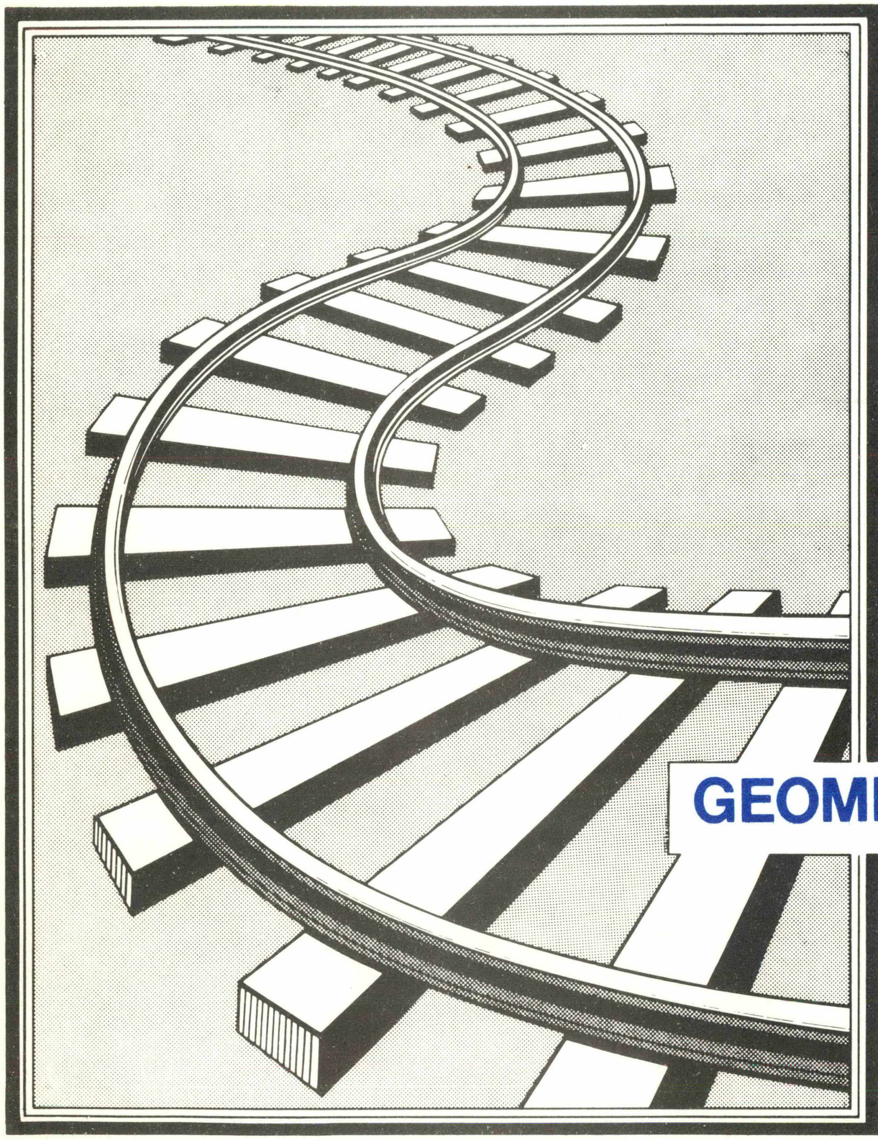
8

TRACK TRAIN DYNAMICS

TRACK STRUCTURES

VOLUME IV

EFFECTS OF TRACK GEOMETRY ON DYNAMIC BEHAVIOR OF A RAILWAY VEHICLE ON TANGENT TRACK



GEOMETRIC STUDY

AAR FRA·RPI·TDC

AN INTERNATIONAL GOVERNMENT - INDUSTRY RESEARCH PROGRAM ON TRACK-TRAIN DYNAMICS

R-377

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1. REPORT NO.	2. REPORT DATE	3. PERIOD COVERED
R- 377	Jan.1979	

4. TITLE AND SUBTITLE

Track Structures - Volume IV.
Effects of Track Geometry (Irregularities) on the Dynamic Behavior of a Railway Vehicle On Tangent Track.

5. AUTHOR(S)

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6. PERFORMING ORGANIZATION NAME AND ADDRESS

ASSOCIATION OF AMERICAN RAILROADS
Track-Train Dynamics Phase II
3140 So. Federal Street,
Chicago, Illinois 60616

7. TYPE OF REPORT

Technical

8. CONTRACT OR GRANT NO.

DOT-FR-64228

9. SPONSORING AGENCY NAME AND ADDRESS

Federal Railroad Administration
Department of Transportation
2100 Second Street, S.W.
Washington, D. C. 20590

10. NO. OF PAGES

54

11. NO. OF REFERENCES

12. SUPPLEMENTARY NOTES

TTD Phase II - Task 1 - Subtask 1.1.1(b)

13. ABSTRACT

The objective of this report is to evaluate the effects of various track irregularities on the dynamic behavior of railway vehicles moving over tangent track which meets the FRA standards. Two different freight cars are used for the study. The peak to peak roll angle of the carbody, center plate loads, wheel loads and accelerations are investigated for different track errors and combinations thereof.

It is found that track classification according to irregularities and speed limit is not appropriate. This type of classification does not identify the severity of dynamic loads to which the track and vehicle components are subjected. Also, in some cases vehicles will perform better at speeds greater than the specified limit and are subjected to undesirable conditions when operated within the speed limit.

14. SUBJECT TERMS

Peak to Peak Roll Angle, Profile Alignment, Gage, Warp, Center Plate Load, Side Bearing Load, Acceleration.

15. AVAILABILITY STATEMENT

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BACKGROUND INFORMATION

ON THE

TRACK-TRAIN DYNAMICS PROGRAM

The Track-Train Dynamics Program encompasses studies of the dynamic interaction of a train consist with track as affected by operating practices, terrain, and climatic conditions.

Trains cannot move without these dynamic interactions. Such interactions, however, frequently manifest themselves in ways climaxing in undesirable and costly results. While often differing and sometimes necessarily so, previous efforts to reasonably control these dynamic interactions have been reflected in the operating practices of each railroad and in the design and maintenance specifications for track and equipment.

Although the matter of track-train dynamics is by no means a new phenomenon, the increase in train lengths, car sizes, and loadings has emphasized the need to reduce wherever possible excessive dynamic train action. This, in turn, requires a greater effort to achieve more control over the stability of the train as speeds have increased and railroad operations become more systematized.

The Track-Train Dynamics Program is representative of many new programs in which the railroad industry is pooling its resources for joint study and action.

A major planning effort on track-train dynamics was initiated in July 1971 by the Southern Pacific Transportation Company under contract to the AAR and carried out with AAR staff support. Completed in early 1972, this plan clearly indicated that no individual railroad had both the resources and the incentive to undertake the entire program. Therefore, AAR was authorized by its Board to proceed with the Track-Train Dynamics Program.

In the same general period, the FRA signaled its interest in vehicle dynamics by development of plans for a major test facility. The design of a track loop for train dynamic testing and the support of related research programs were also pursued by FRA.

In organizing the effort, it was recognized that a substantial body of information and competence on this program resided in the railroad supply industry and that significant technical and financial resources were available in government.

Through the Railroad Progress Institute, the supply industry coordinated its support for this program and has made available men, equipment, data from earlier proprietary studies, and monetary contributions.

Through the FRA, contractor personnel and direct financial resources have been made available.

Through the Transport Canada Research and Development Centre (TDC), the Canadian Government has made a major commitment to work on this problem and to coordinate that work with the United States' effort.

Through the Office de Recherces et D'Essais, the research arm of the Union Internationale des Chemins de Fer, the basis for a full exchange of information with European groups active in this field has been arranged.

The Track-Train Dynamics Program is managed by the Research and Test Department of the Association of American Railroads under the direction of an industry-government steering committee. Railroad members are designated by elected members of the AAR's Operation-Transportation General Committee, supply industry members by the Federal Railroad Administration, and Canadian Government members by the Transport Development Centre. Appropriate task forces and advisory groups are established by the Steering Committee on an ad hoc basis as necessary to pursue and resolve elements of the program.

The staff of the program comprises AAR employees, personnel contributed on a full- or part-time basis by railroads or members of the supply industry, and personnel under contract to the Federal Railroad Administration or the Transportation Development Agency.

The program plan as presented in 1972 comprises:

1) Phase I -- 1972-1974

   Analysis of an interim action regarding the present dynamic aspects of track, equipment, and operations to reduce excessive train action.

2) Phase II -- 1974-1977

   Development of improved track and equipment specifications and operating practices to increase dynamic stability.

3) Phase III -- 1977-1982

   Application of more advanced scientific principles to railroad track, equipment, and operations to improve dynamic stability.

Phase I officially ended in December of 1974. The major technical elements of Phase I included:

a) The establishment of the dynamic characteristics of track and equipment.

b) The development and validation of mathematical models to permit the rapid analysis of the effects on dynamic stability of modifications in design, maintenance, and use of equipment and track structures.

c) The development of interim guidelines for train handling, makeup, track structures, and engineer training to reduce excessive train action.

Reports on all elements of Phase I activities have been completed and are available through the AAR. A list of the Track Train Dynamics publications is available upon request.

The major technical elements of Phase II include:

a) The adaptation of Phase I analytical models to allow for conducting parameter investigations in the area of track, trucks, draft gear and cushion units, and vehicle behavior.

b) The development of fatigue analysis guidelines.

c) The development of a comprehensive program for identifying the loads to which track, vehicles, and vehicle components are subjected.

As research on this program proceeds, reports on other elements of Phase II will be issued, and existing reports updated at appropriate intervals.

ACKNOWLEDGEMENT

The present study is one of the research projects of the Track-Train Dynamics program, which represents a joint effort of the Association of American Railroads, the Railway Progress Institute, the Federal Railroad Administration and the Transport Canada Research and Development Center. The Federal Railroad Administration is acknowledged for funding this research. The Association of American Railroads furnished the required manpower and computational facilities for accomplishing the objective of this study.

This report is the final of the four volumes in fulfillment of an FRA Research Contract on Task I, Track Structure Research. In particular, this study is classified under Subtask 1.1.1 which requires the utilization of a Mathematical Model to evaluate the dynamic response of a typical railway vehicle under various perturbed track geometries.

The authors wish to acknowledge the efforts contributed by Dr. D. R. Sutliff, TTD Phase II Director, for his guidance in conducting this study. The authors appreciate the efforts of Mr. Edward Chang for modifying the "Flexible Carbody Vehicle Model" to incorporate various track irregularities and the wheel and rail creep/friction relationship. Mr. Edward Chang and Suresh Mehta are responsible for the coordination and analysis of the computer runs.

The United States Government assumes no responsibility for the contents of this report or the use thereof.

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EXECUTIVE SUMMARY

The Track-Train Dynamics program encompasses the study of dynamic interactions between railway vehicles and track. This report presents the analysis of the railway vehicle's dynamic response to track perturbation.

To perform the analysis a Mathematical Model for a typical railway freight car was developed by the Association of American Railroads. The model has the capability to simulate vertical track perturbations on tangent track. The computer program was modified to include lateral track perturbations for this analysis.

The report describes the analysis of dynamic response of two railway freight cars on different types of track classified according to Federal Railroad Administration (FRA) standards. Although the track perturbations are random in nature, their characteristics can be described by approximating them as mathematical functions. In this study sine, haversine, rectified sine and linear functions have been used to simulate the track errors.

The track perturbations are not only random in nature but also exist in combinations. Thus it is not possible to investigate every possible track error. Several perturbations such as crosslevel, alignment, warp, gage and their combinations were used in the analysis.

The two railway cars selected for study are a 100-Ton covered Hopper car and a 70-Ton Box car (loaded). The emphasis has been placed on the rock and roll behavior of these vehicles as influenced by simulated track perturbation and resultant loading on the side bearing, center plate, and wheel (vertically and laterally). The vertical and lateral accelerations of the carbody are also analyzed.

The results obtained during this analysis indicate that the FRA specification for operating speed limits for freight cars, based upon the perturbation amplitude alone, is not adequate. The dynamic response of the vehicle is very much influenced by the inertia distribution and the geometric layout of the cars. The study shows that the two cars behaved quite differently to the same amplitude of track perturbation.

1. INTRODUCTION

The study of the dynamic behavior of a railway vehicle travelling on tangent track with periodic or random irregularities is of great importance for track maintenance and vehicle design/maintenance. In the past, attempts to study the problem of vehicle and track interaction were hampered by the limitations of analytical methods, and the cost involved in field testing. Under the Track-Train Dynamics (TTD) program the Association of American Railroads has developed mathematical models to study the vehicle/track interactions for various kinds of track-train related problems.

This report describes the study of the behavior of freight cars moving over different types of tracks classified according to Federal Railroad Administration (FRA) Safety Standards. Based on these FRA track classifications, several types of track irregularities were selected for this study. The rock & roll and bounce behavior of the vehicle, as affected by the track inputs, have been investigated. Other dynamic phenomena, such as lateral instability due to car/truck hunting, are not included in this report.

The track errors (irregularities) selected for study are assumed to be continuous, although FRA standards define the track errors based on a single track segment (62 feet length). It is possible for staggered rails to have continuous segments with irregularities each of which is within the single segment error limit specified by the FRA.

The standard track inputs studied in the report represent a part of all the possible track deformation configurations. In reality, the inputs are of random nature. Any conclusion drawn from the results presented in the report should be viewed in the light of the objective of the study (rock and roll behavior of vehicle) and the limited type of irregularities studied.

The contents of this report are based on results obtained from the Flexible Body Vehicle Model [1,2,3,4]* computer program developed at the AAR to study car rock and bounce behavior due to track inputs. The model was initially developed for vertical track inputs. For the purpose of this investigation the model has been modified to also include lateral input. For the detailed formulation of equations of motion and description of the modified version of the model refer to [5,6]. The model representation is given in figure 1.

* Numbers in [ ] indicate references.

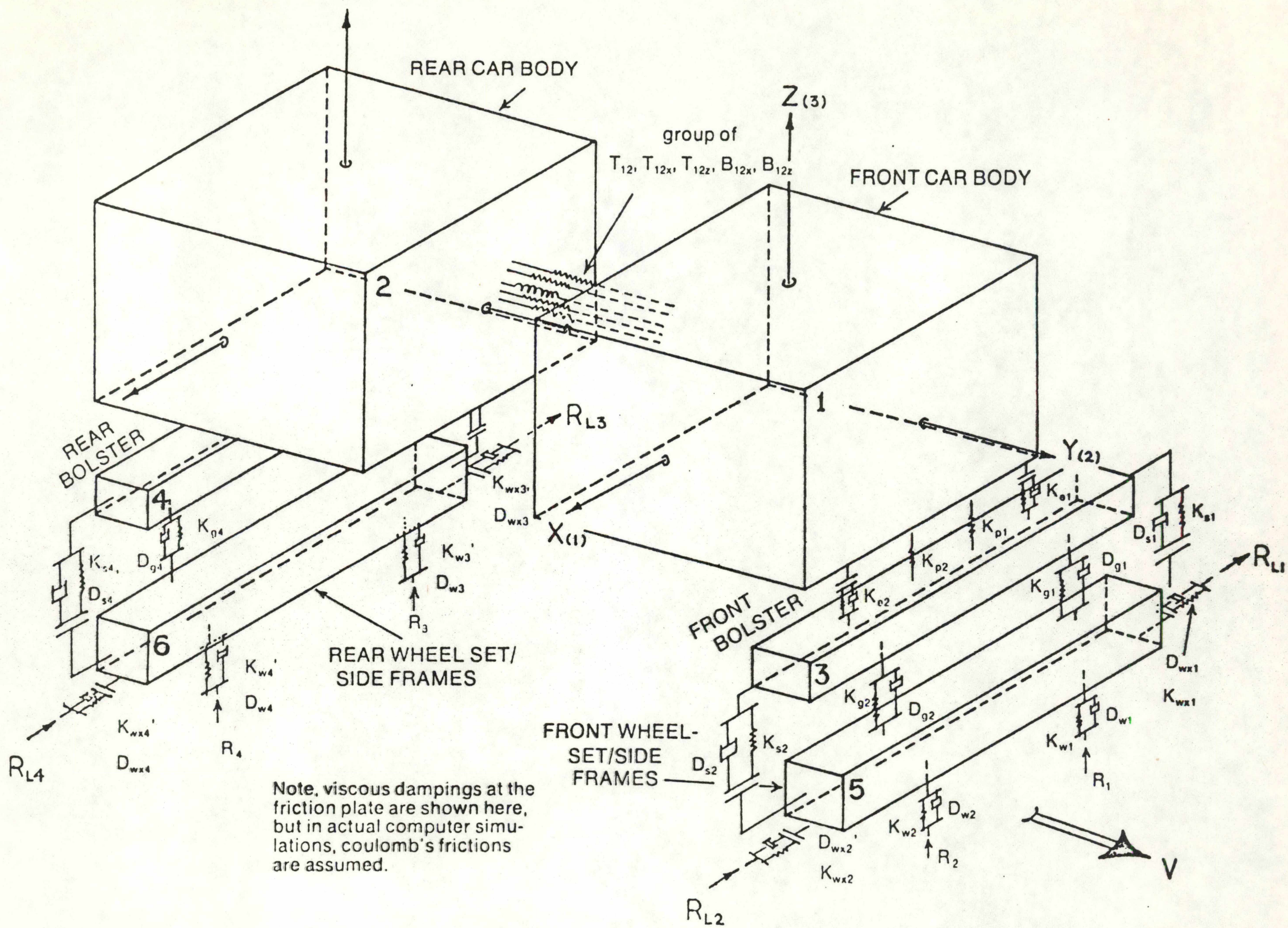


FIGURE 1. FLEXIBLE BODY FREIGHT CAR MODEL

2. TRACK IRREGULARITIES

The track irregularities are defined as the variation in track geometry in either lateral or vertical plane from the designed configuration. Most track errors are of random nature and cannot be prescribed by mathematical functions. However, some of the common irregularities can be approximated by periodic wave forms, e.g. sine wave, haversine wave and rectified haversine wave etc., The mathematical expressions for these wave form are:

a) Sine wave

$$\eta = A \sin\omega t$$

b) Rectified sine wave

$$\eta = A|\sin\omega t|$$

c) Haversine wave

$$\eta = \frac{A}{2} \quad |(1-\cos 2\omega t)|$$

where η = amplitude at time t, A = maximum amplitude and ω = circular frequency.

Four types of track irregularities are included in the study. They are the profile error, warp error, alignment and gage errors. The nature of these errors and some of their combinations are also discussed. The levels of severity of these errors, allowed for a particular class of track by the FRA Track Standards [7], are listed in Table 1. Table 1 includes TTD Phase II Task I-1.1(b) suggestions for allowable track errors obtained using this study.

TABLE 1 SELECTED LEVELS OF PERTURBATIONS

Track Geometry

Operating speed for freight trains for a given class of track as per FRA standards is as follows:

Class of Track	Operating Speed for Freight Trains
1	10 mph
2	25 mph
3	40 mph
4	60 mph
5	80 mph
6	110 mph

Track deviations (vertical and/or lateral) as per FRA standards, TTD-Phase II - Task I 1.1.1 (b) and recommended study are as follows:

(1) Profile deviation

Class of Track	Profile deviation	
1	3"	*,#
2	2-3/4"	*,#
3	2-¼"	*,#,**
4	2"	*,#,**
5	1-¼"	*,#,**
6	1/2"	*,#,**

---

* - suggested levels by FRA standards

\# - suggested levels by TTD-Phase II Task I 1.1.1 (b)

** - suggested levels for study both rail input

(2) Gage deviation

Class of Track	Min.	Max.	Deviation (+)	Deviation (-)
			(Standard Gage 4'8½")	
1	4'8"	4'9-3/4"	1¼"	½" *,#,**
2	4'8"	4'9½"	1"	½" *,
3	4'8"	4'9½"	1"	½" *,#,**
4	4'8"	4'9¼"	3/4"	½" *,#,**
5	4'8"	4'9"	½"	½" *,#,**
6	4'8"	4"8-3/4"	¼"	½" *

(3) Alignment deviation

Class of Track	Alignment deviation	
1	5"	*,#
2	3"	*,#
3	1-3/4"	*,#,**
4	1½"	*,#,**
5	3/4"	*,#,**
6	½"	*,#,**

(4) Warp deviation

Class of Track	Warp deviation	
1	3"	* 4" #
2	2"	*
3	1-3/4"	* 3" #
4	1½"	* 2" #,**
5	1"	* 1" #,**
6	5/8"	*

* - suggested levels by FRA standards

\# - suggested levels by TTD-Phase II Task I - 1.1.1 (b)

** - suggested levels for study

Combination of errors:-

(5) Profile and alignment errors

Class of Track	Profile	Alignment
1	3"	5" #
2	2-3/4"	3" #
3	2¼"	1-3/4"#,**

(6) Profile and Gage Error

Class of Track	Profile	Alignment
4	2"	1½" **
5	1¼"	3/4" **

(7) Profile, Alignment and Gage Error

Class of Track	Profile	Alignment	Gage
1	3"	5"	1¼" #
2	2-3/4"	3"	1" #
-	2¼"	1-3/4"	1" **

(8) Profile, Alignment, Gage and Warp Error

Class of Track	Profile	Alignment	Gage	Warp
3	2¼"	1-3/4"	1"	1-3/4" #,**

---

* - suggested levels by FRA standards

\# - suggested levels by TTD - Phase II Task I - 1.1.1 (b)

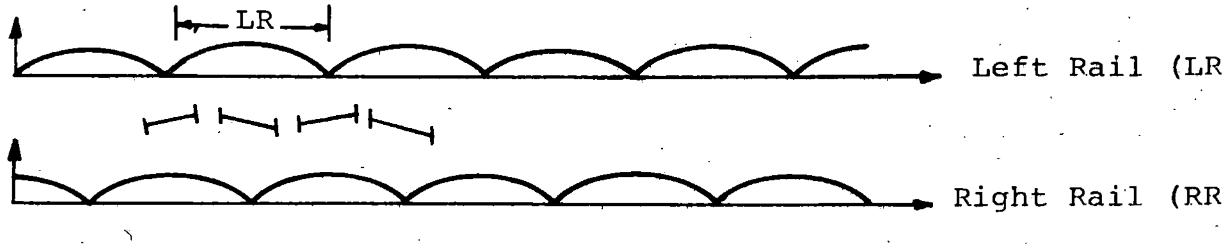
** - suggested levels of study

2.1 PROFILE ERROR

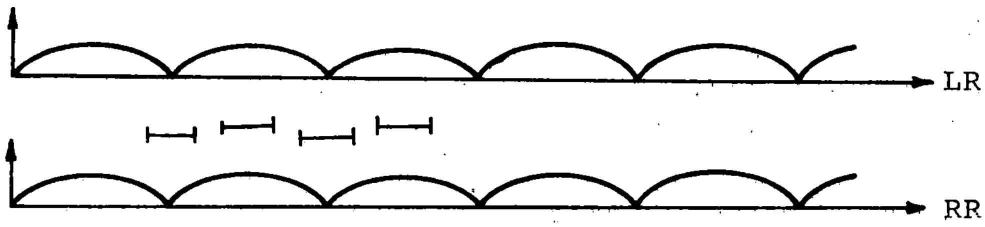
Rail profile error is a surface irregularity of the top surface of the rail in a vertical plane, and results mainly from the existence of low and soft joints or the thermal loads imposed on the track. This error can be represented by a rectified sine wave for the track with rail joints. By half staggering the rail joints (Fig.2a), the rock and roll mode can be simulated. Similarly, if the rail joints are placed in parallel (or full staggered) the bounce mode can be simulated (Fig.2b). The associated wave length is L_R equals the rail length. For the continuous welded rail (CWR), a sine wave should represent the profile error, see Figures 2c and 2d, [8]. The function used to represent a dipped rail joint is comprised of two quarter consine waves. In this study a rectified sine wave of 39' length for 1/2 staggered rail has been used.

2.2 ALIGNMENT ERRORS

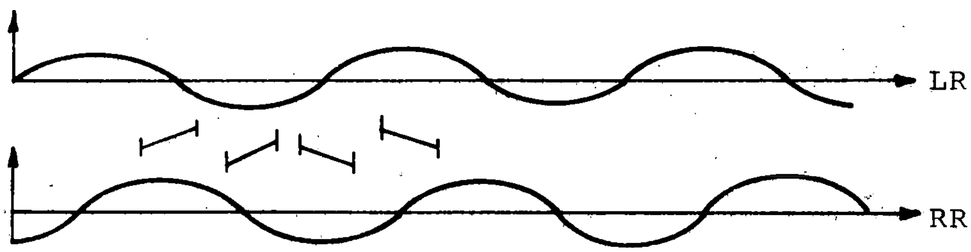
Nominal alignment is defined as the average of lateral positions of two rails (often referred as centerline). The alignment error is the deviation of either rail from a perfectly aligned rail position. This error usually results from an initial imperfection of the rails, faulty track construction, improper maintenance processes and accumulated lateral rail movement due to traffic loads. Some of the alignment errors are shown in Fig.3. Generally, a sinusoidal function is used to represent this error [9,10] as shown in Fig.3a. However, a rectified sine wave, alternated haversine



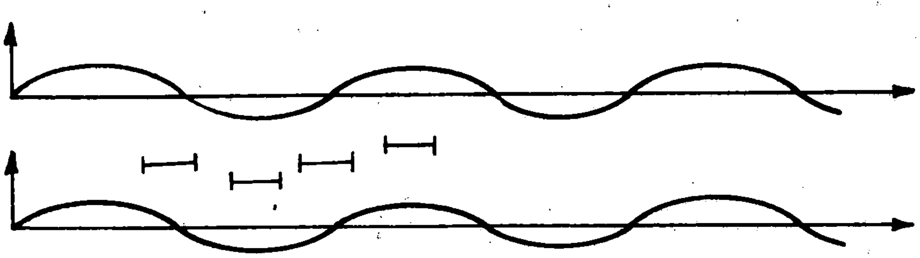
(A) Rail Profile Error - Rock and Roll Mode



(B) Rail Profile Error - Bounce Mode



(C) Rail Profile Error - CWR(Rock and Roll Mode)
CWR - continuously welded rail



(D) Rail Profile Error - CWR(Bounce Mode)

FIGURE 2 TRACK GEOMETRY FOR PROFILE ERRORS

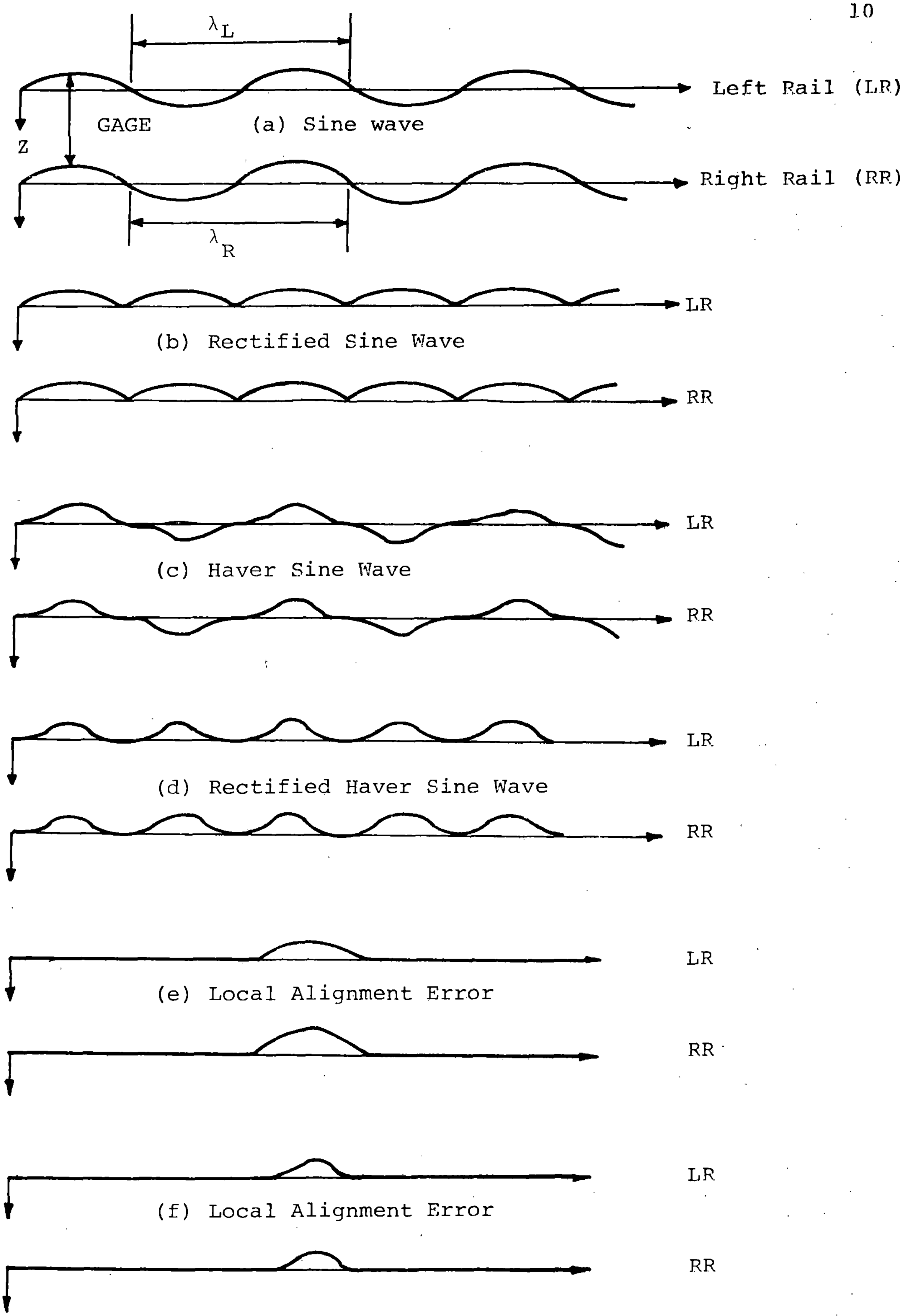


FIGURE 3 RAIL ALIGNMENT ERRORS

wave or rectified haversine wave can also be used to simulate the existence of periodic alignment errors. Using Figures 3b to 3d local errors known as bumps or kinks can also be represented to simulate finite segment of track with error as in Figs.3e and 3f.

As the sine wave type of error is the one most likely to be encountered in track, it is used in the study. Half wave length ($\lambda/2$) used in the study is 62' as defined in the FRA Track Safety Standards [7].

2.3 GAGE ERRORS

Gage is defined as the horizontal distance between two rails, measured between the rail heads in a plane 5/8" below top of the rail head. The gage error is defined as deviation of actual gage from the nominal gage. It may result from either improper track construction, maintenance, or relative movement of rails under traffic loads. The gage error can be represented by sine wave or haversine wave as shown in Figs. 4a to 5b. The effect of local gage widening or narrow gage is shown in Figs.4e and 4f. In this investigation haversine wave with half wave length of 39' has been used for gage errors as shown in Fig.5.

2.4 WARP ERROR

Warp is defined as a rate of change in cross-level over a designated segment of track, Fig.6. It may result from thermal loads on track or the differential settlement of cross ties. It is believed that track warping, together with rail profile error, is responsible for wheel lift.

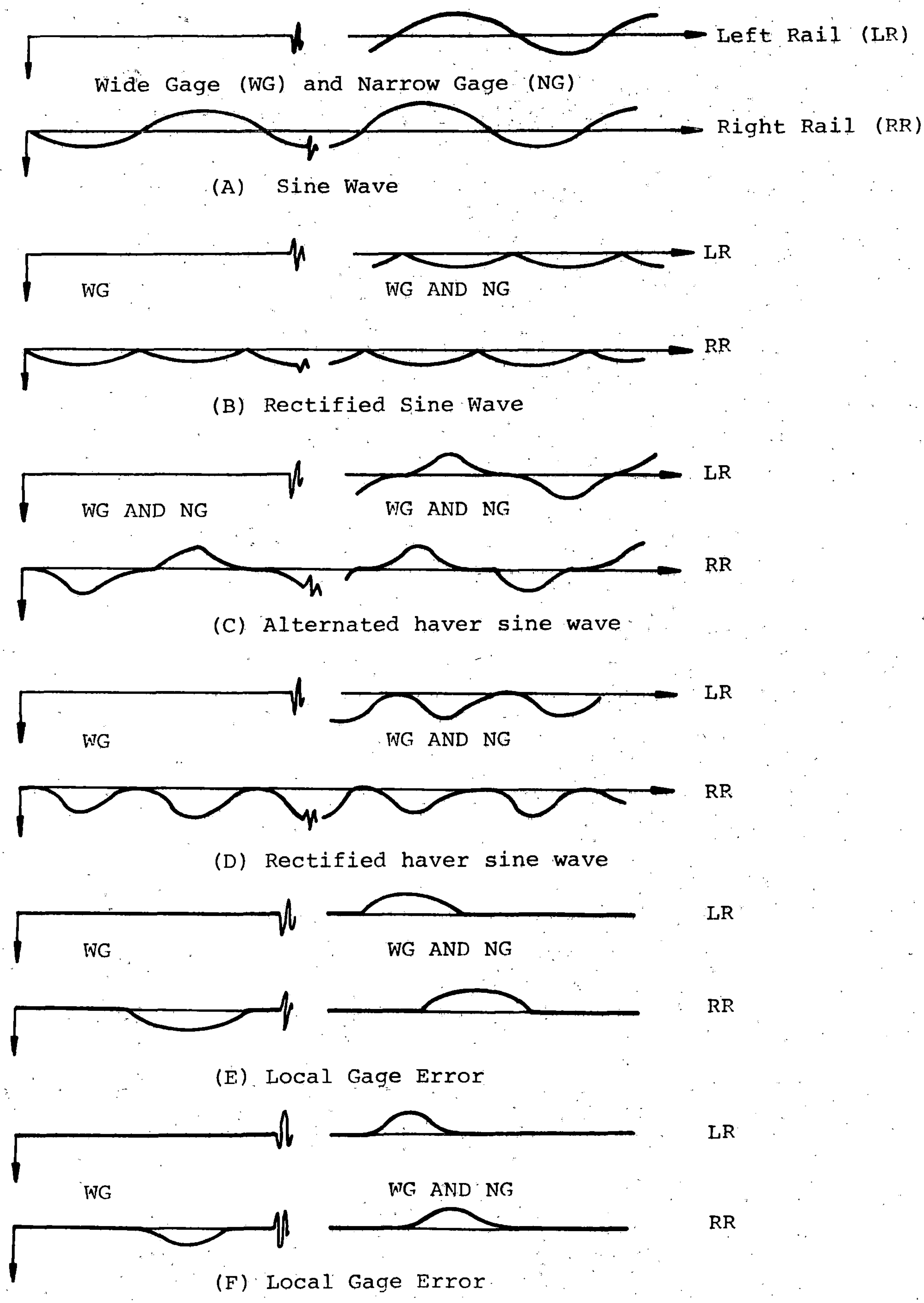
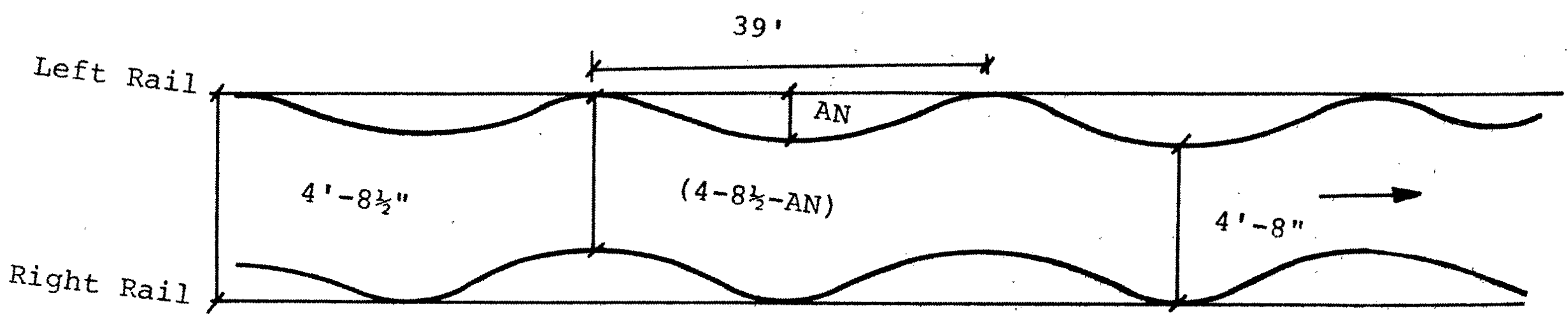
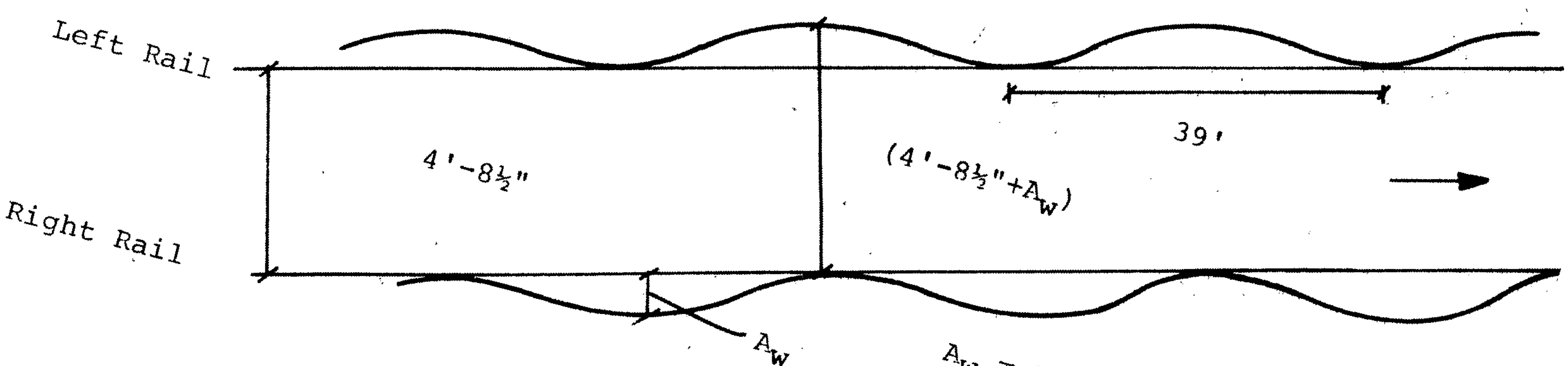


FIGURE 4 TRACK GEOMETRY FOR GAGE ERROR



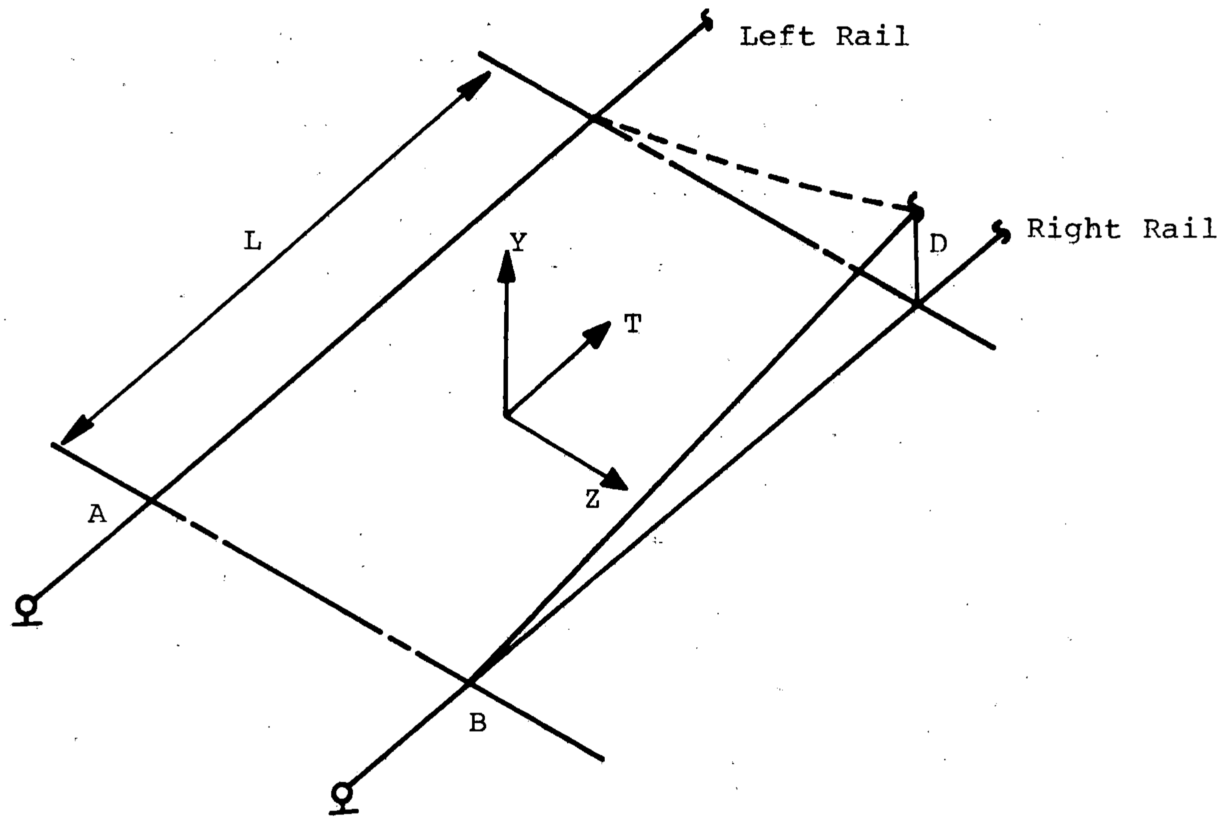
(A) Narrow Gage



(B) Wide Gage

A_w = Amplitude of wide gage
A_N = Amplitude of Narrow gage

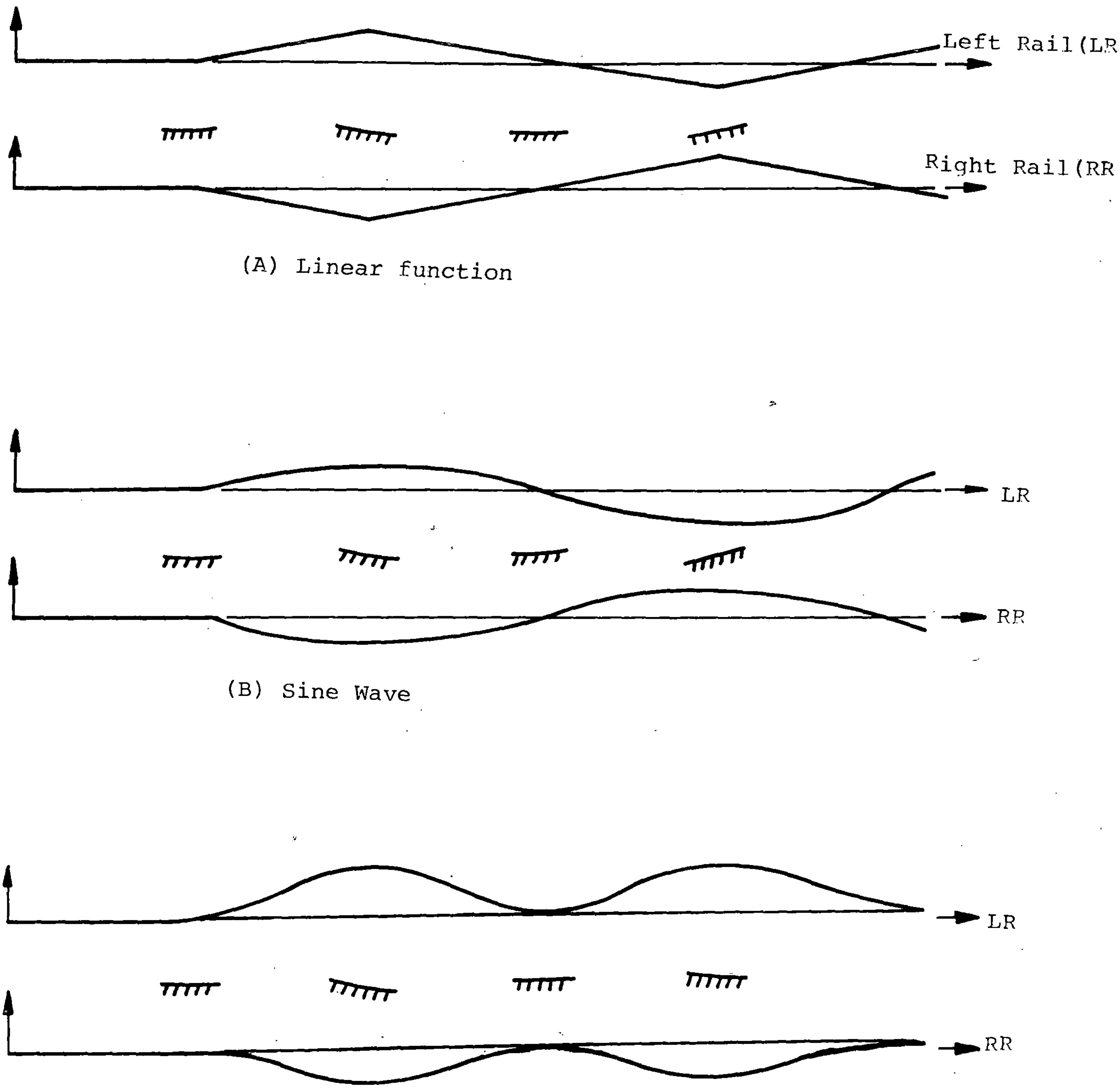
FIGURE 5 TRACK GEOMETRY FOR GAGE ERRORS



$$\text{WARP ERROR} = \frac{(Y_C - Y_A) - (Y_D - Y_B)}{L}$$

Note: Y_A, Y_B, Y_C and Y_D are vertical ordinate of points A, B, C and D

FIGURE 6 TRACK GEOMETRY FOR WARP ERROR



(A) Linear function

(B) Sine Wave

(C) Haver Sine Wave

FIGURE 7 TRACK GEOMETRY FOR WARP ERROR

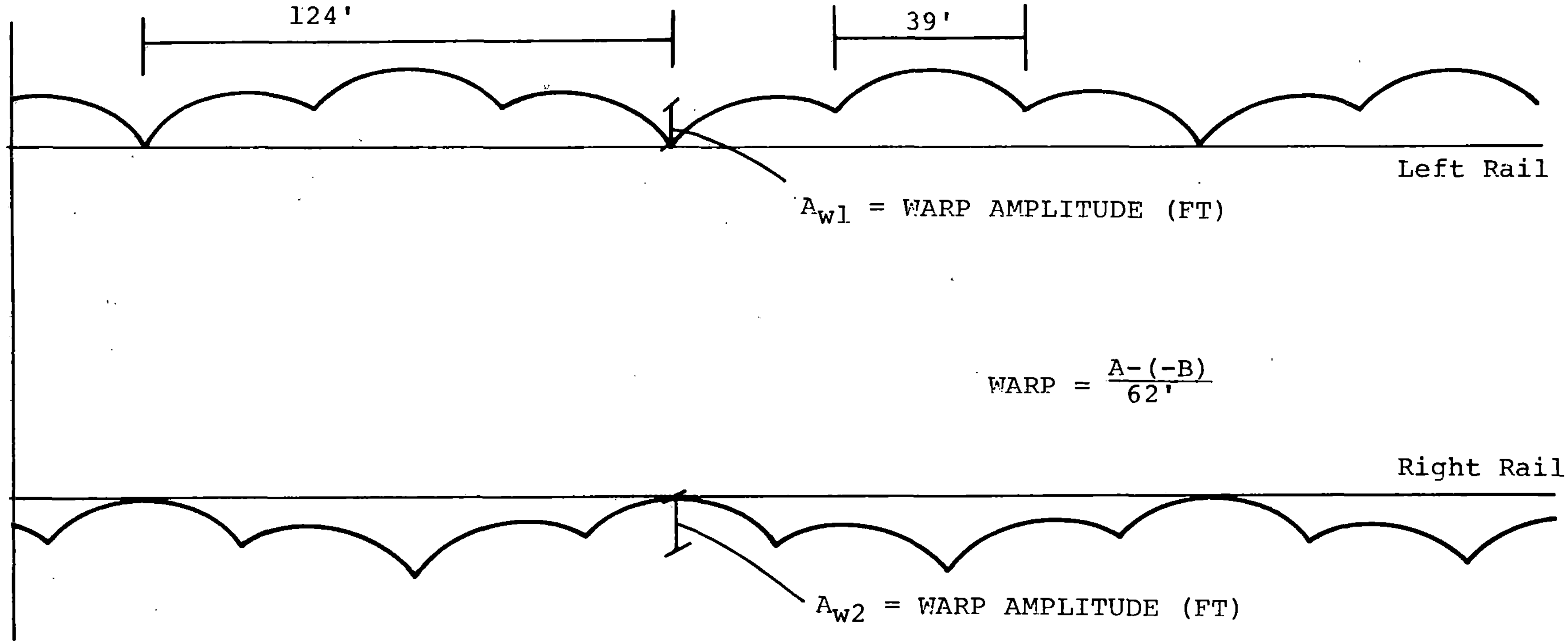


FIGURE 8 TRACK GEOMETRY FOR COUPLED PROFILE AND WARP

The nominal profile of track can change due to a combination of profile and warp error. A railway vehicle with a truck center distance close to a rail length may be excited in the rock and roll motion. In this study a 100-Ton car with truck centers measuring 45 ft. apart has been used to investigate the effect of warp error.

Some of the common representations of warp error are shown in Figs.7a through 7c. As shown in Fig.7, linear function, sine wave, and rectified sine wave can be used to represent a warped track. In this study only rectified linear and continuous linear wave have been used due to limitations of the model [6]. Often warp and profile errors are coupled. Hence, instead of investigating warp error alone, the effects due to warp coupled with profile error have been studied. Warp error is represented by a rectified linear function and 124' half wave length. Warp error is defined for 62' of rail as specified in FRA Track Standards [7]. Fig.8 shows a typical combination of profile and warp error.

2.5 COUPLED PROFILE AND ALIGNMENT ERROR

Track configuration for coupled profile and alignment error is shown in Fig.9. Profile is represented by rectified sine wave of 39'. Alignment error is represented by sine wave with half wave length of 62'. Both rails are arranged parallel in the lateral plane, as if there is no gage error.

2.6 COUPLED PROFILE AND GAGE ERROR

In Fig.10 the track configuration is shown for coupled profile and gage error. The peak amplitude for both errors

A_p = Amplitude of Profile Error

A_L = Amplitude of Alignment Error

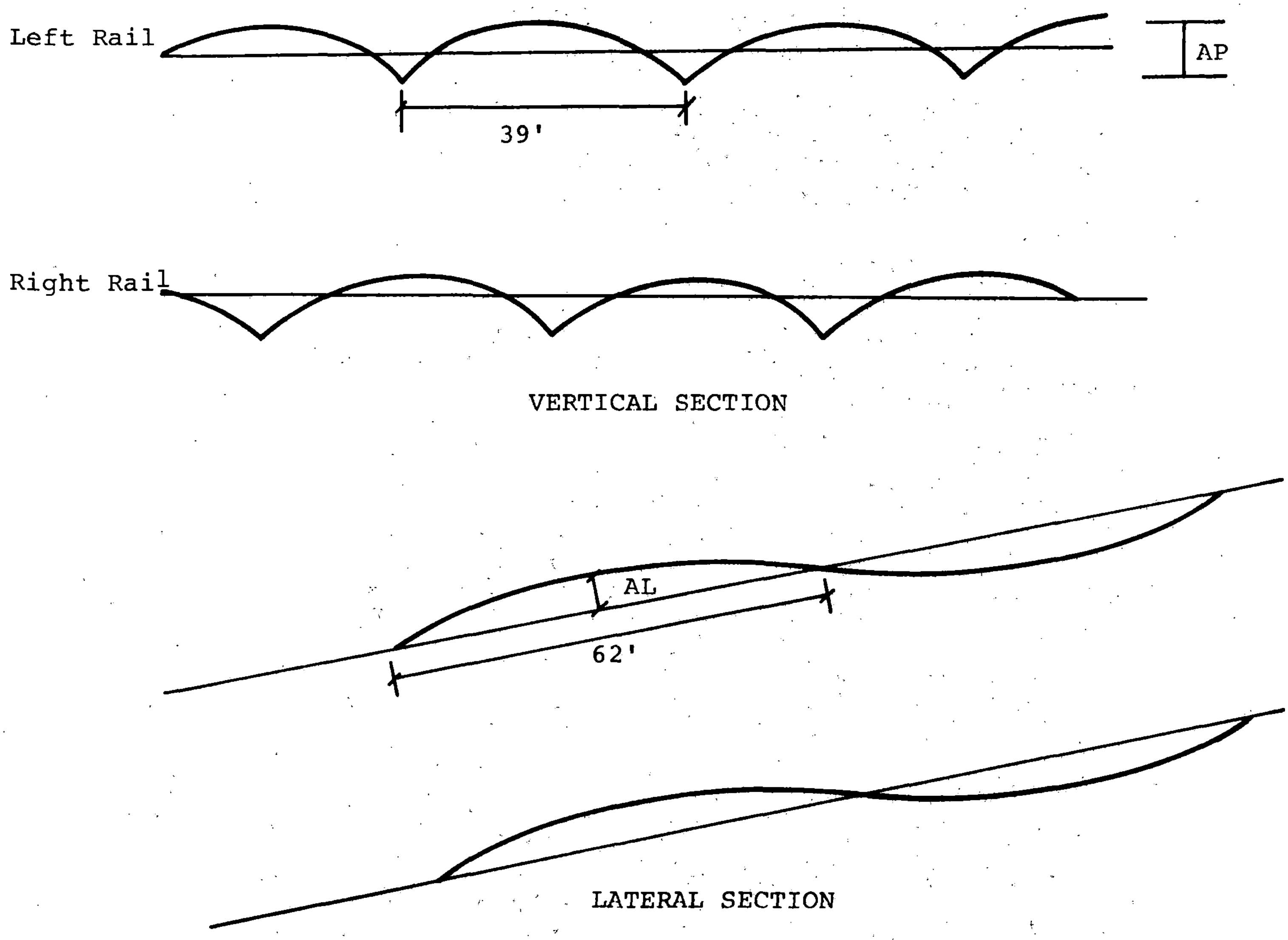


FIGURE 9 TRACK GEOMETRY FOR COUPLED PROFILE AND ALIGNMENT

Profile and Narrow Gage (A) + (B)
Profile and Wide Gage (A) + (C)

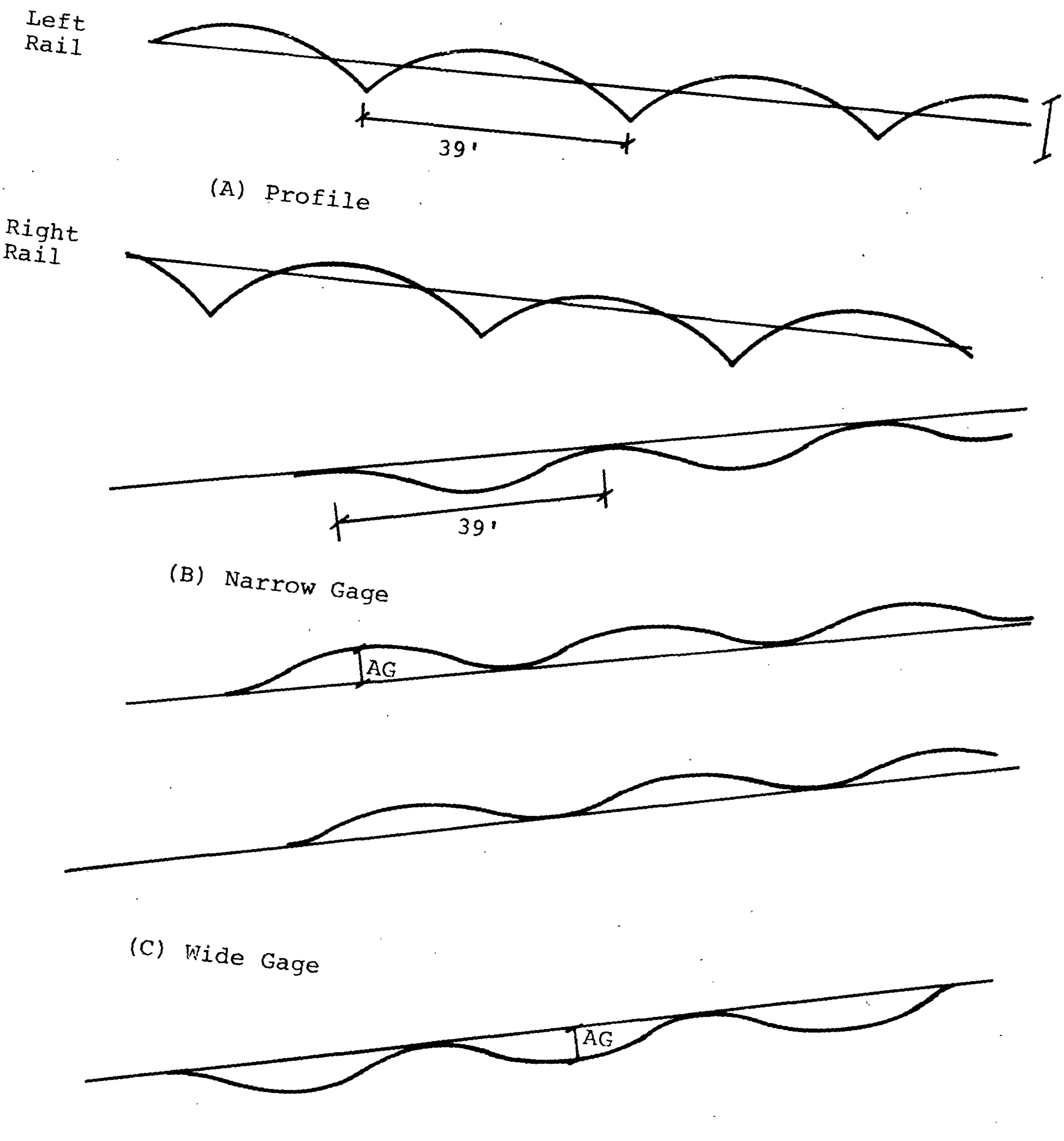


FIGURE 10 TRACK GEOMETRY FOR COUPLED PROFILE AND GAGE

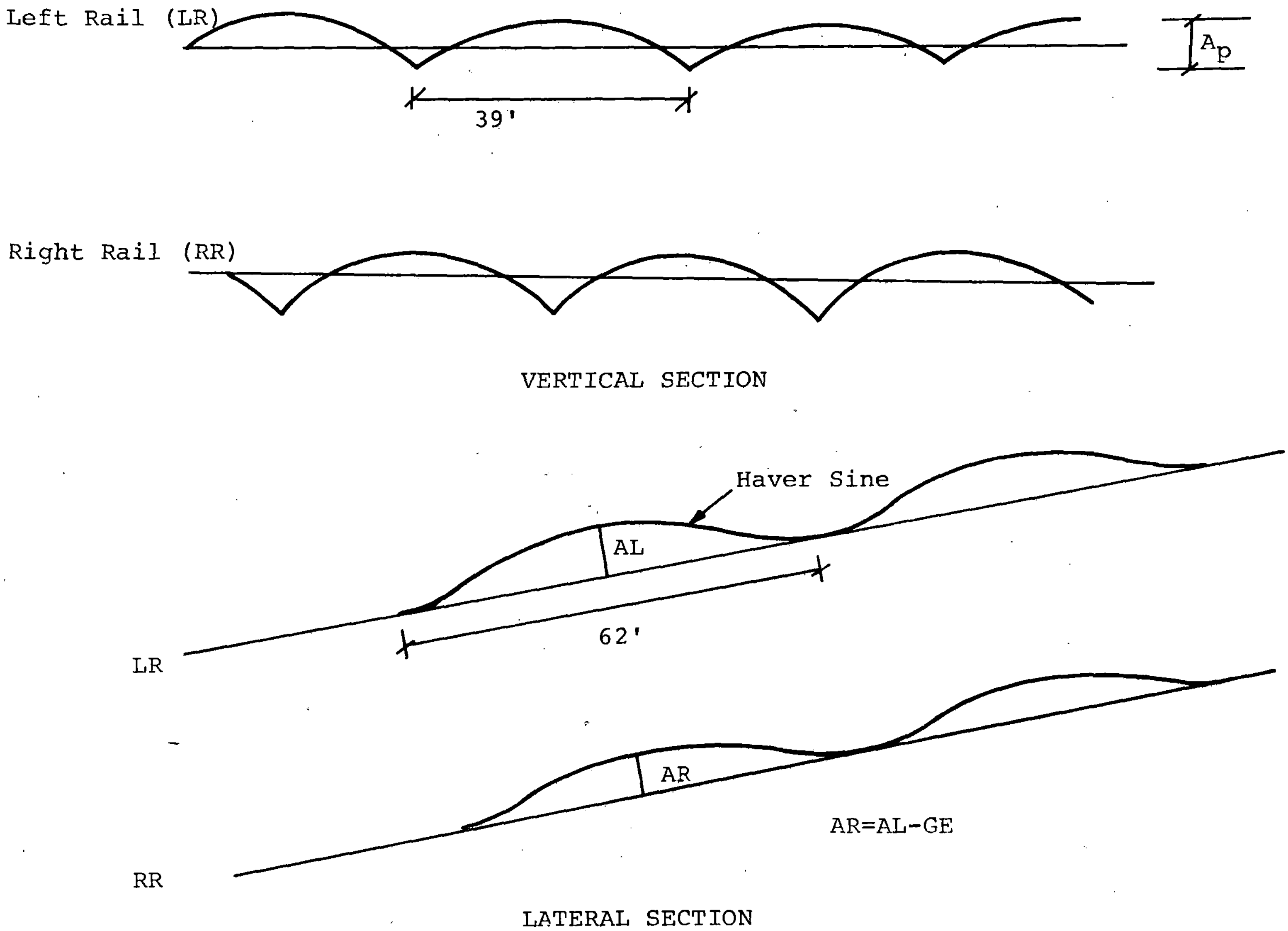
is attained simultaneously as the rails are half-staggered. For alignment error haver-sine of 39' wave length is used.

2.7 COUPLED PROFILE ALIGNMENT AND GAGE ERROR

In Fig.11 the track configuration for coupled profile alignment and gage error is shown. The rails can be staggered with respect to each other. The configuration shown has only the coupled effect of profile and gage error. Wide gage is incorporated by adjusting the amplitude of right rail alignment error such that the difference in lateral error amplitude gives gage error amplitude. Profile error is represented by a rectified sine wave of 39'. A haver-sine wave of 62' is used for lateral errors.

2.8 COUPLED PROFILE, ALIGNMENT, GAGE AND WARP ERROR

In Fig. 12 a combined track profile is shown for all four errors. The configurations of Fig. 8 and Fig. 11 have been superimposed to represent the coupling of profile, gage, alignment and warp error.



AP=Amplitude of Profile Error
AL=Alignment Error left Rail
AR=Alignment Error Right Rail
GE=Gage Error

FIGURE 11 TRACK GEOMETRY FOR COUPLED PROFILE ALIGNMENT AND GAGE

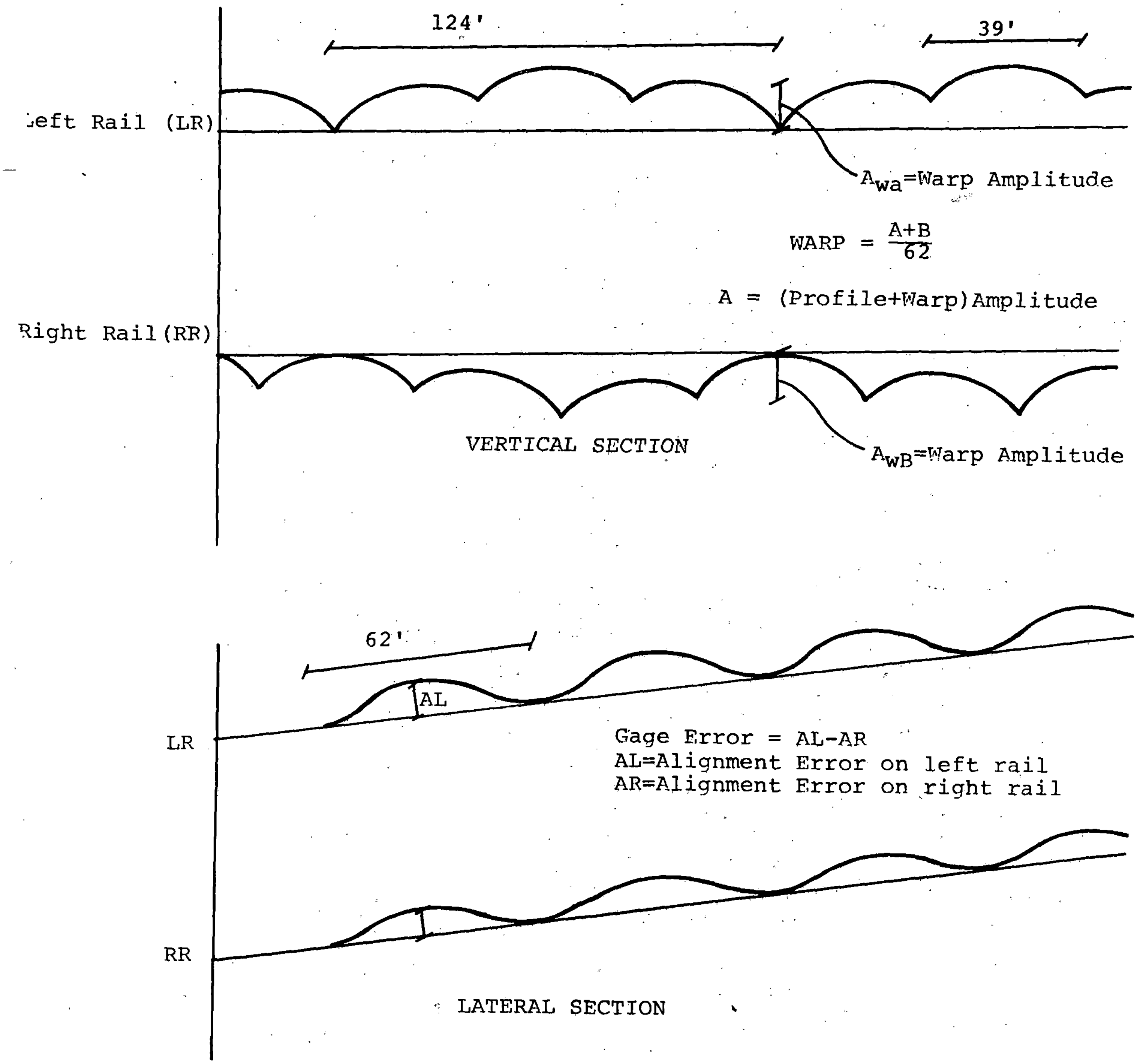


FIGURE 12 TRACK GEOMETRY FOR COUPLED PROFILE, ALIGNMENT GAGE AND WARP

3. RESULTS

A 100-Ton covered hopper car with 97 in. center of gravity height and 45 ft. truck center distance and a 70-ton box car having 98 in. center of gravity height and 39 ft. truck center distance have been selected for simulation.

The following types of track errors have been investigated.

1. Profile (amplitude:½,3/4,1¼,1-3/4, 2,2¼ in.)
2. Profile and warp (amplitude:1¼/1 and 2¼/2 in.)
3. Profile and wide gage (amplitude:1¼/½ and 2-3/4 in.)
4. Profile and narrow gage (amplitude:1¼/½ in.)
5. Profile and alignment (amplitude:2¼/1-3/4 in.)
6. Profile,alignment and wide gage (amplitude: 2¼/1-3/4/1 in.)
7. Profile,alignment wide gage and warp (amplitude:2¼/1-3/4/1/2 in.)

All the above types of track errors have been used in the simulation of 100-Ton Car; for the 70-Ton Car case only a few types of errors have been selected. Data for both the cars are included in the Appendix.

Simulations were made at various speeds at half mile/hour increments. The speed at which maximum roll angle occurs is termed as the roll critical speed. Tables 2 and 3 summarize results for variables of interest, obtained during simulation for the 100-Ton covered Hopper Car and 70-Ton Loaded Box Car, respectively.

TABLE 2

100 TON COVERED HOPPER CAR

ERROR TYPE	SPEED MPH	ROLL ANGLE (Pk.-Pk) DEG.	CARBODY ACCELERATION (0-Pk) L	V	CENTER PLATE LOAD kips	SIDE BEARING LOAD kips	WHEEL LOAD (2 WH. COMBINED	WHEEL LIFT & DURATION in.	sec.	AMPLITUDE OF ERRORS in.
Cross level	14.5*	1.75	0.15	0.05	122.5	0.0	85.0	No.	-	½
Cross level	15.0*	9.25	0.60	0.6	147.5	15.50	160.0	Yes	0.21	3/4
Cross level	14.0*	14.25	2.35	1.8	272.2	190.0	245.0	Yes	0.39	1¼
Cross level	14.0*	14.5	3.05	4.2	290.0	320.0	380.0	Yes	0.99	1-3/4
Cross level	13.0*	16.75	2.25	5.0	247.5	240.0	325.0	Yes	1.12	2
Cross level	12.5*	25.5	3.85	9.4	674.80	420.0	420.0	Yes	1.11	2¼
P/W	13.0	7.2			125.0	140.0	135.0			1¼/1
	13.5	8.9			130.0	160.0	160.0			"
	14.0	11.8			350.0	290.0	335.0			"
	14.5*	14.0			470.0	560.0	400.0			"
	15.0	11.8			200.0	630.0	420.0			"
	15.5	10.4			160.0	180.0	200.0			"
	16.0	8.0			165.0	200.0	240.0			"
P/W	11.0	9.4			338.0	460.0	415.0			2¼/2
	11.5	10.75			378.0	545.0	395.0			"
	12.0*	16.0			778.0	895.0	430.0			"
	12.5	12.0			800.0	800.0	510.0			"
	13.0	14.75			510.0	710.0	460.0			"
	13.5	13.00			660.0	1050.0	505.0			"
	14.0	12.8			1220.0	1380.0	670.0			"
	14.5	12.45			820.0	1190.0	740.0			"
	15.0	11.75			300.0	840.0	630.0			"

* critical speed
P - Profile
W - Warp

100 TON COVERED HOPPER CAR

ERROR TYPE	SPEED MPH	ROLL ANGLE (Pk-Pk) DEC.	CARBODY ACCELERATION (0-Pk) L	V	CENTER PLATE LOAD kips	SIDE BEARING LOAD kips	WHEEL LOAD (2 WH. COMBINED	WHEEL LIFT & DURATION in. sec.	AMPLITUDE OF ERRORS in.
P/W.G.	12.5	6.6			147.5	105.0	142.5		1¼/½
	13.0	8.1			205.0	175.0	287.5		
	13.4	9.3			225.0	226.25	280.0		
	14.0*	13.7			172.5	177.5	189.0		
	14.5	13.0			176.25	167.5	170.0		
	15.0	11.20			161.25	165.5	160.0		
	15.5	10.8			165.75	162.5	164.0		
P/W.G.	12.0	9.0			375.0	490.0	370.0		2-3/4 2-3/4
	12.5	10.75			430.0	375.0	345.0		
	13.0*	16.75			360.0	465.0	395.0		
	13.5	14.75			665.0	200.0	370.0		
	14.0	14.75			445.0	545.0	380.0		
	15.0	11.50			455.0	630.0	405.0		
	15.5	10.75			350.0	365.0	365.0		
	16.0	9.75			315.0	185.0	360.0		
P/N.G.	13.0	6.7			127.5	110.0	127.5		1½/½
	14.0	8.8			140.0	142.5	142.5		
	14.5*	14.3			2220.0	167.5	180.0		
	15.0	12.8			167.5	162.5	202.5		
	15.5	11.9			157.5	165.0	167.5		

* critical speed

P - Profile

W.G. - Wide gage

N.G. - Narrow Gage

100 TON COVERED HOPPER CAR

ERROR TYPE	SPEED MPH	ROLL ANGLE (Pk-Pk) DEG.	CARBODY ACCELERATION (0-PK) L	V	CENTER PLATE LOAD kips	SIDE BEARING LOAD kips	WHEEL LOAD (2 WH. COMBINED) kips	WHEEL LIFT & DURATION	AMPLITUDE OF ERRORS
P/AL	-	-	-	-	-	-	-	-	2¼/1-3/4
	-	-	-	-	-	-	-	-	2¼/1-3/4
	11.5	9.25			345.0	667.5	425.0		
	12.0	10.25			450.0	705.0	450.0		
	12.5	11.25			860.0	532.5	460.0		
	13.0	16.25			775.0	540.0	385.0		
	13.5*	16.50			615.0	315.0	380.0		
	14.0	14.875			455.0	525.0	355.0		
	14.5	16.125			315.0	915.0	520.0		
	15.0	13.125			325.0	307.5	285.0		
P/AL/WG	12.0	10.125			540.0	370.0	430.0		2¼/1-3/4/1
	12.5	11.25			290.0	370.0	405.0		2¼/1-3/4/1
	13.0*	22.0			380.0	280.0	387.0		
	13.5	16.125			700.0	470.0	425.0		
	14.0	13.5			230.0	270.0	440.0		
	14.5	14.75			240.0	180.0	238.0		
P/AL/WG/W	12.0	10.0			645.0	460.0	395.0		2¼/1-3/4/1/2
	12.5	14.0			760.0	550.0	470.0		
	13.0*	17.125			900.0	1130.0	955.0		
	13.5	14.75			630.0	1560.0	890.0		
	14.0	12.125			900.0	760.0	550.0		
	14.5	13.375			805.0	1010.0	695.0		
	15.0	11.25			510.0	870.0	560.0		

* critical speed

P - Profile

AL - Alignment

W - Warp

WG - Wide gage

TABLE 3

50"-70 TON BOX CAR LOADED

ERROR TYPE	SPEED MPH	ROLL ANGLE (Pk-Pk) DEG.	CARBODY ACCELERATION (0-Pk) L	V	CENTER PLATE LOAD kips	SIDE BEARING LOAD kips	WHEEL LOAD (2 WH. COMBINED) kips	WHEEL LIFT & DURATION in.	sec.	AMPLITUDE OF ERRORS INCHES
Cross level	14.5	4.75	0.6	0.1	107.5	45.0	85.0	NO	-	½
Cross level	15.5	11.0	0.95	0.6	150.0	145.0	135.0	Yes	0.57	5/8
Cross level	15.0	16.25	1.35	1.0	147.5	145.0	150.0	Yes	1.30	3/4
Cross level	14.5	15.0	1.09	1.2	162.5	152.5	140.0	Yes	0.60	1
Cross level	14.0	17.5	1.25	1.2	140.0	160.0	160.0	Yes	0.87	1¼
Cross level	13.0	22.0	1.1	2.8	225.0	200.0	220.0	Yes	1.14	1-3/4
Cross level	12.5	23.75	1.45	2.4	195.0	225.0	240.0	Yes	0.87	2
Cross level	12.0	24.5	1.25	2.8	240.0	255.0	245.0	Yes	1.38	2¼
P/WG	13.0	8.5			125.0	143.75	155.0			1¼/½
	13.5	9.4			147.5	155.0	151.25			
	14.0*	16.0			225.0	177.5	175.00			
	14.5	15.0			155.0	160.0	157.5			
	15.0	13.1			151.25	182.5	172.25			
	15.5	11.2			127.5	175.0	152.50			
P/WG	11.5	10.0			195.0	190.0	212.5			2-3/4
	12.0	12.0			265.0	230.0	190.0			
	12.5*	21.0			190.0	240.0	230.0			
	13.0	19.75			200.0	220.0	202.5			
	13.5	18.0			197.0	255.0	197.5			
	14.0	17.0			165.0	225.0	167.5			
P/AL	11.0	8.625			110.0	80.0	105.0			2¼/1-3/4
	11.5	9.5			128.0	112.5	123.0			
	12.0*	23.875			200.0	255.0	224.0			
	12.5	22.625			180.0	255.0	237.0			
	13.0	20.875			195.0	262.5	225.0			
	14.0	18.0			155.0	307.5	235.0			
	14.5	16.875			155.0	232.5	183.0			

* critical speed   P - Profile , AL - alignment, W - Warp, WG - Wide gage

3.1 EFFECT OF PROFILE ERROR

Tables 2 & 3 show the effect of profile error magnitude on the critical speed of 100-Ton and 70-Ton cars. The general trend is that the critical speed is reduced as profile error is increased. In the case of the 100-Ton car the critical speed reduces from 14.5 to 12.5 mph and for the 70-Ton car it reduces from 14.5 to 12 mph with an increase in the profile error from 1/2 to 2½ in.

It can be seen from Fig.13 that the roll angle for both cars increases with an increase in the profile error. Peak to peak roll angle increases from 1.75 to 25.5 degrees for the 100-Ton Car; for the 70-Ton Car it increases from 4.75 to 24.5 degrees as the profile error increases ½ to 2¼ in. Except at 2¼ in. profile error the roll angles for the 70-Ton Car are higher than the 100-Ton Car. The primary reason for this is that the amount of damping used for the 70-Ton car simulation is representative of worn wedge condition whereas for the 100-Ton Car it is fairly equal to the amount of damping provided by a new wedge.

Figs.14 and 15 show the carbody lateral and vertical accelerations as affected by the profile error. The increase in carbody lateral acceleration for the 70-Ton Car is from 0.6 g to 1.25 g for an increase in profile error from ½ to 2¼ in. whereas for the 100-Ton Car the increase is from 0.15 g to 3.85 g. For the same increase in the profile error, vertical carbody acceleration for the 70-Ton Car increases from 0.1 g to 2.8 g and for the 100-Ton Car it increases from 0.05 g to 9.4 g.

100 TON COVERED HOPPER CAR

70 TON LOADED BOX CAR

FIGURE 13

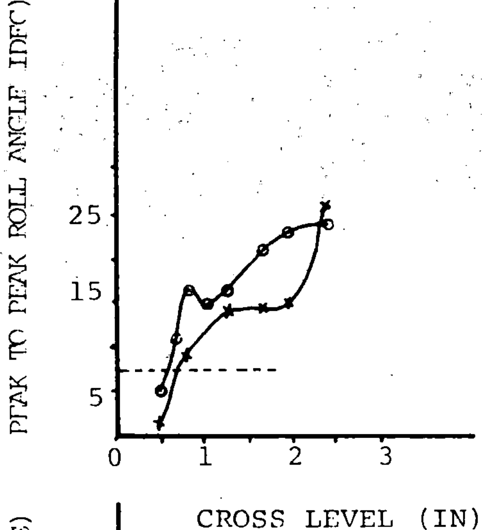


FIGURE 14

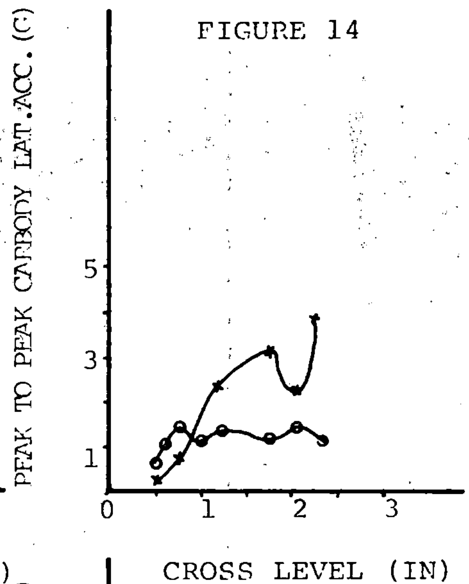
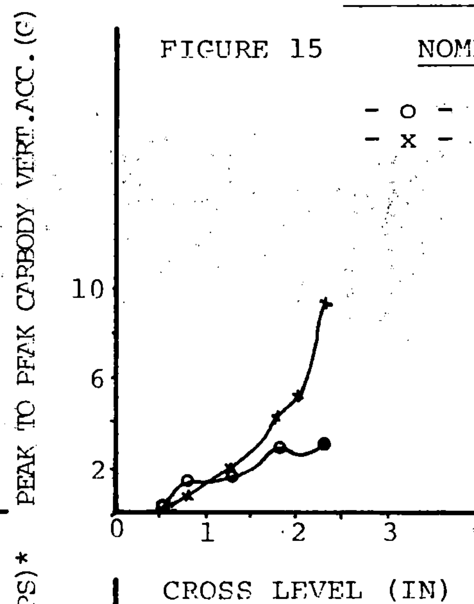


FIGURE 15



NOMENCLATURE

- o - 70 ton box car
- x - 100 ton hopper car

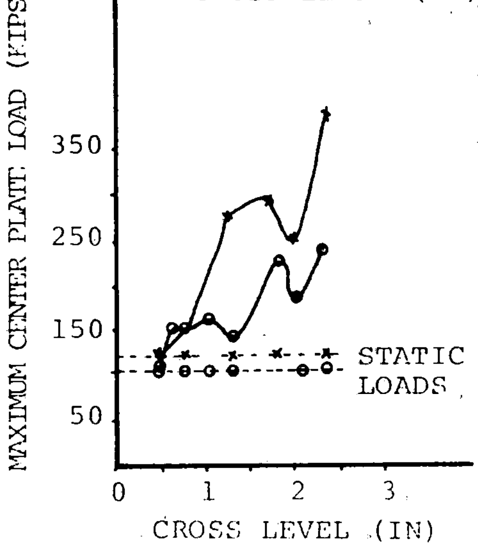


FIGURE 16

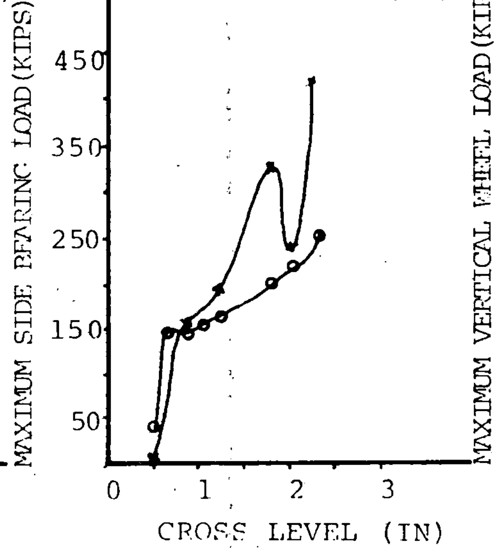


FIGURE 17

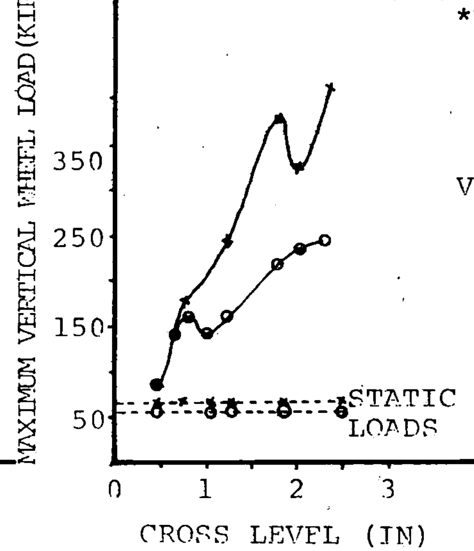


FIGURE 18

*WHEEL LOAD REPORTED IS FOR TWO WHEELS COMBINED ON TRUCK SIDE

LAT.ACC. = LATERAL ACCELERATION

VERT.ACC. = VERTICAL ACCELERATION

G = GRAVITATIONAL ACCELERATION

Figs.16 and 17 show the centerplate and side bearing loads for the 70-Ton and 100-Ton cars. The model outputs give the left and right components of the centerplate load. The centerplate load reported here is the maximum of the front and rear centerplates obtained by the addition of the left and right load components. The centerplate load for the 70-Ton Car increases from 107.5 to 240 kips whereas for the 100-Ton Car it increases from 122.5 to 674.8 kips corresponding with an increase in profile error from $\frac{1}{2}$ to $2\frac{1}{4}$ in. Comparison between the maximum loads of 70-Ton and 100-Ton Cars does not reveal a vivid picture of the difference in their dynamic behavior. The ratio between the static and the dynamic loads of the cars gives a clear comparison of their dynamic behavior. The static centerplate load for the 70-Ton Car is 102.5 kips whereas for the 100-Ton Car it is 121.9 kips. The ratio between the static and dynamic centerplate load for the 70-Ton Car is 1.05 at $\frac{1}{2}$ in. profile error and 2.34 at $2\frac{1}{4}$ in. profile error, whereas for the 100-Ton Car these values are 1.01 and 5.54, respectively. The side bearing load at $\frac{1}{2}$ in. profile error for the 70-Ton Car is 45 kips and at $2\frac{1}{4}$ in. profile error it is 255 kips, whereas the values for the 100-Ton Car are 0 and 440 kips, respectively.

Fig.18 shows the vertical wheel load for both cars. In the model, the two wheels on each side of the truck are combined together. Therefore, the wheel load output from the model is approximately twice the actual wheel load. At $\frac{1}{2}$ in. profile error the wheel load for both the 70-Ton and

100-Ton Cars is 85 kips whereas at 2¼ in. profile error they are 245 and 440 kips, respectively. The corresponding ratios between dynamic and static wheel loads are 1.55 and 1.29 (70-Ton and 100-Ton Car at ½ in. profile error) and 4.46 and 6.39 (70-Ton and 100-Ton Car at 2¼ in. profile error). Tables 2 & 3 also show occurences of wheel lift. Except for ½ in. profile error, wheel lift occurs for both cars at all the profile errors studied. For the 100-Ton Car the trend is that the wheel lift duration increases with the profile error, whereas for the 70-Ton Car this is not the case. Wheel lift is an important factor in derailment tendency. Longer wheel lift duration enchances the possibility of derailment.

From Figs. 13 to 18 it can be concluded that the 100-Ton car performs better than the 70-Ton car when the profile error is low. When the profile error increases the deteriorat-ion of performance for the 100-Ton Car is more pronounced than the 70-Ton Car. For a 6 deg. peak to peak roll angle limit the allowable profile error for the 70-Ton car with worn wedges is 0.6 in. It should be noted that for the same 70-Ton car with new wedges the allowable profile error would be more than 0.525 in. and for the same 100-Ton car with worn wedges it would be less than 0.6 in.

FRA track classification limits the operating speed according to track error and is not dependent upon car type. It is suggested that the critical speed limitations should be taken into consideration along with car type. There is no doubt that different cars will have different critical speeds.

Usually the critical speed of the majority of the loaded cars operating on one-half staggered rails (major part of the track in North American Sub-Continent is one half staggered) lies between 12 and 22 mph. For example class 2 track limits the operating speed to 25 mph. A car operating on class 2 track whose critical speed lies around 22 mph will perform better if its speed is increased. Lowering its speed for below 22 mph will also improve its performance, but this may result in an uneconomical operating practice. Computer simulation runs were made to determine the dynamic behavior of both 100-Ton and 70-Ton Cars travelling on track with wide and narrow gages and alignment errors. These types of errors did not have any significant effect on the rock and roll behavior of the cars. In reality the track errors mentioned above are usually coupled with the profile errors. Hence, five additional track conditions have been simulated. They are:

1. Profile coupled with warp
2. Profile coupled with wide gage
3. Profile coupled with narrow gage
4. Profile coupled with alignment
5. Profile coupled with warp, wide gage and alignment.

3.2 EFFECT OF PROFILE COUPLED WITH WARP ERROR

Figs.19 and 23 show the peak to peak roll angle of a 100-Ton Car travelling on track with $1\frac{1}{4}$ in. profile error and $1\frac{1}{4}$ in. profile coupled with 1 in. warp error and $2\frac{1}{4}$ in.

100 TON COVERED HOPPER CAR

PERTURBATION NOMENCLATURE

- o - PROFILE

- x - PROFILE WARP

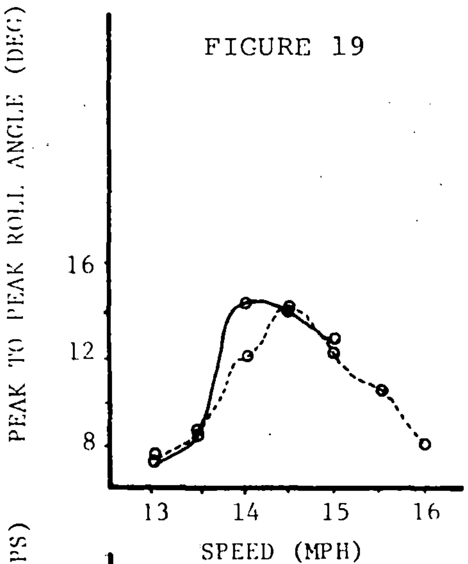


FIGURE 19

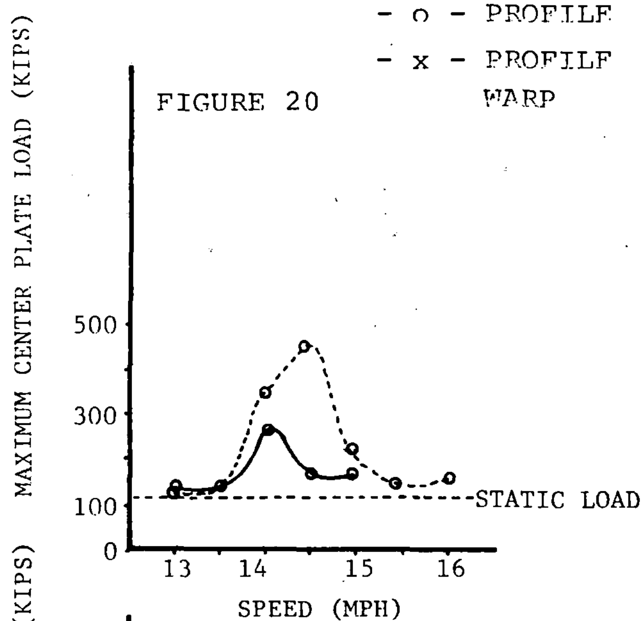


FIGURE 20

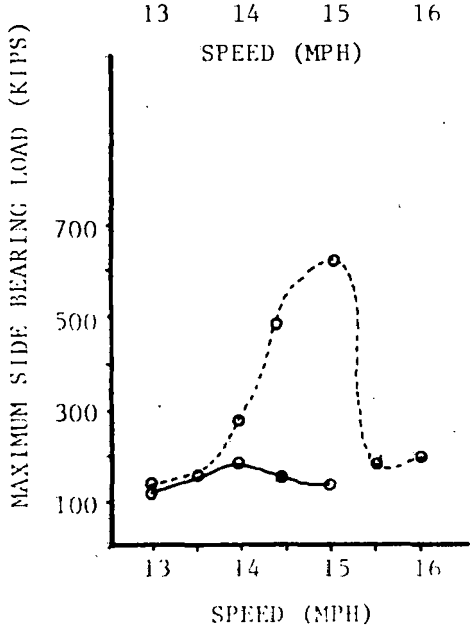


FIGURE 21

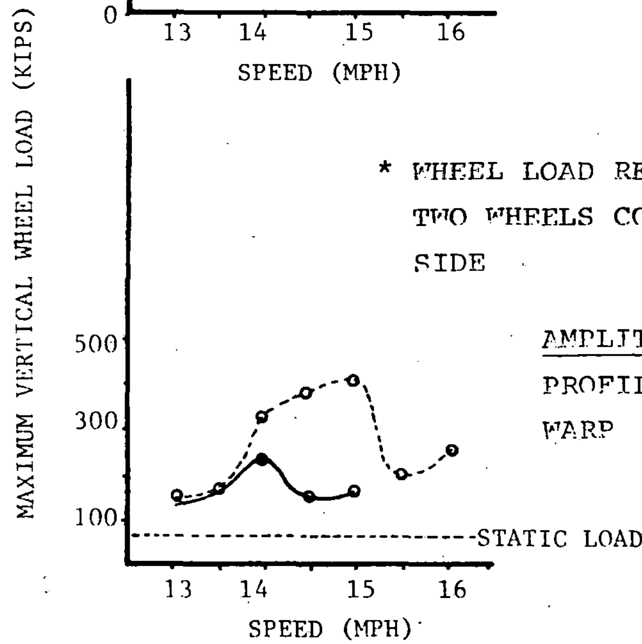


FIGURE 22

* WHEEL LOAD REPORTED IS FOR TWO WHEELS COMBINED ON A TRUCK SIDE

<u>AMPLITUDE</u>

PROFILE = 1½"

WARP = 1"

100 TON COVERED HOPPER CAR

PERTURBATION NOMENCLATURE

- o - PROFILE

- x - PROFILE WARP

FIGURE 23

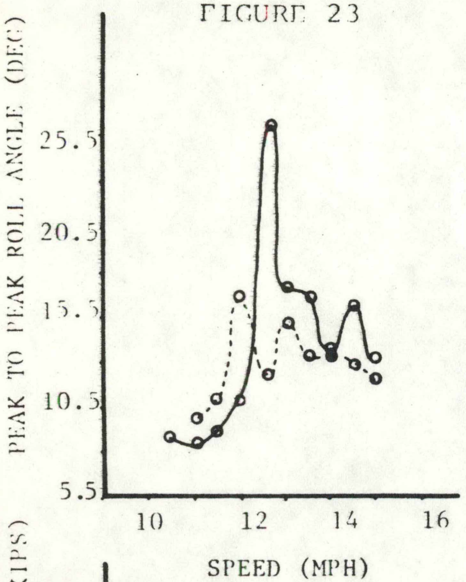
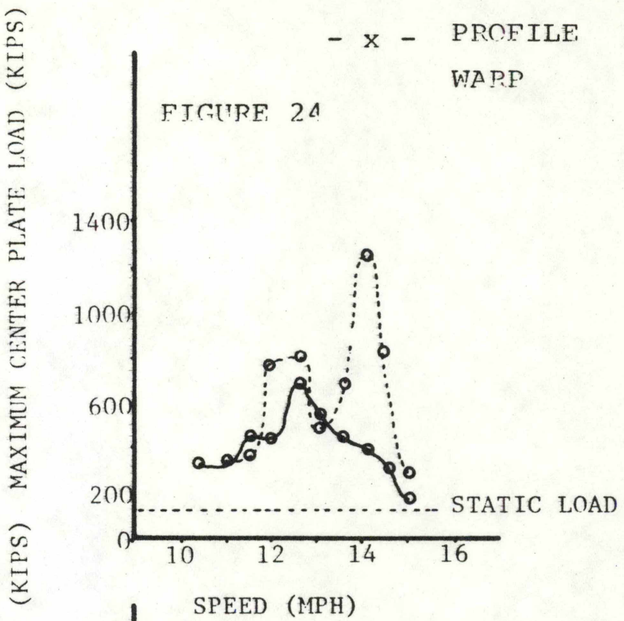


FIGURE 24



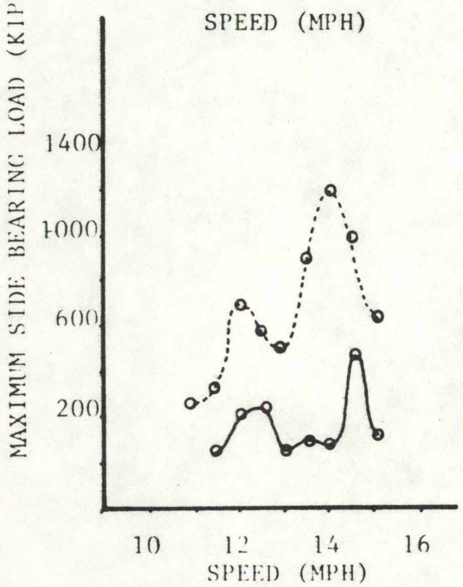


FIGURE 25

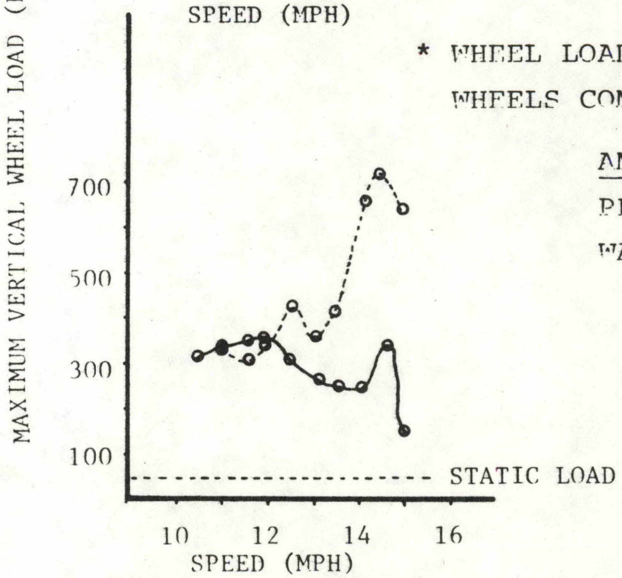


FIGURE 26

* WHEEL LOAD REPORTED IS FOR TWO WHEELS COMBINED ON A TRUCK SIDE

AMPLITUDE

PROFILE = 2½"

WARP = 2"

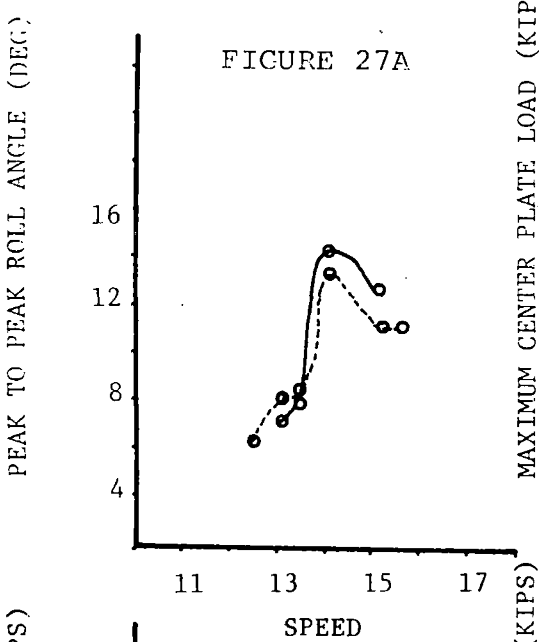


FIGURE 27A

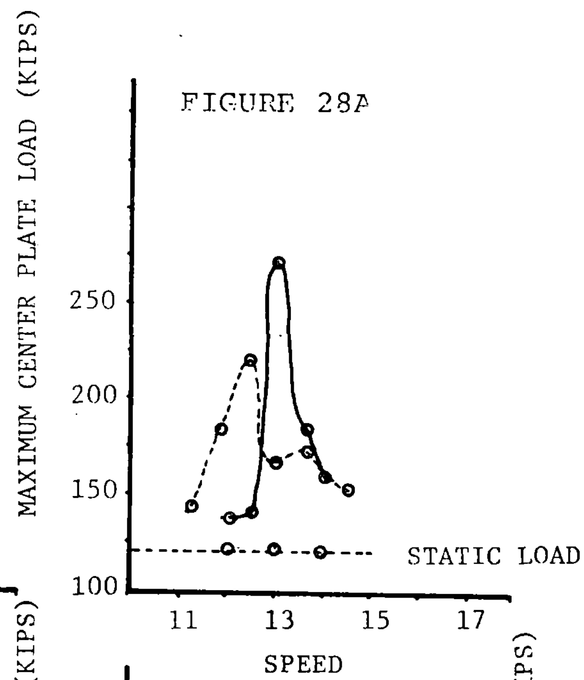


FIGURE 28A

100 TON COVERED HOPPER CAR

PERTURBATION NOMENCLATURE

- o - PROFILE

- x - PROFILE WIDE GAGE

* WHEEL LOADS REPORTED ARE FOR TWO WHEELS COMBINED ON A TRUCK SIDE

AMPLITUDE

PROFILE = 1½"

WIDE GAGE = ½"

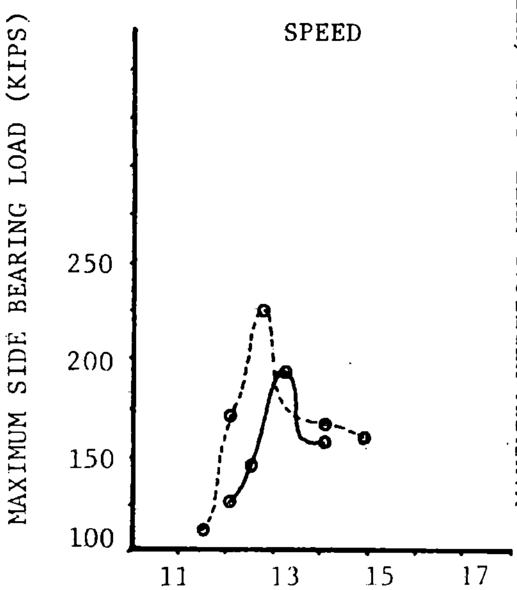


FIGURE 29A

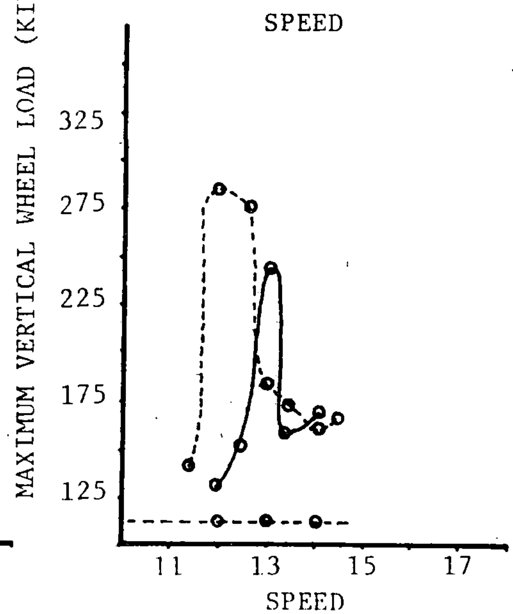


FIGURE 30A

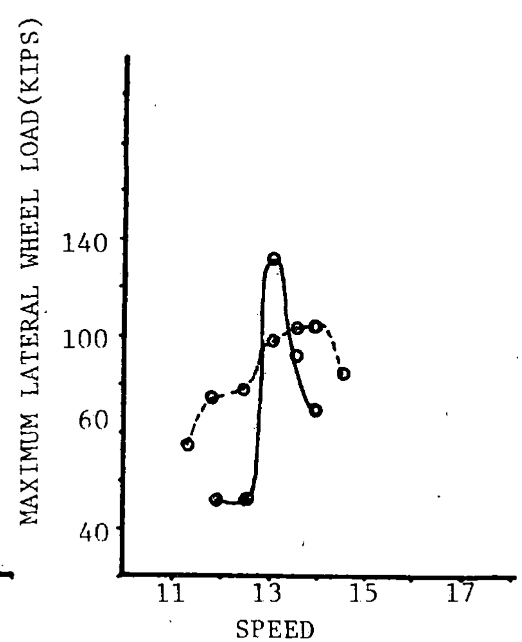


FIGURE 31a

70 TON LOADED BOX CAR

PERTURBATION NOMENCLATURE

- o - PROFILE

- x - PROFILE
WIDE GAGE

* WHEEL LOADS REPORTED ARE FRO TWO WHEELS COMBINED ON A TRUCK SIDE

AMPLITUDE

PROFILE = 1¼"

WIDE GAGE = ½"

FIGURE 27B

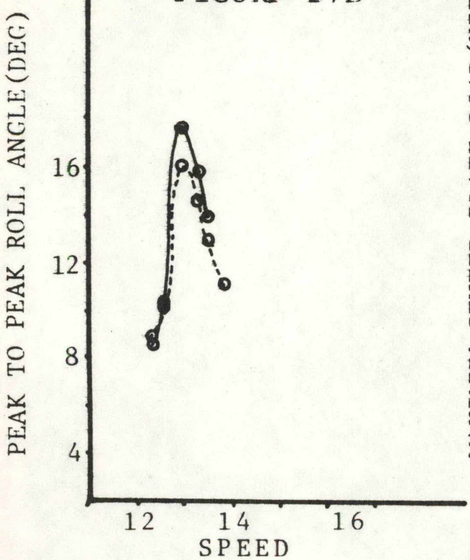
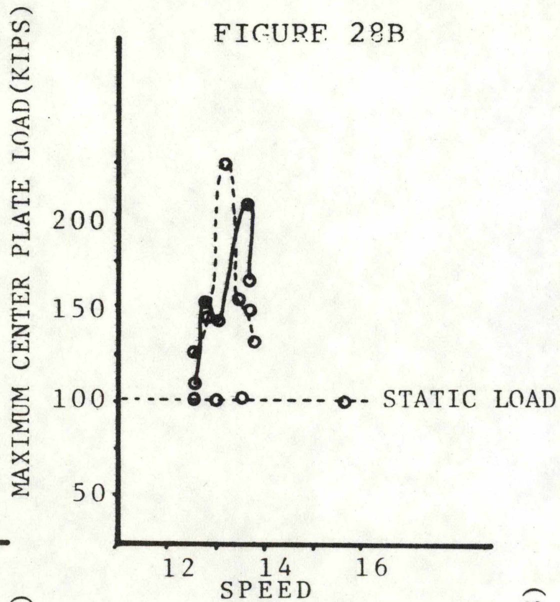


FIGURE 28B



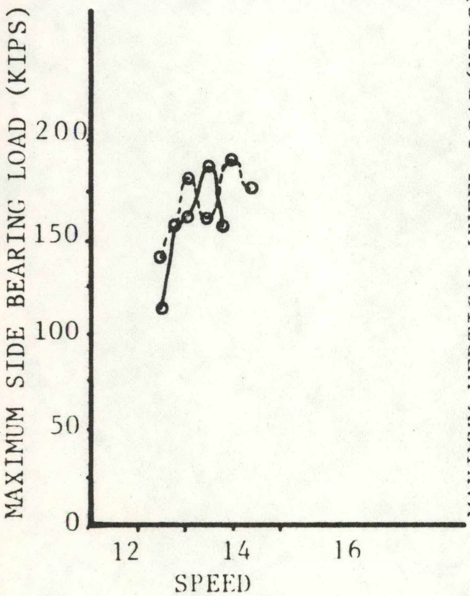


FIGURE 29B

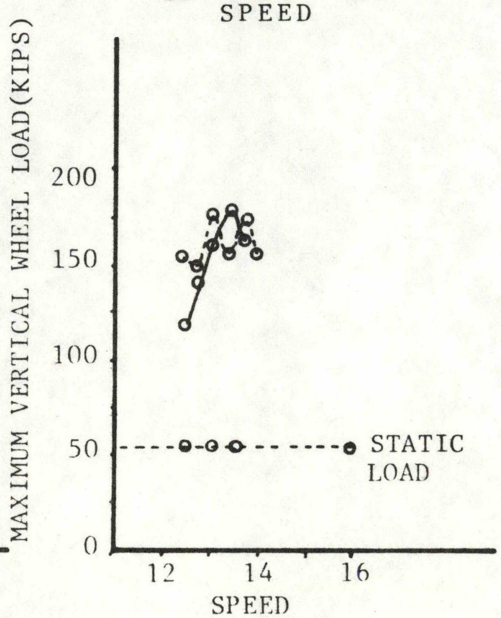


FIGURE 30B

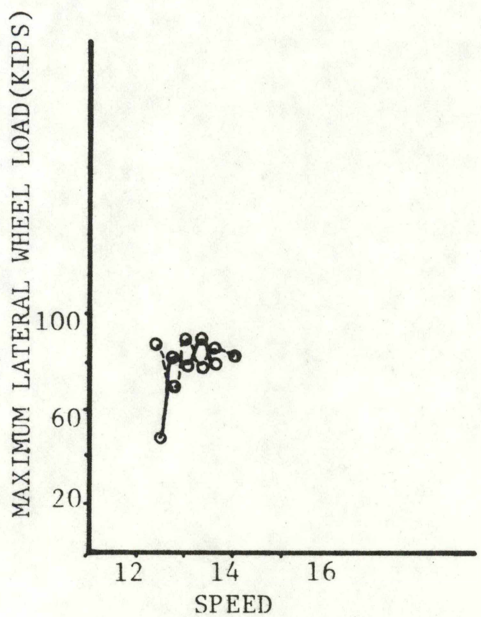


FIGURE 31B

profile error. In the first case, introduction of warp error in a track with profile error has no effect on the peak to peak roll angle of the car whereas in the second case the roll angle appreciably reduces from 25.5 to 14.8 deg.

Figs. 21, 22 and 24 to 26 show that for both the above cases the various loads increase with superposition of warp error to 1¼ in. profile error. The centerplate, side bearing and wheel loads increase from 272 to 470 kips, 190 to 560 kips and 245 to 400 kips, respectively. Similar increases are obtained for 2 in. warp error on 2¼ in. profile error.

3.3 EFFECT OF PROFILE COUPLED WITH WIDE GAGE ERROR

Figs.27A and 27B show the peak to peak roll angle of 100-Ton and 70-Ton Cars travelling on track with profile and profile with wide gage errors. For both cars there is no appreciable change in the roll angle with the above change in track errors. For case 1 where the profile error is 1¼ in. and profile/wide gage errors are 1¼/½ in. the change in peak to peak roll angle is from 14.3 to 13.7 deg. for the 100-Ton Car and from 17.5 to 16 deg. for the 70-Ton Car. For case 2 where the track is changed from 2 in. profile error to 2 in. profile error with 3/4 in. wide gage error, the roll angle for the 100-Ton Car remains the same (16.75 deg.) and for the 70-Ton Car there is a small decrease from 23.75 to 21 deg.

Figs. 28A through 31A, 28B through 31B, and Figs. 32 through 36 show the various component loadings of 70-Ton and 100-Ton Cars for both the above cases. In general, the coupling of wide gage error to profile error for a 70-Ton Car

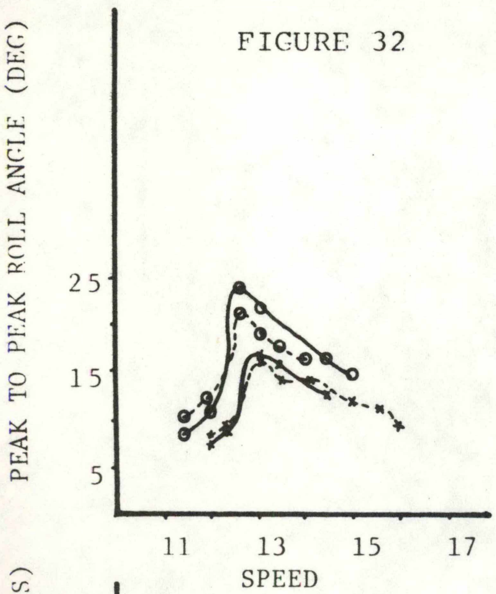


FIGURE 32

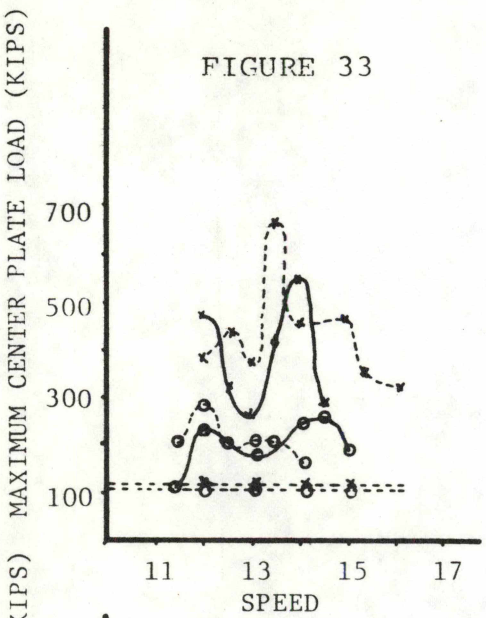


FIGURE 33

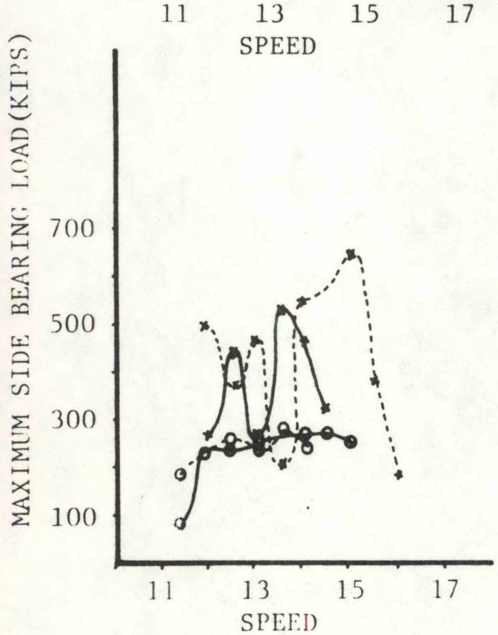


FIGURE 34

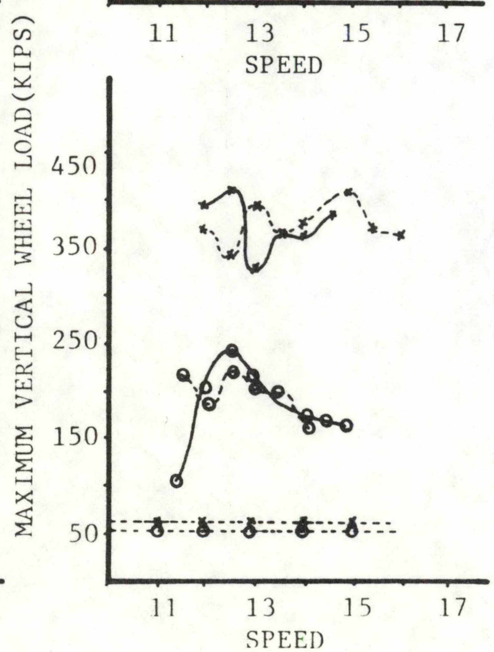


FIGURE 35

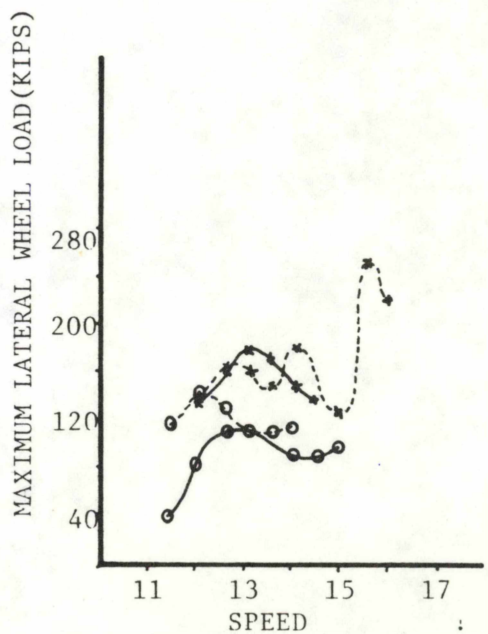


FIGURE 36

100 TON COVERED HOPPER CAR

70 TON LOADED BOX CAR

PERTURBATION NOMENCLATURE

100 TON HOPPER CAR

——x—— PROFILE

---x--- PROFILE GAGE (WIDE)

70 TON BOX CAR

——o—— PROFILE

——x—— PROFILE GAGE(WIDE)

* WHEEL LOADS REPORTED ARE FOR TWO WHEELS COMBINED ON A TRUCK SIDE

AMPLITUDE

PROFILE = 2"

GAGE(WDIE) = 3/4"

increases centerplate load in both cases 1 and 2. The side bearing, vertical wheel loads and the lateral wheel load also remain about the same in both cases. Introduction of wide gage error increases the lateral wheel load. The coupling of wide gage error to profile error for the 100-Ton Car reduces the centerplate and lateral wheel load in case 1. The reverse is true for case 2. Gage error when coupled with profile error increases the side bearing loads for both 1 and 2 cases, whereas the vertical wheel load increases for case 1 but remains about the same for case 2.

3.4 EFFECT OF PROFILE COUPLED WITH NARROW GAGE ERROR

Fig. 37 shows that the peak to peak roll angle of the 100-Ton Car is unaffected by the introduction of narrow gage error to the profile error. Figs. 38 to 41 show the various loadings. Loads on all components are reduced with the coupling of narrow gage error to profile error. The center-plate reduces from 272 to 220 kips. Also the side bearing load, vertical and lateral wheel loads reduce from 190 to 168 kips, 245 to 180 kips and 132 to 88 kips respectively.

3.5 EFFECT OF PROFILE COUPLED WITH ALIGNMENT ERROR

The effect of 2¼ in. profile error and the same error coupled with 1-3/4 in. alignment error have been investigated for both 70-Ton box and 100-Ton covered hopper cars. Figs 42 shows that the introduction of alignment error to profile error has no effect on the carbody roll for the 70-Ton Car whereas the carbody roll of the 100-Ton Car significantly reduces from 25.5 to 16.5 deg. Figs. 43 to 45 show the various loadings on components of 70-Ton and 100-Ton Cars.

FIGURE 37

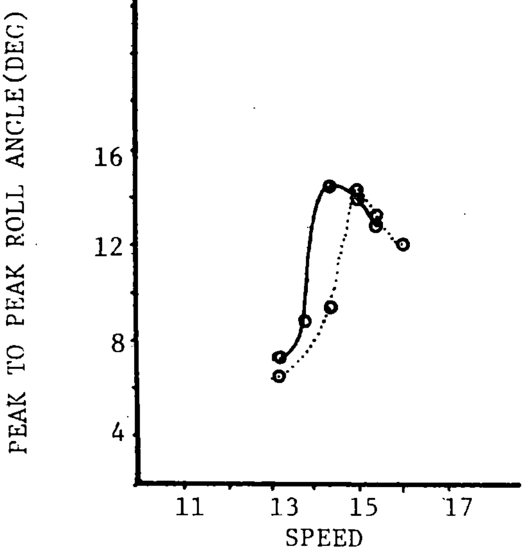
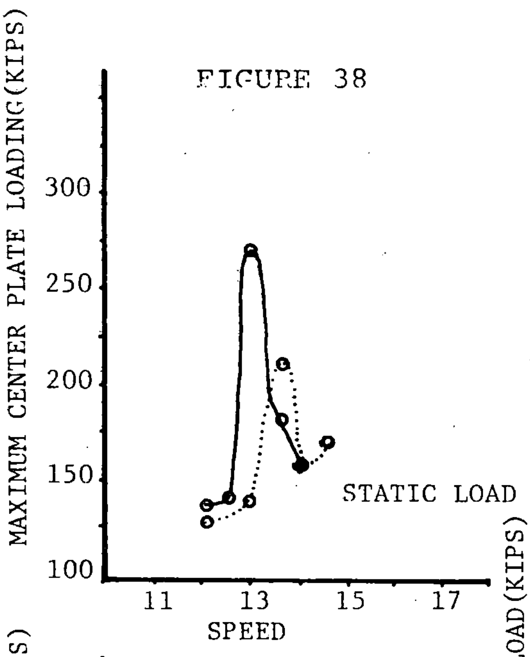


FIGURE 38



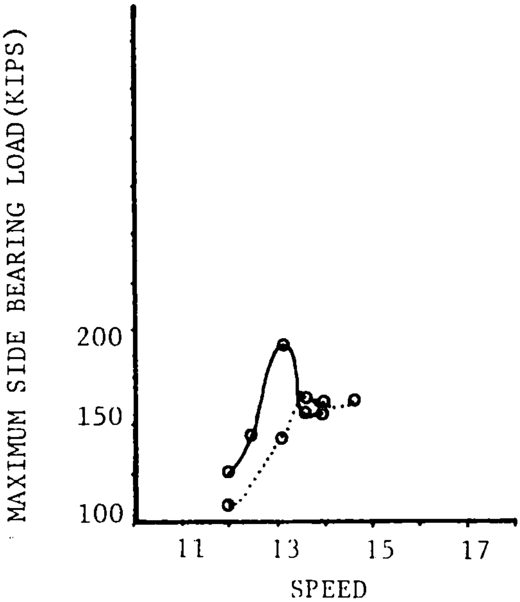


FIGURE 39

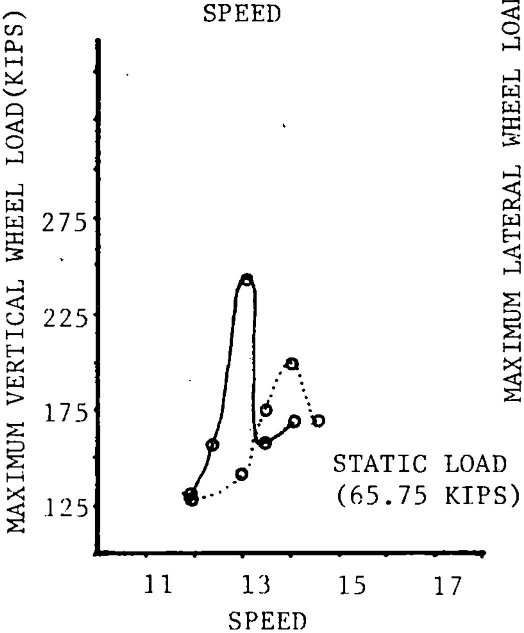
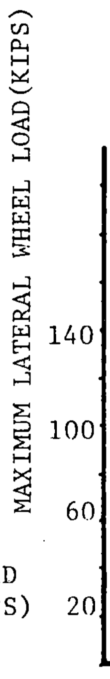


FIGURE 40



100 TON COVERED HOPPER CAR

PERTURBATION NOMENCLATURE

——o—— PROFILE

---o--- PROFILE NARROW GAGE

* WHEEL LOADS REPORTED ARE FOR TWO WHEELS COMBINED ON A TRUCKSIDE

AMPLITUDE

PROFILE = $1\frac{1}{4}$"

NARROW GAGE = $\frac{1}{2}\frac{1}{4}$

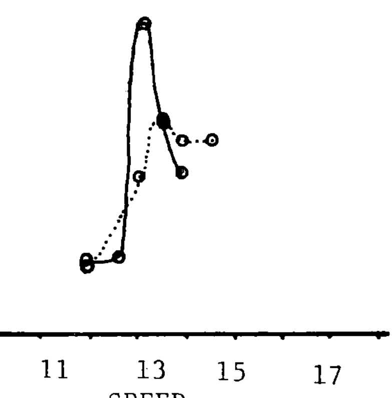


FIGURE 41

FIGURE 42

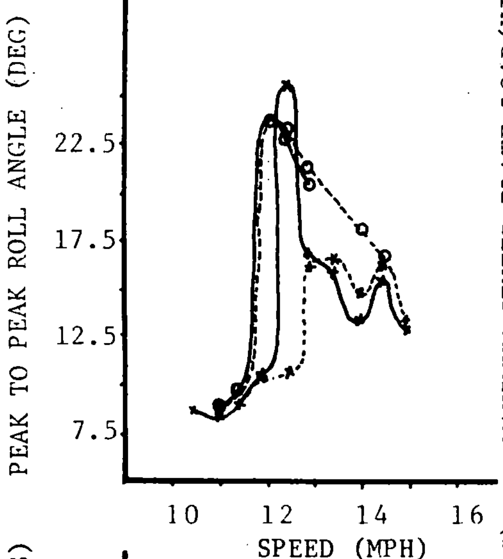
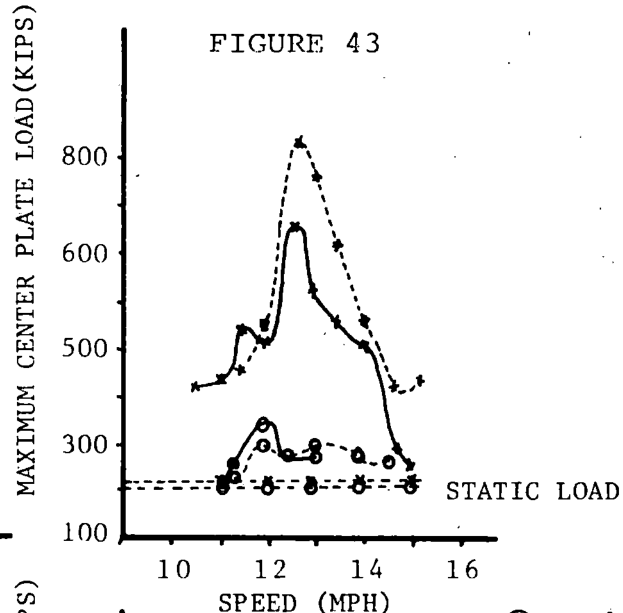


FIGURE 43



100 TON COVERED HOPPER CAR

70 TON LOADED BOX CAR

PERTURBATION NOMENCLATURE

100 TON HOPPER CAR

——x—— PROFILE

---x--- PROFILE ALIGNMENT

70 TON BOX CAR

——o—— PROFILE

---o--- ALIGNMENT

* WHEEL LOADS REPORTED ARE FOR TWO WHEEL COMBINED ON A TRUCK SIDE

AMPLITUDE

PROFILE = 2¼"

ALIGNMENT=1-3/4"

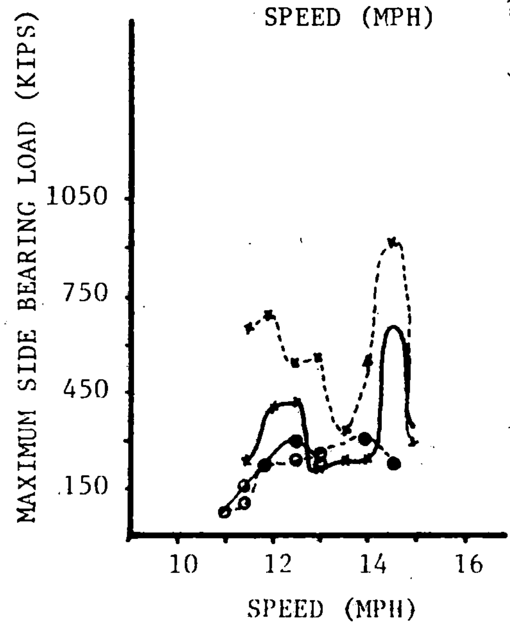
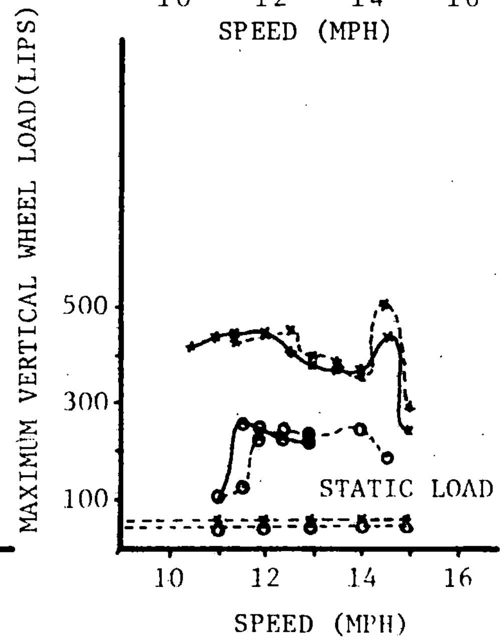


FIGURE 44



FIGURE

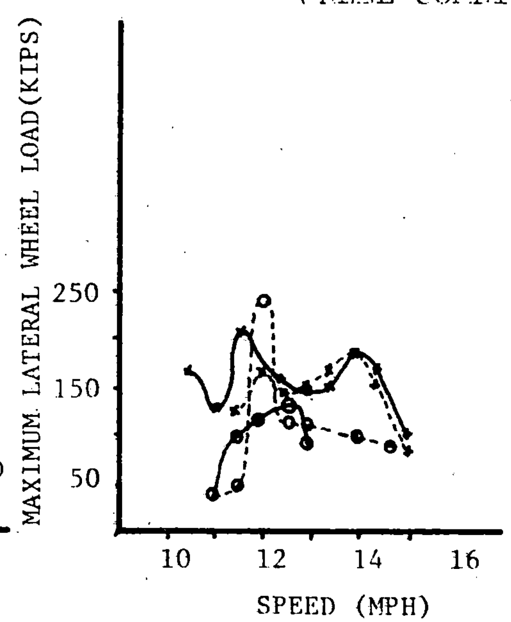


FIGURE 46

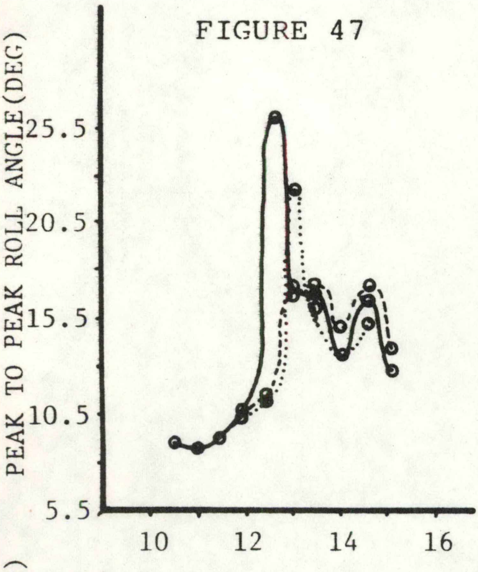


FIGURE 47

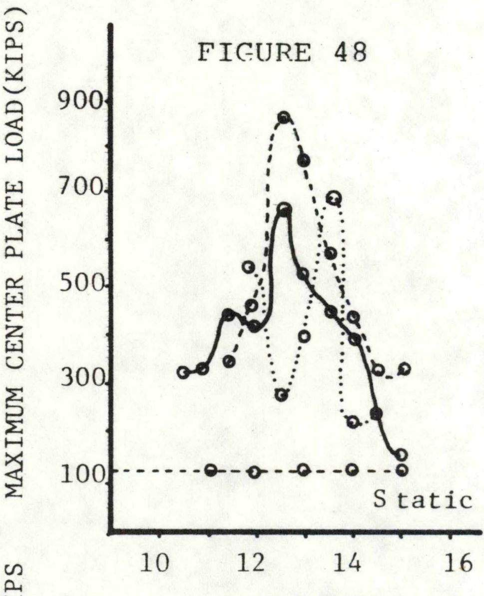


FIGURE 48

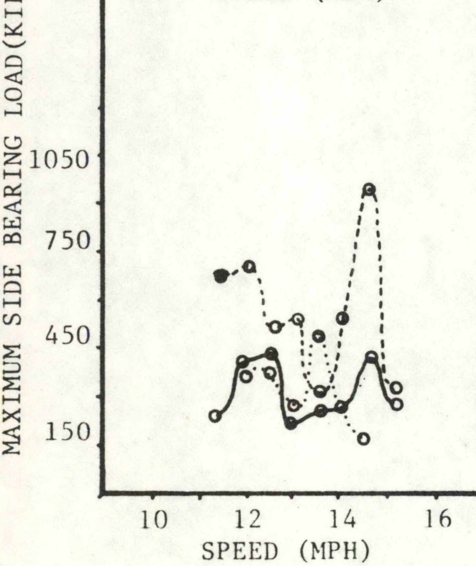


FIGURE 49

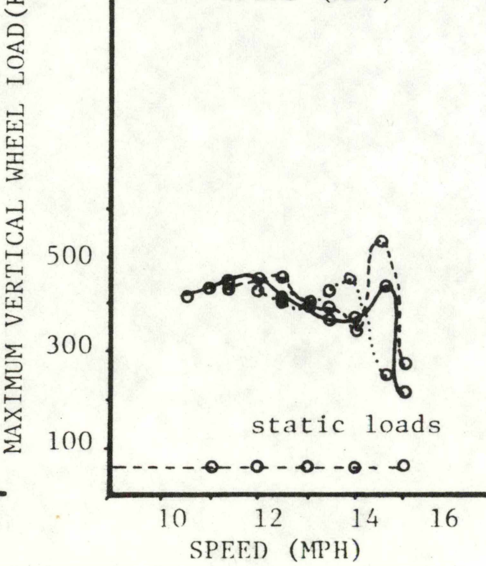


FIGURE 50

100 TON COVERED HOPPER CAR

PERTURBATION NOMENCLATURE

——o—— PROFILE

---o--- PROFILE
ALIGNMENT

...o... PROFILE
ALIGNMENT
GAGE (WIDE)

* WHEEL LOADS REPORTED ARE FOR TWO WHEELS COMBINED ON A TRUCK SIDE

loads

AMPLITUDE

PROFILE = 2¼"

ALIGNMENT = 1-3/4"

GAGE (WIDE) = 1"

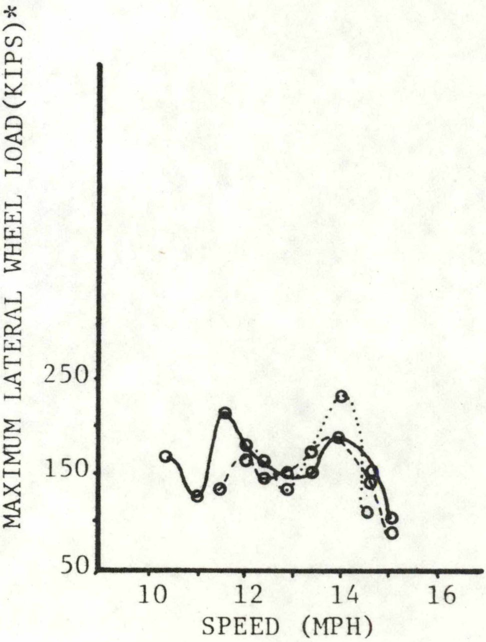


FIGURE 51

Alignment coupled with profile error increases the various loads for the 100-Ton car as compared to loads due to profile error alone. In the case of 70-Ton Car the reverse is true. For the 100-Ton Car the centerplate load increases from 675 to 860 kips and the side bearing and vertical wheel loads increase from 440 to 915 kips and 420 to 520 kips, respectively. The lateral wheel load reduces from 245 to 190 kips. For the 70-Ton Car the decrease in centerplate and vertical wheel load is from 240 to 200 kips and 245 to 237 kips. There is no change in side bearing load, but the lateral wheel load increases from 140 to 245 kips.

3.6 EFFECT OF PROFILE COUPLED WITH ALIGNMENT AND WIDE GAGE ERRORS

For this part of the parametric study the profile error of 2¼ in. is coupled with 1-3/4 in. alignment error and 1 in. wide gage error. The results obtained with profile error (Case 1, 2¼ in.), profile error coupled with alignment error (Case 2,2¼/1-3/4 in.) and profile error coupled with alignment and wide gage errors (case 3, 2¼/1-3/4/1 in.), will be discussed only for the 100-ton covered hopper car. Fig.47 shows the peak to peak carbody roll angle for cases 1, 2 and 3. The carbody roll angle for case 1 is 25.5 deg. and for case 2 it reduces to 16.5 deg. For case 3, it increases to 22 deg. As far as the roll angle is concerned, track conditions of case 2 are better than that of cases 1 and 3. Figs. 48 to 51 show the various loadings for cases 1, 2 and 3. There is no significant difference between the centerplate loads for cases 1 and 3 which are 675 and 700 kips, respectively. For case 2 the centerplate load increases to 860 kips. The largest side bearing load is experienced on class 2 track, 915 kips. For case 3 it reduces to 470 kips and further decreases to 420 kips for case 1. The same trend is observed for vertical wheel load; the corresponding values are 520, 440 and 420 kips. The lateral wheel loads are approximately 225 kips for both cases 1 and 3 and it decreases to 190 kips for case 2.

3.7 EFFECT OF PROFILE COUPLED WITH ALIGNMENT, WIDE GAGE AND WARP ERRORS

A total of 5 cases have been investigated and are discussed here. Case 1 : Profile error of 2¼ in., Case 2 : Profile error 2¼ in. coupled with alignment error 1-3/4 in., Case 3 : Profile error 2¼ in. coupled with warp error 2 in., Case 4 : Profile error 2¼ in., coupled with alignment 1-3/4 in., wide gage errors 1 in. and finally Case 5 : Profile error 2¼ in. coupled with alignment 1-3/4 in., wide gage 1 in. and warp errors 2 in. Fig. 52 shows the peak to peak roll for all the five cases. The largest roll angle is 25.5 deg. for case 1; it reduces to 22 deg. for case 4 and further reduces to 27, 16.5 and 16 deg. for cases 2, 5 and 3 respectively. Fig. 53 shows the centerplate load. The worst case is No.3 (profile/warp) where the centerplate load is 1220 kips. It reduces to 900, 860, 700 and 675 kips for cases 5, 2, 4 and 1, respectively. Figs. 54, 55 and 56 show the side bearing, vertical and lateral wheel loads for all five cases. The largest side bearing and vertical wheel load occurs in case 5 where the profile, alignment, wide gage and warp errors are coupled. Their

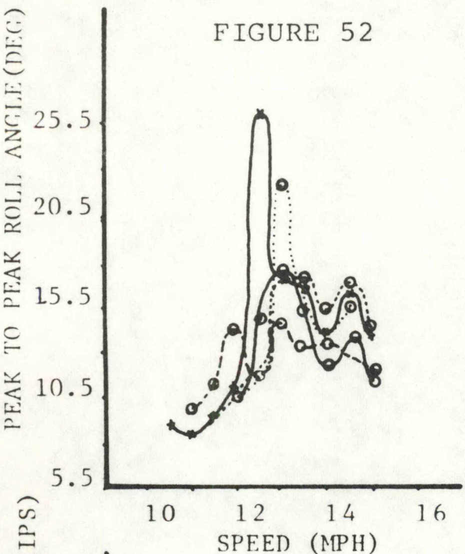


FIGURE 52

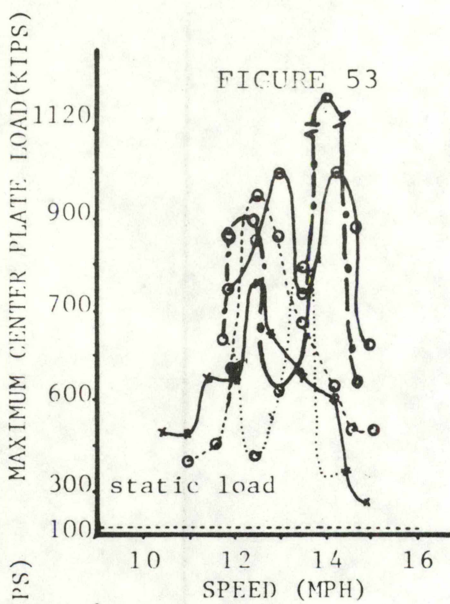


FIGURE 53

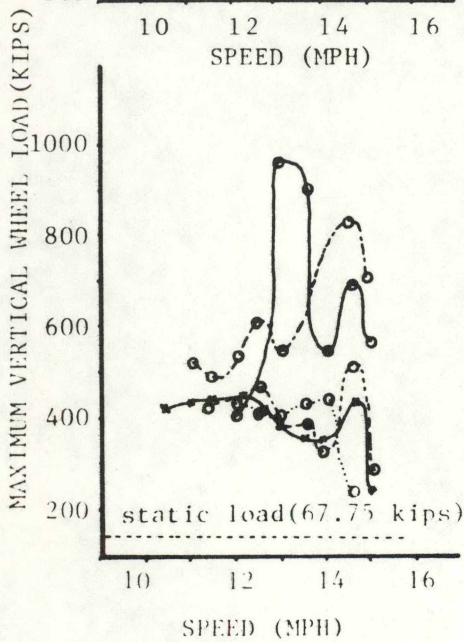


FIGURE 55

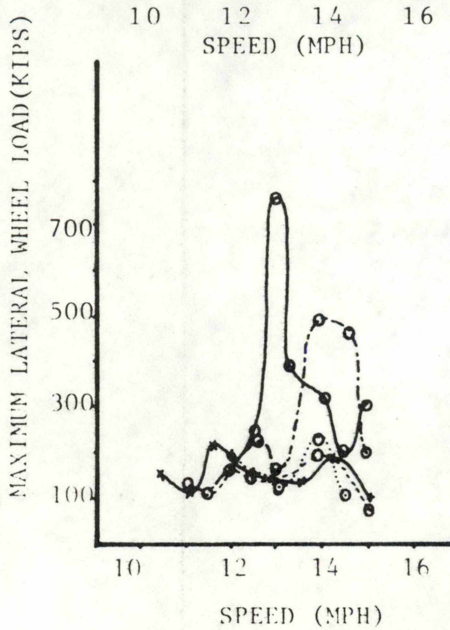
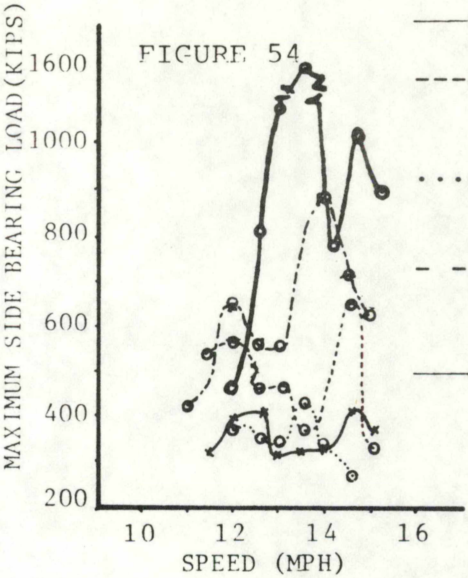


FIGURE 56

100 TON COVERED HOPPER CAR



PERTURBATION NOMENCLATURE

——x——	PROFILE
---o---	PROFILE ALIGNMENT
···o···	PROFILE ALIGNMENT GAGE (WIDE)
- -o- -	PROFILE WARP
——o——	PROFILE ALIGNMENT GAGE (WIDE) WARP

* WHEEL LOADS REPORTED ARE FOR TWO WHEELS COMBINED ON A TRUCK SIDE

AMPLITUDE

PROFILE = 2½"

ALIGNMENT = 1-3/4"

GAGE (WIDE) = 1"

WARP = 2"

magnitudes are 1560 and 955 kips, respectively. The side bearing load reduces to 1380, 915, 470 and 420 kips for cases 3 (profile/warp), 2 (profile/alignment), 4(profile/alignment/ wide gage) and 1 (profile), respectively. The vertical wheel loads follow the same pattern. The largest lateral wheel load of 760 kips occurs in case 5 followed by cases 3 (500 kips), 4(228 kips), 1(220 kips) and 2(190 kips).

4. SUMMARY AND CONCLUSIONS

The objective of this study is to evaluate the FRA allowable track irregularities on tangent track and to investigate the vehicle response due to these irregularities. The FRA Track Standards specify allowable track irregularities and operating speed limits for freight cars for each class of track (Table 1). However, the speed specified for each level of severity does not take into account some critical situations especially when the track contains a large number of consecutive irregularities (gage, alignment, surface etc) each of which may satisfy the FRA Safety Standards. The vehicle responses due to track input are obviously of a harmonic resonant nature. One example is the 'rock and roll' behavior which usually occurs below the FRA specified speed limits. It is the intent of this study to investigate the possible undesirable situations which may occur due to track inputs which are within FRA track standards. A mathematical model developed under Phase I of the AAR/TTD Program was used for this investigation. Based on the results obtained from the computer simulations, some conclusions are summarized below:

(1) Considering the effect of profile error and assuming a limit of 6 deg. peak to peak roll, the allowable track cross level difference would be 0.525 in. for the 70-Ton loaded box car (with worn wedges) and 0.6 in. for the 100-Ton hopper car (with fairly new wedges). Corresponding speed limit for these profile errors is approximately 14 mph for both the 70-Ton loaded box car and 100-Ton hopper car.

It should be noted that the cars would perform better outside the harmonic roll critical speed envelope. The critical speed envelope for the above cars with the profile errors studied is between 11.5 and 16 mph.

(2) There is a strong coupling between profile and warp errors. Centerplate, side bearing and vertical wheel load are significantly increased when track has both the profile and warp error.

(3) Presence of wide gage error in combination with profile error does not affect the response of the vehicle in any appreciable manner when the magnitudes of perturbation are small. As the amplitudes increase the load levels on centerplate, side bearing and wheel are increased significantly. It suggests that there is strong coupling between the two perturbations. Narrow gage when coupled with profile error appears to reduce the component loading.

(4) A combination of the alignment and profile errors is more severe than the profile error alone. The components considered in this study showed a consistent trend of higher loading for coupled profile and alignment errors.

(5) When profile error appears in combination with more than one type of track perturbation, it reduces the loading on a particular component as compared to profile alone or profile with another perturbation. However, this is not true for other components which may be severly loaded. This was observed for the combination of profile and warp errors as opposed to the profile, warp, alignment and wide gage errors.

The centerplate load was reduced whereas the side bearing and vertical wheel loads were increased for the same speed range.

This study has indicated that FRA specifications for defining the critical speeds should take into account the coupling of various track perturbations. The magnitude of individual track error is not adequate to define the speed range; it is the coupling of different perturbations which is more important as far as the dynamic behavior of the vehicle is concerned.

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APPENDIX

The data used in the simulation for the 70-Ton 50 ft. box car are:

The pitching moment of inertia for the carbodies	Loaded	239344.25 lb-ft-sec^2
The rolling moment of inertia for the carbodies	Loaded	53700.00 lb-ft-sec^2
The yawing moment of inertia for the carbodies	Loaded	230131.75 lb-ft-sec^2
The rolling moment of inertia for the bolsters		180.00 lb-ft-sec^2
The rolling moment of inertia for the wheel-sets/side frames combined		1363.20 lb-ft-sec^2
The mass of the carbodies	Loaded Empty	3182.15 slugs
The mass of the bolsters		36.00 slugs
The mass of the wheelsets/side frames combined		198.00 slugs
Diameter of the centerplate		1.167 ft.
Distance of the side bearing from the vertical center line of the bolsters		2.0833 ft.
Distance of the suspension group from the vertical center line of the bolsters		3.29 ft.
Rail gage		4.70 ft.
Center of gravity of wheelsets from the lateral springs		0.17 ft.
Center of gravity height of the car bodies above rail	Loaded	98.0 in.
Distance between the center of gravity of the carbodies and the center of gravity of the bolsters in the vertical direction	Loaded	6.63 ft.

Distance between the center of gravity of the carbodies and the center of gravity of the bolsters in the longitudinal direction		6.557 ft.
Gib clearances		0.03125 ft.
Flange clearances		0.0339 ft.
Rail length		39.00 ft.
Truck center distance		39.00 ft.
Wheel base		5.67 ft.
Center plate stiffness		25.44×10^6 lb/ft.
Side bearing stiffness		42.96×10^6 lb/ft.
Side bearing viscous damping coefficient		500 lb/ft/sec.
Suspension stiffness vertical	D-5 springs	243960 lb/ft.
Suspension stiffness lateral	D-5 springs	111600 lb/ft.
Track stiffness (2 wheels combined)	Vertical	3×10^6 lb/ft.
	Lateral	2×10^6 lb/ft.
Coulomb's friction coefficient at the column		2000 lb.
Torsional stiffness between the carbodies	Loaded	6×10^6 ft-lb/rad.
Bending stiffness about vertical and lateral axis between the carbodies	Loaded	240×10^6 ft-lb/rad.
Shearing stiffness of the vertical axis between the carbodies	Loaded	240×10^6 ft-lb/rad.
Length of spring travel	D-5	0.3073 ft.
Coefficient of friction at the gib		0.3
Side bearing clearance		0.02083 ft.

The data used in the simulation for the 100-Ton covered hopper are:

Pitching moment of inertia for the carbodies	750000 lb-ft-sec^2
Rolling moment of inertia for the carbodies	76000 lb-ft-sec^2
Yawing moment of inertia for the carbodies	750000 lb-ft-sec^2
Rolling moment of inertia for the bolsters	183.5 lb-ft-sec^2
The rolling moment of inertia for the wheelset/side frame combined	1534.3 lb-ft-sec^2
Mass of the carbodies	3784.63 slugs
Mass of the bolsters	42.39 slugs
Mass of the wheelsets/side frames combined	256.83 slugs
Diameter of the centerplate	14.0 in.
Side bearing spacing from car centerline	25.0 in.
Spring group spacing from car centerline	3.293 ft.
Rail gage	4.70 ft.
Center-of-gravity of wheelsets below the lateral springs at equilibrium	0.5 ft.
Distance between the center-of-gravity of the carbodies and the center-of-gravity of the bolsters in the vertical direction	6.567 ft.
Side bearing clearance	1/4"
Wheel diameter	36"
Distance between the center-of-gravity of the carbodies and the center-of-gravity of the bolsters in the longitudinal direction	9.0 ft.

Gib clearances	0.03125 ft.
Flange clearances	0.0339 ft.
Rail length	39.0 ft.
Truck center distance	45.0 ft.
Wheel base	5.8333 ft.
Centerplate stiffness	25440000 lb/ft.
Side bearing stiffness	42960000 lb/ft.
Side bearing viscous damping coefficient	500 lb/ft/sec
Suspension group stiffness	298800 lb/ft
Lateral spring stiffness	150000 lb/ft
Vertical track stiffness (two wheels combined)	2520000 lb/ft
Lateral track stiffness (two wheels combined)	2000000 lb/ft
Coulomb's friction coefficient at the column	4000 lb.
Torsional stiffness between the carbodies	$8.0x10^{7}$ ft/lb/rad
Bending stiffness about vertical axis between the carbodies	$2.4x10^{8}$ ft-lb/rad
Shearing stiffness of the vertical axis between carbodies	2.4x10 ft-lb/rad
Length of the spring travel	0.3073 ft.
Lateral side frame stiffness	1000000 lb/ft.
Vertical side frame stiffness	38520000 lb/ft.
Coefficient of friction at the gib	0.3

Track Structures: Volume IV: Effects of Track Geometry (Irregularities) on the Dynamic Behavior of a Railway Vehicle on Tangent Track - Track Train Dynamics, 1979

S. Singh, Suresh C. Mehta, SMA Hussain