement of the slider and rail would be restrained by the elastic pad.

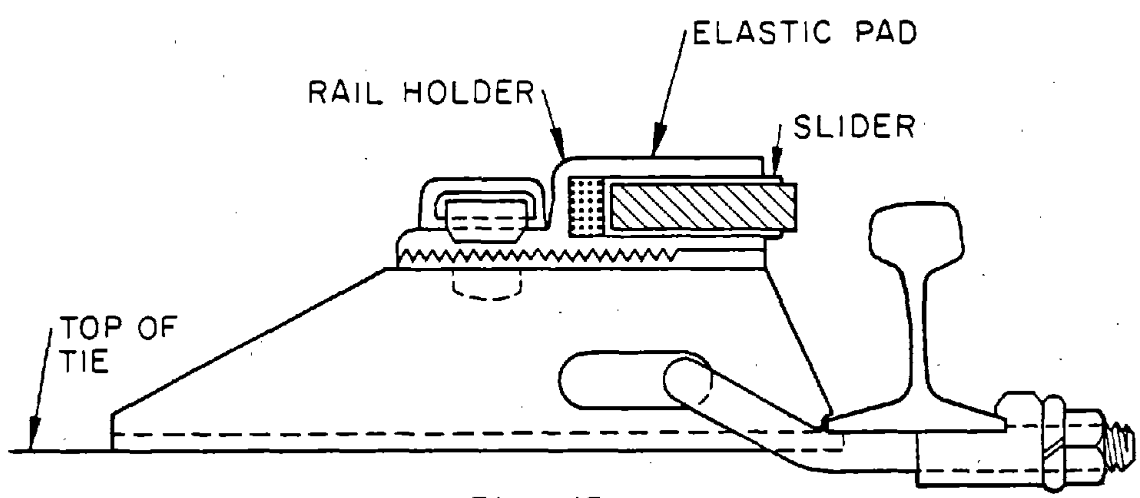


Fig. 47
Wear-Resistant Restraining Rail with Elastic Lateral Support.

Under lateral load, this arrangement would allow some distribution of the flange force over adjacent fasteners. It would also permit wheels to flange against the high rail when large lateral forces are encountered where the wheels do not normally contact the high rail.

Since the pad increases the distance from the face of the restraining rail to the pin that fastens the rail holder to the brace, it will increase the moments developed by any vertical forces on the restraining rail, which must be resisted by the pin. Accordingly, it appears that this type of restraining rail support will require further modification to make it suitable for slab track installations where the fasteners are widely separated.

The essential function of the connectors for vertical restraining rails is to prevent independent lateral deflection of the unbraced rail. The connectors now in use (high strength bolts and rigid chocks or heavy clamps and rigid spacers, some of which are adjustable) prevent independent vertical deflection of the rails as well as independent lateral deflection or rotation. As indicated, in Section 6 large forces may be transferred between the rails which would increase the maximum stresses. In addition, if the nuts loosen, large combinations of shear and tension tend to develop in the bolts when wheel forces are imposed.

Smaller bolts with rounded washers and pipe spacers could be used, which would transfer lateral forces and permit small vertical deflections without overstressing the bolts even when loose. These connectors would still prevent independent lateral deflection of the restraining rail, and (when coupled to lateral resistance between the rail flanges at the support) would limit the rotation of the restraining rail. If clearances permit, the connectors could be installed above the vertical supports.

9. CONCLUSIONS AND RECOMMENDATIONS

9.1 CONCLUSIONS

Restraining rails are desirable in mainline curves to a 500-foot radius and in longer radius curves that show excessive wear or a tendency of wheels to climb the high rail. This excludes turnouts and crossings where problems are controlled by speed restrictions. The economic value of a restraining rail value depends mostly upon savings to be made by reducing the wear of the high rail and the frequency of its replacement.

Tests of lubrication effects in short-radius curves showed substantial reductions in drift resistance.

Taking the life factor for dry, high rail as 1.0 (when used in a curve without restraining rail) the tests indicated an average life factor of 2.9 for a lubricated high rail in a curve of a 150-foot radius and 0.5 inch superelevation. In the same curve, a lubricated restraining rail (with a larger wear area) had an indicated life factor of 4.2, almost 50% more than the lubricated high rail.

At the test speeds (all below 15 mph), increases in curve radius or superelevation reduced the relative benefits of lubrication. In a curve of 340-foot radius and 0.5-inch superelevation, a lubricated high rail had an indicated life factor of 1.3. In a curve of 150-foot radius and 1.5-inch superelevation, a lubricated high rail had a life factor of 1.9, while a lubricated restraining rail (again considering the larger wear area) had a life factor of 3.1.

While the results of the lubrication tests cannot be extrapolated to other curves and cars and operating conditions, somewhat similar benefits may be expected from lubrication.

The costs of vertical restraining rail structures fabricated and installed under current designs could be reduced by using simple tube spacers. These should not transmit vertical loads and should permit small independent vertical deflections of either the running rail or the restraining rail. With these spacers, smaller bolts than those now used will be suitable, since large, combined stress concentrations will be avoided even when nuts loosen.

The maximum forces developed in a smooth curve when the cars are operated at the highest feasible speed would not exceed the capacity of restraining rails worn to limits specified by the properties. Accordingly worn restraining rails could be rotated and relaid to effect savings in some cases where other factors would not add greatly to the cost; such factors may include shearing the flange of the worn rail, drilling many new holes for connectors, and rebending rail for a curve having a different radius than the curve from which the rail was removed.

The remote hazard of a car wheel striking the end of the rail, if a car were to derail as it approached a restraining rail, could be reduced further by redesign of the facing end of the restraining rail installations.

Car characteristics and operating practices at each Transit Property affect the local suitability of the limits stated in the Guidelines (Volume II of this report) for measurable track conditions. Car characteristics include the resonant frequencies of the carbodies and trucks, truck suspension, distance between truck centers, numbers and locations of powered wheels, wheel base, wheel diameters and tread contour.

The wide scatter of the drift resistance data (Appendix B) indicated irregular dynamic behavior of the car, with intermittent flanging of the wheels, even in the drift mode and at low speeds.

Such behavior increases the rate of wear and energy requirements, and actions to reduce it would be highly desirable. Unfortunately the dynamic phenomena involved are not fully understood, and the effects of changes in speeds or operating modes and other factors cannot be foreseen completely.

Increasing the height of the restraining rail above the low rail has some obvious advantages in that it eliminates unfavorable step-type wear, provides a more effective guard, and increases the wear area so as to reduce the rate of wear. It also moves the contact point farther forward of the axle, which increases the leverage the flange force has in turning the truck through the curve, and reduces the radius of the wheel contact patch, which reduces its tangential velocity slightly.

The clearances of some cars permit moving the restraining rail to a height several inches above the surface of the running rails. All of the effects, however, cannot be foreseen, and some may be unfavorable.

Additional information is needed to define exactly the reactions that occur at flange and tread contacts between wheels and rails and to determine the effects of adjustments (such as the optimum height of the restraining rail) that could be made in track and car. Fully instrumented tests would be necessary to determine such things as exactly when and at what speeds wheels oscillate between free curving and flanging on either the restraining rail or low rail under the several operating modes, the flange forces that develop, and the lateral accelerations of the car. Such information would assist in determinations of track adjustments for best performance with existing cars, and it may also be of value in adjusting car designs for best performance.

Implementation of the Guidelines will be assisted by the development of low-cost automatic inspection equipment which is being done under programs in support of track maintenance.

The concepts presented in Section 8 provide valid opportunities for new cost-effective designs of restraining rails. However, the paths between concepts and useful products are long and involved. The development of useful products will require detailed investigation, design and fabrication of sample lengths of restraining rail and fasteners, installation and tests in track, modification of designs and installation procedures based on the results of tests, and funding for the installation of the tested restraining rails where new installations or replacements are needed.

9.2 RECOMMENDATIONS

Detailed tests should be conducted to obtain additional information with which to make further improvements in track adjustment (such as the optimum height of a restraining rail) and practices to reduce rail wear and smooth the ride of transit cars in curve track.

Modifications to connectors for vertical restraining rail should be designed, fabricated and tested; and other modifications to components with potential for reducing costs should be investigated.

Designs should be developed for the facing end of restraining rail installations that will reduce potential damage in the remote event that any approaching cars derail.

The concepts for new restraining rail designs presented in this report should be investigated in detail; preliminary and final designs should be completed for the two most promising concepts; and prototypes should be fabricated and tested in laboratories and in transit track.

APPENDIX A

DATA FROM TRANSIT TRACK CURVES

A.1 NOTATION

The collection of curve data is discussed in Section 3.2 of the report.

In order to compress the data and simplify their arrangement, abbreviations and symbols are used in the tables. The notation used is as follows:

A - Automatic
B - Bolted Joints
C - Control-Cooled Rail
H - High Rail
L - Low Rail
M - Manual
R - Restraining Rail
S - Alloy Steel
T - Heat-Treated
W - Continuous Welded Rail
a - accelerating
b - bolted to low rail
c - constant speed or drifting
f - direct fixation fastener
h - horizontal
k - kip, 1000 pounds
m - mixed
p - tie plate
s - braking

v - vertical
x - brace
MGT - Million Gross Tons
ft - feet
in - inches
lbs - pounds
max - maximum
mph - miles per hour
yr - year

Blank Space - the omitted data are the same as the last entry on the left

A.2 DATA TABLES

TABLE 5. DATA FROM CURVES WITH RESTRAINING RAILS

CURVE NUMBER			1	2	3	4	5	6	7	8	9	10
Radius, ft			115	115	168	217	420	212	422	200	140	500
R Rail Length, ft			109		150	220	62	74	112	325	143	296
Superelevation, in			1.8		2.9	2.3	2.4	0		5.0	2.8	3.0
R Rail	weight, lbs		100									
	position		V									
	type		TW		TB							
R above L, in			0.5		0.3			1.0	0.3	1.0	0.3	
Fastener Spacing, in			xp 19.8		b26 p24						b26 p28	
Flangeway Width, in			2.1	2.0	1.8	1.9	1.7		1.8	1.6	2.0	1.7
Lubrication			AHR	AHR	MHR							
H Rail	weight, lbs		100									
	type		TW		TB							
Gage Wear, max, in		R	0.30	0.15	0.05	0.00		0.05				
		H	0.05		0.30	0.40	0.55	0.65	0.50	1.00	0.50	
Useful Life, yrs		R	16	30	2		7	4	8	4	24	21
		H	30	40	2		7	4	8	4	24	21
Design Speed, mph			8		13	4	19	7	10	18	12	23
Mode			m	a	c	a		c			a	c
Traffic, MGT/yr			31	15	10			12		10	4	8
Car Weight, max, k			84.0		151						77.3	
Car Length, ft			51.3		67.5						55.3	
Truck Centers, ft			33.0		47.5						38.0	
Wheel Base, ft			6.83		6.70						6.67	
Wheel Diameter, in			28		36						28	28
Wheel Conicity			0		1:20							

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

TABLE 5. (cont.)

CURVE NUMBER			11	12	13	14	15	16	17	18	19	20
Radius, ft			575	215	441	400	450	407	300	994	1006	994
R Rail Length, ft			362	234	200	56	55	322	170	155	158	136
Superelevation, in			2.8	5.8	4.8	4.0	2.5	1.9	3.5	2.0	2.5	1.9
R Rail	weight, lbs		100		115	80						
	position		v			h						
	type		B									
R above L, in			1.0		0.3	1.0	0.5			0.0		
Fastener Spacing, in			b26. p23			x36 p18		x48 p24	x36 p18			
Flangeway Width, in			1.6	1.5	1.8	2.8	2.6	2.0	2.1	2.0	2.3	
Lubrication			MR	MHR		AHR						
H Rail	weight, lbs		100		115	100				90		
	type		TB		W	TW		TB				
Gage Wear, max, in		R	0.05			0.35	0.10	0.05				0.20
		H	0.40	0.05		0.30	0.25	0.05	0.20	0.25	0.30	0.20
Useful Life, yrs		R	30	20	24	30	35	40	35			15
		H	30	20	24	30	35	40	25	20		30
Design Speed, mph			24	18	26	30	28	25	35			
Mode			a	s		c			a	c		
Traffic, MGT/yr			8			10			9	10		
Car Weight, max, k			51.3			51.5						
Car Length, ft			55.3			48.0						
Truck Centers, ft			38.0			33.7						
Wheel Base, ft			6.67			6.50						
Wheel Diameter, in			28									
Wheel Conicity			1:20			0						

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

TABLE 5. (cont.)

CURVE NUMBER		21	22	23	24	25	26	27	28	29	36
Radius, ft		1006	206	194	394	406	637	815	205	229	700
R Rail Length, ft		157	266	251	372	383	260	360	250	265	250
Superelevation, in		2.3	3.5	4.0	2.8	2.5	2.4	4.1	5.0		4.0
R Rail	weight, lbs	80					100				115
	position	h					v				h
	type	B					TB				
R above L, in		0.0					0.5				0.3
Fastener Spacing, in		x36 p18					b26 p24				x30 f30
Flangeway Width, in		2.1	2.8	2.4	2.1	1.9	1.8			1.6	2.0
Lubrication		AHR									0
H Rail	weight, lbs	90					100				115
	type	TB					TW	TB			TW
Gage Wear, max, in	R	0.35	0.10	0.25	0.05		0.10	0.20	0.05		0.35
	H	0.20	0.25	0.30	0.10	0.50	0.30		0.25	0.10	0.35
Useful Life, yrs	R	40		35	40			35	40		5
	H	40	30	25	30	40	25	25	20	35	8
Design Speed, mph		35	20		25		40	15	20		40
Mode		m	c					b		a	m
Traffic, MGT/yr		9.8	5.1				5.3				14
Car Weight, max, k		51.5	58.0				105				103
Car Length, ft		48.0	48.3				67.8				75.0
Truck Centers, ft		33.8					47.5				52.0
Wheel Base, ft		6.50	6.00				7.50				7.50
Wheel Diameter, in		28	26				28				
Wheel Conicity		0					1:20				0

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

TABLE 5. (cont.)

CURVE NUMBER		37	38	39	40	41	42	43	44	45	46
Radius, ft		700	530	500	600	148	147	900			
R Rail Length, ft		270	817	788	550	250	250	500	600	880	
Superelevation, in		4.0		5.0	4.5	2.0	3.0		6.1	3.5	3.0
R Rail	weight, lbs	115	90			100					
	position	h				v					
	type	TB									
R above L, in		0.3									
Fastener Spacing, in		x30 f30	x60 f30			x36 p18		b52 p21	x36 p21	x48 p24	b52 p24
Flangeway Width, in		2.0				2.3	2.2	2.0	2.4	2.2	2.1
Lubrication		0	AR			AHR		MHR			
H Rail	weight, lbs	115	119			100					
	type	TW				TB					
Gage Wear, max, in	R	0.35	0.05			0.10			0.05	0.10	0.05
	H	0.5	0.10	0.05	0.20	0.25	0.30	0.10		0.20	0.05
Useful Life, yrs	R	5	40	35	40	20		15	30		
	H	7	30	35	20	9	5	10	20	6	12
Design Speed, mph		40	30		40	10		25			
Mode		m	c				a		s		a
Traffic,		14	7		6	21					
Car Weight, max, k		103	81.3			112					
Car Length, ft		75.0				60.5					
Truck Centers, ft		52.0	50.0			44.6					
Wheel Base, ft		7.30	7.00			6.80					
Wheel Diameter, in		28	30			34					
Wheel Conicity		0				1:20					

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

TABLE 5. (cont.)

CURVE NUMBER			47	48	49	50	51	52	53	54	55	56
Radius, ft			350		500	290	500	235	216	120	350	1400
R Rail Length, ft			280	400	150	180	150	210	250	460	300	400
Superelevation, in			5.5	6.5	2.0	4.1	2.0	2.5	1.5	2.0	19.9	3.8
R Rail	weight, lbs		100					85				
	position		v					h				
	type		TB					B		SB		
R above L, in			0.3		0.5	1.0	0.5				0.6	0.5
Fastener Spacing, in			x52 p21	b52 x48				x24 p24		x18 p18	x36 p18	x48 p24
Flangeway Width, in			2.5	2.2	3.0	2.8	3.0	1.9		2.2	1.9	2.0
Lubrication			MHR			AHR	MHR	AHR				
H Rail	weight, lbs		100					85		115		
	type		TB					TW		SW	W	
Gage Wear, max, in		R	0.10	0.05						0	0.10	0
		H	0.30	0.20	0.30					0.10	0.55	0
Useful Life, yrs		R	30	30	25			20	18	10	20	35
		H	6	20	10	8	10	25	20	10	15	35
Design Speed, mph			23	25	16	18	16	20	20	10	20	40
Mode			s	c	s	a	s	a	b	s		c
Traffic, MGT/yr			21					4.7		8.8		
Car Weight, max, k			112		101			72.5		98.7		
Car Length, ft			60.5		51.3			48.8		69.8		
Truck Centers, ft			44.6		36.0			33.3		51.0		
Wheel Base, ft			6.80					6.50		6.80		
Wheel Diameter, in			34					28				
Wheel Conicity			1:20									

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

TABLE 5. (cont.)

CURVE NUMBER		57	58	59	60	61	62	63	64	65	66
Radius, ft		800	500	75	500	516	512		1500	1515	379
R Rail Length, ft		350	340	280	250	240	460	260	739	790	528
Superelevation, in		1.8	5.3	2.0	5.6	2.0	6.9	2.8	4.0		
R Rail	weight, lbs	85									
	position	h		v	h						
	type	B							SB		
R above L, in		0.6		0.5	0.6				0.4		
Fastener Spacing, in		x48 p24		x24 p24	x48 p24		x24 p24				
Flangeway Width, in		2.0		2.3	2.1	2.0	2.3		2.0	2.1	
Lubrication		AHR							AR		
H Rail	weight, lbs	115		85	115				100		
	type	W							SW		
Gage Wear, max, in	R	0.05	0	0.10	0.05	0	0.05				
	H	0.10	0.35		0.75	0.25	0.05	0.40	0.35	0.25	0.05
Useful Life, yrs	R	25		10	25	20	25	18	25	23	10
	H	25	20	8	20	16	20	15	10		5.5
Design Speed, mph		40	25	6	25				50		20
Mode		a	c			a	c	s	s	cs	s
Traffic, MGT/yr		10.4							23		
Car Weight, max, k		98.7							121		
Car Length, ft		69.8							57.0		
Truck Centers, ft		51.0							38.0		
Wheel Base, ft		6.80							7.00		
Wheel Diameter, in		28							28		
Wheel Conicity		1:20									

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

TABLE 5. (cont.)

CURVE NUMBER		67	68	69							
Radius, ft		416	692	1100							
R Rail Length, ft		996	767	1247							
Superelevation, in		3.0	4.0								
R Rail	weight, lbs	85	100								
	position	h									
	type	SB	B								
R above L, in		0.4	0.3								
Fastener Spacing, in		x24 p24	x60 f30								
Flangeway Width, in		2.4	2.0								
Lubrication		AR									
H Rail	weight, lbs	100	115								
	type	TW	W								
Gage Wear, max, in	R	0.25	0.05								
	H	0.35	0.20								
Useful Life, yrs	R	10	15	20							
	H	5	10	15							
Design Speed, mph		25	27	33							
Mode		ca	cb	ac							
Traffic, MGT/yr		23									
Car Weight, max, k		121									
Car Length, ft		57.0									
Truck Centers, ft		38.0									
Wheel Base, ft		7.0									
Wheel Diameter, in		28									
Wheel Conicity		1:20									

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

TABLE 6. DATA FROM CURVES WITHOUT RESTRAINING RAILS

Curve Number		70	71	72	73	74	75	76	77	78	79
Radius, ft		1500		955	1002	1002	2860	955	955	601	584
Superelevation, in		4.5	5.0		2.3	2.4	4.0	3.0		4.0	2.0
H Rail	Weight, lbs	119	115	100			115	100	115	100	
	Type	TB					TW	W		B	
Fasteners Spacing, in		p24					f30	p24			
Lubrication		O	AH	O			MH	AH			
Max Gage Wear		0.10	0.25	0.05	0.20	0.35	0.20	0.05	0.10	0.25	
Useful Life		10	8	15	10	6	10	20		15	
Design Speed, mph		35	40	30	30		40	35			30
Mode		c	a		s	a		c			
Traffic, MGT/yr		15.4		8.0	10.0						
Car Weight, Max, k		85		77	151			71			
Car Length, ft		51.3		55.3	67.5			48			
Truck Centers, ft		33.0		38	47.5			33.8			
Wheel Base, ft		6.8		6.7				6.6			
Wheel Diameter, in		28			36			26			
Wheel Conicity		0		1:20				0			
Curve Number		80	81	82	83	84	85	86	87		
Radius, ft		318	800	755	299	314	600	1400	5993		
Superelevation, in		1.8	8.0	3.0	1.0		5.0	1.6	1.5		
H Rail	Weight, lbs	100	132	115			119	115			
	Type	B	W				TW	W			
Fasteners Spacing, in		p24	f30		p24		f30	p24	f30		
Lubrication		O	AH	O			MH	AH	O		
Max Gage Wear		0.40	0.35		0.60	1.10	0.20	0.05	0.05		
Useful Life		13	25	8	4	2	6	35	25		
Design Speed, mph		20	40	30	12		20	40	50		
Mode		c		m	c	m	c	a			
Traffic, MGT/yr		5.3		6.0			5.9	7.5	23		
Car Weight, Max, k		105		103			81.3	98.7	121		
Car Length, ft		67.8		74.8			75.0	69.8	57.0		
Truck Centers, ft		47.5		52.0			50.0	51.0	38.0		
Wheel Base, ft		7.50		8.50			7.00	6.80	7.00		
Wheel Diameter, in		28		28			30	28			
Wheel Conicity		1:20		0			1:20				

Note: Repetitive data are omitted. Blank spaces indicate that the data are the same as the last entries to the left.

APPENDIX B
TEST RESULTS

B.1 GENERAL

A test car was run through one tangent and three curve test zones in the drift (coasting without power or braking) mode at the Transportation Test Center during the period September-November 1980. In each run, the car entered a test zone close to its target speed of 15, 10 or 5 miles per hour. Tests were run in two directions on tangent track only. The tests were run under track conditions as follows:

1. Dry rails; no restraining rail
2. Water spray on both rails; no restraining rail
3. Light application of lubricant type 1 on the gage side of the high rail; no restraining rail
4. Same as condition 3, with water spray added
5. Heavy application of lubricant type 1 on the gage side of the high rail; no restraining rail
6. Dry rails with restraining rail installed
7. Water spray on all rails with restraining rail installed
8. Light application of lubricant type 1 to restraining rail
9. Same as condition 8 plus water spray on all rails
10. Heavy application of lubricant type 1 to restraining rail
11. Light application of lubricant type 2 to restraining rail
12. Heavy application of lubricant type 2 to restraining rail

The tests are described in Section 3.3 of the report.

Tests with a sliding friction device pulled against the gage side of the high rail indicated a coefficient of static friction of 0.58 for a dry rail and 0.36 for a lubricated rail.

B.2 NOTATION

The notation used in the tables of data is as follows:

s Distance in feet through a section of a test zone measured to 0.001 foot. Zeros are omitted after decimal points. The distance from the centers of the last speed trap passed to the end of the rollout, was measured to 0.01 foot.

v Speed in feet per second at the start of the test section or at the last speed trap passed before rollout. It is the speed at the center of the trap obtained from the trap distance and the times at the trap entry and exit markers, recorded to 0.0001 second.

$\bar{v}$ Average speed in feet per second over the distance obtained by dividing the distance by the elapsed time. A few exceptions are noted in the tables for rollouts where the time was not recorded and v was taken as half the entry speed, v/2. In a few cases where time was recorded with apparent error, $\bar{v}$ is calculated both ways.

t Elapsed time over the distance was obtained from the centers of the speed traps at the ends of a section. For the few rollouts where the time was not recorded or in error, it was taken as 2s/v.

F Drift resistance force was found from:

$$F = m\Delta v/2$$

where m is the mass of the car, 95,500/32.2 in slugs and Δv is the difference between the entry and exit speed.

At the few rollouts where t was not available (mentioned under $\bar{v}$ above and noted in the tables), F was obtained from the entry velocity and the rollout distance using the relation:

$$F = mv^2/2s$$

This calculation is less accurate, because it includes a presumption that the deceleration rate was constant.

B.3 TABLES OF DRIFT RESISTANCE FORCES

The tables of drift resistance forces in this appendix are arranged under the test conditions numbered and described in Section B.1. Data sets from separate test runs under a single test condition are separated by blank lines. The data in the last line of each set were obtained from a rollout to a complete stop.

TABLE 7. ZONE I, TANGENT TRACK

Condition 1

s	v	$\bar{v}$	t	F
50	20.93	20.25	2.47	1640
195	19.56	16.09	12.12	1700
50	12.61	11.27	4.44	1710
*89	10.04	5.02	17.73	1680
50	15.54	14.68	3.40	1520
174.8	13.80	6.81	25.69	1590
50	8.22	6.14	8.14	1510
17	4.08	1.99	8.56	1410
50	21.26	20.35	2.46	1980
195	19.62	15.42	12.65	2000
50	11.11	9.24	5.41	2050
*39.25	7.37	3.69	10.64	2060
50	15.57	14.45	3.46	1820
143.5	13.45	6.65	21.59	1850
44.17	6.23	3.08	14.32	1290

Condition 2

s	v	$\bar{v}$	t	F
50	21.57	21.00	2.38	1530
195	20.34	17.36	11.23	1560
50	14.42	13.40	3.73	1520
*153.7	12.51	24.55	6.26	1510
50	16.27	15.49	3.23	1405
195	14.74	10.65	18.31	1300
50	6.73	4.37	11.45	1180
*4.5	2.16	1.08	4.17	1530
50	8.95	7.17	6.98	1480
31.17	5.47	2.71	11.51	1410
50	21.02	20.11	2.49	1950
195	19.38	15.96	12.22	1630
50	12.68	11.56	4.33	1560
101.2	10.40	5.15	19.65	1570
50	15.90	14.90	3.36	1730
164.7	13.95	6.82	24.16	1710
51.75	8.46	3.99	12.96	1940

Note: *Time not recorded, $\bar{v} = v/2$ is used to calculate t and F.

TABLE 8. TEST ZONE II, CURVE, 340-FT. RADIUS, 0.5 IN. SUPERELEVATION

Condition 1

s	v	$\bar{v}$	t	F
50	22.62	21.68	2.31	2690
100	20.53	17.17	5.83	3410
83	13.84	6.74	12.31	3330
50	15.30	13.67	3.66	2760
63.25	11.90	5.20	12.17	2900
34.17	7.61	4.12	8.30	2720

Condition 5

s	v	$\bar{v}$	t	F
50	21.54	20.66	2.42	2280
100	19.69	16.91	5.91	2760
95.75	14.17	6.52	15.31	2740
50	13.83	12.28	4.07	2380
58.3	10.56	5.38	10.70	2930
43	8.17	4.12	10.44	2190

Condition 3

s	v	$\bar{v}$	t	F
50	22.13	21.24	2.35	2380
100	20.24	17.37	5.76	2960
103	14.49	7.17	14.37	2990
50	14.59	13.37	3.74	2080
80.5	11.97	5.90	13.65	2600
50	10.03	6.16	8.12	2900
3.0	2.08	0.83	3.60	1710

TABLE 9. ZONE III, CURVE, 150-FT. RADIUS, 0.5 IN. SUPERELEVATION

Condition 1

s	v	$\bar{v}$	t	F
50	21.33	20.10	2.49	3350
100	18.52	13.51	7.40	3990
+27.75	8.55	4.28	6.48	3910
		(3.95)	(7.02)	(3610)
50	14.56	11.97	4.18	3890
28.08	9.08	4.36	6.43	4180
24.33	7.82	4.37	5.57	4160

Condition 2

s	v	$\bar{v}$	t	F
50	22.70	21.78	2.30	2610
100	20.68	18.21	5.49	2630
50	15.81	14.33	3.49	2550
*95	12.81	6.41	14.83	2560
50	16.92	15.45	3.24	2780
51.5	13.88	4.44	11.61	3540
32	8.36	4.20	7.62	3250

Condition 3

s	v	$\bar{v}$	t	F
50	20.17	19.32	2.59	2020
100	18.41	16.15	6.19	2170
50	13.88	12.42	4.02	2160
82.25	10.95	5.72	14.39	2260
50	20.98	20.20	2.47	1920
100	19.39	17.35	5.76	2110
50	15.27	14.01	3.57	2110
50	14.59	13.02	3.84	2420
75.75	11.45	5.85	12.95	2500
23.67	5.71	3.06	6.92	2450

Condition 4

s	v	$\bar{v}$	t	F
50	21.97	21.20	2.36	1930
100	20.43	18.54	5.39	2100
50	16.62	15.50	3.23	2040
147	14.40	7.18	20.49	2080
50	14.49	12.84	3.89	2420
77.34	11.29	5.83	13.21	2520
25.25	5.92	3.00	8.39	2090

Condition 5

s	v	$\bar{v}$	t	F
50	21.08	20.17	2.48	2360
100	19.11	16.50	6.06	2560
50	13.88	11.41	4.38	2450
58.25	10.25	5.02	11.62	2620
50	13.92	12.20	4.10	2200
78.5	10.88	5.44	14.43	2240
35	7.74	3.70	2.61	2430

Condition 6

s	v	$\bar{v}$	t	F
50	21.32	20.49	2.44	2260
100	19.46	16.89	5.92	2170
50	15.14	12.50	4.02	2660
69.73	10.69	5.73	12.13	2610
50	15.97	13.85	3.61	2310
94.42	13.14	6.38	14.79	2630
29.5	6.91	3.39	8.70	2360

NOTES:
*Time was not recorded, $\bar{v} = v/2$ is used to calculate t and F.
+Recorded time appears to be in error (data are included in line below in parenthesis), $\bar{v} = v/2$ is used to calculate t and F.

TABLE 9. (CONTINUED)

	Condition 7					Condition 10			
s	v	$\bar{v}$	t	F	s	v	$\bar{v}$	t	F
50	20.17	19.38	2.58	2170	50	21.53	20.75	2.41	2200
100	18.28	15.50	6.45	2610	100	19.74	17.18	5.82	2600
50	12.61	10.42	4.80	2760	50	14.63	12.95	3.86	2600
35.45	8.15	4.34	8.16	2960	+71.1	11.24	5.62	12.65	2630
							(8.29)	(8.58)	(3890)
50	14.27	12.56	3.98	2510	50	14.67	13.37	3.74	2090
65.83	10.90	5.54	11.88	2720	94	12.03	6.62	14.21	2510
46.25	8.04	4.01	11.53	2070	34.5	7.47	4.42	7.80	2840
	Condition 8					Condition 11			
50	22.30	21.74	2.30	1840	50	20.64	19.69	2.54	2250
100	20.87	18.76	5.33	2370	100	18.71	16.18	6.18	2370
50	16.61	15.29	3.27	2410	50	13.77	12.05	4.15	2390
117.4	13.95	7.49	15.67	2640	71.9	10.42	5.45	13.19	2340
50	13.77	12.50	4.00	1900	50	14.62	12.76	3.92	2830
89.2	11.21	5.96	14.97	2220	59.3	10.88	5.28	11.23	2870
31.5	7.27	3.67	8.59	2510	38.3	7.79	3.81	10.06	2300
	Condition 9					Condition 12			
50	18.75	18.38	2.72	980	50	20.70	19.84	2.52	2140
100	17.85	16.13	6.20	1660	100	18.88	16.56	6.04	2210
50	14.38	13.26	3.77	1750	50	14.38	12.85	3.89	2270
119.6	12.15	6.40	18.68	1930	86.8	11.40	5.51	15.76	2150
50	14.00	12.85	3.89	1750	50	14.56	13.02	3.84	2250
100	11.70	7.75	12.90	1820	96.8	11.65	6.80	14.24	2430
+10.7	3.80	1.90	5.63	2000					
		(2.37)	(4.52)	(2490)	43	7.77	4.12	10.42	2210
43.15	7.48	4.10	10.53	2110					

NOTES:
+Recorded time appears to be in error (data are included in line below in parenthesis), $\bar{v}$ = v/2 is used to calculate t and F.

TABLE 10. ZONE IV, CURVE 150-FT. RADIUS, 1.5 IN. SUPERELEVATION

Condition 1

s	v	$\bar{v}$	t	F
50	20.52	19.12	2.61	3430
100	17.50	12.73	7.86	3540
27.13	8.11	4.19	6.48	3710
50	14.45	12.06	4.15	3420
41.25	4.81	1.13	8.58	3340
24.75	7.57	3.75	6.61	3400

Condition 2

s	v	$\bar{v}$	t	F
50	21.24	19.64	2.55	3700
100	18.06	14.45	6.92	3050
60.17	10.96	5.67	10.69	3040
50	14.55	12.46	4.01	3020
59.58	10.46	5.28	11.27	2760
20.33	6.45	3.32	6.13	3120

Condition 3

s	v	$\bar{v}$	t	F
50	22.97	22.20	2.25	2570
100	21.02	17.92	5.58	3310
97.5	14.79	7.38	13.21	3320
50	14.19	12.56	3.98	2440
71.75	10.92	5.56	12.91	2510
26.58	7.40	3.84	6.93	3160

Condition 4

s	v	$\bar{v}$	t	F
50	20.96	20.02	2.50	2280
100	19.03	16.76	5.97	2270
125	14.47	9.93	12.58	2100
23.52	5.55	2.79	8.43	1950
50	14.14	12.58	9.25	2330
2.58	11.02	5.72	3.14	2370
50	9.15	5.41	9.25	2400

Condition 5

s	v	$\bar{v}$	t	F
50	21.54	20.26	2.47	2240
100	19.68	17.63	5.67	2290
125	15.31	11.35	11.01	2190
31.02	7.18	2.98	10.42	2070
50	14.75	13.27	3.77	2430
82.67	11.66	5.67	14.58	3370

Condition 6

s	v	$\bar{v}$	t	F
50	19.64	18.45	2.71	2580
100	17.28	14.10	7.09	2560
75.56	11.15	5.78	13.07	2530
50	14.13	12.35	4.05	2620
+65.25	10.55	5.28	12.36	2530
		(4.92)	(13.26)	(2360)
37.33	8.70	4.26	8.76	2950

NOTE: +Recorded time appears to be in error (data are included in line below in parenthesis), $\bar{v} = v/2$ is used to calculate t and F.

TABLE 10. (CONTINUED)

Condition 7

s	v	$\bar{v}$	t	F
50	19.98	17.16	2.91	2250
100	17.77	14.93	6.69	2430
95.98	12.29	6.52	14.71	2480
50	13.26	11.44	4.37	2490
52.58	9.59	4.72	11.14	2550
47.92	8.22	4.74	10.10	2410
37	8.15	4.04	9.16	2640

Condition 8

s	v	$\bar{v}$	t	F
50	21.51	20.58	2.43	2380
100	19.56	16.98	5.89	2560
125	14.48	8.77	14.25	2370
5.6	3.09	1.86	3.00	3050
50	13.58	11.66	4.29	2720
48.7	9.64	5.19	9.38	3050
19.4	5.71	3.29	5.90	2870

Condition 9

s	v	$\bar{v}$	t	F
50	21.66	21.28	2.35	1410
100	20.54	18.48	5.41	2200
125	16.52	13.28	9.41	2040
70.8	10.05	5.55	12.75	2340
50	15.37	14.29	3.50	2050
100	12.95	8.29	12.08	2290
+8.3	3.64	1.82	4.56	2370
		(2.36)	(3.51)	(3080)
48.2	7.83	4.24	11.38	2040

Condition 10

s	v	$\bar{v}$	t	F
50	20.08	19.16	2.61	2190
100	18.15	15.80	6.33	2170
125	13.51	8.17	15.29	2060
5.8	2.87	1.42	4.08	2090
50	14.59	13.16	3.80	2360
78.3	11.56	6.57	11.92	2880
+22.9	6.53	3.27	7.00	2760
		(3.74)	(6.12)	(3160)

Condition 11

s	v	$\bar{v}$	t	F
50	20.47	19.53	2.56	2270
100	18.51	16.10	6.21	2240
125	13.82	8.33	15.00	2160
6.5	2.92	1.47	4.41	1970
50	13.31	11.49	4.35	2470
55.1	9.69	4.69	11.75	2450
25.6	6.69	3.30	7.75	2560

Condition 12

s	v	$\bar{v}$	t	F
50	19.36	18.32	2.73	2280
100	17.26	14.53	6.88	2310
96.78	11.90	6.80	14.23	2480
50	13.31	11.68	4.28	2270
66.5	10.03	5.96	11.15	2670
+36.7	7.92	3.96	9.62	2530
		(5.21)	(7.05)	(3330)

NOTE: +Recorded time appears to be in error (data are included in line below in parenthesis), $\bar{v}$ = v/2 is used to calculate t and F.

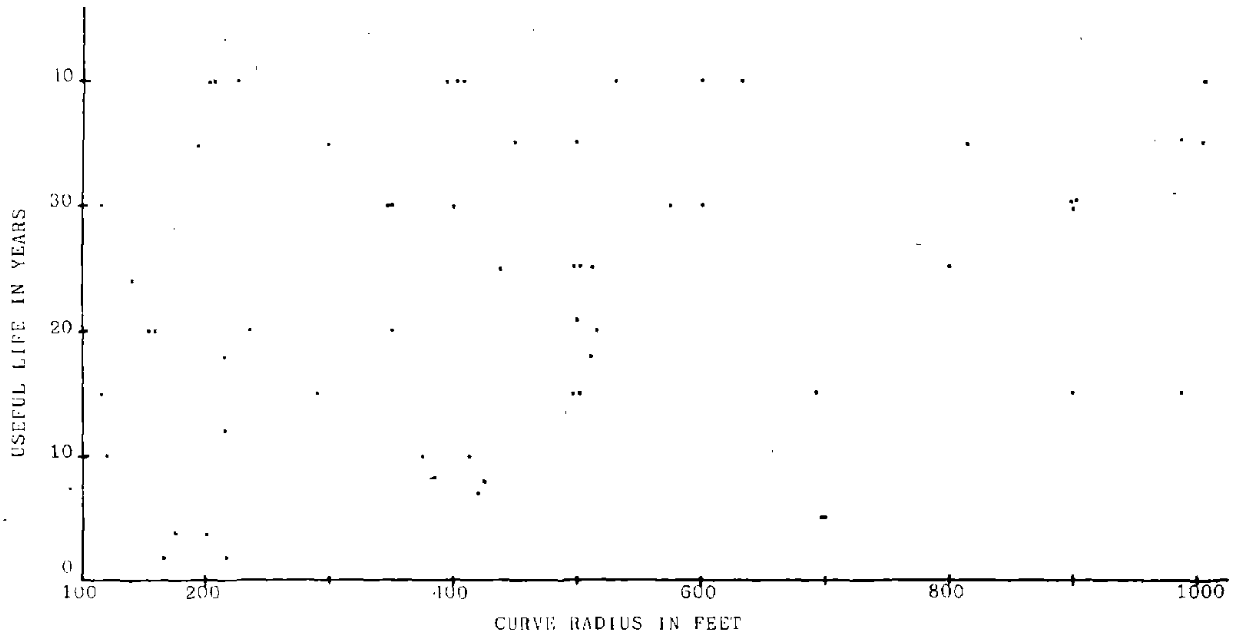


Fig. 48

Useful Life of Restraining Rail Versus Curve Radius.

The scatter of data points indicated no consistent patterns. Plots of useful life versus other data (such as gross tonnage, maximum wheel loads, type of rail steel, operating mode and weighted combinations) also produced random scatters of data points.

APPENDIX C

ANALYSIS OF THE COSTS OF ALTERNATIVE ACTIONS IN THE MAINTENANCE OF CURVE RAILS

C.1 GENERAL

As indicated in Table 11, many different actions are possible in attempting to correct problems or improve the maintenance of curve rails. Pricing out these options helps direct attention to those that are most cost effective and supports good decisions.

TABLE 11. GENERAL PROBLEMS AND POSSIBLE ACTIONS IN THE MAINTENANCE OF CURVE RAILS

Problem	Indicators	Actions
Safety	Derailments Severe Scuffing of high rail Low angle of wear Step-type wear	Adjust operating mode. Increase superelevation. Lubricate high rail. Install restraining rail.
Noise	Observations Measurements Complaints	Any of the above, plus: Keep restraining rail lubricated and adjusted to prevent flanging on high rail. Add water spray.
High maintenance costs; excessive wear of rail	Cost and frequency of replacement Measurements	Any of the above, plus: Increase wear limits. Install automatic lubricator. Improve job planning, and control.
Wear reaches allowable limit.	Observations Measurements	Evaluate limit and increase if safe. Install or adjust restraining rail to prevent further wear. Replace or transpose rail. Evaluate need and, if favorable, remove restraining rail.

As a prerequisite to reliable cost analysis, cost records should be sorted, evaluated and updated; cost factors should be developed for job sizes, site conditions and track availability; and current costs for rail maintenance and replacement work should be estimated. The work required in these efforts is outlined in Figure 24.

Estimating costs for standard units of rail maintenance and replacement will simplify and speed all the later work required to estimate costs for specific jobs. The costs of standard units amount to simple cost bases; with good data, they can be easily applied to specific jobs. Replacing a standard, 300-foot length of high rail is suggested as a cost base. One base cost would have to be estimated for each distinct type of track. The work site should be in an easily accessible location, and the job should be estimated for good working conditions without traffic interruptions. Items in the estimate should include the following:

The rail.
Drilling holes for joint bars, if used.
Weld materials, if used.
Removing and hauling out the worn rail.
Hauling, fastening and lining the new rail.
Welding joints, if CWR is used.
Equipment and fuel.
Overhead on direct labor.

All other costs such as adding ballast, tamping, replacing broken bolts and worn pads and replacing joint bars should be considered extras and not included in the simplified base. Such work would be required with or without a restraining rail. When the current base costs have been derived, estimates should be made for each of the common alternatives in the installation and replacement of rail, as listed in Table 2, each for a standard length of 300 feet, and the value of each should be calculated as a percent of the base cost. This will greatly facilitate estimating in future years, since only the cost bases will have to be adjusted by changes in labor, materials, equipment and energy costs.

Factors should be derived from cost records and local conditions in the track system for job size (efficiency) and site and time. The site and time factors should cover access, exposure to weather, track time availability, site clearances, availability of utilities, and other conditions that affect the work. These factors can quickly convert the base cost, multiplied by the percentage for each acceptable alternative, to good estimates for any specific job.

In addition to the one-time cost of the alternatives, the recurrent replacement and maintenance costs that will be associated with it in the future have to be estimated. Most Properties have estimate forms suitable for this use. Others are available from commercial sources.

In order to simplify the comparisons of the alternative work being considered, all calculations should be in present year dollars, using present replacement costs and present values of work and materials in the estimates of future costs over the life cycle of the track. The effects of inflation are considered included in the discount rate, since costs and values generally tend to rise together. Of course, if ways can be found to reduce future costs, such as increasing labor or material to save energy, these should be investigated and planned; and their estimated effects should be applied to future costs. In every case, the effects of planned changes in traffic and equipment on rail wear should be evaluated and applied to estimates of future maintenance and replacement costs. The future costs should then be discounted to present values.

For example, consider the case of a 860-foot length of high rail in a 550-foot radius curve in slab track, that has worn to a prescribed limit of 0.63 inch on the gage side in a period of 5 years. Surface wear of the 100-pound heat-treated rail has been less than 0.10 inch. Traffic is expected to increase 20% during the next 5 years and then level off because of the construction of another line. Acceptable alternatives include:

- Replace the high rail.
- Install a restraining rail.
- Install a restraining rail and a rail lubricator.
- Replace the high rail now, and install a restraining rail and rail lubricator.

Based on knowledge of the rate of rail wear, future traffic and other factors, certain assumptions and estimates can be made:

- A new high rail will be worn to the limit and have to be replaced at 4 year intervals unless changes are made to reduce the rate of wear.
- Installation of an adjustable restraining rail to protect the high rail from gage wear will extend the worn rail (now in place) 24 years before surface wear necessitates its replacement.
- Installation of an adjustable restraining rail will give a new high rail a useful life of 30 years.
- An adjustable restraining rail will last 8 years with manual lubrication and 14 years with automatic lubrication.
- The annual cost for routine maintenance of the high rail alone (inspection bolt tightening) is $860 and manual lubrication costs $830 per year. Costs will continue at this rate in 1980 dollars.
- Annual maintenance of a high rail and restraining rail will cost $1550, 1980 dollars per year.
- The installation of an automatic lubricator will increase the amount of lubricant used from the present 100 pounds to 500 pounds per year.

In this example, the current (base year) cost for replacing a standard 300-foot length of high rail is \$6780, the 1980 base cost for replacing high rail on slab track, as developed in Section 5.1.3. This base cost is multipled by the percentages in the lower range of Table 2 to estimate the base costs of the alternatives.

The job site and time factor is 120% because of the small size of the tunnel and the distance to good access.

The job size factor is 1.00/1.15. Replacement of the high rail at the same time as the restraining rail gives a job size factor of 1.00/1.05. Then for this job, the cost of replacing the high rail is estimated to be:

$$\$6780 \times 860/300 \times 1.20/1.15 = \$20,300$$

If it is replaced at the same time as the restraining rail, this becomes:

$$20,300/1.05 = \$19,300$$

Installation of a new restraining rail at the time the high rail is replaced is estimated:

$$20,300 \times 1.30 = \$26,400$$

Installation of a restraining rail alone will have the added cost of raising the high rail to install braces, replacing and lining it. This is estimated to add an additional 10% cost of:

$$26,400 \times 1.10 = \$29,000$$

Future costs of maintenance and replacement are figured in base year (1980) dollars and are discounted at 12% to obtain their present values. The equations used are:

$$PV = C(1 + i)^{-y}$$

where, C is a future cost in base year dollars

i is 12%

y is the year in which the cost is incurred and, $$PV = A\left(\frac{1 - (1 + i)^{-30}}{i}\right)$$

where A is the annual recurring cost

30 is the number of years the cost is incurred

First alternative: the high rail is replaced now at a cost of 20,300 and at 4 year intervals in the future.

The present values of future costs are:

$$PV = 20{,}300(1.12^{-4} + 1.12^{-8} \ldots + 1.12^{-28}) = \$33{,}900 \text{ replacement}$$

$$PV = 860\left(\frac{1 - 1.12^{-30}}{0.12}\right) = 6930 \text{ maintenance}$$

$$PV = 830\left(\frac{1 - 1.12^{-30}}{0.12}\right) = 6690 \text{ lubrication}$$

The second alternative: a restraining rail is installed now at \$29,000. This increases the cost of inspection and maintenance to an estimated \$1550 per year.

The adjustable restraining rail will prevent flanging on the high rail and thus eliminate the need for lubricating it; but, for best results, the restraining rail will receive more frequent manual lubrication than the high rail did. The cost of lubricating the restraining rail is estimated at \$1330 per year.

The restraining rail will be replaced alone at \$17,300 in years 8 and 16, and with the high rail at \$15,200 in year 24. The high rail will be replaced in year 24 at \$19,300.

The present values of these future costs are:

Restraining Rail, $PV = 17{,}300\ (1.12^{-8} + 1.12^{-16}) = \$\ 9810$

$15{,}200 \times 1.12^{-24} = 1000$

High Rail, $PV = 19{,}300 \times 1.12^{-24} = 1270$

Inspection & Maintenance, $PV = 1550\left(\frac{1 - 1.12^{-30}}{0.12}\right) = 12{,}500$

Lubrication, $PV = 1330\left(\frac{1 - 1.12^{-30}}{0.12}\right) = 10{,}700$

Third alternative: a restraining rail is installed now at $29,000, and a lubricator and controls are installed at an estimated $3600. The well-lubricated restraining rail will require replacement in years 14 and 28, and the high rail will require replacement in 24 years. The cost of inspection and maintenance is estimated at $1550 per year, and the cost of lubrication (grease and maintenance of the lubricator) is estimated at $1080 per year.

The present values of the future costs are estimated to be:

Restraining rail, $PV = 17{,}300(1.12^{-14} + 1.12^{-28}) = \$\,4260$

High rail, $PV = 20{,}300 \times 1.12^{-24} = 1340$

Inspection & Maintenance as under the second alternative, $= 12{,}500$

Lubrication $PV = 1080\left(\frac{1 - 1.12^{-30}}{0.12}\right) = 8700$

Under the fourth alternative, the high rail is replaced at $19,300; a restraining rail is installed at $26,400; and a lubricator is installed at $3600. Maintenance and lubrication will cost $1550 and $1080 per year as under the third alternative. The restraining rail will have to be replaced at years 14 and 28, but the high rail will not require replacement during the 30-year life cycle.

The present values of the future costs for replacing the restraining rail, maintenance and lubrication are estimated to be the same as for the third alternative. The estimated costs are summarized in Table 12.

TABLE 12. COST ANALYSIS SUMMARY

Present Values of Current and Future Costs
Curve Maintenance and Replacement Over 30 Years
Urban Line Baker. Station 13785 to 14645

Alternatives	Replace High Rail	Install Restraining Rail	Install Restraining Rail and Lubricator	Replace High Rail; Install Restraining Rail and Lubricator
Installation/ Replacement Costs. This Year	20,300 20,300	29,000 29,000	29,000 3,600	19,300 26,400 3,600
Replacement Costs	33,900	10,800 1,270	4,260 1,340	4,260
Maintenance Costs	6,930	12,500	12,500	12,500
Lubrication Costs	6,690	10,700	8,700	8,700
Totals	67,800	64,300	59,400	74,800

The accumulation of cost data and the use of simplified procedures and forms will make the analyses increasingly easy, so that they will take less and less time. They will also have other benefits that will pay you for your trouble, such as contributing to job planning and control, as well as supporting cost-effective decisions.

APPENDIX D
REPORT OF NEW TECHNOLOGY

The new concepts for the design of restraining rail structures presented in this report were developed in accordance with the statement of work. Accordingly, they are not considered patentable under the patents rights clause of the terms and conditions of the contract.

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