track-train dynamics

Harmonic Roll Series

VOLUME 3

Freight Car Models
Design Parameter Study
Freight Car Simulation Series

An International Government—Industry Research Program on Track-Train Dynamics

AAR

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R-174

track-train dynamics Harmonic Roll Series Volume 3

An International Government—Industry Research Program on Track-Train Dynamics

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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 phenomena, 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 problem resided in the railroad supply industry and that significant technical and financial resources were available in government.

Through the Railway 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 Transportation Development Agency, 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 Recherches 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 were designated by elected members of the AAR's Operation-Transportation General Committee, supply industry members by the Railway Progress Institute, U. S. Government members by the Federal Railroad Administration, and Canadian Government members by the Transportation Development Agency. 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 comprised:

1) Phase I -- 1972-1974

   Analysis of and 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 is nearing completion with a projected two-year budget of about $4 million supported about 27% by FRA, 20% by RPI and its member companies, 10% by TDA, and 43% by AAR and its member railroads.

The major technical elements of Phase I include:

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.

The attached report presents reference information to be used in recognizing and solving "Rock and Roll" problems with certain freight cars. The information contained in this document is derived from computerized mathematical models of common railroad freight cars.

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

ACKNOWLEDGEMENT

The following reference manual on freight car dynamic modeling was developed as a joint effort of the Association of American Railroads, the Railway Progress Institute, the Federal Railroad Administration, and the Transportation Development Agency of Canada under the auspices of the International Government-Industry Research Program on Track-Train Dynamics. The document represents the completion of a segment of the program designated to provide insight into solving rock and roll problems.

With these thoughts in mind, the editor wishes to gratefully acknowledge the A. Stucki Co., who performed the box car simulation series. In particular, the editor expresses his appreciation to Mr. Donald Wiebe, Manager-Engineering of A. Stucki Co., who personally directed the simulation series and who provided so much advice and insight into the development of the Track-Train Dynamics vehicle model.

The editor also acknowledges the work directed by Dr. Gregory C. Martin of the AAR through which the Track-Train Dynamics vehicle model was developed. In particular, Mr. Yan Hai Tse should be acknowledged for his work in developing the model, and performing the parameter study under Dr. G. C. Martin's direction. Mr. S. Chawla should be acknowledged for his assistance in performing the parametric study. The model is the culmination of three years of concentrated effort by Dr. Martin and his associates and it reflects the diligence and professionalism with which they approached the task. Dr. Martin's contributions in this field will long be of great significance to the railroad industry.

The editor also recognizes Mr. Edward F. Lind, Project Director, for his contribution to the successful completion of this manual. Finally, the editor wishes to gratefully acknowledge the Louisville and Nashville Railroad for allowing the contribution of his time to the project.

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Approved:
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FOREWORD

In recent years, service demands on the railroad industry have undergone significant changes. As a result, a number of important design innovations have been introduced to the freight car fleet. The resultant has been the construction of a large number of heavy, high volume freight cars. As the utilization of this type of equipment has increased, so have the operating problems. One important problem is that of harmonic roll, most commonly referred to as "Rock and Roll."

There are several synonymous terms for harmonic roll that may be used throughout the text. They are:

Harmonic Roll - Rock and Roll
Car Rocking
Lateral Instability

In general, the rock and roll problem is related to the operation of high capacity, high center of gravity freight cars over track that has an uneven surface. The problem is most predominant on track that has surface variation due to alternately staggered joints. When operating over rough track of this nature at speeds usually between 15-25 mph, excessive carbody roll may be developed. Energy is added to the moving system with each roll cycle, and, if the car suspension does not have adequate damping, extreme carbody roll will develop, resulting in wheel lift and probable derailment.

A general solution to the rock and roll problem involves an extensive study of the entire system consisting of track, suspension, and carbody design. A study of this scope results in a better understanding of the mechanics and dynamic interactions of a freight car in its operating environment. The result can be a freight car that not only meets marketing requirements, but one that can be operated safely with minimal problems.

In order to begin investigation of track-train dynamics problems, the Southern Pacific under contract to the AAR sent a questionnaire to sixteen selected railroads in mid-1971. From the responses received, it was realized that two serious problems faced the railroads. These problems were

(a) Rock and Roll
(b) Sudden gage widening and rail roll-over

Keeping this in view and realizing work was needed in these areas, the planners of Phase I of the track-train dynamics research program designated Task 13, Special Projects, to handle these areas. With respect to rock and roll, the primary objectives of Task 13 were to:

(1) Develop guidelines which could be used by individual railroads to assist them in minimizing rock and roll problems
(2) Develop a document to be used as a reference in solving rock and roll problems
(3) Develop a model for computer simulation of freight car dynamics
(4) Develop characteristics of common freight car trucks and their related components
(5) Develop a log of freight car characteristics critical in designing stable freight cars
(6) Develop comparisons of important parameters controlling rock and roll by using existing computer simulations.

By using the information developed from the primary objectives, certain secondary goals could be accomplished during Phase II of the overall research program. These goals will be to:

(1) Develop car design specifications
(2) Develop specifications for truck design
(3) Aid in developing laboratory procedures for evaluation of damping devices
(4) Define the function of the freight car truck within the dynamic system as related to transfer of energy from track to carbody
(5) Perform simulations to evaluate new car designs

To present the findings related to the primary objectives of Task 13 outlined above, it was decided to present a series of harmonic roll related documents to the industry to be used as reference material in a similar manner as the Track-Train Dynamics Bibliography. This document, Volume III of the Harmonic Roll Series, presents technical information about computerized mathematical models used to simulate the dynamic response of freight cars to various operating conditions. Particular emphasis is placed on studying changes in certain basic freight car design parameters and how these changes affect the dynamic response. It is hoped that this information will be used as guidelines by the AAR Mechanical Division for establishing car design specifications and setting maintenance standards that insure the dynamic stability of freight cars.

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INTRODUCTION

The rock and roll problem is now well known in the railroad industry. Basically, it is a resonance phenomenon that occurs when a railroad freight car traverses track constructed with half-staggered rails or track that contains low joints or a series of rough depressions. This problem has become more severe with the advent of high capacity, large volume cars in recent years.

The manner in which a freight car establishes a resonant condition is illustrated in Fig. 1.

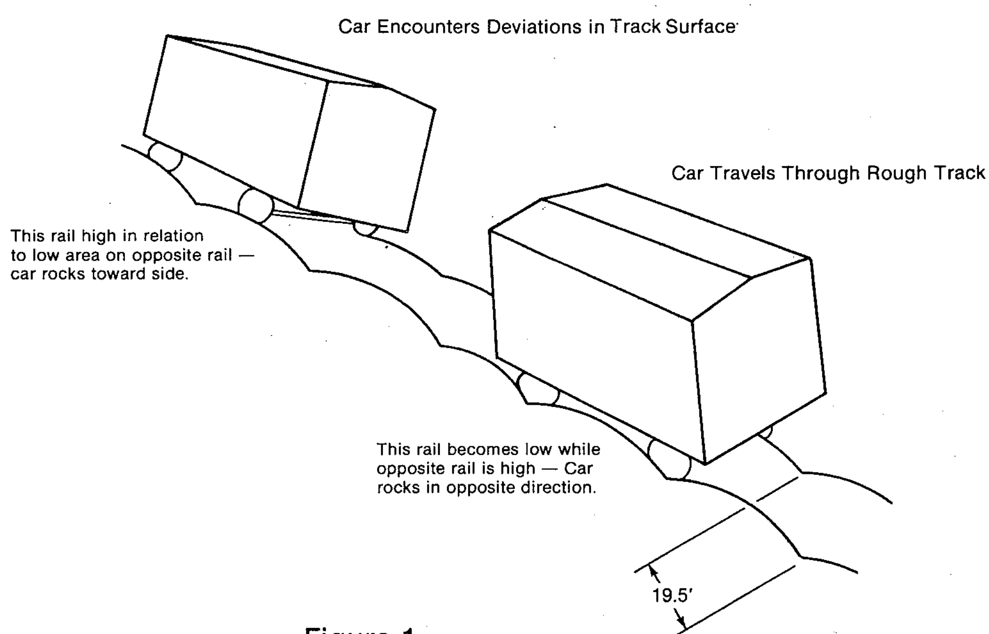


Figure 1
Freight Car Rocking
(70 TON CAR)

As the rail car encounters alternately low and high areas in rail profile, it begins to rock from side to side. Note that the relation between the truck spacing and half-staggered rail joints is emphasized. If there are a number of low areas in succession, the rocking amplitude will increase. If the rail car encounters enough of these depressions while moving at "critical speed" a resonant condition will prevail as it would with any other vibrating or oscillating structure operating at critical frequency. In this same sense, the term critical speed is that train speed at which the freight car system is excited at its critical frequency.

Once a rail car has reached its resonant rocking frequency, the rocking amplitude will increase until spring bottoming and centerplate separation occurs. In the end, the wheels lift from the rail and a derailment may occur.

Volume I of the Harmonic Roll Series contains a discussion of the various track and freight car design parameters and how they affect car rocking.

Various attempts have been made to reduce the severity of the rock and roll problem. These attempts have included several modifications to improve the track structure. Also, many attempts have been made to alter car design to improve the dynamics of the rail car. Some attempts have been successful while others have not. For example, some excellent motion dampers or snubbers have been developed to absorb the rocking energy once it begins to develop. These methods are not foolproof, however, since the basic physical dimensions of a sensitive car remain unchanged. Thus most solutions so far have served to minimize the rock and roll problem but not eliminate it.

The purpose of the work described in this manual was to give car designers a detailed look into the resultant effects of changing key freight car design parameters. A second objective was to model a basic freight car whose characteristics could be readily associated to other car designs. In this manner, the information herein could be used to establish primary dynamic design criteria for freight cars. The means for developing this type of data was through mathematical modeling of the freight car structure. In turn the mathematical models were solved by digital and analog-digital computers for comparison of track conditions, design parameters, speeds, etc. The following sections describe these models, the case studies investigated, and the results from the case studies.

FLEXIBLE CAR BODY MODEL

DESIGN PARAMETER STUDY

In early 1972 under the direction of Dr. Gregory C. Martin of the Association of American Railroads, a research effort was initiated in the area of computer simulations. One of the goals of this effort was to develop a mathematical model for solving the dynamic equations of motion of a conventional freight car by digital computer. The model and computer program resulting from that study was utilized to conduct the parameter investigation described in subsequent pages of this manual. The details of the model and computer program are described in the documentation of Track-Train Dynamics Program Task 7, Mathematical Modeling. What follows is a general description of the way the model was programmed for the design parameter study.

In all, eight case studies were investigated. A typical 70 ton box car was run throughout all cases except in case 7, in which a heavier car, a 100 ton covered hopper was run to produce the truck spring bottoming effect. The input computer data are described in Appendix I.

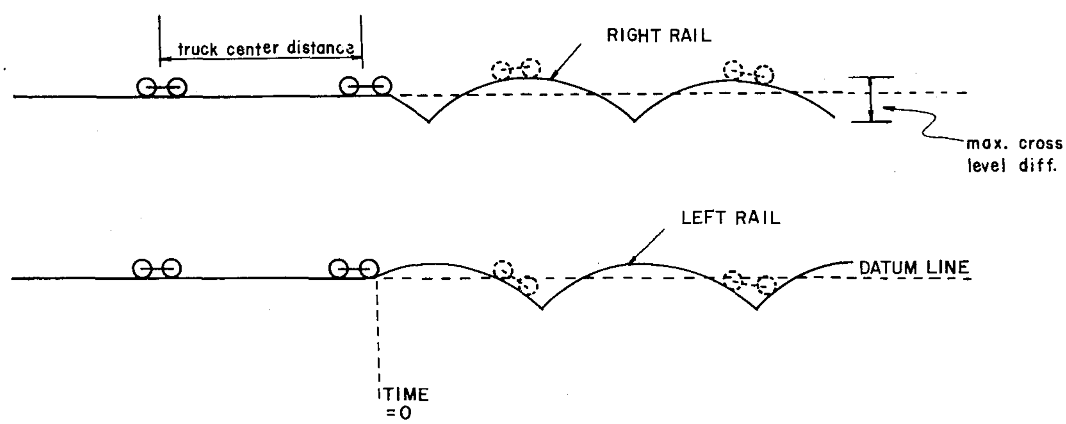


Figure 2
Input Functions of Half-Staggered Rails with Soft Joints

(70 TON CAR)

The forcing input function to the model was from the surface variations of the track. Initially, the car rests with all eight wheels on level track at static equilibrium (Fig. 2). As the car is started, the wheels of the front truck follow the traces in the figure moving up and down while the wheels of the rear truck remain on the flat track. As the car moves, the rear wheels follow the front wheels separated by the truck center distance. Since the rails are half-staggered, there is a phase lag between the input function of the right rail and that of the left. For all runs, the car is assumed to start instantaneously at the given speed and maintain that speed throughout the entire run.

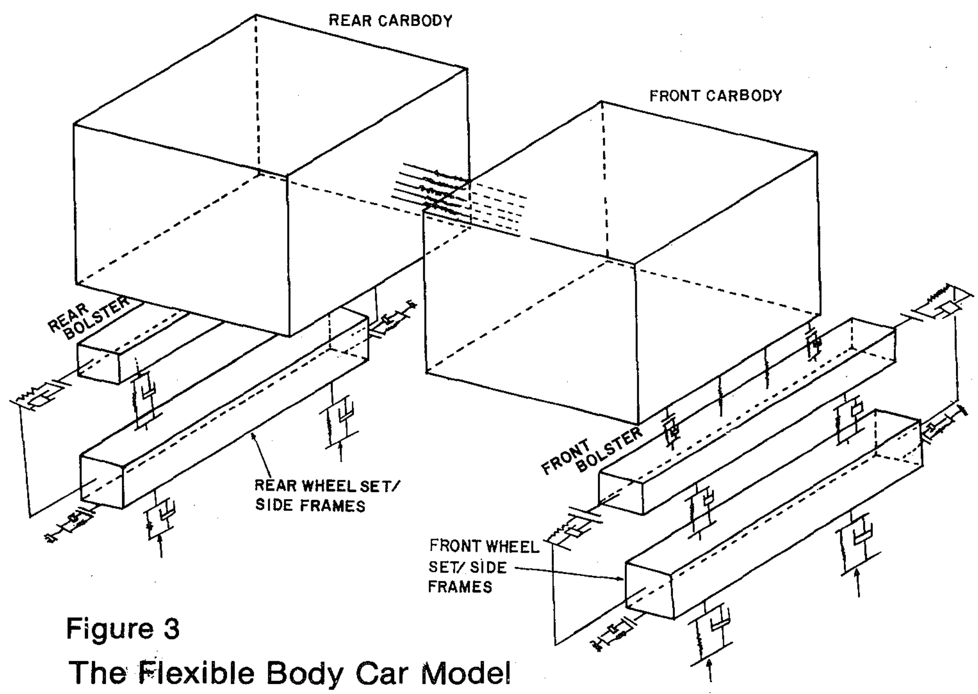


Figure 3
The Flexible Body Car Model
Design Parameter Study
(70 TON CAR)

There are a total of twenty two degrees of freedom described in the model. Fig. 3 shows the springs and the dashpots (or coulomb dampers) connecting the masses of the model with various clearances indicated.

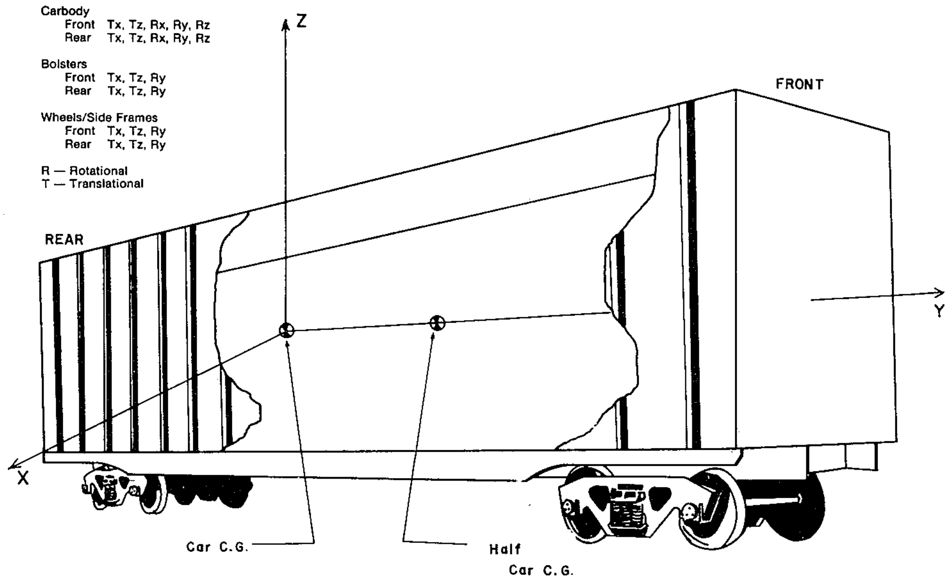


Figure 4
Coordinates and Degrees of
Freedom of the Freight Car Model
(70 TON CAR)

Fig. 4 shows the degrees of freedom assumed for each mass. The reader should note that this model assumes a flexible carbody. The carbody is, therefore, cut into two rigid bodies, connected by a torsional spring and other flexural springs. The detailed equations of motion are not given in this manual, but they are available for reference in the technical documentation of this model associated with Task 7, Mathematical Modeling of the Track-Train Dynamics Program.

DESIGN PARAMETER INVESTIGATION

DESCRIPTION OF CASE STUDIES

As mentioned previously, eight case studies were conducted to determine the effects of making changes in certain key freight car design parameters. These studies were made according to the following descriptions.

Case 1. Determine the effect of center of gravity:

Track Conditions: 39' rail, 3/4" maximum cross level difference.
Car Conditions: 39½' truck center distance
Center of gravity 5', 6.5', 8.5', 10.5' and 11.5' above the top of the rail. (ATR)

Case 2. Determine the effect of truck center spacing:

Track Condition 1: 39' rail, 3/4" max. cross level difference.
Car Conditions: 98" center of gravity height ATR
Track Condition 2: 33' rail, 3/4" max. cross level difference.
Car Conditions: 98" center of gravity height ATR
Vary truck center distance from 28' to 49' in 3' increments.

Case 3. Determine the effect of speed:

Track Conditions: 39' rail, 3/4" max. cross level difference.
Car Conditions: 98" center of gravity height ATR, 39½' truck center distance.
Vary speed from 10 mph to 30 mph in 2 mph increments.

Case 4. Determine the effect of cross level variations in track surface:

Track Conditions: 39' rail
Car Conditions: 98" center of gravity height ATR 39½' truck center distance.
Vary maximum cross level difference from 1/8" to 1" in 1/8" increments.

Case 5. Determine the effect of carbody stiffness:

Track Conditions: 39' rail, 3/4" max. cross level difference.
Car Conditions: 98" center of gravity height 39½' truck center distance.

Runs based on values taken from stiffness measurements of actual freight cars.

Case 6. Determine the effect of side bearing clearance:

Track Conditions: 39' rail, 3/4" max. cross level difference.
Car Conditions: 98" center of gravity height ATR, 39½'truck center distance.

Vary side bearing clearance from 1/8" to 1/2" in 1/8" increments.

Case 7. Determine the effect of spring travel:

Track Conditions: 39' rail, 3/4" max. cross level difference.
Car Conditions: 98" center of gravity height ATR, 39½'truck center distance.

Vary springing with AAR D-3, D-5, and proposed D-7 springs.

Case 8. Determine the effect of centerplate diameter:

Track Conditions: 39' rail, 3/4" max. cross level difference.
Car Conditions: 98" center of gravity height ATR, 39½'truck center distance.

Vary diameter from 14" to 16" in 1" increments.

In all cases, all parameters except the one under investigation were held constant in order to isolate the individual effects. Again, all other constants may be found in Appendix I.

The rectified fine curve surface profile used in the case study comparisons represents one particular type of track surface variation between joints. Other common profile shapes will occur in practice that can result in more severe vehicle response for the same cross level difference changes. The Stucki model case studies presented later employ a smoother transition at the rail joints and as a result, fewer 3/4" rail joints variations are required to achieve maximum response.

DESIGN PARAMETER INVESTIGATION

DISCUSSION OF RESULTS

Case 1 - Effect of the Carbody Center of Gravity Height

As Fig. 5 shows, the carbody center of gravity height is a significant factor in determining the critical speed of a freight car. Within a range of 60" to 144" center of gravity height above the rail, the critical speed of almost all the operating freight cars falls within the range of 10.0 to 22.0 mph. Also, as shown in the figure, the center of gravity, just like the length of a pendulum, is the determinant of the natural frequency of the freight car dynamic system.

The center of gravity height, however, does not affect the various loadings and maximum carbody roll angles significantly, as shown by Table 1. The maximum loadings at different center of gravity heights differ by at most ten percent. In spite of this, spring bottoming is found to occur at the heights of 11.5 and 10.5 ft., but not 8.5, 6.5 or 5.0 ft. For the two higher cars, the durations of wheel-lift around the critical speed is longer than the other three (Fig. 6). This may be explained by the slower period of the motion or vibration due to the higher center of gravity. Thus, the moment of the vertical restoring force is similar since the rocking inertia is greater and the lateral overturning force induced by this acceleration occurs over a longer time period. The longer the zero wheel load duration, the greater the probability of derailment.

In general, the critical speed decreases with an increase in center of gravity. This helps explain why some extremely high cars such as wood chip cars or extremely high open-top loads such as transformers, boilers, etc. mysteriously derail at speeds below 15 mph. Also minimum wheel loads tend to decrease as center of gravity increases. Reduction in wheel loads reduces the ability of a car to negotiate changes in track curvature. Slight rocking of a high car or high load could result in low wheel loads, a higher L/V ratio, and a tendency to split switches in low speed yards for example. There are many instances where a yard derailment occurs, the track foreman finds the frog and switchpoints in good condition and the mechanical foreman finds all wheels within wear tolerances. Theoretically, the derailment should not have occurred. In these cases, one or two rough spots in close proximity to the switch could have induced rocking and resultant minimum wheel loads, thus causing the car to split the switch.

A good criterion for car stability would be limitation of minimum wheel load. For example, wheel load would not be allowed to get below 25% of the static wheel load. Further investigation into this type of specification could probably produce the most cost saving benefits.

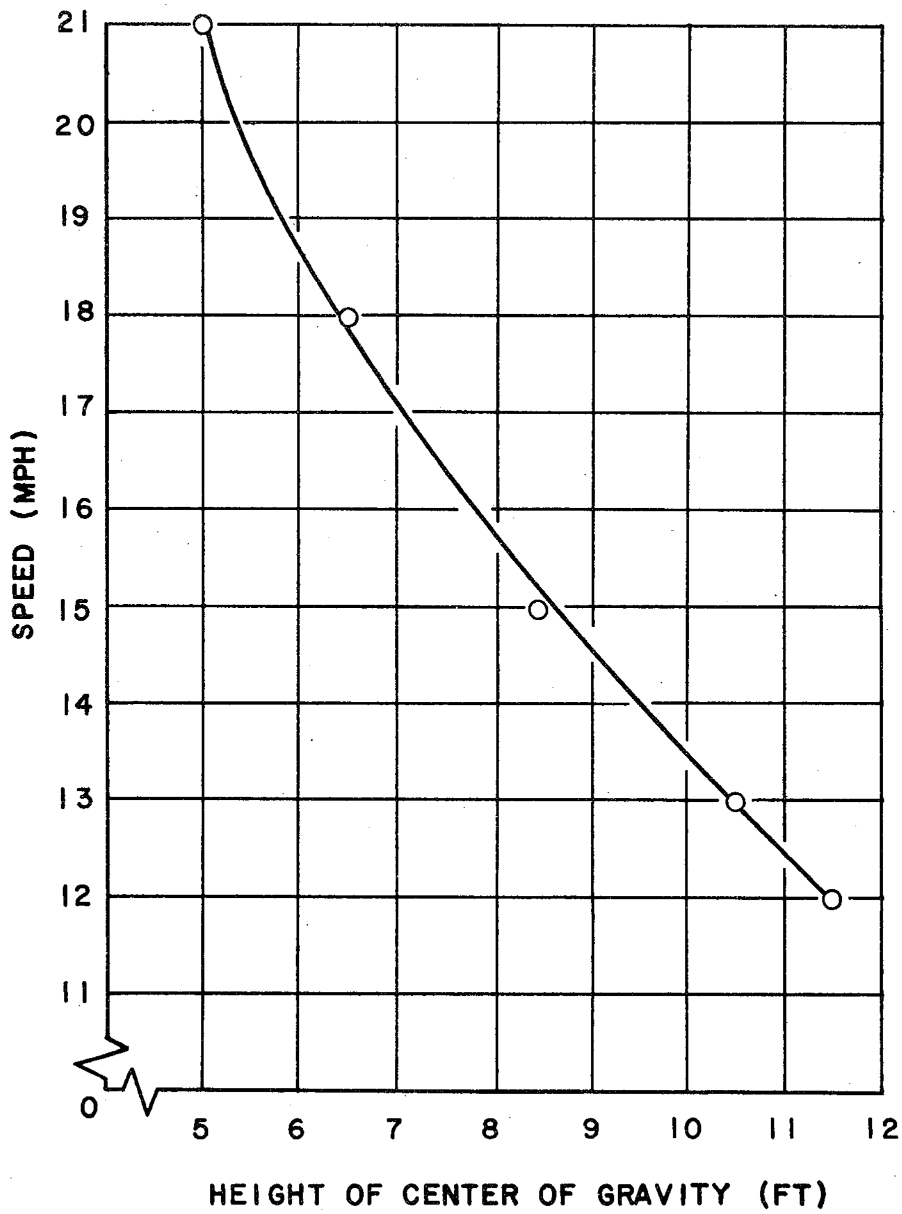


Figure 5
Effects of the Carbody Center-of-Gravity
Height on the Critical Speed
(70 TON CAR)

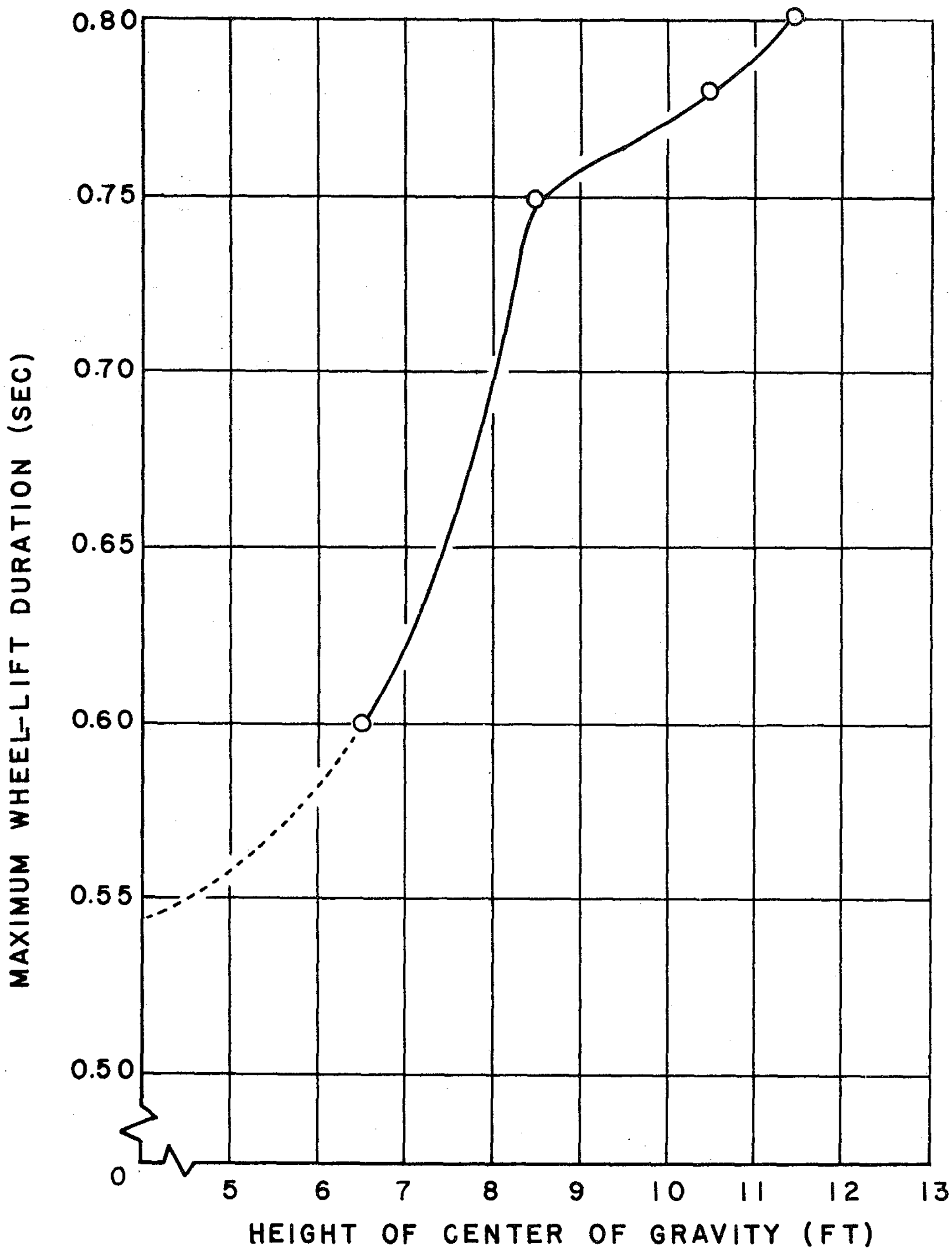


Figure 6
Effects of the Carbody Center-of-Gravity Height on the Wheel-lift Duration
(70 TON CAR)

Table 1. Effect of the Center-of-Gravity height on Various Loadings and Carbody Roll Angle

C.G. Height	Speed	Maximum center-plate loading	Maximum side bearing loading	Maximum wheel loading (two wheels)	Maximum roll angle (peak-to-peak)
5.0 ft	19 mph	90900#	51500#	76100#	4.8°
	20	92000	70100	91200	5.6
	21	118700	83100	97000	9.4*
	22	113500	81200	96700	8.0*
6.5	16	90000	42600	73600	5.2
	18	108000	85000	98000	11.4*
	20	96800	84600	93700	6.0*
8.5	12	87000	0	60000	2.7
	14	90000	86000	91000	7.0
	15	105000	90000	100000	11.4*
	16	103000	87000	95000	8.0*
	18	88000	42600	69500	3.0
	20	88000	20000	60000	1.6
10.5	12	102185	106000	110000	9.2*
	13	100000	99000	101000	10.2*
	14	96000	90000	95000	6.5*
	15	89000	46000	74000	3.2
11.5	10	89250	44000	73000	5.2
	12	95000	97500	101000	10.2*
	13	94000	91700	97000	6.1*
	14	86000	52000	74400	3.2

*Wheel-lift occurs at this speed

Case 2 - Effect of Truck Center Distance

The car was simulated at various speeds while the truck center distance was varied from 28 to 49 ft. in 3 ft. increments on 39 ft. jointed rail and then 33 ft. jointed rail. The critical speed was found to be 15 to 16 mph for the case of 39 ft. rail and 13 mph for the 33 ft. rail. Fig. 7 shows the maximum peak-to-peak roll angles at these critical speeds for different truck center distances. As can be seen in the figure, cars with their truck center distances which are close to the rail length give much higher roll angles. For an example, on 33 ft. rail length track, cars with truck center distances from 28 ft. to 39 ft. roll with a maximum peak-to-peak angle of about 12.0 degrees while a car with 49 ft. truck center distance rolls with a maximum of as low as half a degree in this particular rail profile shape.

Fig. 8 shows a maximum wheel-lift duration at the truck center distance close to or the same as the rail length as would be expected. On a 33 ft. rail length track, cars with truck center distances above 43 ft. will not experience wheel lift even at critical speed.

Figures 9, 10, and 11 show the effects of the truck center distance on various maximum loadings. Truck center distances away from the rail length will result in lower loadings.

In sum, it appears that as the forcing function is a combination of the deflections of the rail on the eight wheels, truck center distances coinciding with the rail length lead to a greater forcing amplitude and thus give a greater roll angle. The frequency of this forcing function also depends on these truck center distances, making the critical speed shift between 15.0 and 16.0 mph for the case of 39 ft. rail.

It is interesting to note the maximum roll angles and loadings for a car traveling on 39 ft. rail. Rule 88 of the Office Manual of AAR Interchange Rules sets a critical truck spacing range from 28' to 45' for cars with 6 1/2 x 12 or larger journals. This rule should be reviewed because the simulated car (70 ton, 6 x 11 axles) with a truck spacing of 45' and above had a roll angle and maximum forces still within a significant range. This explains why some cars with 46' or greater truck spacing have experienced slight wheel lifts on actual test tracks.

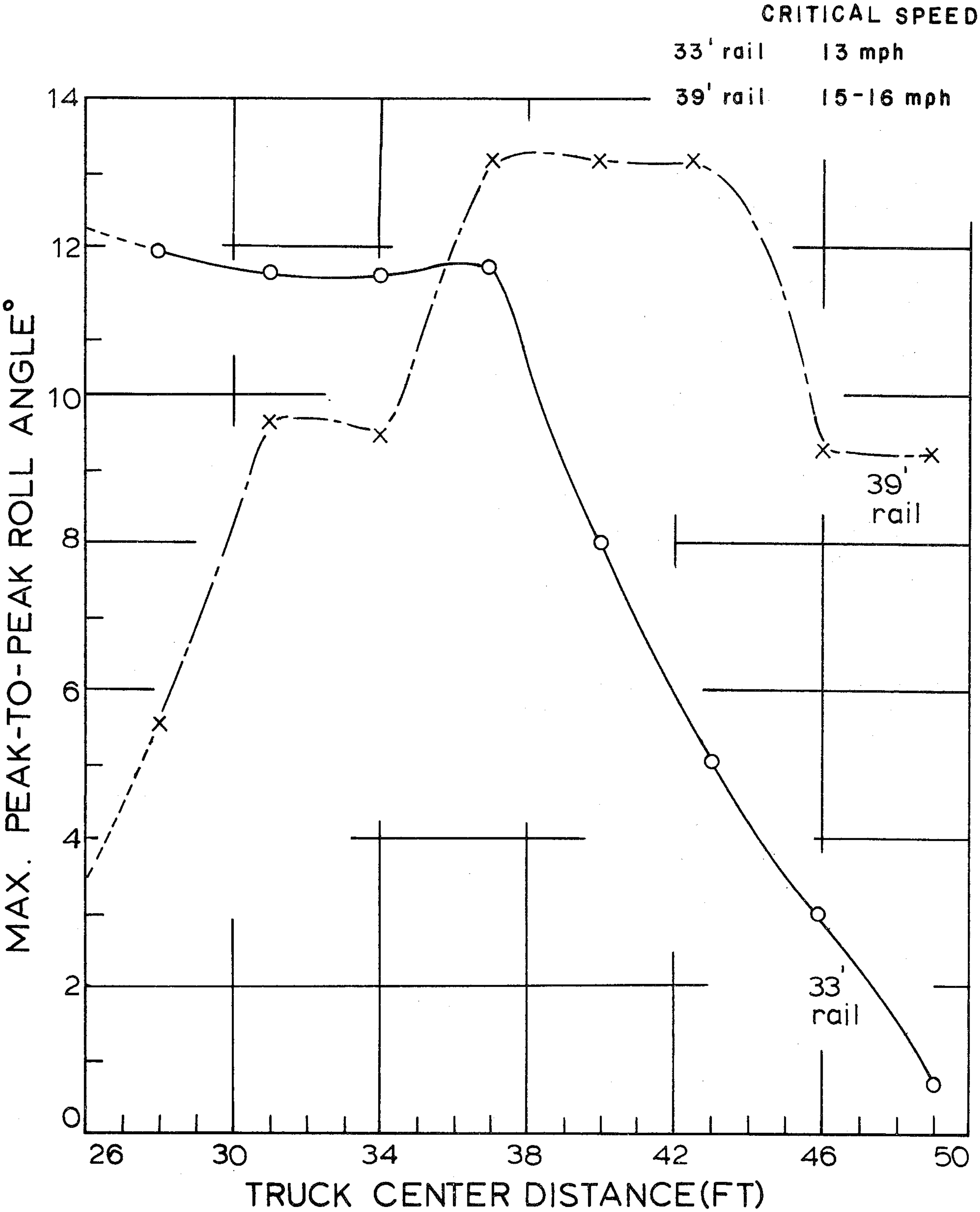


Figure 7
Effects of Truck Center Distance on the Maximum Peak-to-Peak Roll Angle
(70 TON CAR)

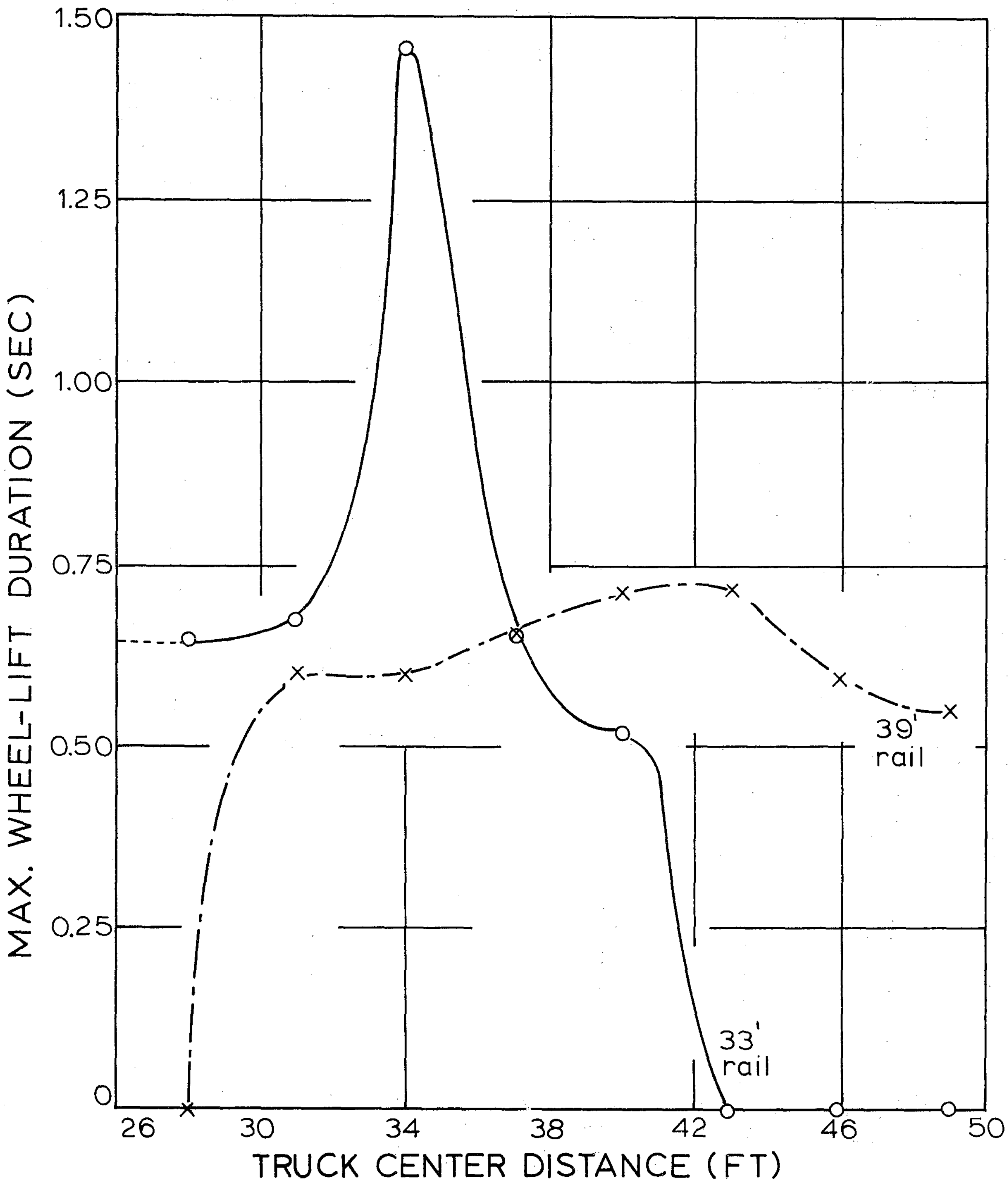


Figure 8
Effects of Truck Center Distance on Wheel-lift Duration.
(70 TON CAR)

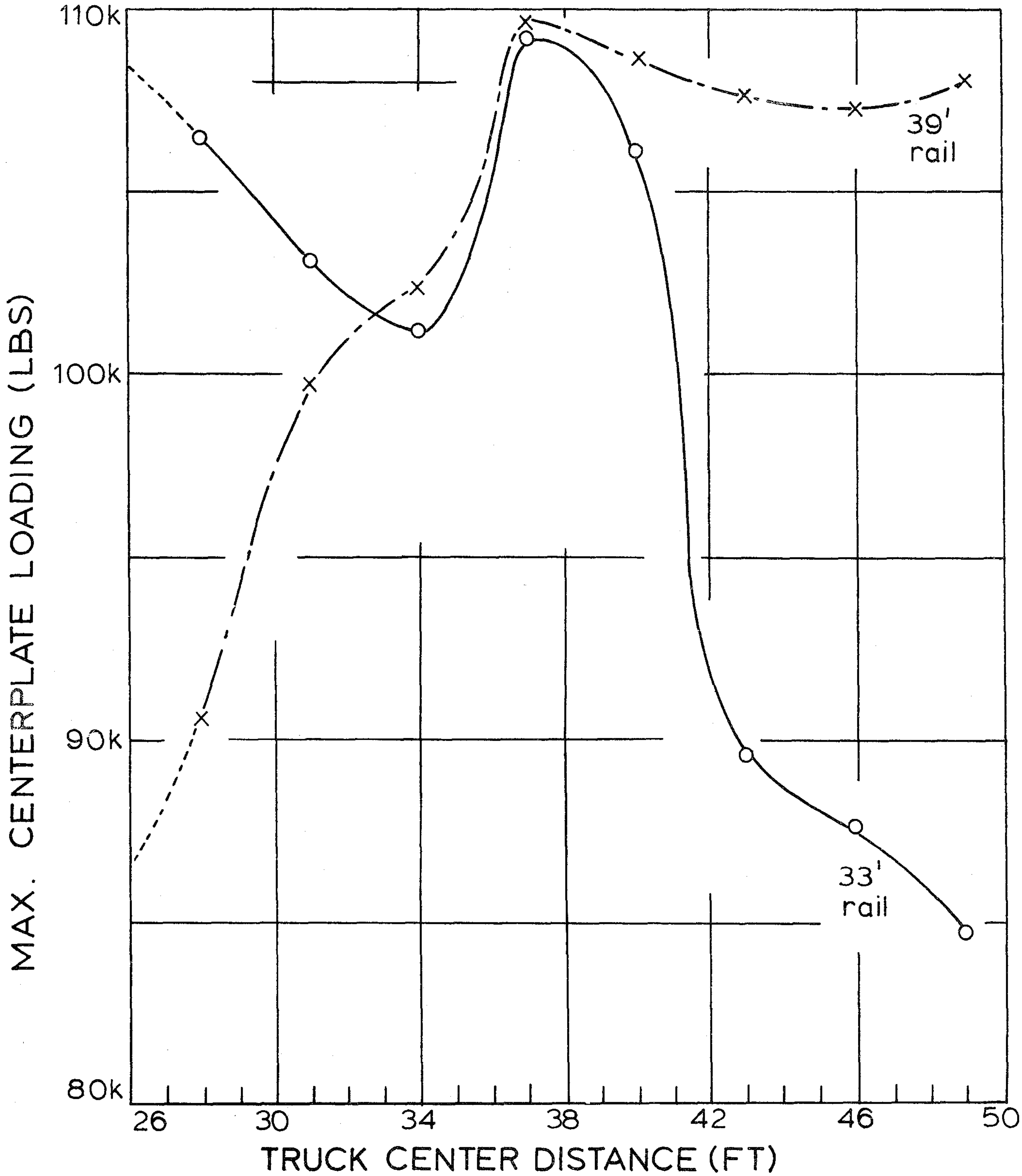


Figure 9
Effects of Truck Center Distance on Maximum Centerplate Loading
(70 TON CAR)

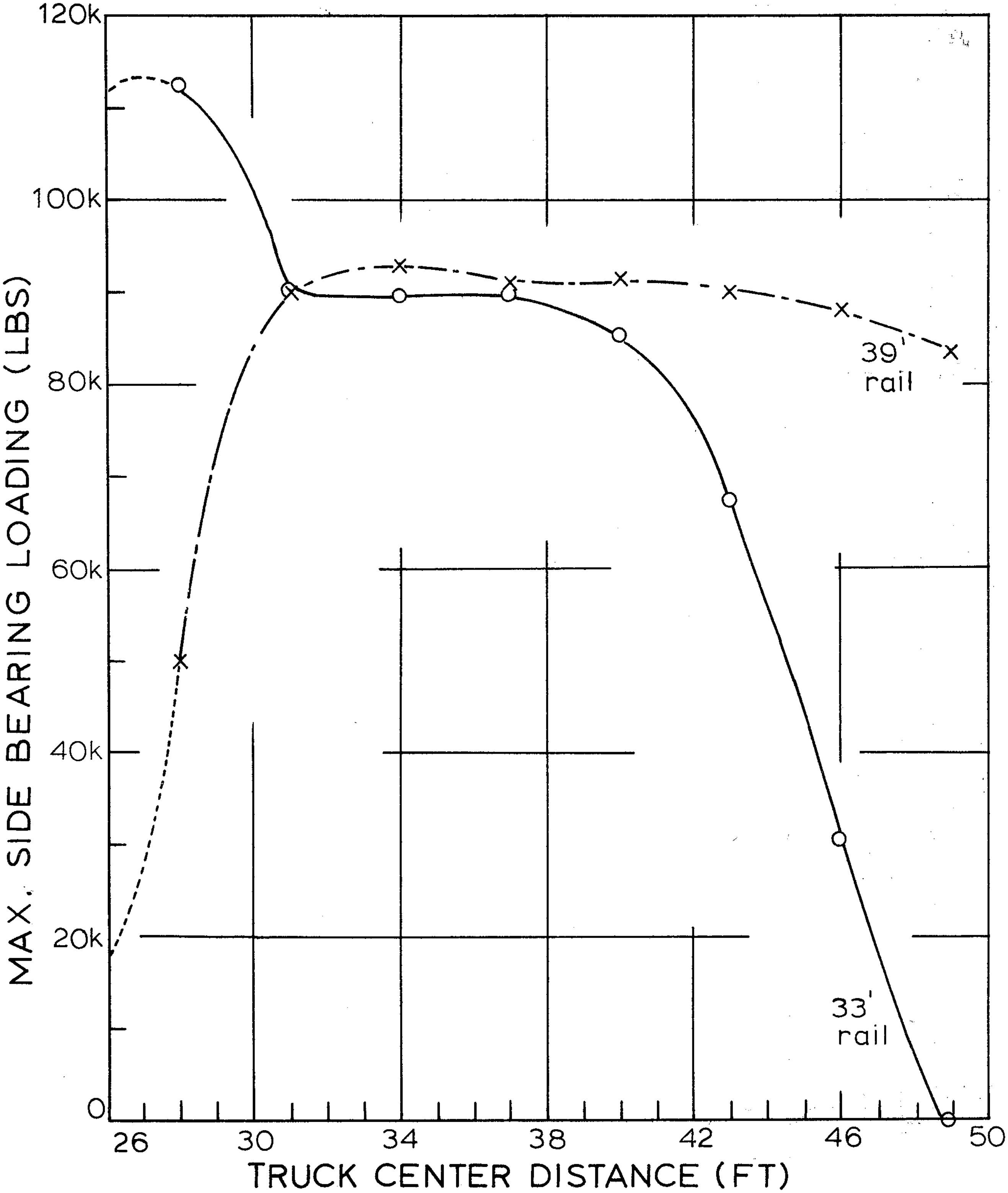


Figure 10
Effects of Truck Center Distance on Maximum Side Bearing Loading
(70 TON CAR)

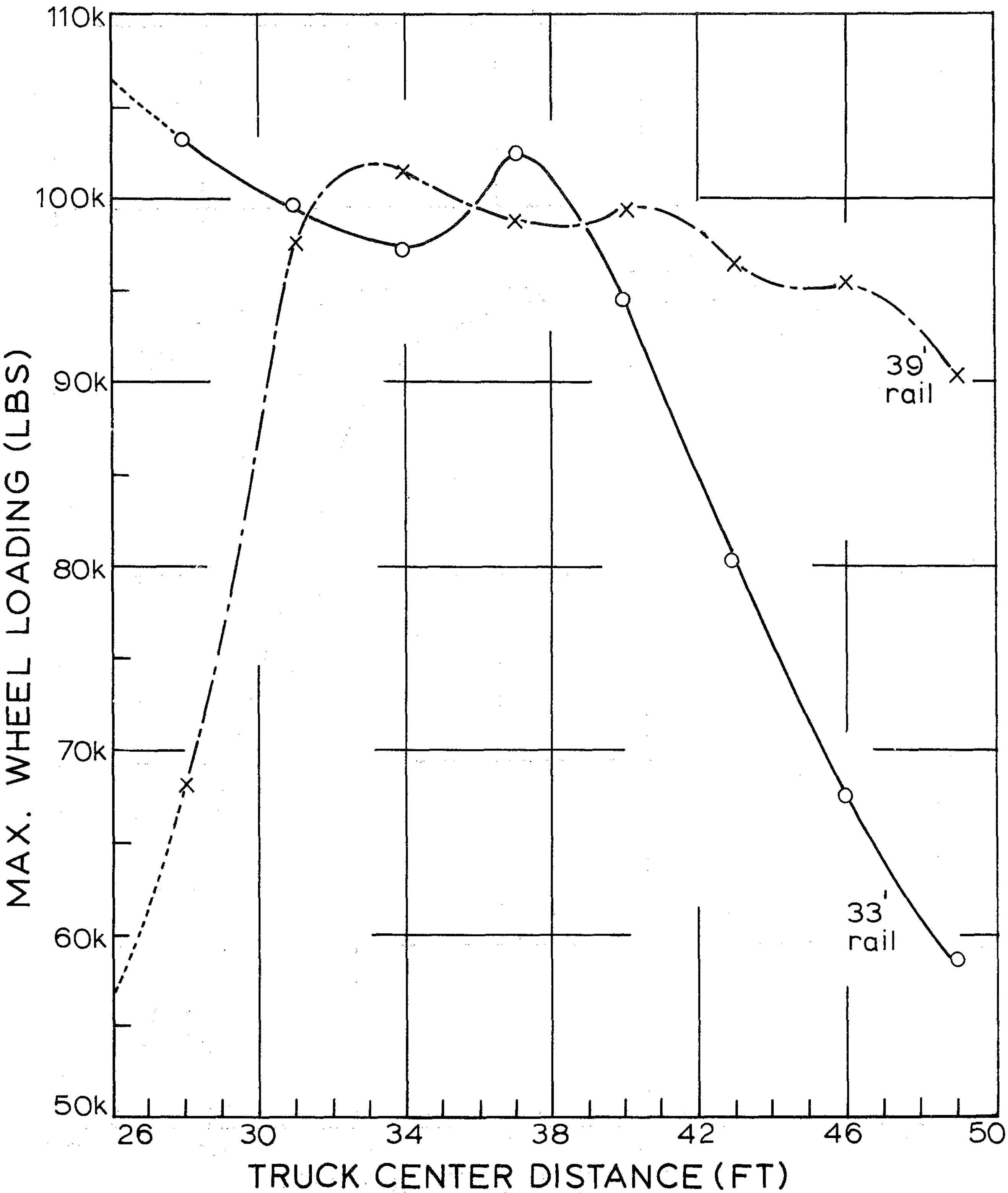


Figure 11
Effects of Truck Center Distance on Maximum Wheel Loading
(70 TON CAR)

Case 3 - Effect of Speed

The car was simulated at speeds between 10.0 to 30.0 mph. The maximum peak-to-peak roll angle was found at 15.0 mph (Fig. 12). At high speeds, the roll angle is less than one degree. Figures 12, 13, 14, and 15 show the highest loadings at the speed of 15.0 mph. Note that the maximum roll angles correspond to maximum loadings. At high speeds, as shown in Fig. 16, the maximum wheel loading is about the same as the static loading, and is about half of the loading at 15.0 mph.

Fig. 12 is included to show that the maximum roll angle is associated with the maximum lateral acceleration. The peak of the lateral acceleration is at 15.0 mph, the critical speed. As a matter of fact, the equations of motion reveal that the lateral motion is coupled with the carbody roll dynamically.

Table 2 indicates wheel-lift occurs at the critical speed. At this speed, the system is actually at resonance with the input function, the frequency of which is determined by how fast the freight car moves on the half-staggered rails. Theoretically, the resonance occurs when this frequency coincides with the system natural frequency. Due to some damping present, the critical speed may shift slightly away from the theoretical resonant speed.

These speed tables indicate the importance of moving trains rapidly through the critical speed range. They also emphasize setting slow orders at 10 mph if 25 mph cannot be maintained.

Although it was not within the scope of this study, the model and program can be used to study freight car bouncing at higher speeds.

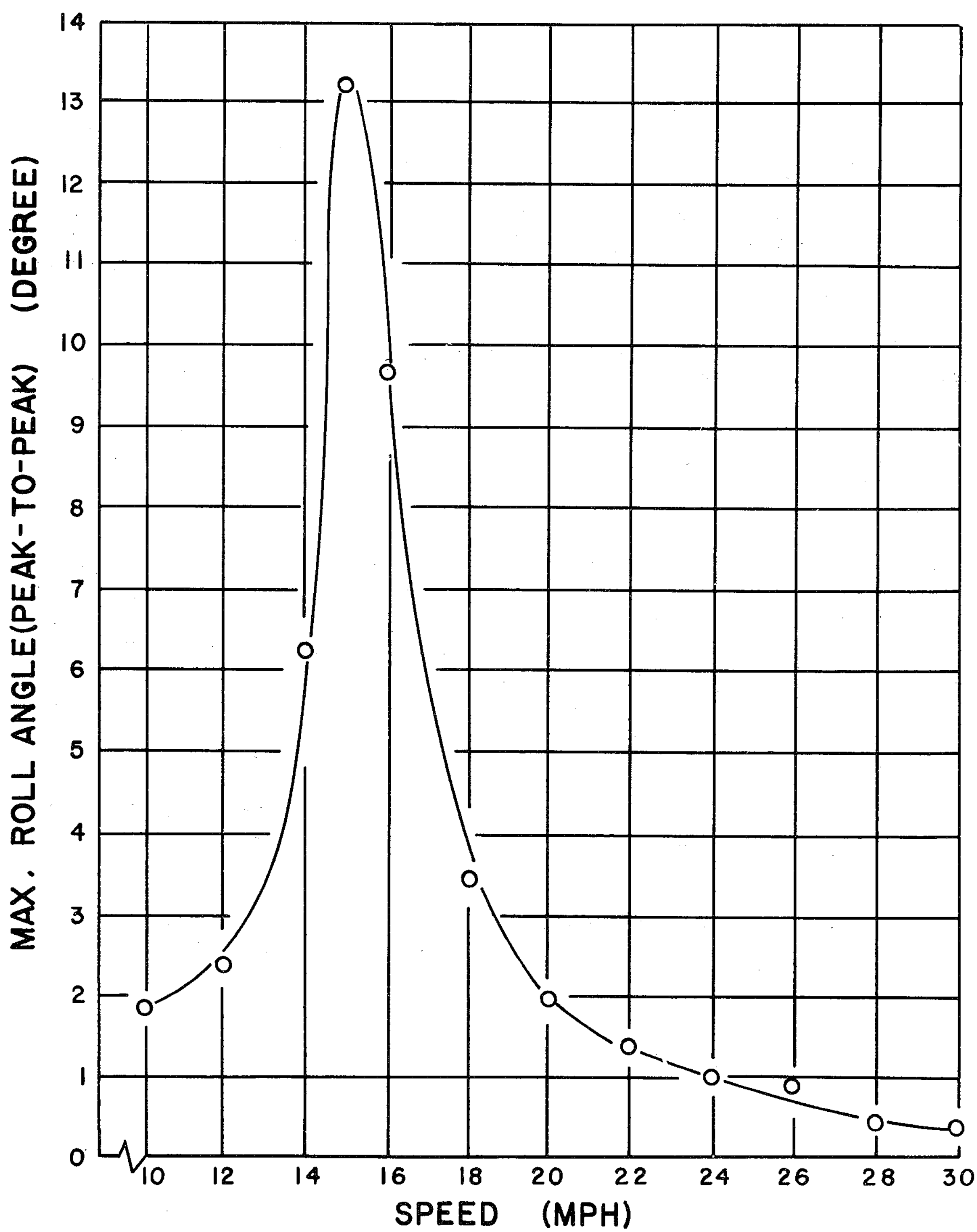


Figure 12
Effects of Speed on the Carbody
Peak-to-Peak Maximum Roll Angle

(70 TON CAR)

Table 2. Effect of Speed on Wheel-lift

Speed	Wheel-lift occurrence	Duration
10 mph	-	-
12	-	-
14	-	-
15	yes	0.75 sec
16	yes	0.64
18	-	-
20	-	-
22	-	-
24	-	-
26	-	-
28	-	-
30	-	-

Case 4 - Effect of Maximum Cross Level Difference

To increase the maximum cross level difference is to increase the amplitude of the input. Correspondingly, a higher amplitude will be produced for the outputs too. Figures 17-20 show this effect on the roll angle and various loadings. Below 1/2" maximum cross level difference, no excessive roll is induced as shown in Fig. 17, and as a result, no wheel-lift occurs (Fig. 21). As this cross level difference is increased, the maximum roll angle (peak-to-peak) and the various maximum loadings will, in turn, increase.

It was also found that as maximum cross level difference increases, excessive roll is produced more rapidly. By the same token, less cycles are required to build up the maximum roll angle. This relationship is shown in Table 3. For the two smallest values, practically no carbody roll is seen to build up. That means the carbody roll stayed minimal throughout the time of simulation.

The results of this case study may be used to determine track maintenance standards. Page 5, Volume 1 of the Harmonic Roll Series recommends track maintenance standards predicated on the experience of many railroads. These standards are:

a. 1/2" cross level deviation on several consecutive joints will induce rocking, but generally will not cause wheel lift.

b. 5/8" cross level deviation on several consecutive joints will put a car on the verge of wheel lift.

c. 3/4" cross level deviation occurring over three to four consecutive joints will make derailment probable.

Actually, the simulations verify that these are good maintenance standards to practice. More specifically, a maintenance standard could be written as follows:

a. No cross level difference (twist) greater than 1½" on a 62' chord or less should be allowed on any track having speed limits above 10 mph.

b. No more than eight consecutive joints (joints counted on both rails) having 5/8" or greater deviations should be allowed.

c. Maximum 1/4" deviation optimum goal.

Careful selection of cross level standards can reduce forces on both the track structure and truck components that result in early fatigue failures.

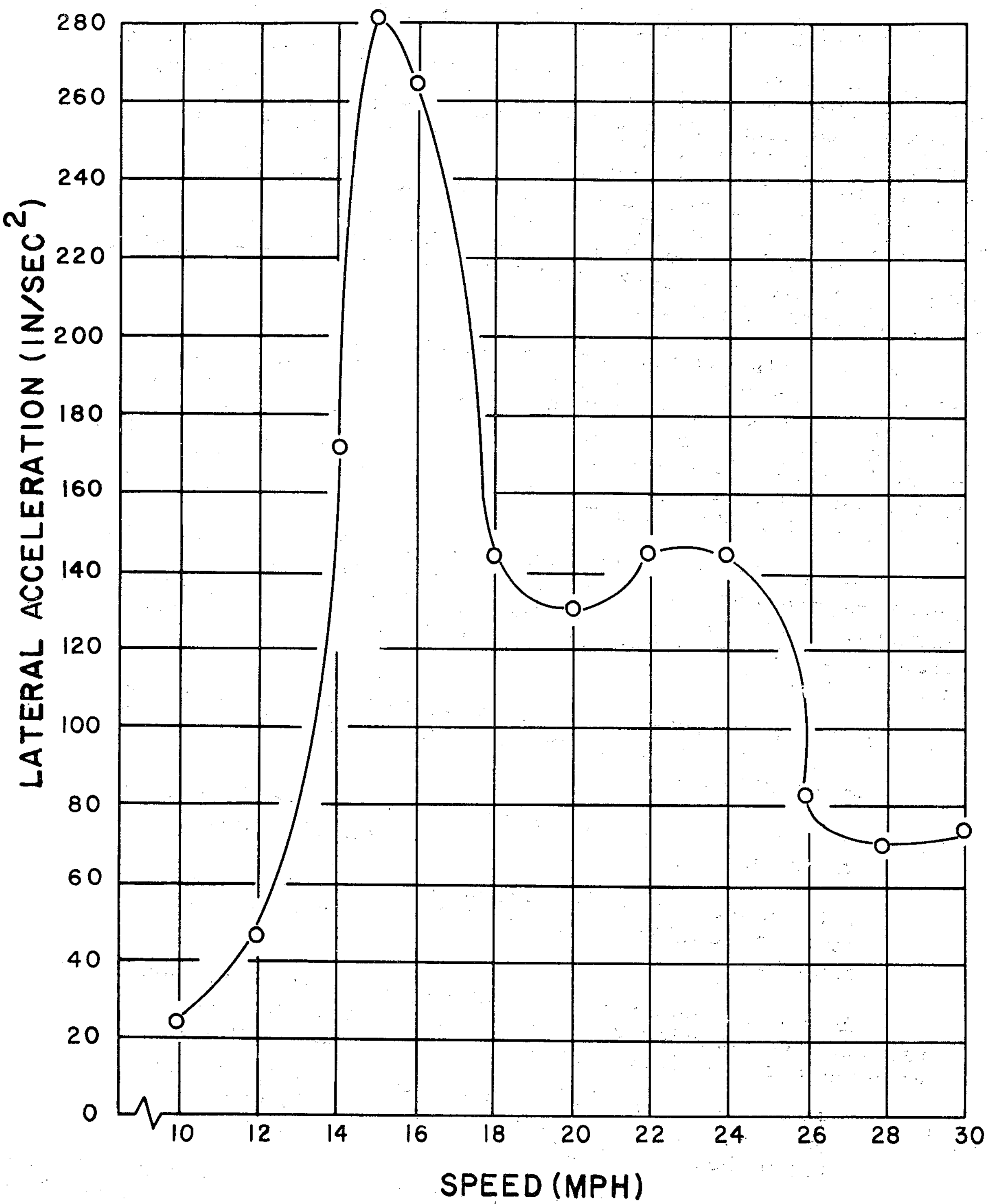


Figure 13
Effects of Speed on Carbody Lateral Acceleration
(70 TON CAR)

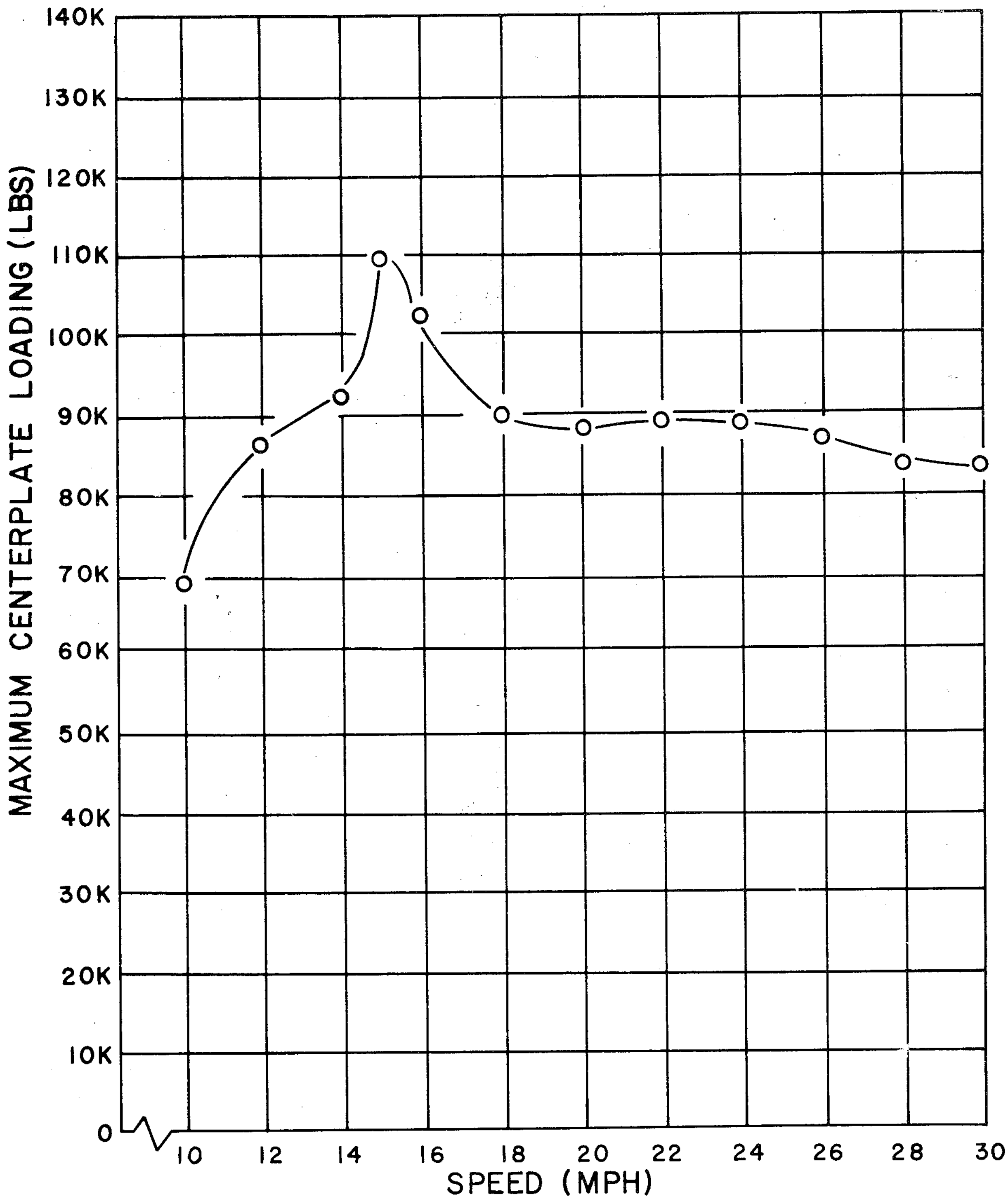


Figure 14
Effects of Speed on Maximum Centerplate Loading
(70 TON CAR)

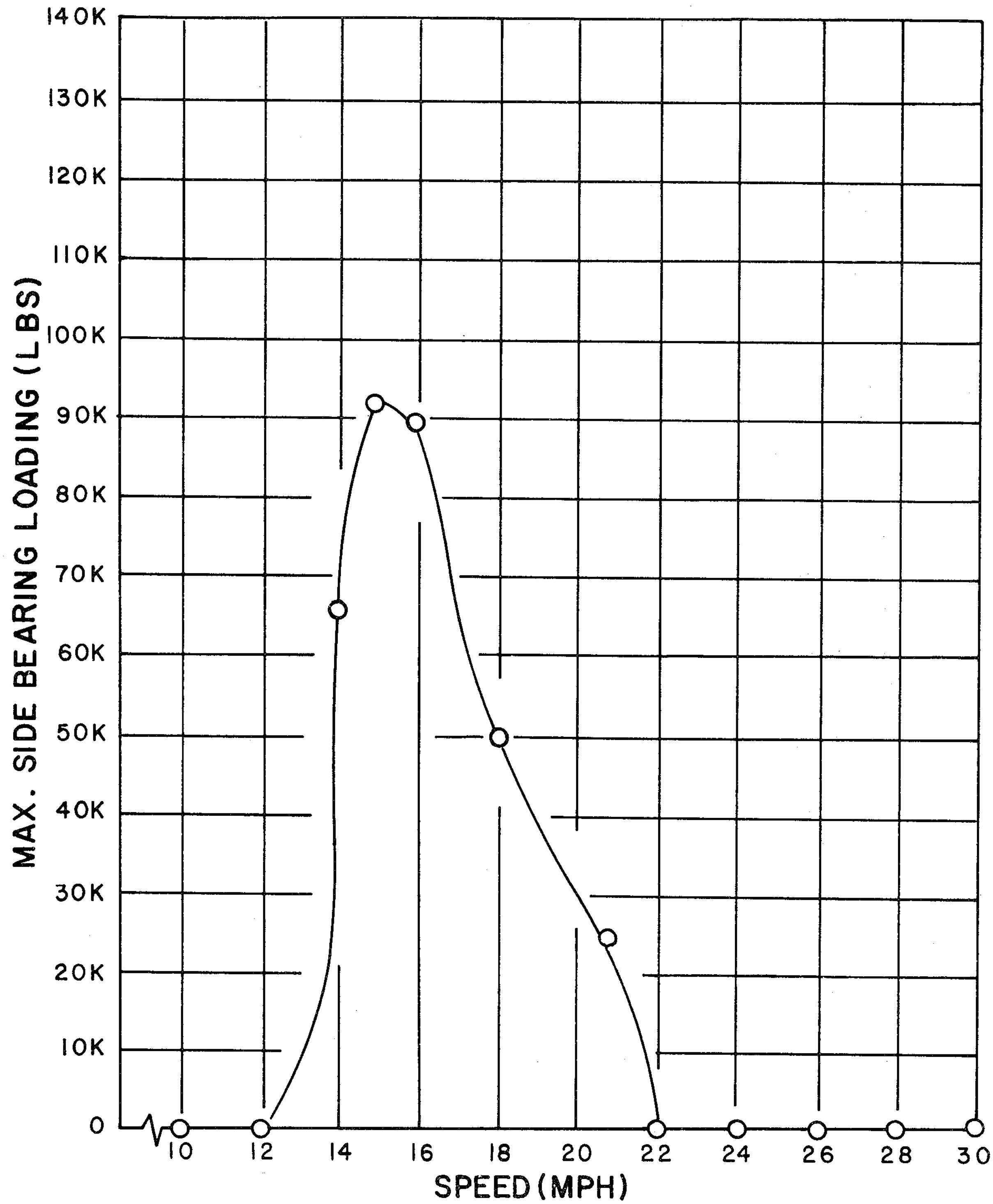


Figure 15
Effects of Speed on Maximum Side Bearing Loading
(70 TON CAR)

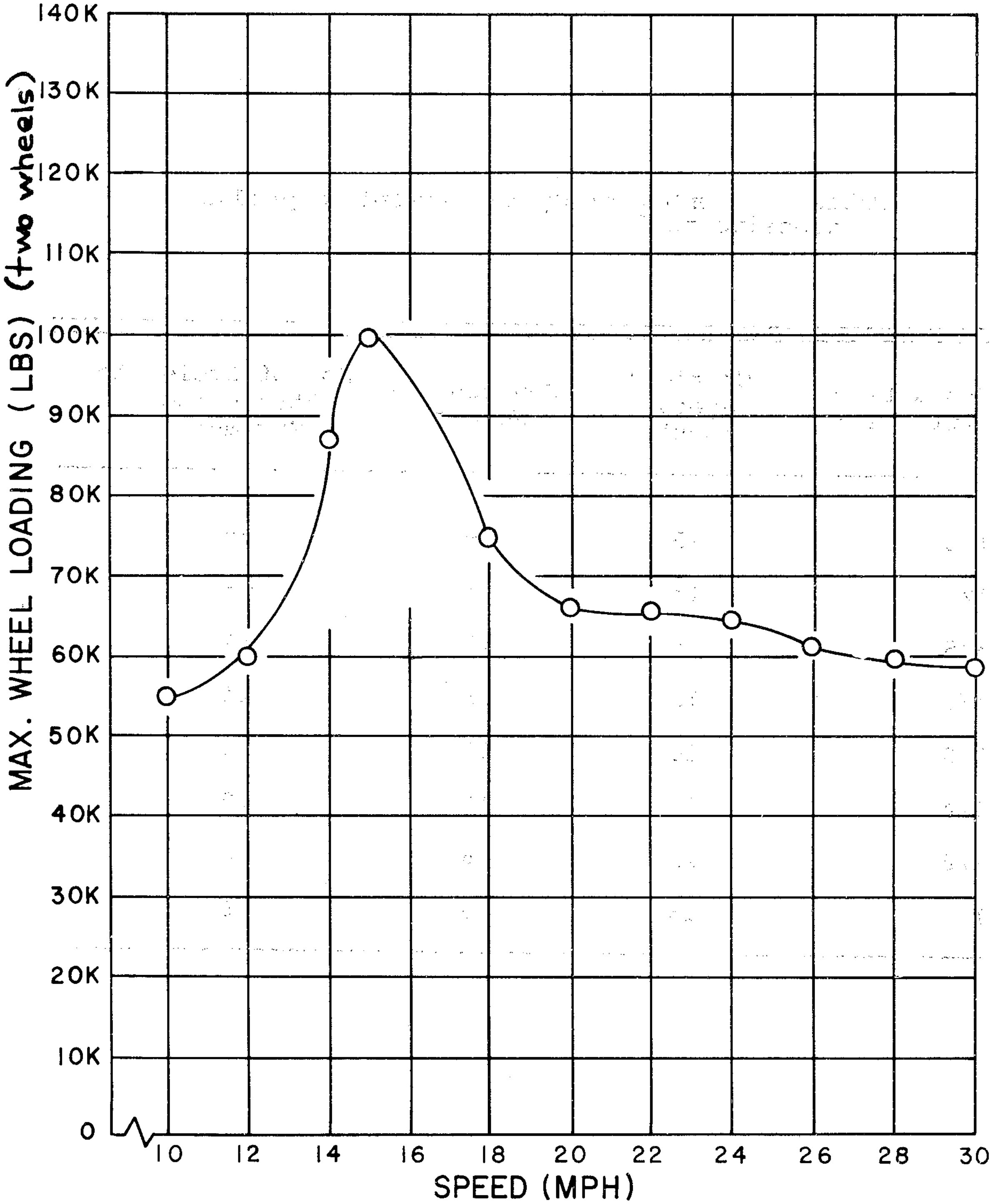


Figure 16
Effects of Speed on Maximum Wheel Loading
(70 TON CAR)

Table 3. Number of Cycles needed to produce excessive roll

Cross Level Diff. (IN)	Critical Speed (MPH)	Consecutive Joints	No. of Cycles to Attain the Maximum
1/8	15	--	--
1/4	15	--	--
3/8	15	14	14
1/2	15	11	11
5/8	16	11	11
3/4	15	10	10
7/8	15	6	6
1	15	6	6

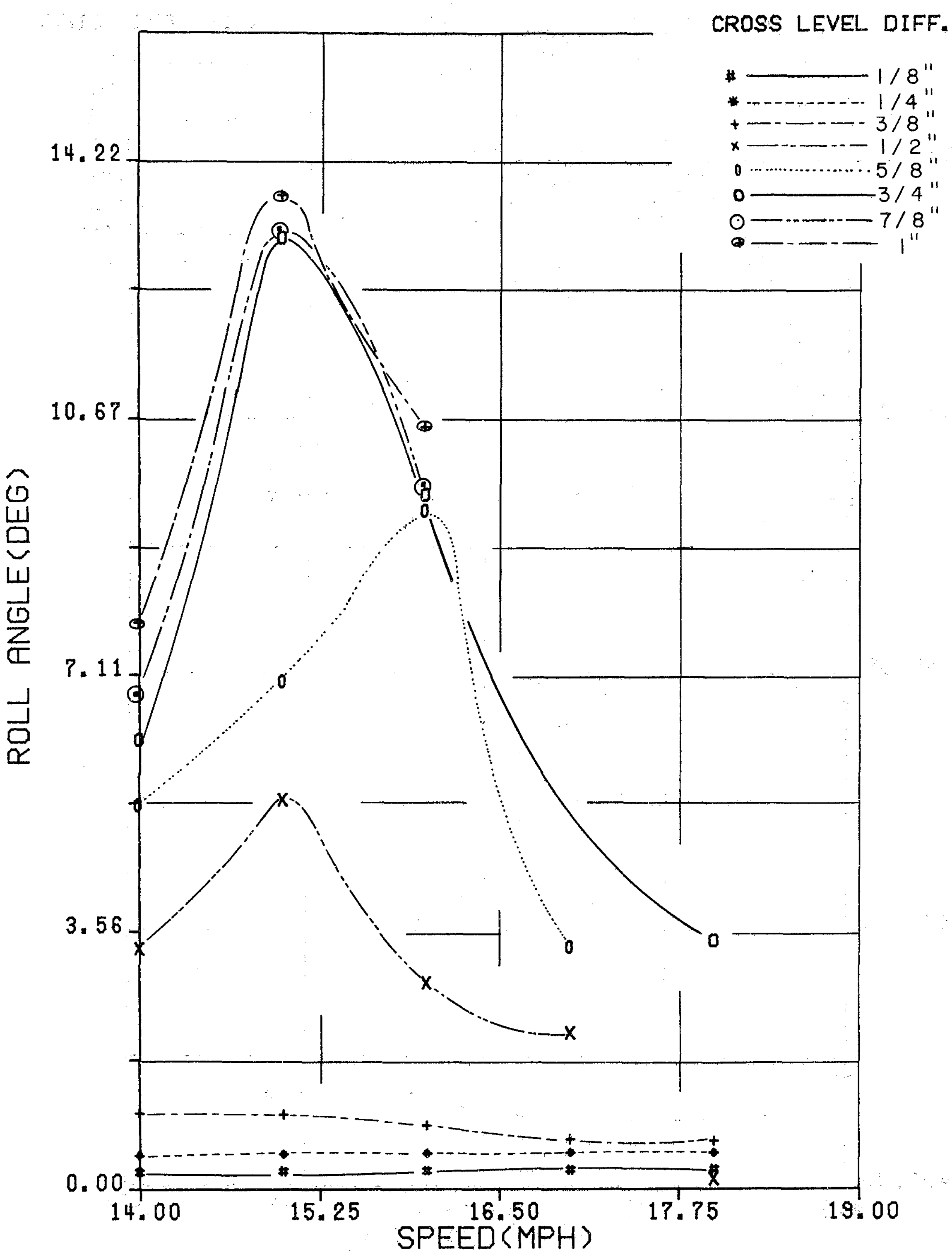


Figure 17
Effect of Cross Level Difference on the Maximum Peak-to-Peak Roll Angle
(70 TON CAR)

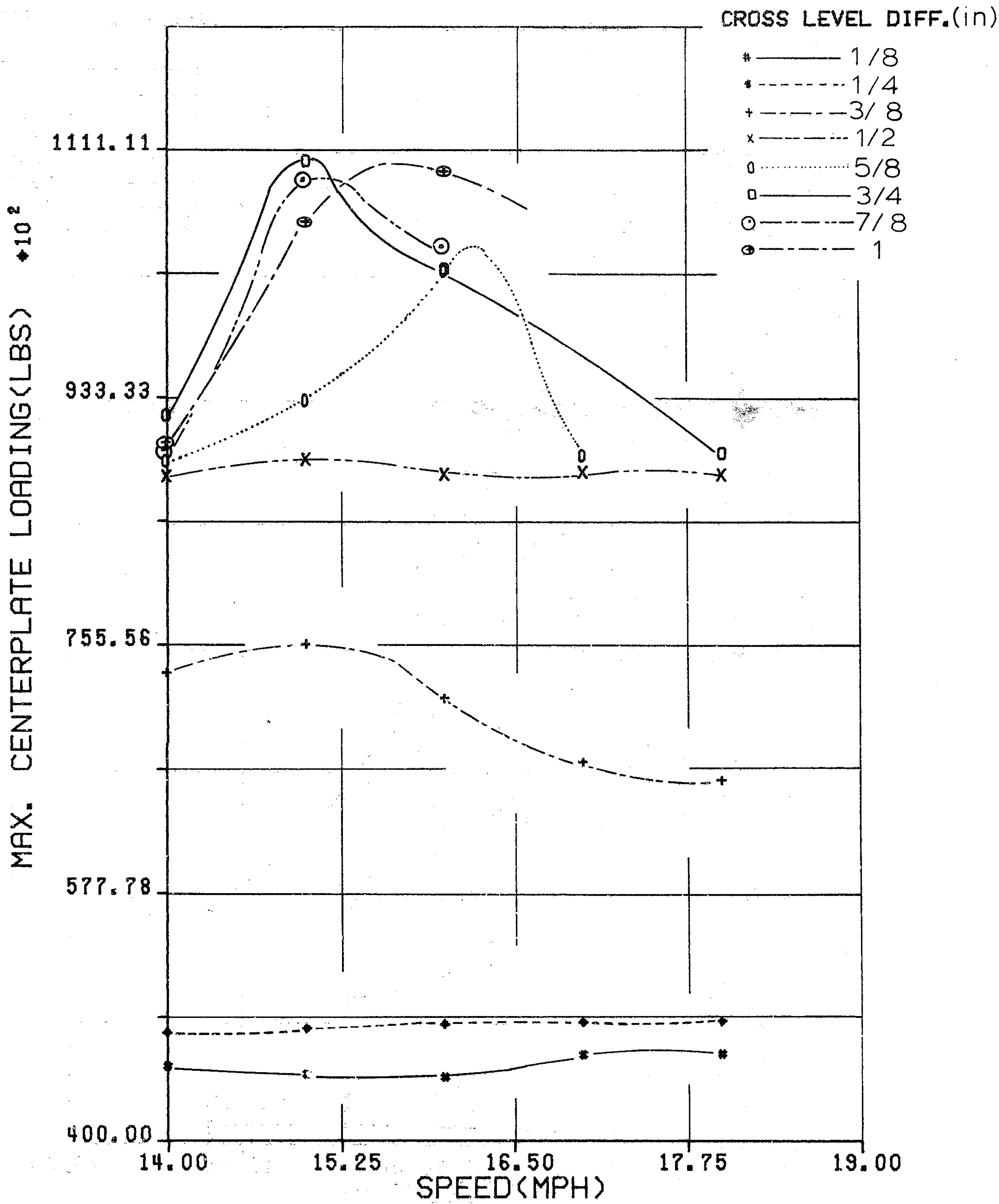


Figure 18
Effect of Cross Level Difference on
Maximum Centerplate Loading
(70 TON CAR)

(CURVE NOT SHOWN HERE INDICATES ZERO SIDE BEARING LOADING AT ANY SPEED)

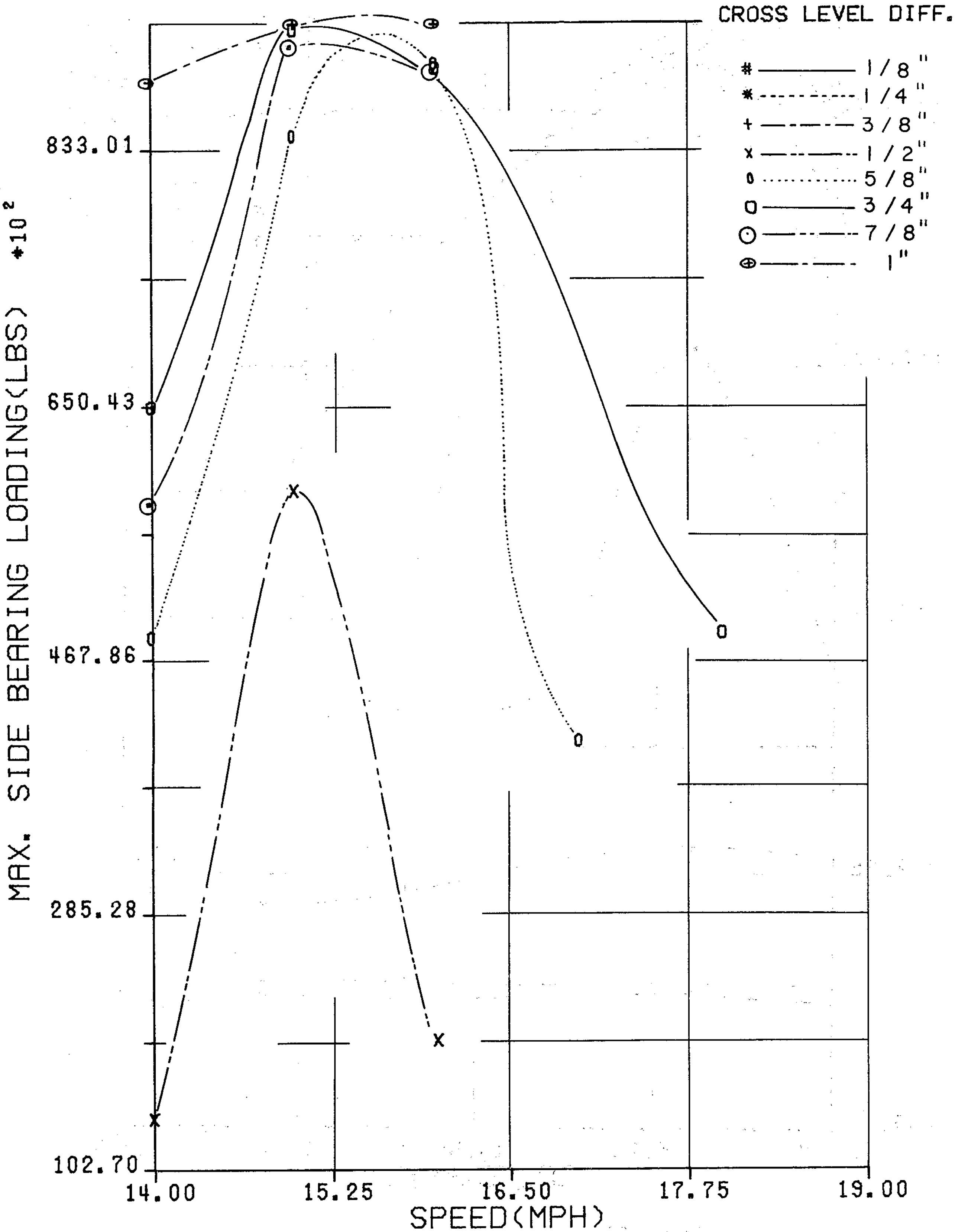


Figure 19
Effect of Cross Level Difference on Maximum Side Bearing Loading
(70 TON CAR)

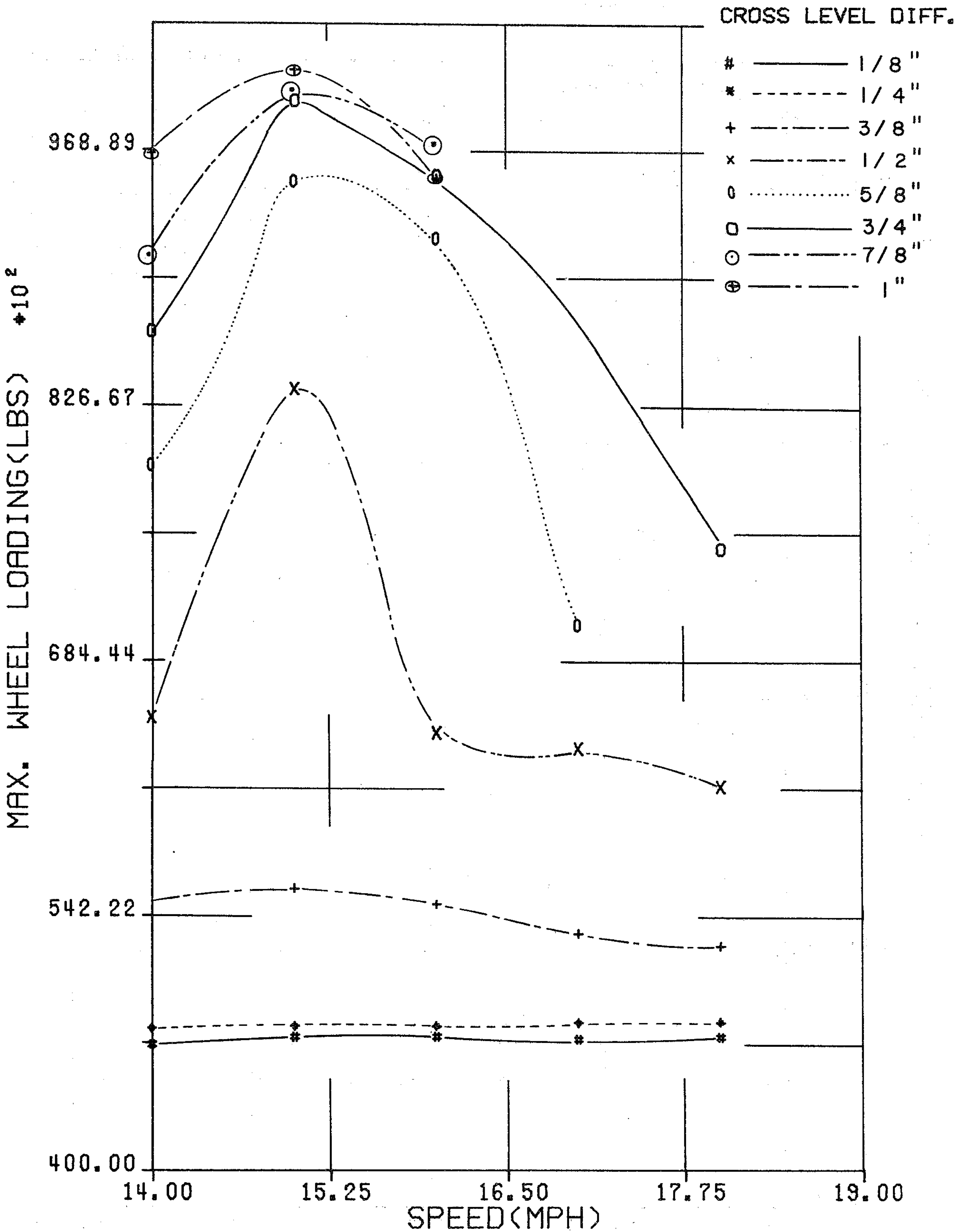


Figure 20
Effect of Cross Level Difference on Maximum Wheel Loading
(70 TON CAR)

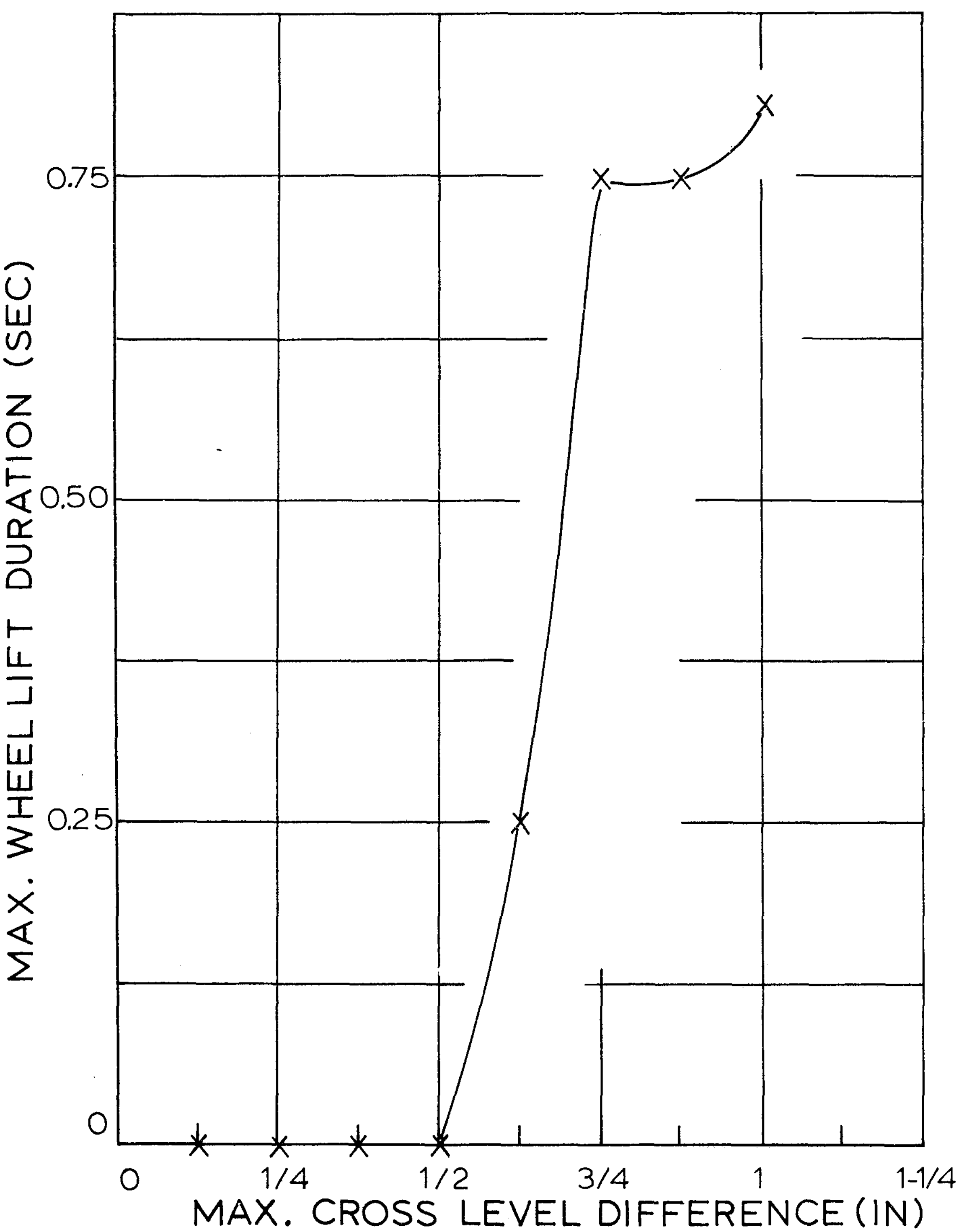


Figure 21

Effect of Cross Level Difference on Wheel-lift

(70 TON CAR)

Case 5 - Effect of Carbody Torsional Stiffness

This case is based on five actual values of carbody torsional stiffness found in different cars. They are 1.25×10^6, 8×10^6, 1.1×10^7, and 8×10^7 ft-lb/rad. Although the highest value differs from the lowest by as much as 8×10^7 ft-lb/rad, practically no difference is seen in the responses. Fig. 22 shows that all values give about thirteen degrees of maximum peak-to-peak roll angles. Also, loadings are about the same.

The reason for this is that the 39½ ft. truck center distance car is running on 39 ft. rail length track. When the truck center distance coincides with the rail length, the forcing functions at the front wheels and the rear wheels are identical. Thus, there is no relative torsional (rotational) displacement between the front and the rear carbodies, and the torsional spring between them gives no restoring moment. The slight variation on these response curves is due to the initial conditions of the input function. When each simulation is started, the front wheels follow the trace shown in Fig. 22 while the rear wheels stay on level track prior to following the rectified wave traces in succession.

As the effect of the carbody torsional stiffness cannot be seen for the 39½ ft. truck center distance car, this distance is first changed to 45 ft. and then 33 ft. The results show more about this effect. As seen in Fig. 23, the most flexible car has a lower peak-to-peak roll angle. As the carbody becomes more rigid, the roll angle will in turn increase until a certain stiffness. From that point, the roll angle starts to drop down. This tendency occurs in both cases of 45 ft. and 33 ft. Though the lower bound value of the five actual carbody torsional stiffness results in a lower roll angle, it is found that the rocking of a car with this value builds up more rapidly (Table 4). Of course, it is more desirable to have the car roll angle built up its maximum in 12 cycles than 9 cycles. A compromise has to be made to attain an optimum stiffness, which will enable the car to build up its rocking slowly and yet the roll angle is not too large.

It is wished to point out that as the truck center distance deviates more from the rail length, the carbody stiffness will certainly impose more effects on the car rocking. It is due to the fact that a bigger difference between the inputs on the front truck and the rear one will occur.

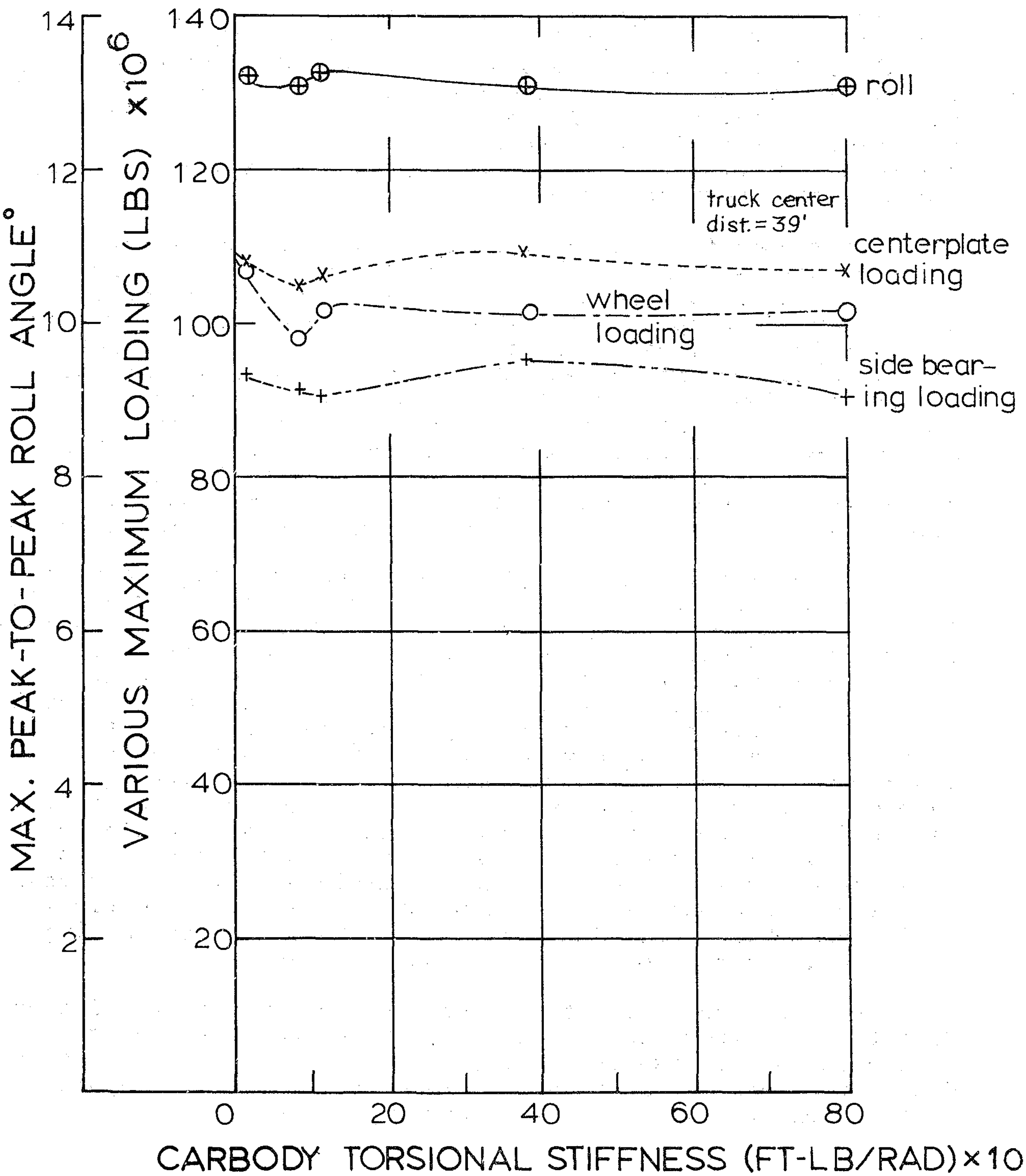


Figure 22
Effects of Carbody Torsional Stiffness
(70 TON CAR)

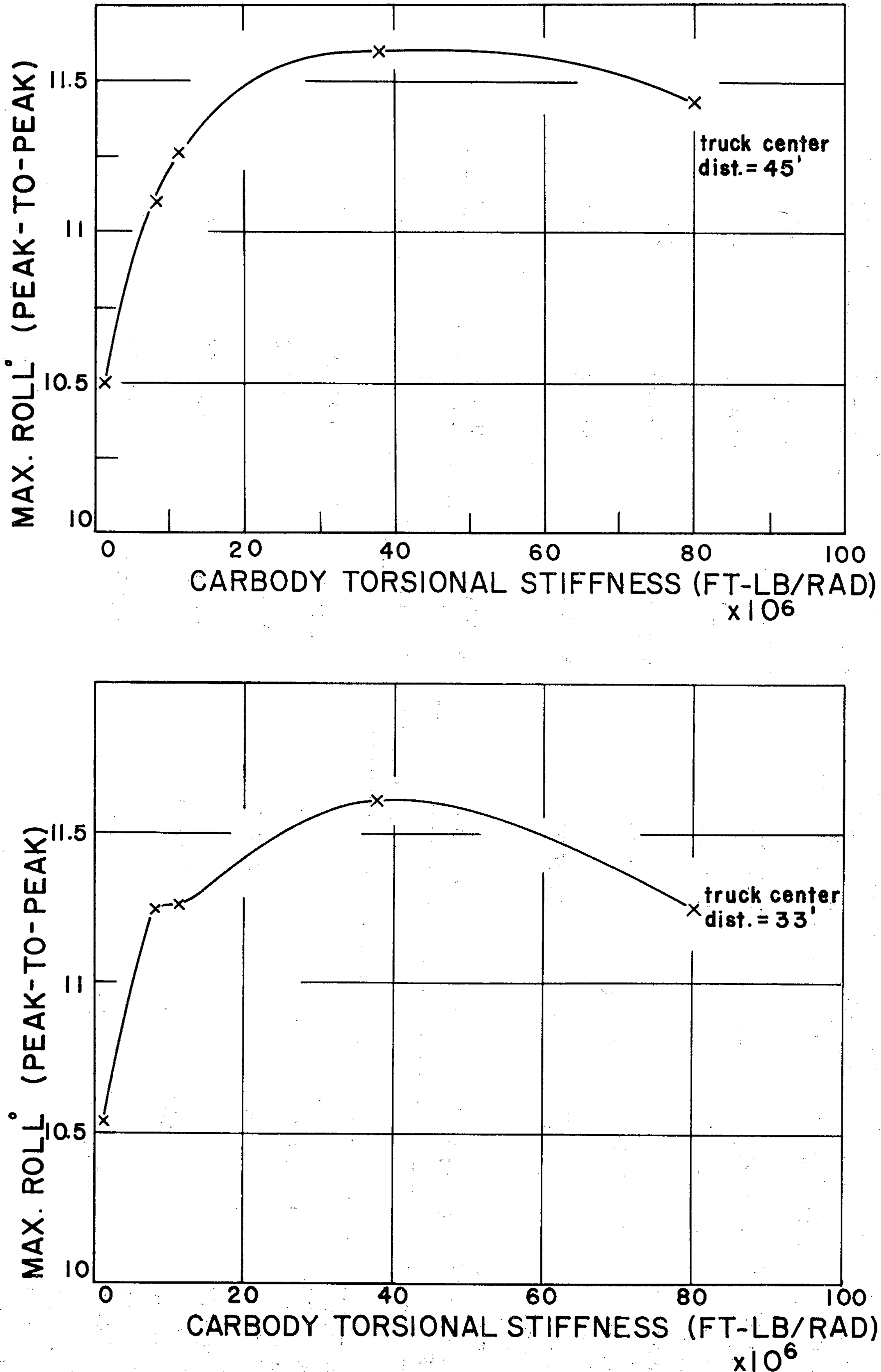


Figure 23

Effects of Carbody Torsional Stiffness

(70 TON CAR)

Table 4. Effect of the Carbody Torsional Stiffness

Type of Car	Torsional Stiffness (lb-ft/rad)	Critical Speed (mph)	Max. cent. plate loading (lbs)	Max. side bearing loading (lbs)	Max. wheel loading (lbs)	# of cycles to attain max. roll angle
45' truck center distance						
100 ton open hopper (empty)	1.25×10^6	15	111000	85000	94000	9
100 ton box car	8.00×10^6	15	109000	89000	97000	12
ACF(TOFC-COFC) STOX-763	1.10×10^7	15	108000	85000	94000	12
100 ton Hi-Cube box car	3.76×10^7	15	109000	92000	97000	11½
100 ton covered twin hopper car (empty)	8.00×10^7	15	108000	89000	95000	11½
33' truck center distance						
100 ton open hopper (empty)	1.25×10^6	15	99000	124000	120000	9
100 ton box car	8.00×10^6	15	100000	94000	98000	12
ACF(TOFC-COFC) STOX-763	1.10×10^7	15	99000	109000	103000	12
100 ton Hi-Cube box car	3.76×10^7	15	101000	93000	99000	11½
100 ton covered twin hopper car (empty)	8.00×10^7	15	100000	93000	98000	11½

Case 6 - Effect of Side Bearing Clearance

The reader should note in Fig. 24 that for the three larger clearances (1/2", 3/8", and 1/4"), the maximum peak-to-peak roll angle at the critical speed is about the same. But, for the smallest clearance of 1/8", the roll angle at the critical speed is about 11.5 degrees, 1.5 degrees lower than the other three. The side bearing clearance can, therefore, aid in suppressing carbody rolling action. Nevertheless, a smaller side bearing clearance will result in a higher side bearing loading as well as wheel loading (Fig. 25).

Moreover, by varying the side bearing clearance, the critical speed will shift too. Studying Fig. 24, one finds that as the side bearing clearance is decreased, the critical speed increases. Therefore, side bearing clearance may affect the natural frequency of the system.

Test track as well as road test experience has shown that the roll angle increases proportionately to the side bearing clearance. Also, the period of rocking motion is reduced in a like manner. Test track experience has also shown that although maximum side bearing reaction force levels do not change materially with wider clearance, longer time wheel load zeroes and larger car body rocking angles do occur. This means the resonant speed will be lower and will persist over a wider speed range. Thus, larger side bearing clearance will result in more numerous reactions over the wider and lower speed range with a resultant increase in derailment probability on curved, superelevated track.

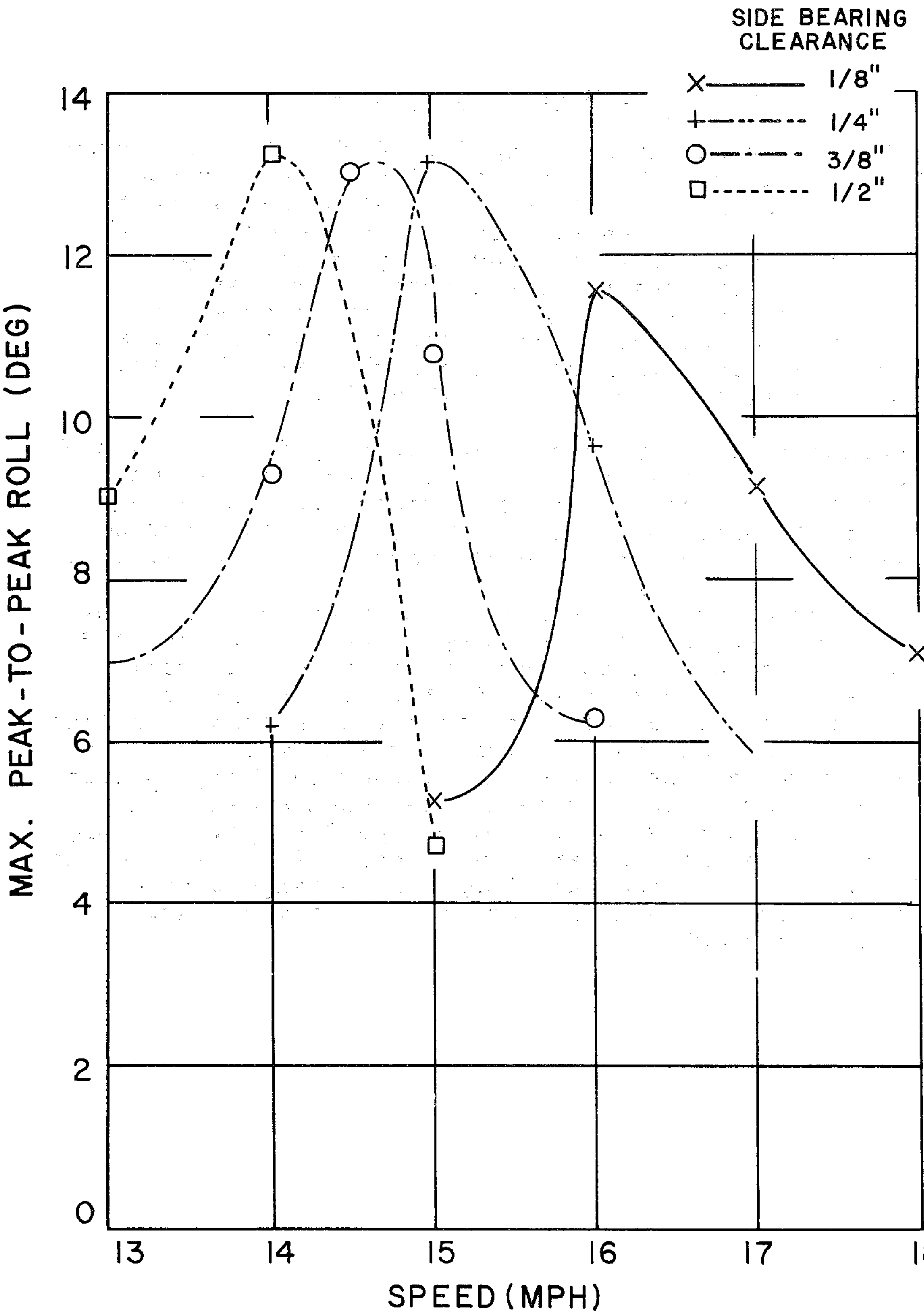


Figure 24

Effects of the Side Bearing Clearance on Maximum Peak-to-Peak Roll Angle

(70 TON CAR)

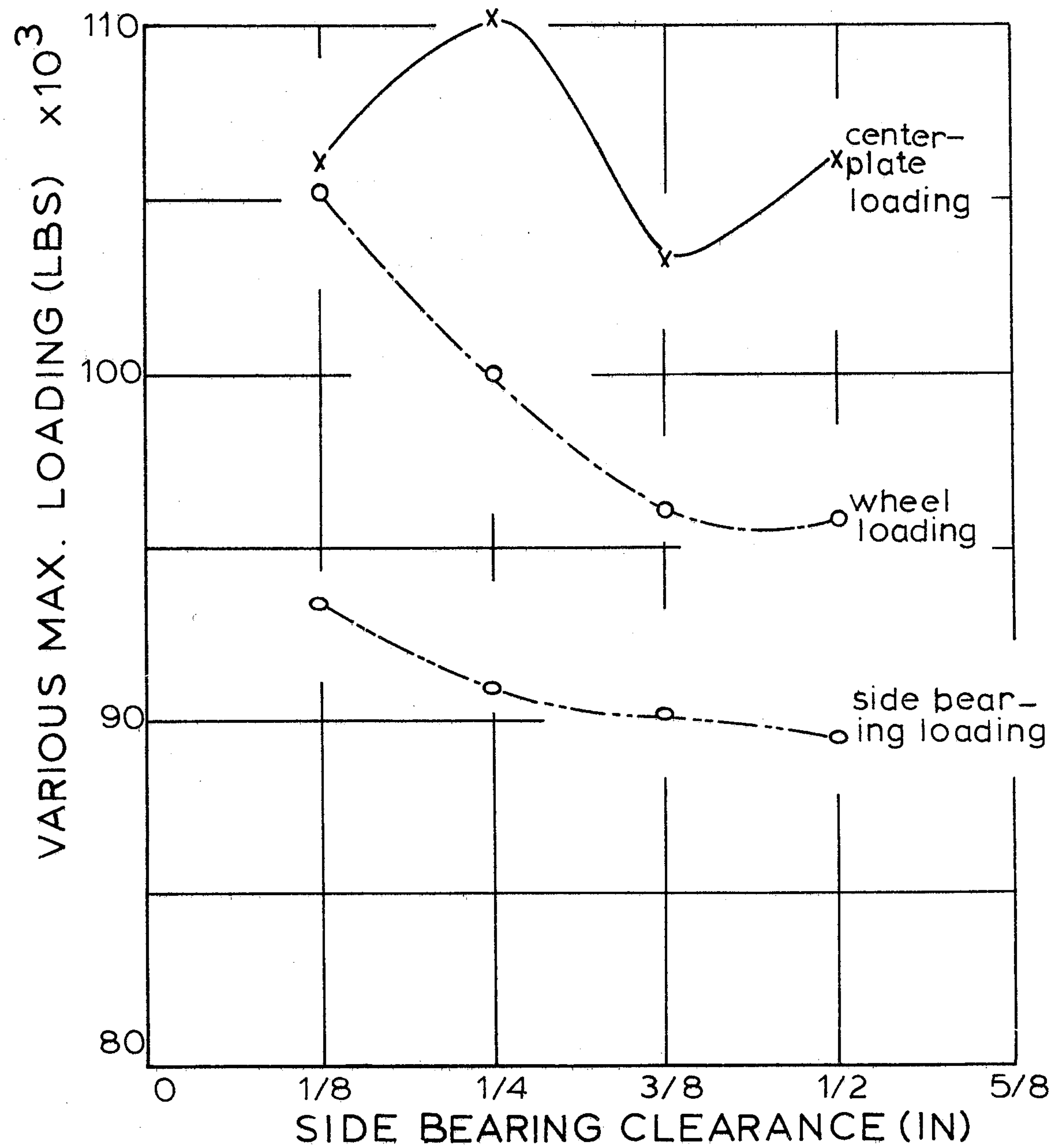
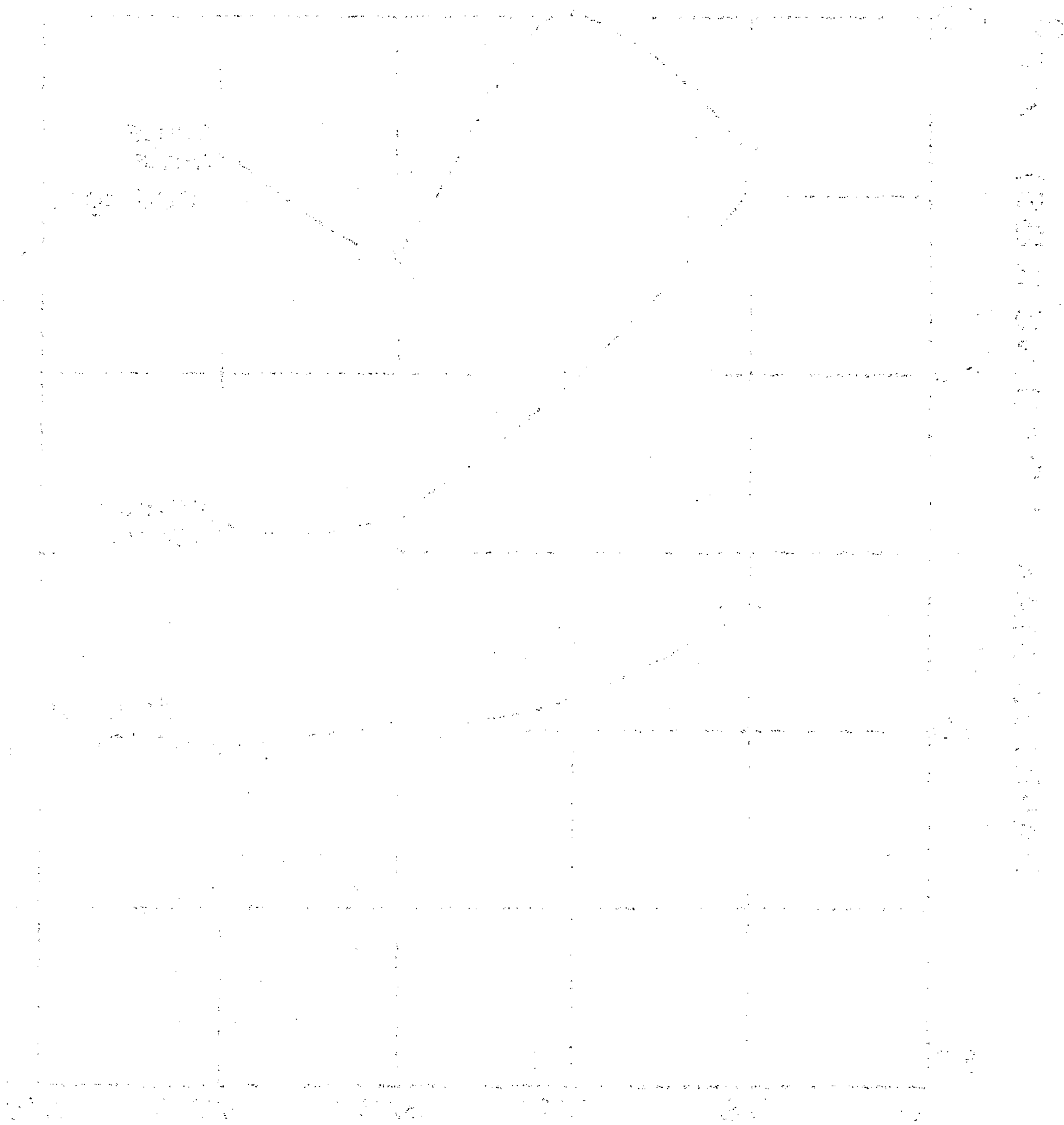


Figure 25
Effects of Side Bearing Clearance on Various Loadings
(70 TON CAR)



Case 7 - Effect of Spring Travel

The characteristics of the three types of spring groups used in this case study are as follows:

AAR Spring Type	Spring Group Constant	Free Travel Length
D-3	382880 lb/ft	2 1/2 in.
D-5	243960	3 11/16
D-7	218400	4 1/4*

* Proposed

First, Fig. 26 shows the shift in the critical speed as the spring group is changed. This shift is expected since the suspension spring constant is one of the determinants of the natural frequency of a suspended system. A stronger suspension will give a higher frequency. Therefore, the stiffest spring type, D-3, has the highest reasonant speed.

Studying Table 5, we find that the roll angle is increased slowly as the free travel length of the spring groups is increased. Nevertheless, the loadings of the longer travel spring groups, D-5 and D-7 may be as high as, or even higher than the D-3 group after the maximum roll angle is established. However, the longer travel springs help preclude reaching maximum roll amplitudes because the probability of having enough low spots in succession to produce maximum roll angle, is less likely.

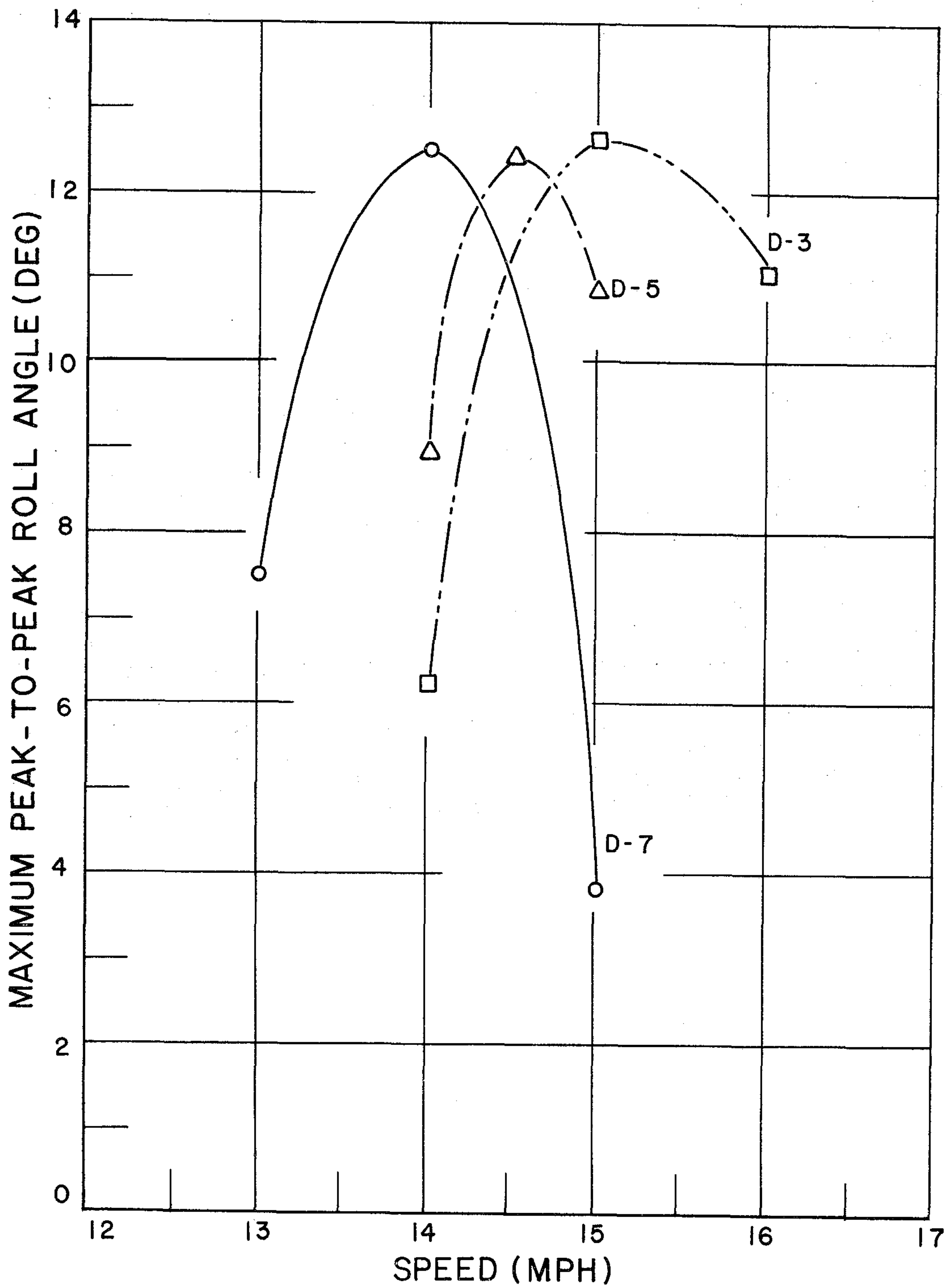


Figure 26
Effect of Spring Group on Maximum
Peak-to-Peak Roll Angle

(70 TON CAR)

Table 5. Effect of Spring Groups on Various Loadings, Wheel-lift and Bottoming of the Springs at Critical Speed

AAR Spring type	Spring group arrangement per nest	Critical speed	No.of roll cycles run to reach maximum	Max. centerplate loading (lbs)	Max. side bearing loading (lbs)	Max. wheel loading (lbs) (two wheels)	Max. wheel-lift duration (sec)
D-3	7 outer 2 inner	15 mph	9	167,000	157,000	158,000	.55
D-5	7 outer 5 inner	14.5	10	179,000	160,000	168,000	.60
D-7*	7 outer 5 inner	14	14	157,000	186,000	182,000	.64
		14	9	148,000	145,000	138,000	-

* Proposed - 4-1/4" travel

Case 8 - Effect of Centerplate Diameter

Although the centerplate diameter is varied from 14 in. to 16 in., Fig. 27 shows that the maximum peak-to-peak roll angles are the same for all three sizes. Loadings on the centerplate differ only by about 8000 lbs. Therefore, varying the centerplate diameter will not affect the excessive rocking problem, but it may serve to reduce some high centerplate stresses. A reduction of stresses may be more significant with respect to longitudinal loads rather than lateral loads.

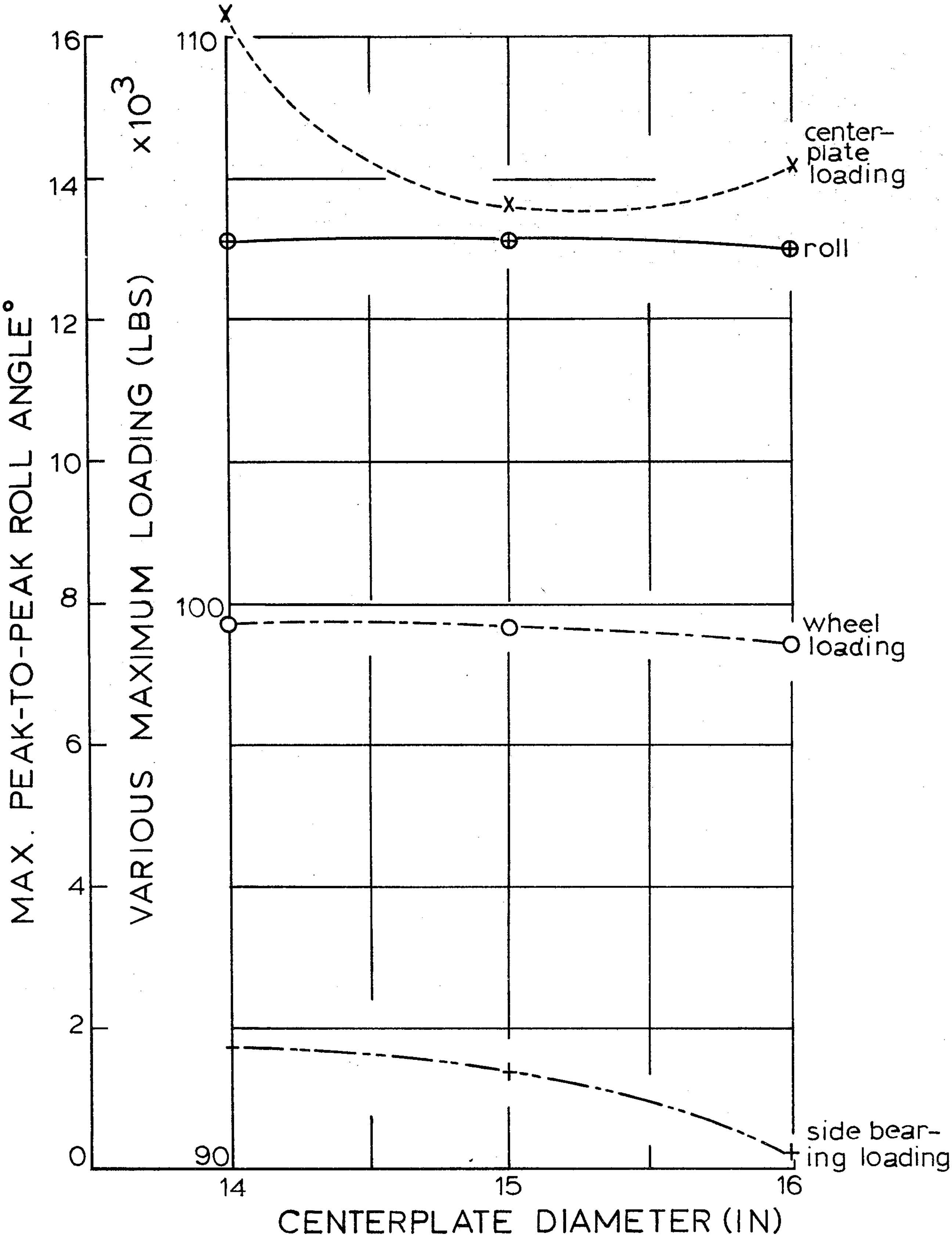


Figure 27
Effects of Centerplate Diameter
(70 TON CAR)

A. STUCKI CO. FLEXIBLE CARBODY MODEL

FREIGHT CAR SIMULATION SERIES

In the mid-sixties, the A. Stucki Co. of Pittsburgh began developing a damping unit which could be placed in a standard freight car truck to absorb the roll energy developed by a rocking freight car. This development work required extensive field tests in order to determine force levels and absorption requirements. In 1968, Donald Wiebe of the A. Stucki Co. described these field tests in his technical paper "The Effects of the Lateral Instability of High Center-of-Gravity Freight Cars," found in Transactions of the American Society of Mechanical Engineers Journal of Engineering for Industry, November 1968, pp. 727-740. In conjunction with these field tests, Mr. Wiebe developed a mathematical model of a freight car which could be solved on a hybrid computer system. The field test results were in fact used to validate the model. Since its development, the model has been used to study the dynamics of freight cars with a high degree of accuracy and confidence.

In general, the model may be described as a system of dynamic equations involving the use of approximately one hundred constants, including all dimensional geometry and the relative vertical and lateral constraints required to compute the motion of the wheel sets, sideframes, bolsters, and carbody. The model is a three dimensional system as shown in Fig. 28.

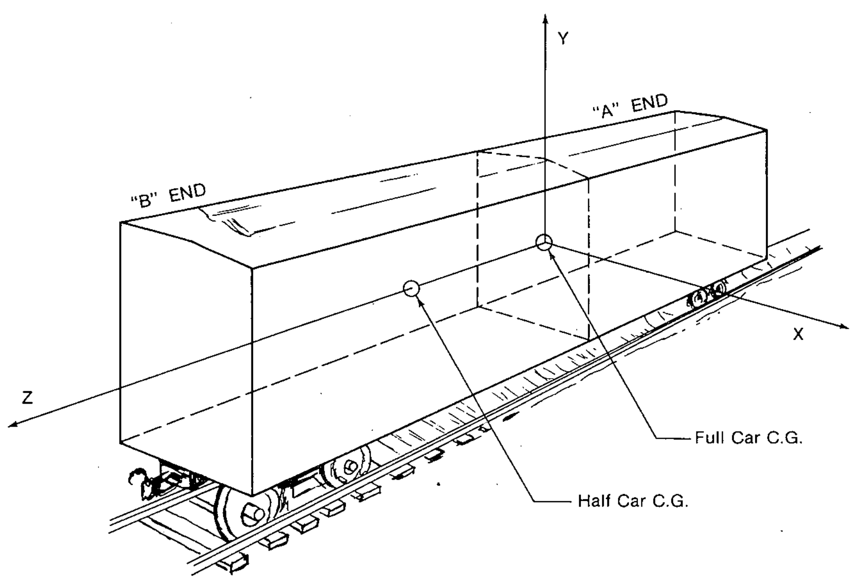


Figure 28
Freight Car Model Coordinates
(70 TON CAR)

This figure demonstrates the coordinate axis reference as it relates to the car and direction of motion. As the observer faces the B end of the car, the left rail determined from this vantage point becomes the superelevated rail for the curved track simulations. The car moves away from the observer during the simulations.

The computer drawn traces illustrated in the following sections are results from a hybrid computer arrangement using digital computation interfaced with a digital to analog converter producing the resulting analog plot. The traces shown are referenced with respect to time, starting at a zero time point where the car is in static equilibrium with all eight wheels on flat tangent track at the rail profile level indicated on the profile trace at time zero. The leading wheel set and each succeeding wheel set in turn start to follow the constant curvature at that point in the case of curved track simulations. The rail profile trace shows the undeflected surface of the left rail. The right rail profile is identical to the left rail along the zero elevation level with the profile out of phase representing half stagger of the rail joints. The car is started instantaneously at the given speed and proceeds at constant speed for the entire run. All runs were continued to the point of maximum response.

The degrees of freedom are listed in Table 6. Again, the detailed equations of motion are not listed because of their availability in other reference sources. All stiffness characteristics are described by table valued, piecewise, linear, elements. Bolster-sideframe friction in both lateral and vertical directions was modeled as a fixed force with stik-slip logic. The wheel sets are laterally fixed to the path of the rails but can separate vertically from the rail in order to represent wheel lift. The carbody can separate from the bolster at the center plate and side bearing contact points. The two carbody halves are combined with the linear deformational constraints necessary for the degrees of freedom indicated in Table 6. Complete degrees of freedom are unnecessary for the sideframes and axles in the response traces included. This is true whenever the wheels and side-frames are referenced to the deflected track profile and experience no independent low frequency response. However, the sideframe-axle degrees of freedom are necessary for displaying wheel lift and the effect of wheel lift in the total response result. The load inside the car is modeled with a separate degree of lateral freedom coupled with the carbody. The constraint is similar to that measured for paper rolls.

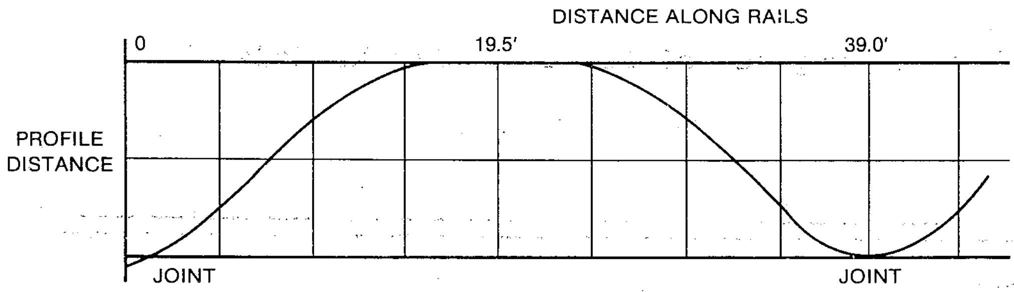


Figure 29
Vertical Profile Shape for Jointed Rail
(70 TON CAR)

A smooth, modified sine function, shown in Fig. 29 was used to describe the vertical profile of the undeflected rails for all of the runs. This function is a good representation of typical jointed rail on average road bed, as has been determined from experimental track profile measurements on various test tracks. In actual practice, the deflected track profile is much different from the undeflected profile especially when high dynamic reactions are indicated. This profile shape also results in maximum rocking or vertical energy input for a given depth of depression or cross level change between successive joints.

Table 6. Degrees of Freedom - Stucki Model

R - Rotational	
T - Translational	
<u>CARBODY</u>	
A - half car	T_x, T_y, R_x, R_y, R_z
B - half car	T_x, T_y, R_x, R_y, R_z
<u>TRUCK BOLSTER</u>	
A end	T_x, T_y, R_z
B end	T_x, T_y, R_z
<u>SIDEFRAMES</u>	
A_l	T_y, R_x
A_r	T_y, R_x
B_l	T_y, R_x
B_r	T_y, R_x
<u>AXLES</u>	
#1	T_y, R_z
#2	T_y, R_z
#3	T_y, R_z
#4	T_y, R_z

STUCKI FREIGHT CAR SIMULATION SERIES

DESCRIPTION OF CASE STUDIES

A 70 ton and 100 ton box car typical of modern construction techniques was selected for the simulation series. The constants or parameters describing these two cars are given in Appendix II. The values found in that appendix are actual values taken from car designs that are now in service on certain AAR member roads. Basically, the following table describes the general configuration of each car for the reader's ease in comparison.

Table 7. General Description of Box Cars Used in Stucki Simulation Series

	gross rail length	inside length	inside height	inside width	truck spacing	loaded center of gravity	truck type
70 ton car	185,400 lbs.	52'5"	11'3"	9'5"	39'6"	86.5"	constant column guide friction
100 ton car	252,150 lbs.	50'7"	12'8"	9'6"	40'10"	98.4"	

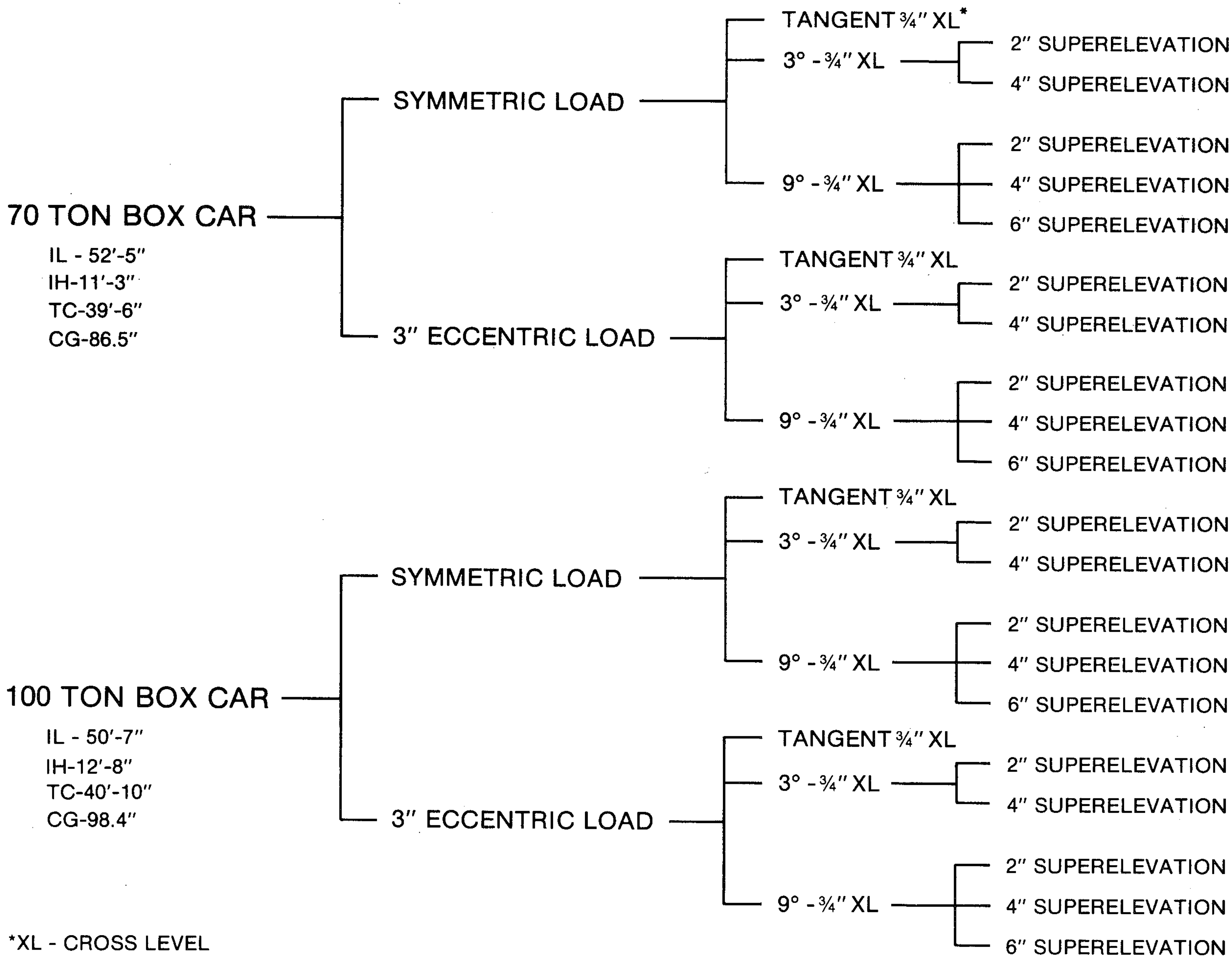
The rationale behind selecting a box car for simulation was that it represents a general configuration of freight car. Essentially, the box car is modeled as a rectangular box shape. An open top hopper or covered hopper of the same general dimensions would be modeled in almost the same manner with the primary difference being the carbody stiffness. Case 5 in the parameter study shows that a difference in stiffness renders only a subtle change in dynamic response for cars with truck centers closely matching the rail joints. Therefore, the response characteristics of the box cars simulated in this series would generally represent all high center of gravity cars of the same general dimensions.

The load utilized for the simulation runs was modeled after a typical rolled paper load in order to create a high density, high center of gravity load condition. However as with the carbody model, the load is represented mathematically by rectangular box shapes. Therefore, the reader can again consider the modeled load representative generally of any load filling the same volume as the paper load.

Half of the runs were made with the load symmetrically arranged around the longitudinal centerline of the car. In order to determine the effects of eccentric loading, the other half of the simulation runs were made with the center of the load shifted 3" laterally from the longitudinal centerline of the car. All the runs except Figures 31 and 35 include a lateral degree of freedom between the carbody and the load. The load can also move in the car with a restraint characteristic of a paper roll load. This added degree of freedom generally results in a somewhat slower rocking resonant speed than would occur with a fixed load condition.

With two load configurations for each car, a series of track conditions were simulated. These are shown in Fig. 30. Each track consisted of 39' rail lengths with half-staggered joints. The cross level variation was 3/4" for each simulation run in order to approximate the testing standards set forth on page D-65 of the AAR Manual of Standards and Recommended Practices. In addition to tangent track simulations, 3° and 9° curves were simulated. The 3° curve closely approximates the test track at Hollidaysburg, Pa. that was used in early studies of the rock and roll problem. The 9° curve closely approximates the Frankfort, Ky. test track that is currently in use for approving devices for control of rocking action. In order to show the effects of superelevation on the car response, 2" and 4" superelevations were used on the 3° curve, and 2", 4" and 6" superelevations were used on the 9° curve. The track surface profile was similar to that resulting from the recommended shimming described in the AAR rocking test specification.

Some of the simulation runs are validated by field measurements taken during actual testing of box cars containing paper loads on the Frankfort, Ky. test track. These tests were described by D. G. Orr during a panel discussion at the November, 1974 ASME Rail Transportation Meeting.



*XL - CROSS LEVEL

Figure 30
Freight Car Simulation Series Matrix of Runs

FREIGHT CAR SIMULATION SERIES

DISCUSSION OF RESULTS

The actual computer drawn traces are presented for the reader's analysis according to the following schedule of figures. The speed given is the critical speed for that particular simulation except as noted.

70 TON BOX CAR - SYMMETRIC LOAD

Tangent Track	18 mph	Fig. 31
3 deg. curve, 2" elev.	18 mph	Fig. 32
3 deg. curve, 4" elev.	18 mph	Fig. 33
9 deg. curve, 2" elev.	18 mph	Fig. 34
9 deg. curve, 4" elev.	19 mph	Fig. 35
9 deg. curve, 6" elev.	18 mph	Fig. 36

70 TON BOX CAR - 3" ECCENTRIC LOAD

Tangent Track	18 mph	Fig. 37
3 deg. curve, 2" elev.	17 mph	Fig. 38
3 deg. curve, 4" elev.	17 mph	Fig. 39
9 deg. curve, 2" elev.	17 mph	Fig. 40
9 deg. curve, 4" elev.	17 mph	Fig. 41
9 deg. curve, 6" elev.	18 mph	Fig. 42
9 deg. curve, 6" elev.	17 mph (critical)	Fig. 43

100 TON BOX CAR - SYMMETRIC LOAD

Tangent Track	13.5 mph	Fig. 44
3 deg. curve, 2" elev.	13.5 mph	Fig. 45
3 deg. curve, 4" elev.[1]	13 mph	Fig. 46
3 deg. curve, 6" elev.	13.5 mph	Fig. 47
9 deg. curve, 2" elev.	14 mph	Fig. 48
9 deg. curve, 4" elev.	13 mph	Fig. 49
9 deg. curve, 6" elev.	14 mph	Fig. 50

100 TON BOX CAR - 3" ECCENTRIC LOAD

Tangent Track	13 mph	Fig. 51
3 deg. curve, 2" elev.	13 mph	Fig. 52
3 deg. curve, 4" elev.	13 mph	Fig. 53
9 deg. curve, 2" elev.	13.5 mph	Fig. 54
9 deg. curve, 4" elev.	13.5 mph	Fig. 55
9 deg. curve, 6" elev.	13.5 mph	Fig. 56

1. Not included in matrix shown in Fig. 30.

Each figure is labelled to show the resulting forces and displacements concurrently as the car progresses through the simulation. Each parameter is plotted against time with the top trace showing the rail profile. The critical factor of the carbody roll angle display is the peak-to-peak value of roll. The slight upward shift of the trace is due to the superelevation of the curves. The spring group compression trace is labelled to show bottoming of the springs. In most runs, the springs are almost fully extended. The right front spring travel display represents the grouping on the low side of the car during curved runs. Thus, spring bottoming of the right side group occurs more often as superelevation increases. The trace of the left front wheel load governs the loading of the lead wheel on the lead truck. On curved runs, this is the outer wheel. Wheel lift generally occurs when the left front wheel load is zero. The longer the wheel load is zero, the greater the potential for a large wheel lift. On runs showing the centerplate reaction, a zero load indicates impending centerplate separation.

Some relationships can be drawn by comparing traces of individual parameters. For example, the side bearing reaction reaches a peak when the carbody roll is maximum. Also note that the carbody roll goes out of phase with the rail profile trace. If the simulation were carried far enough, the roll angle trace would appear to have beats because of constant speed and the slight difference between truck spacing and rail length. Theoretically if the car would decelerate slightly after reaching peak roll, it would continue to increase in roll angle.

Generally, all runs proved the 50', 70 and 100 ton loaded, high center of gravity cars to be particularly sensitive to 39' jointed, half-staggered track. In all simulations, sufficient zero wheel loads were developed to make wheel lift highly probable.

An examination of the carbody roll shown in Figures 57 and 58 reveals an interesting phenomenon. A shifted load will develop lower roll angles than a symmetric load. Also, tangent track roll is significantly greater than some instances on curved track. The apparent reason for this is the non-symmetry from the eccentric load and superelevation resulting in spring motion cycles on one side near the solid level and therefore less energy storage capacity of springs on that side. Superimposing the 100 ton values on the 70 ton values shows that the center of gravity greatly increases the rocking tendency.

The fact that the carbody roll decreases with eccentricity and increasing superelevation is misleading. The real criteria for car stability is the duration of zero wheel load since this is the period the car is most susceptible to derailment. Table 8 illustrates this criteria. For the 100 ton eccentric load, the wheel load zero is more than 1-1/2 seconds!

Needless to say, poor shipping practices can be very detrimental to freight car dynamic response.

The results suggest that the test requirements for a rocking control device could be revised to have testing conducted on a tangent track for more safety. Indications are the freight car response magnitudes would be slightly greater. Duration of wheel load zero could become the qualifying parameter rather than unpredictable wheel lifts. The simulation results could also be used to help develop new, untried car design dynamic test requirements.

It must be recognized that the character of rocking motion on tangent track is quite different than that occurring on curved track. Tangent test response will not result in the severe non-linearities that account for higher reactions and larger deviations in wheel loads that occur with superelevated track. In many typical carbody-spring combinations, springs will not go solid during a tangent rocking test. In comparison, the low side springs will remain solid for 1/3 to 1/2 the total rocking cycle during a test on superelevated track.

In summary, the traces are most conclusive. The 50', 70 ton and 100 ton box cars are most susceptible to the rock and roll problem and therefore should require some form of rocking control. It is interesting to note that the 100 ton car does fit the criteria for supplemental snubbing according to Rule 88 of the Office Manual of AAR Interchange Rules. In truth, the actual car that was modelled does have rocking control devices in service although they were not simulated in this investigation. On the other hand, the 70 ton car does not require supplemental snubbing according to Rule 88. The instability of the 70 ton car demonstrated by the simulations points to the need for modification of these requirements.

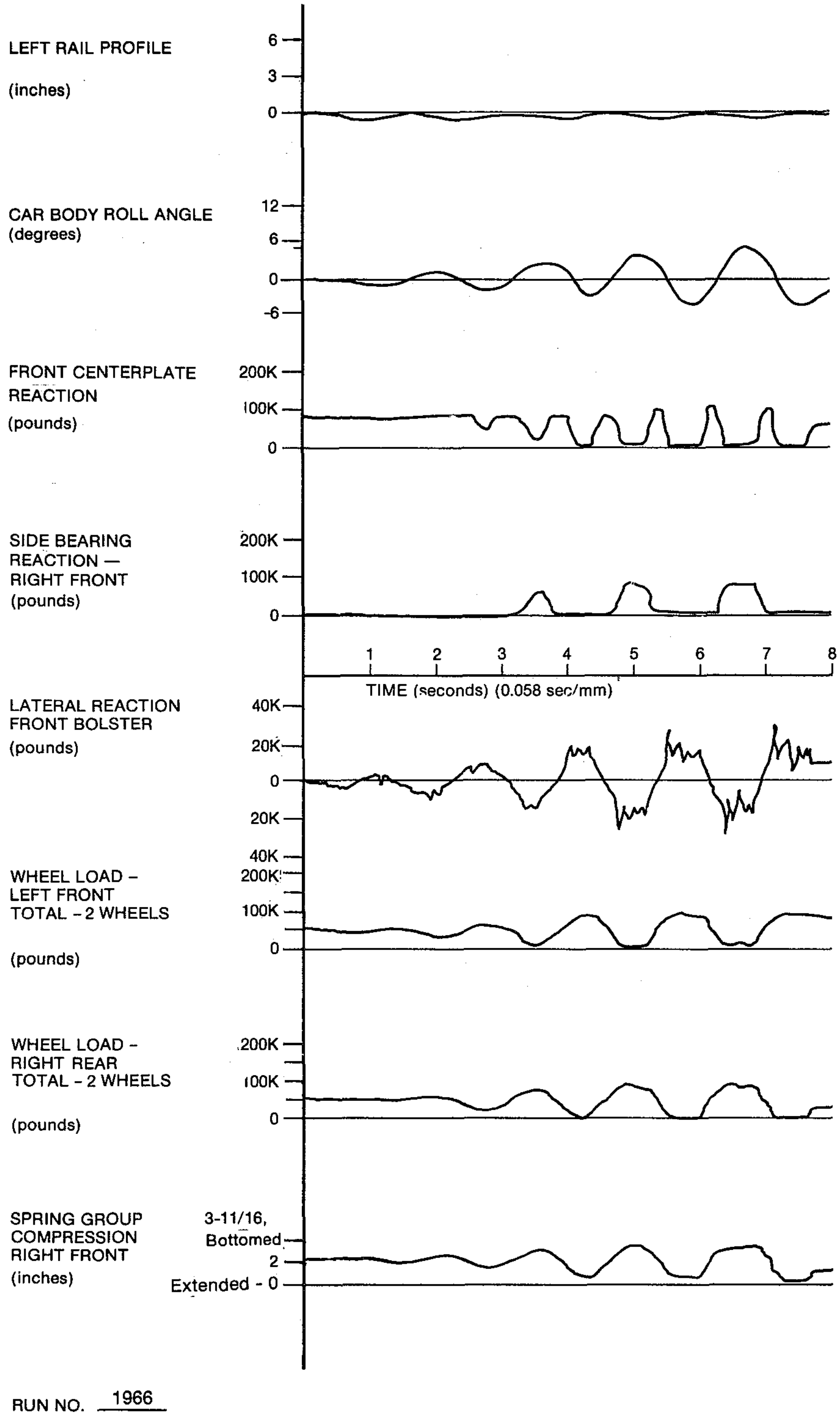
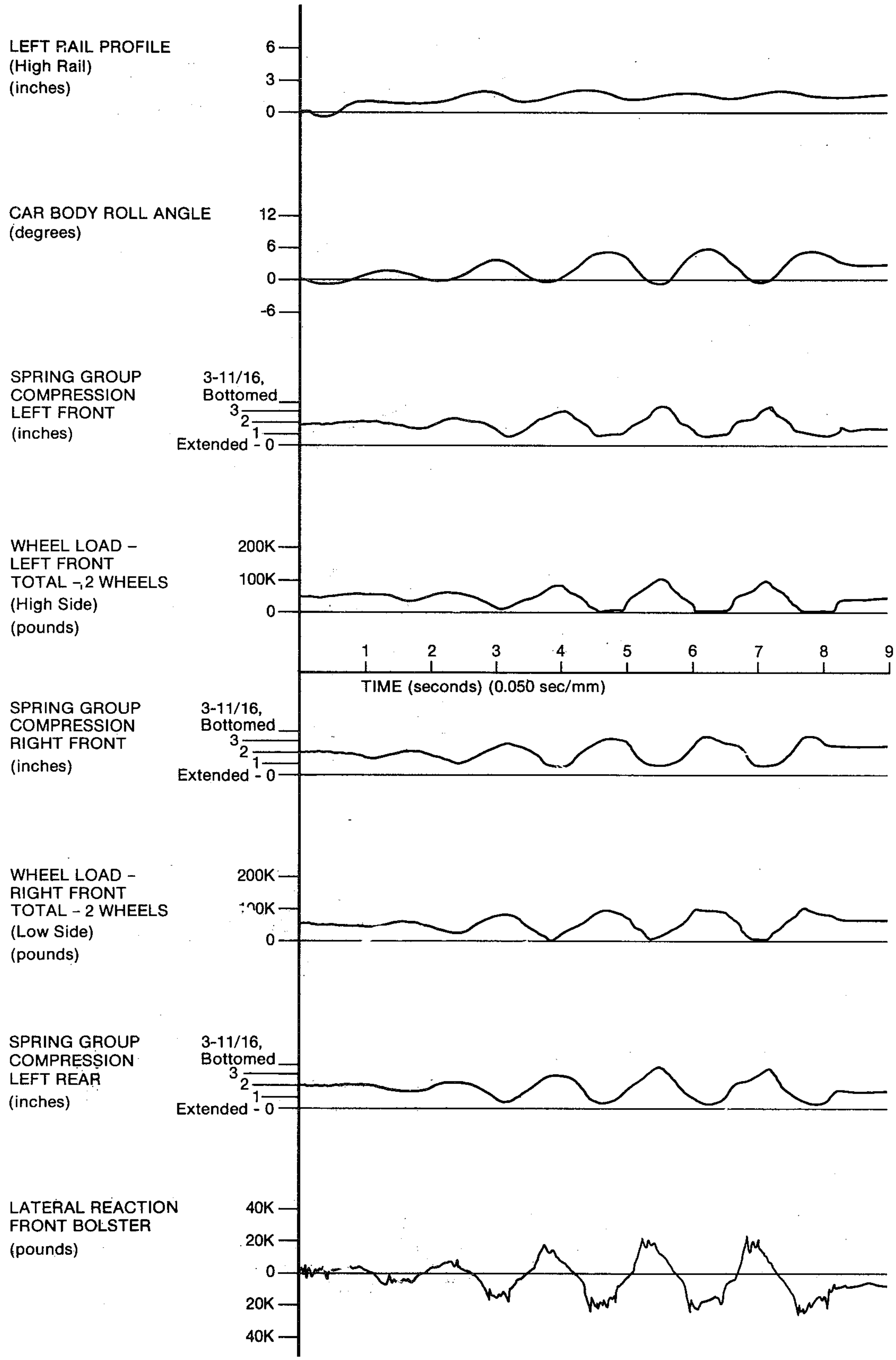
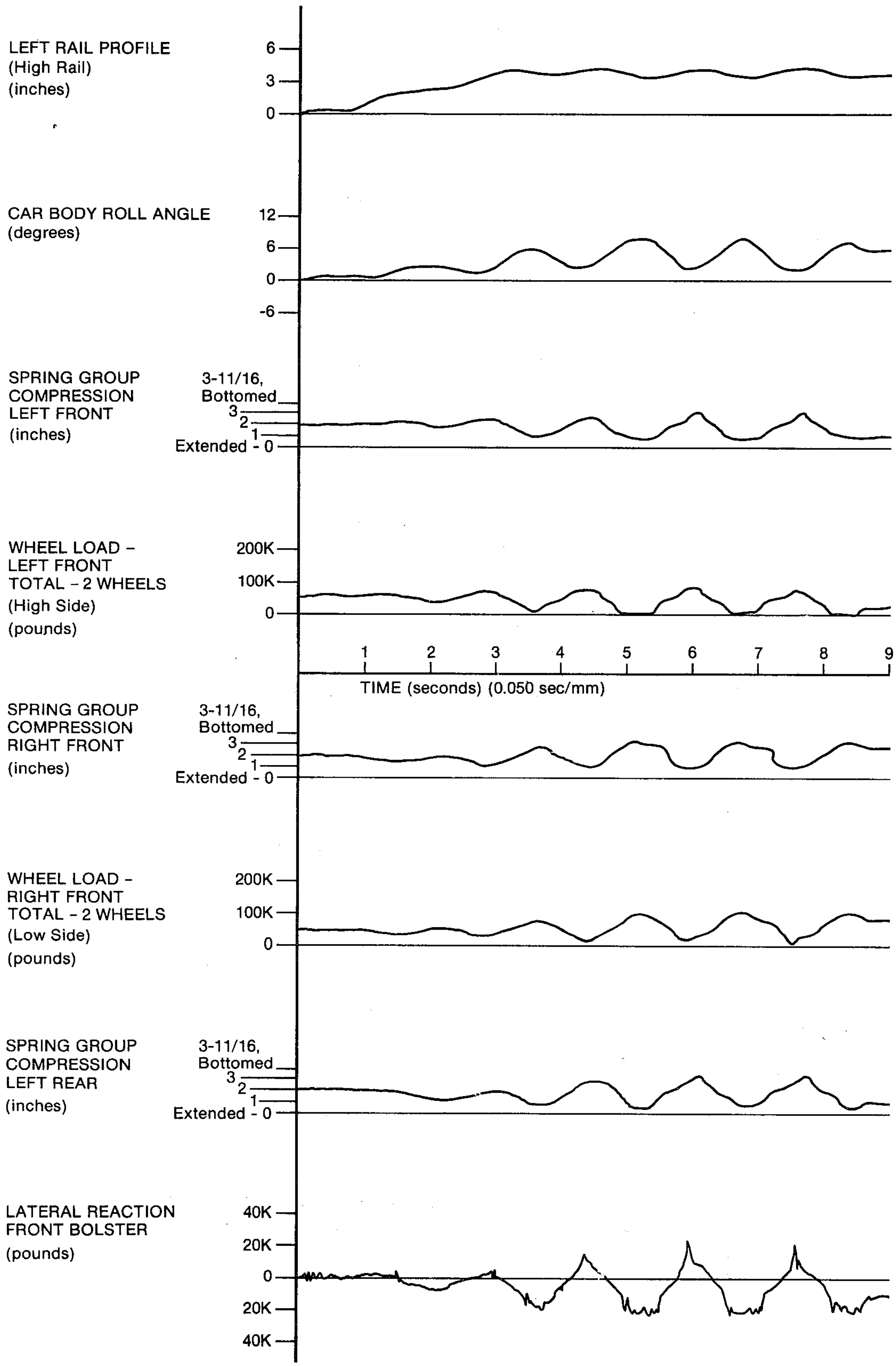


Figure 31
70 Ton Box Car
Symmetric Load
Tangent Track
18 mph



RUN NO. 2089

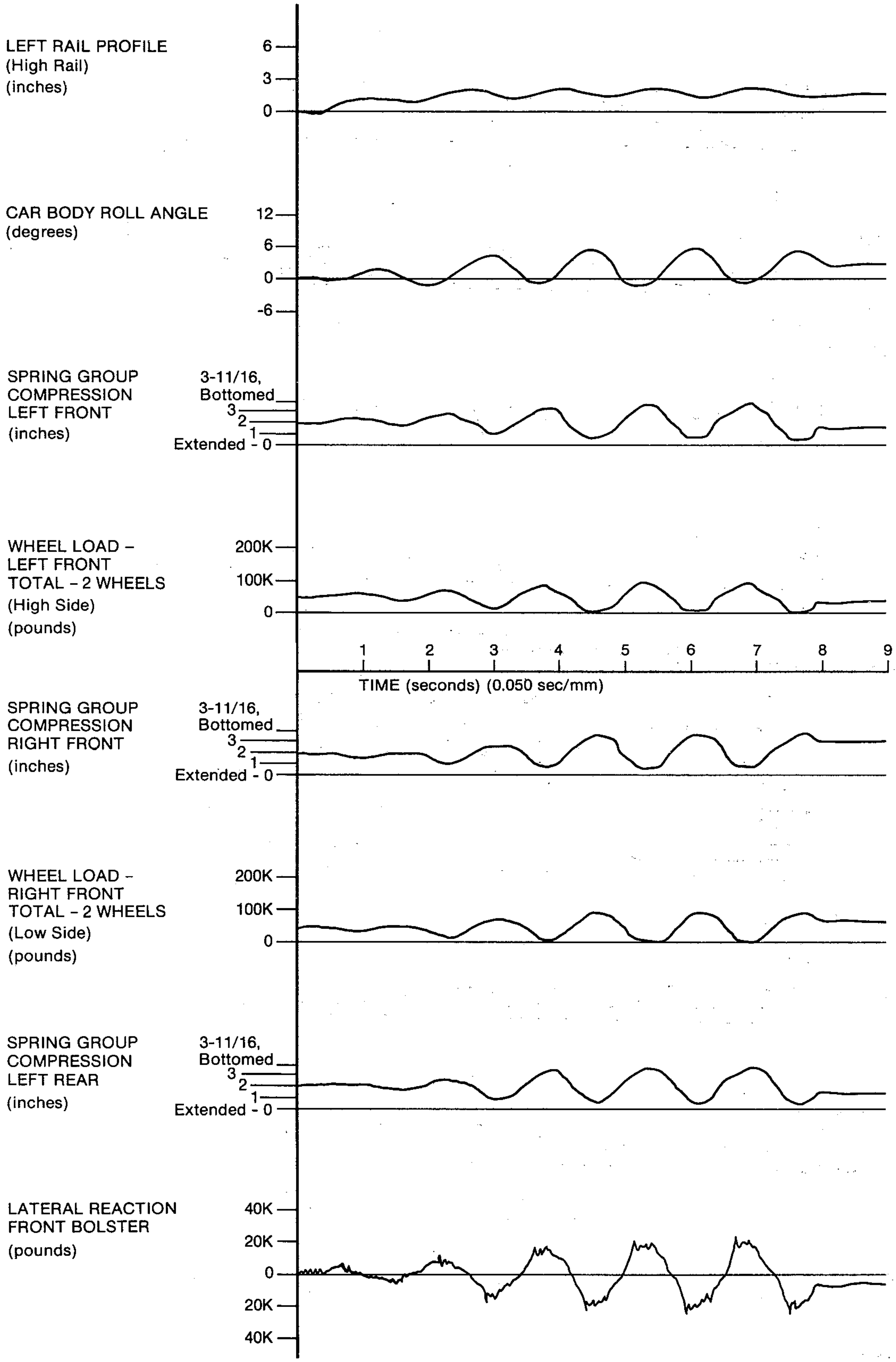
Figure 32
70 Ton Box Car
Symmetric Load
3° Curve
2" Superelevation
18 mph



RUN NO. 2090

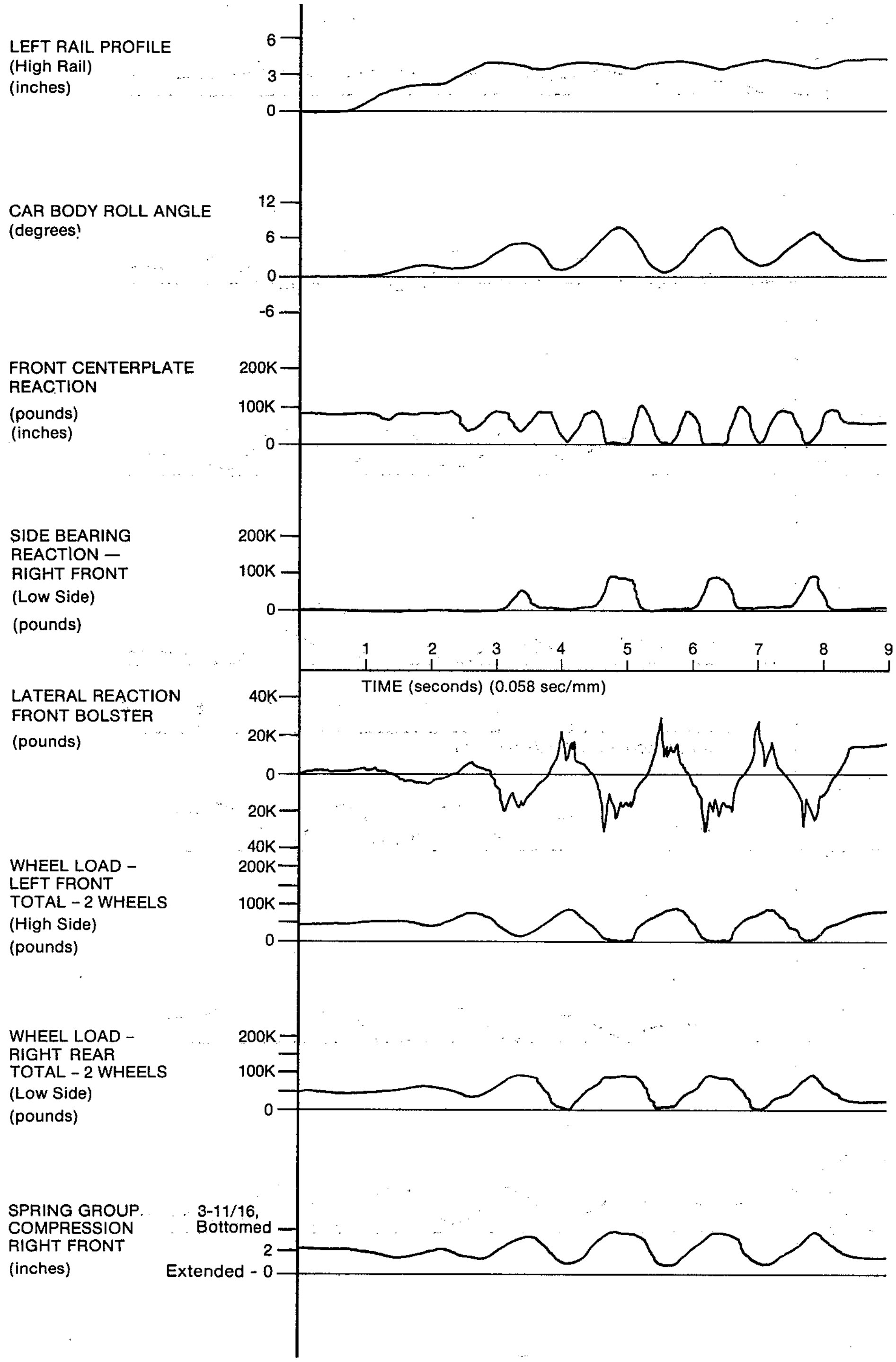
Figure 33

70 Ton Box Car
Symmetric Load
3° Curve
4″ Superelevation
18 mph



RUN NO. 2093

Figure 34
70 Ton Box Car
Symmetric Load
9° Curve
2" Superelevation
18 mph



RUN NO. 1982

Figure 35
70 Ton Box Car
Symmetric Load
9° Curve
4" Superelevation
19 mph

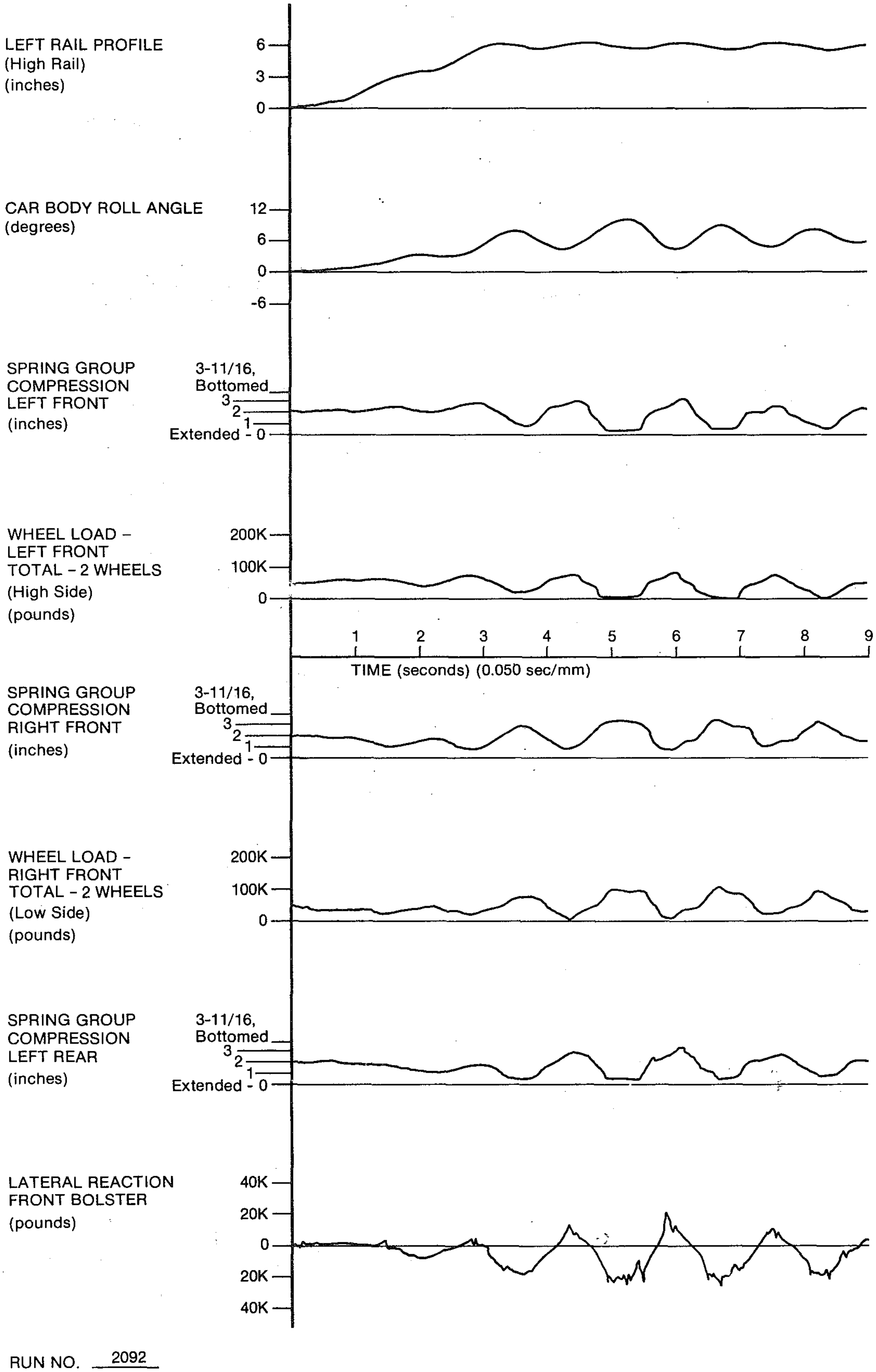
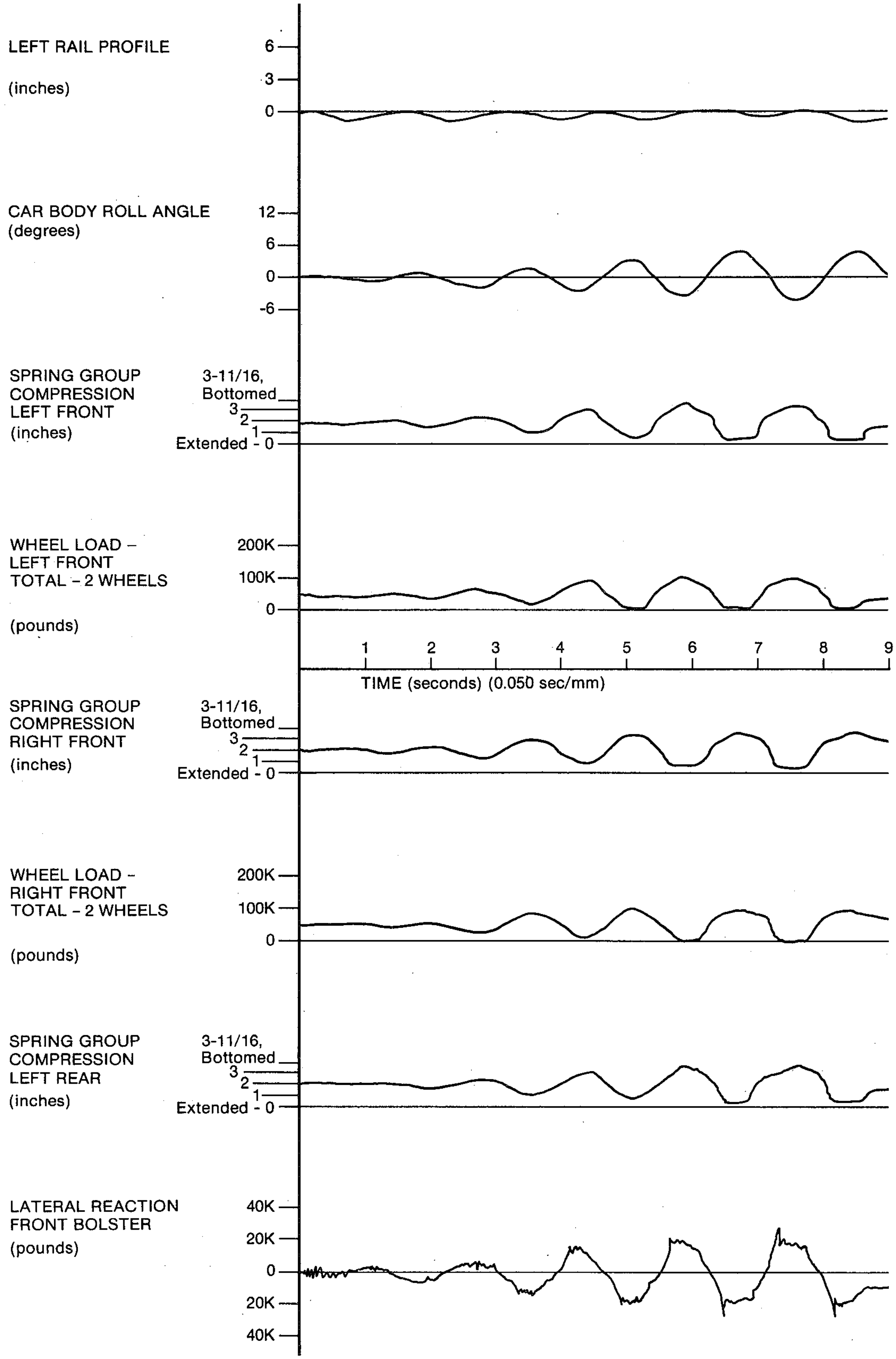


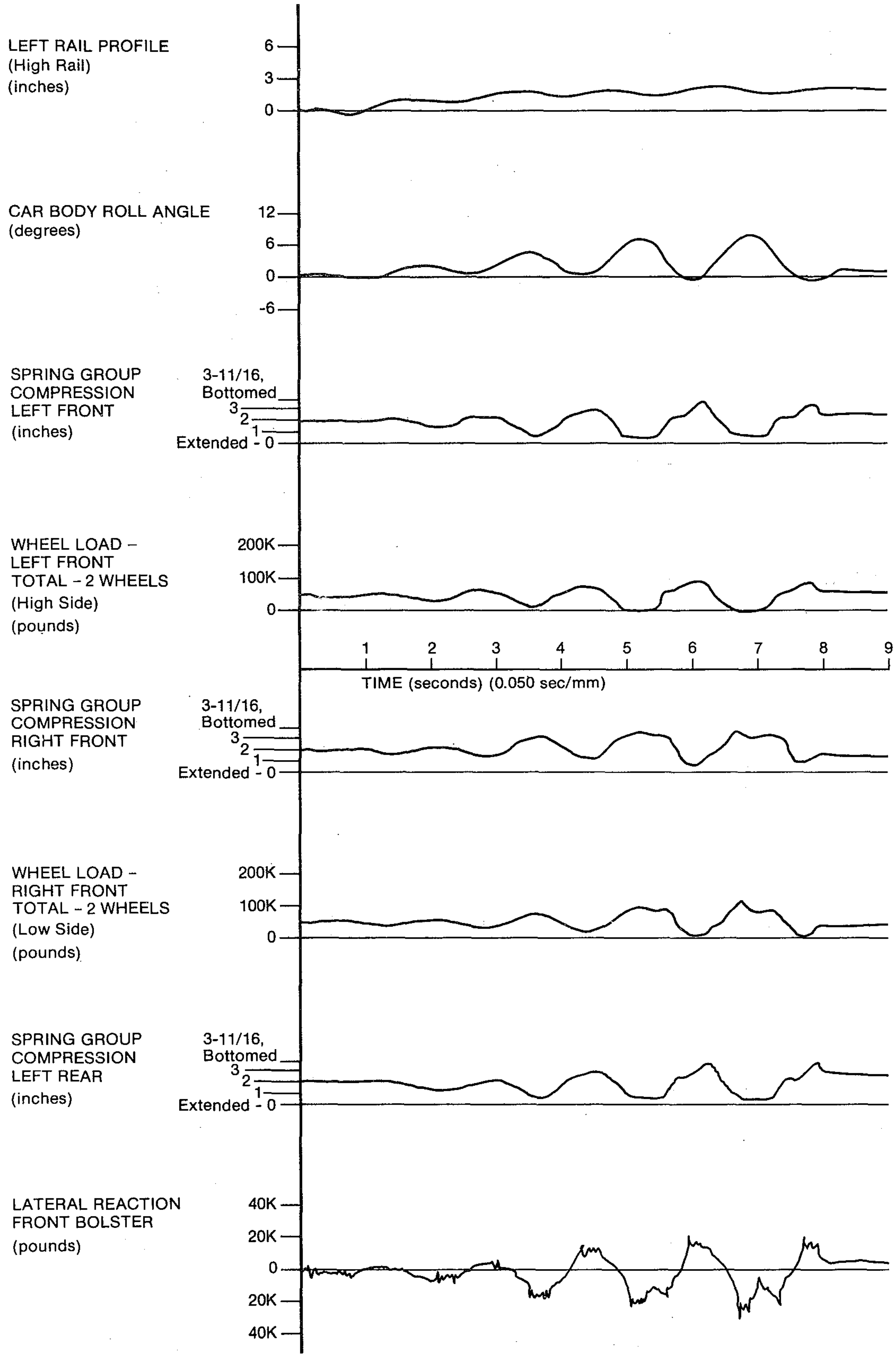
Figure 36
70 Ton Box Car
Symmetric Load
9° Curve
6″ Superelevation
18 mph



RUN NO. 2099

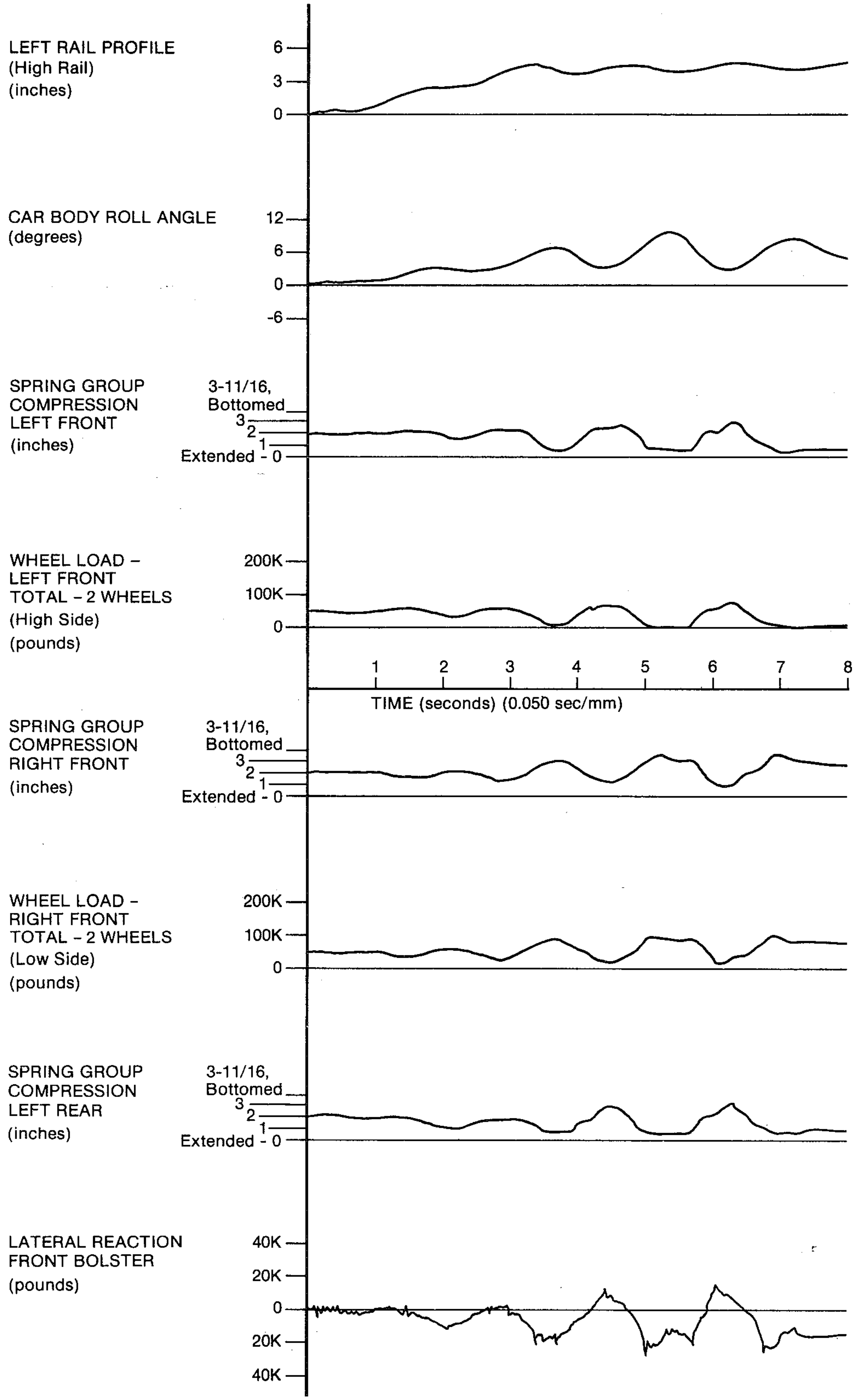
Figure 37

70 Ton Box Car
3" Eccentric Load
Tangent Track
18 mph



RUN NO. 2097

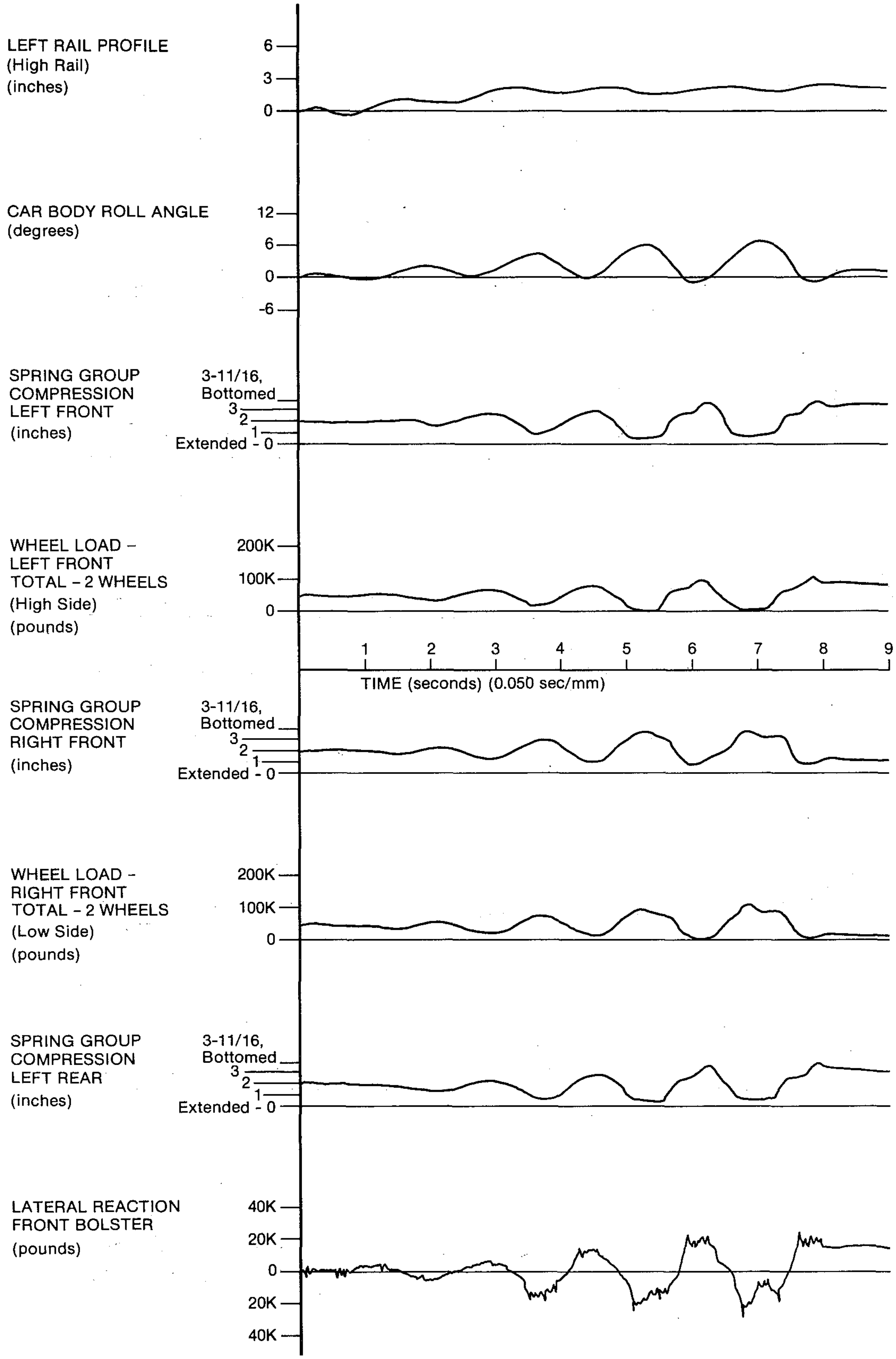
Figure 38
70 Ton Box Car
3″ Eccentric Load
3° Curve
2″ Superelevation
17 mph



RUN NO. 2098

Figure 39

70 Ton Box Car
3" Eccentric Load
3° Curve
4" Superelevation
17 mph



RUN NO. 2096

Figure 40
70 Ton Box Car
3" Eccentric Load
9° Curve
2" Superelevation
17 mph

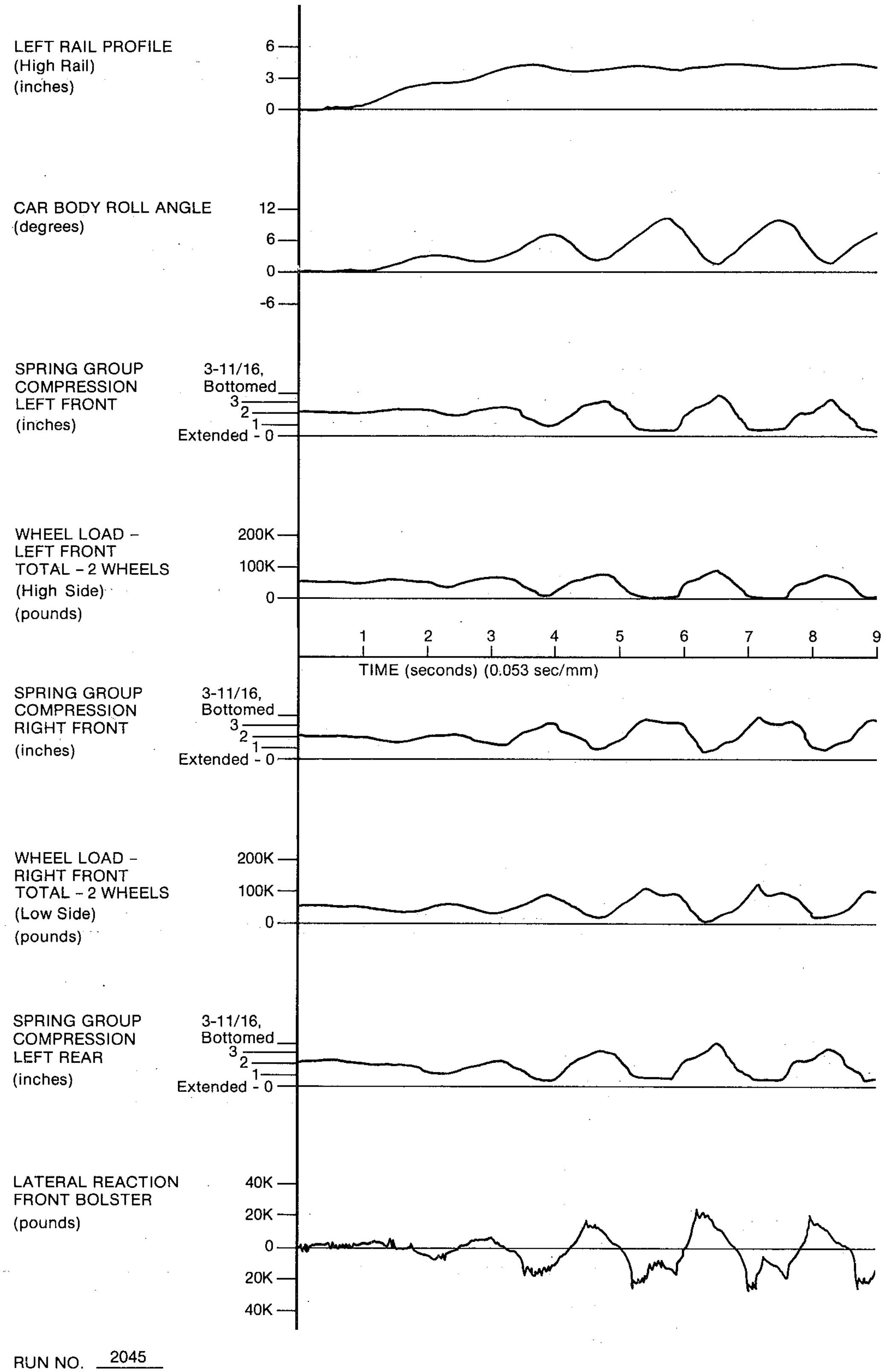
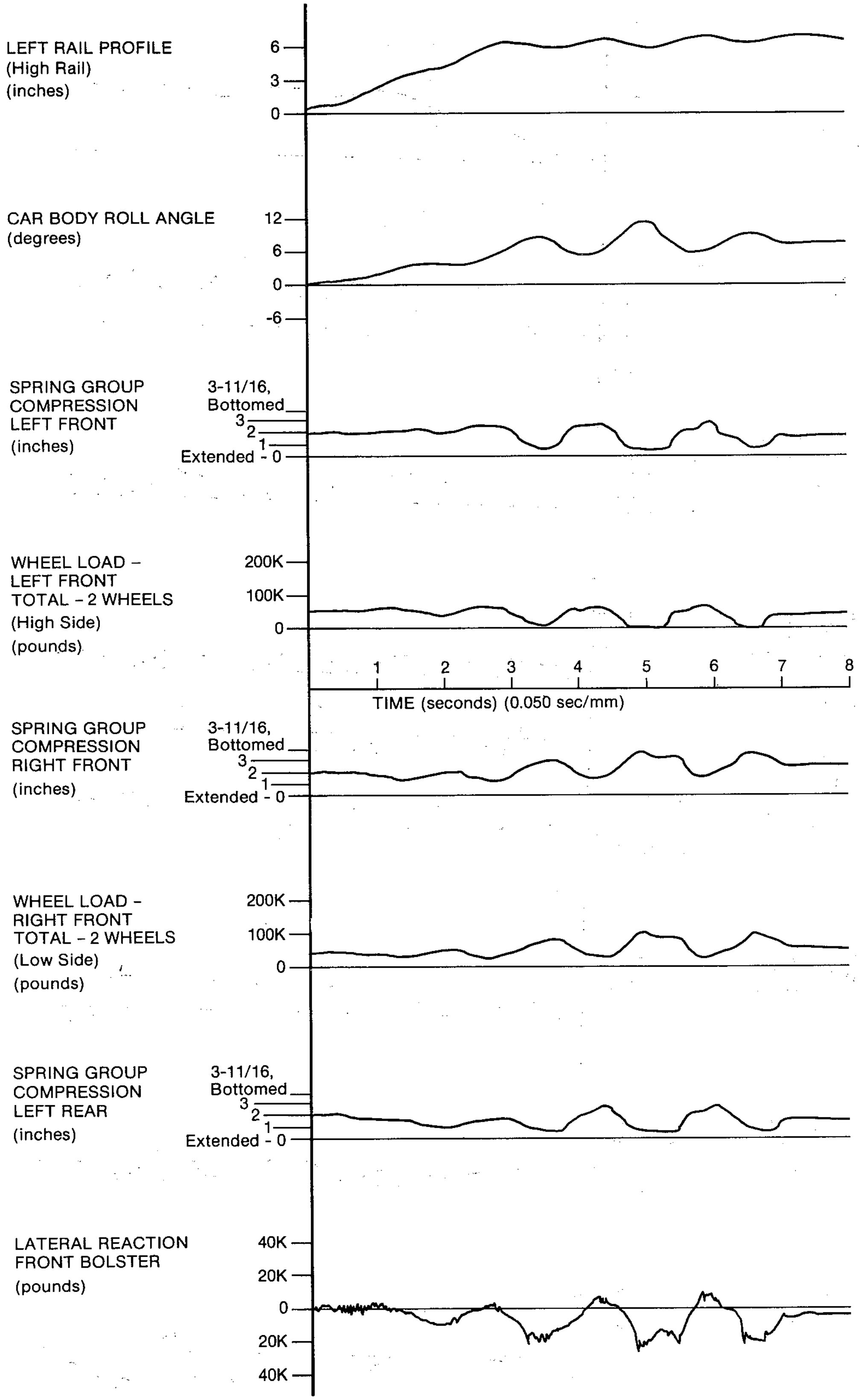


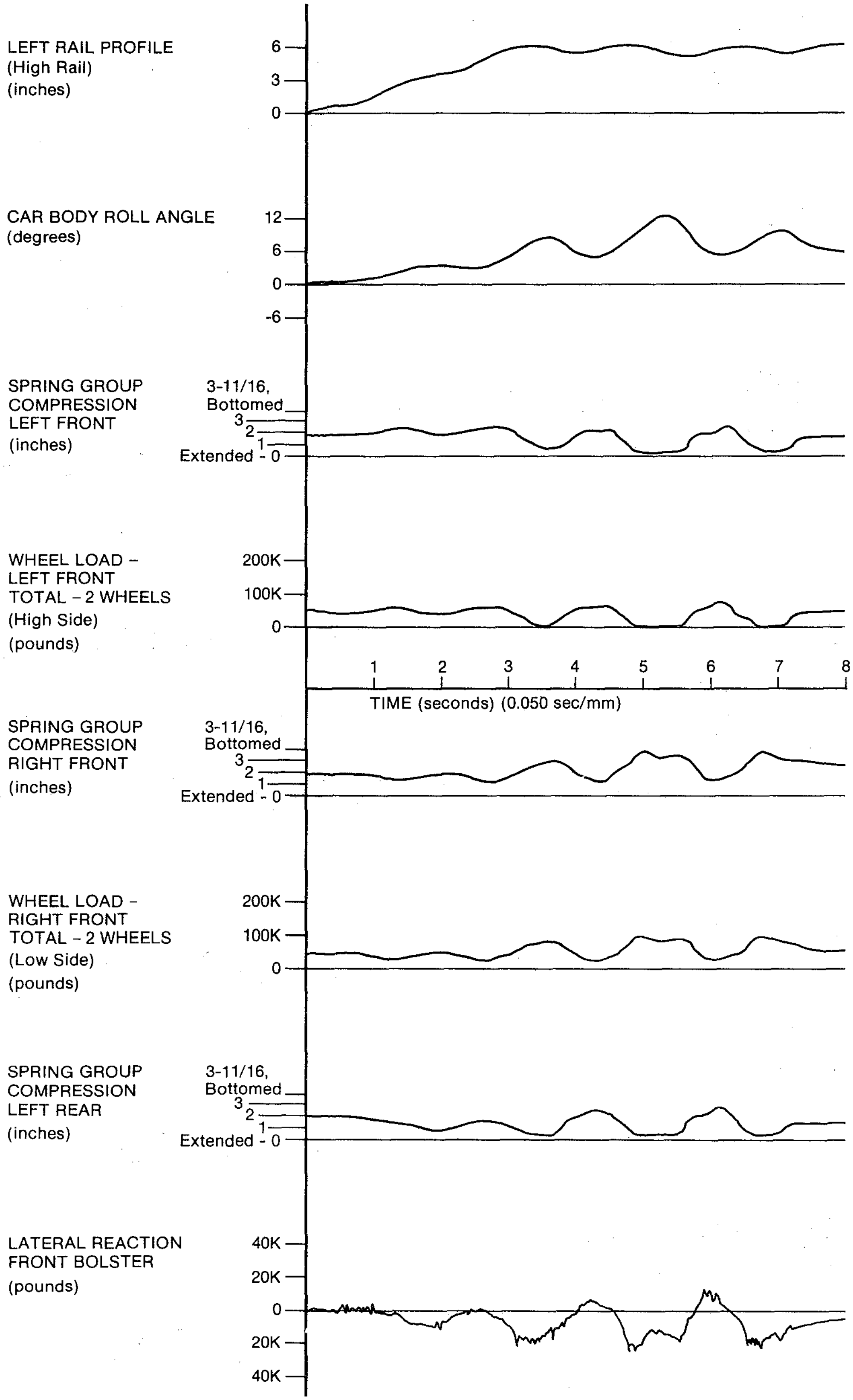
Figure 41

70 Ton Box Car
3" Eccentric Load
9° Curve
4" Superelevation
17 mph



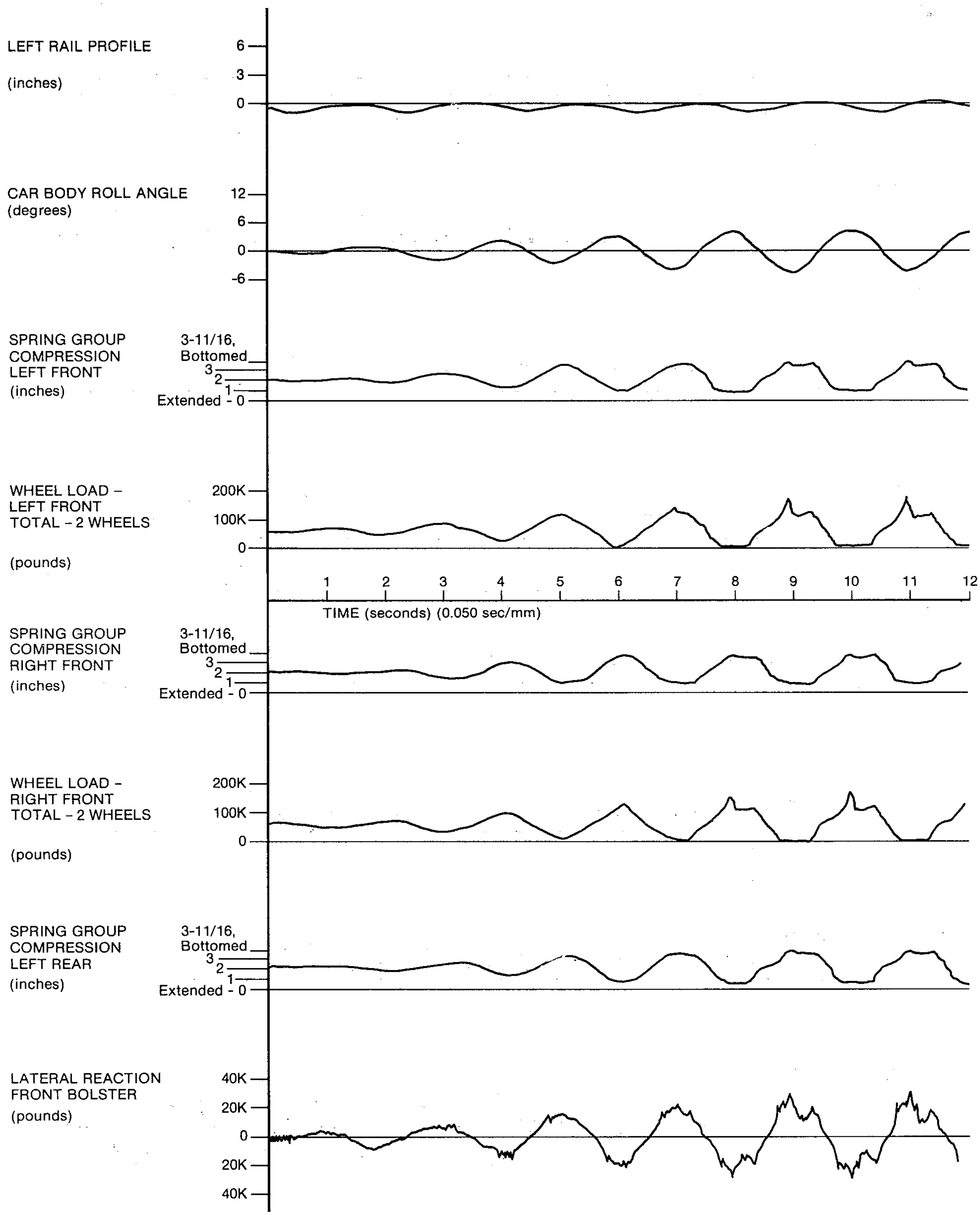
RUN NO. 2095

Figure 42
70 Ton Box Car
3" Eccentric Load
9° Curve
6" Superelevation
18 mph



RUN NO. 2100

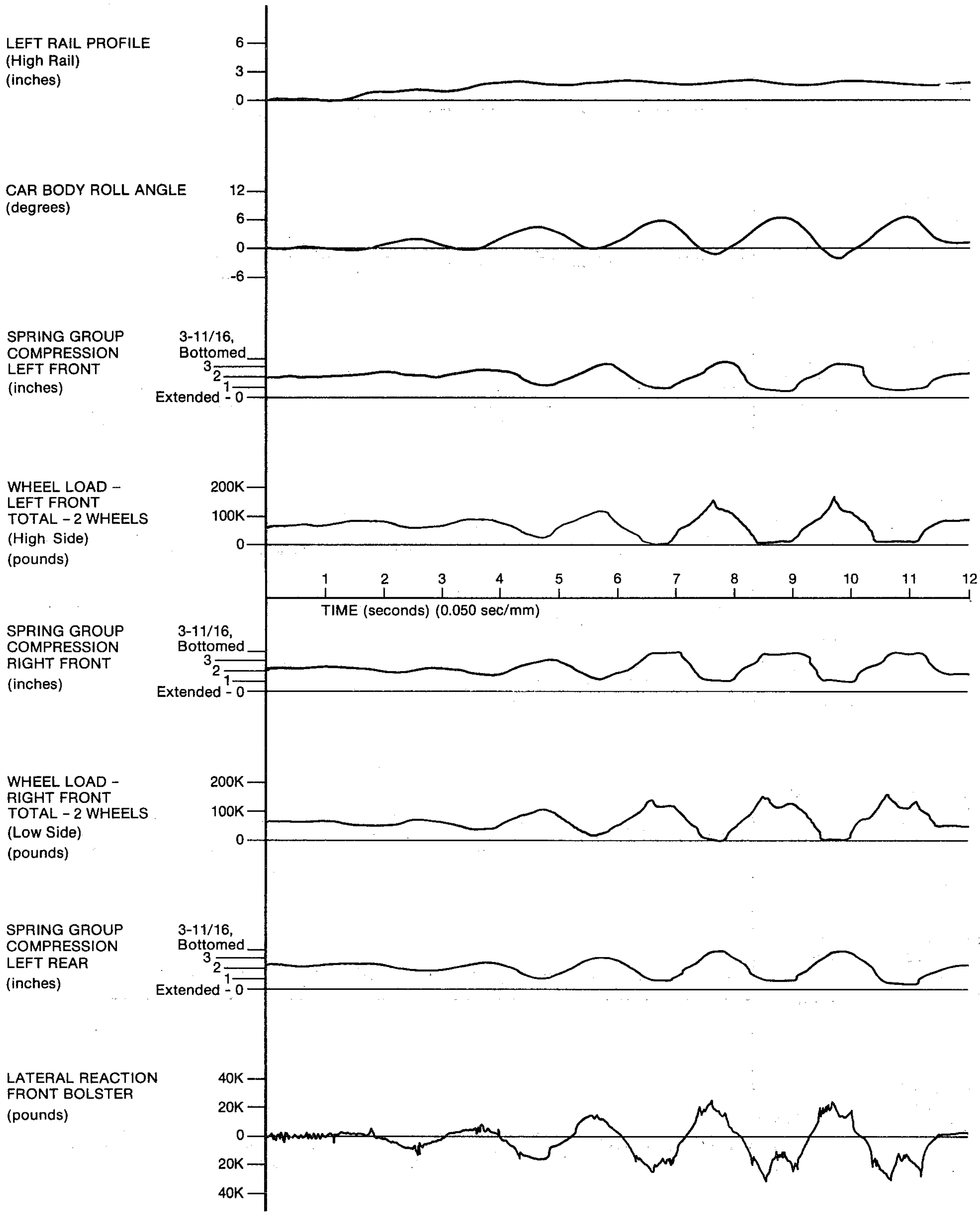
Figure 43
70 Ton Box Car
3" Eccentric Load
9° Curve
6" Superelevation
17 mph



RUN NO. 2087

Figure 44

100 Ton Box Car
Symmetric Load
Tangent Track
13.5 mph



RUN NO. 2080

Figure 45
100 Ton Box Car
Symmetric Load
3° Curve
2" Superelevation
13.5 mph

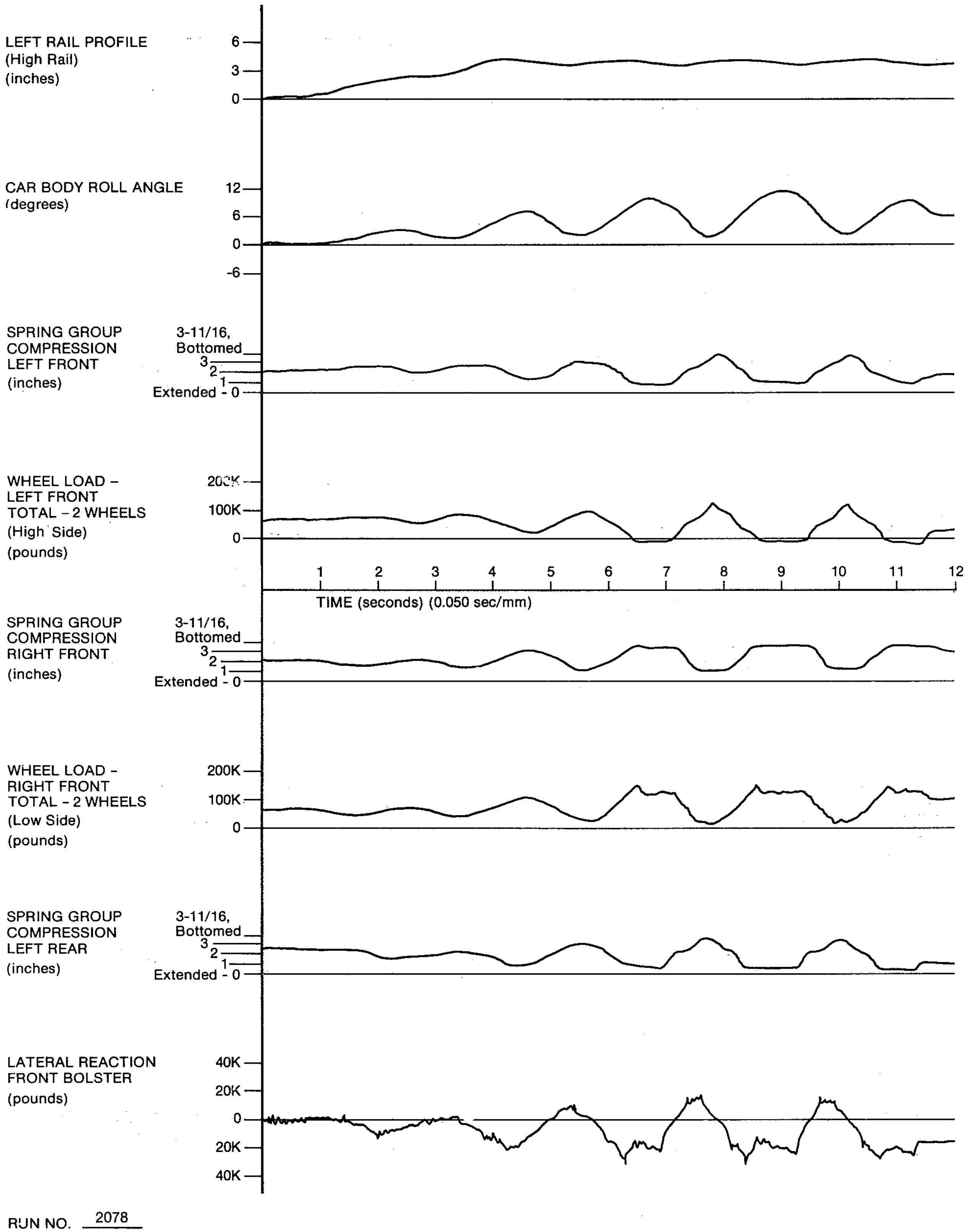
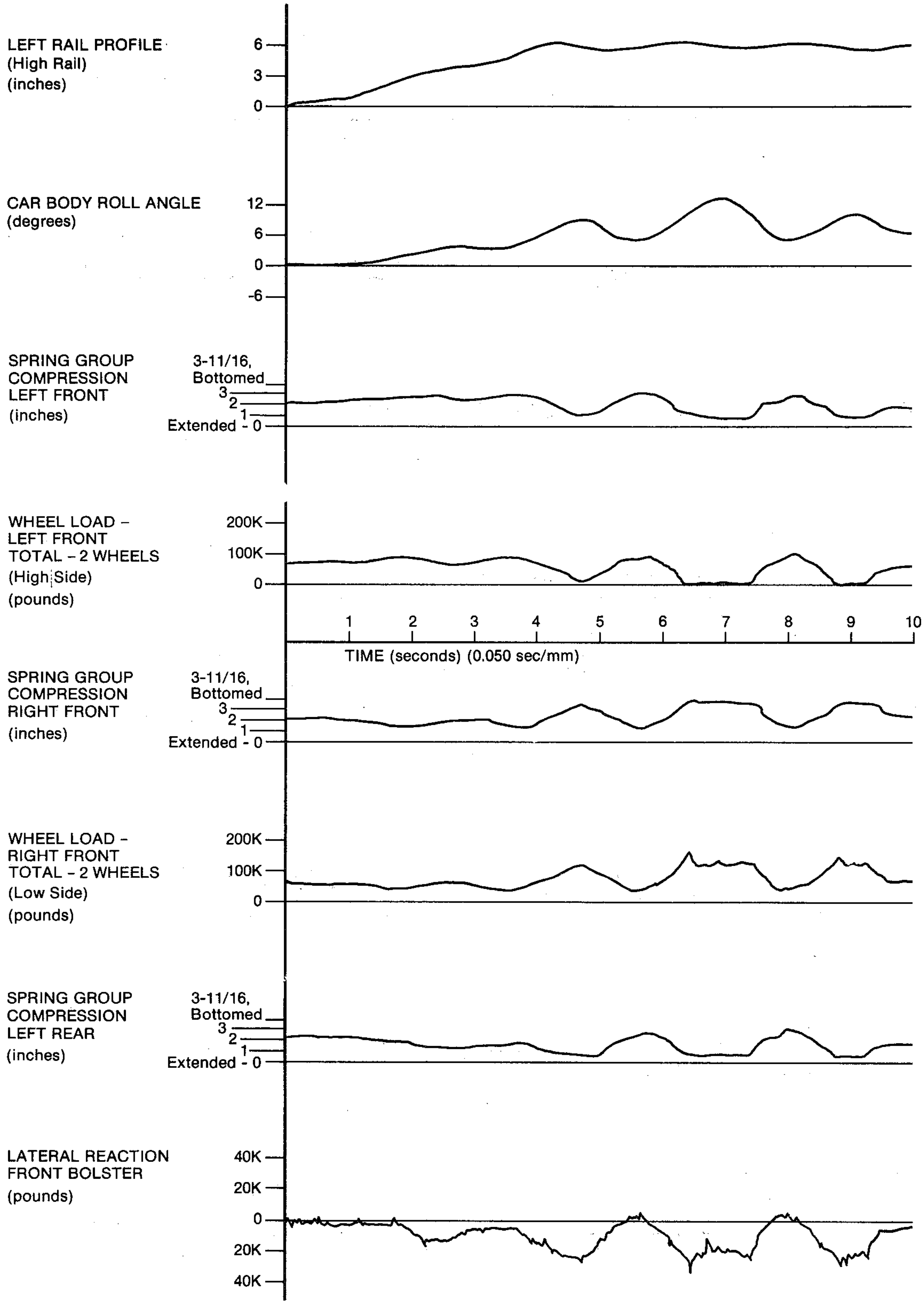


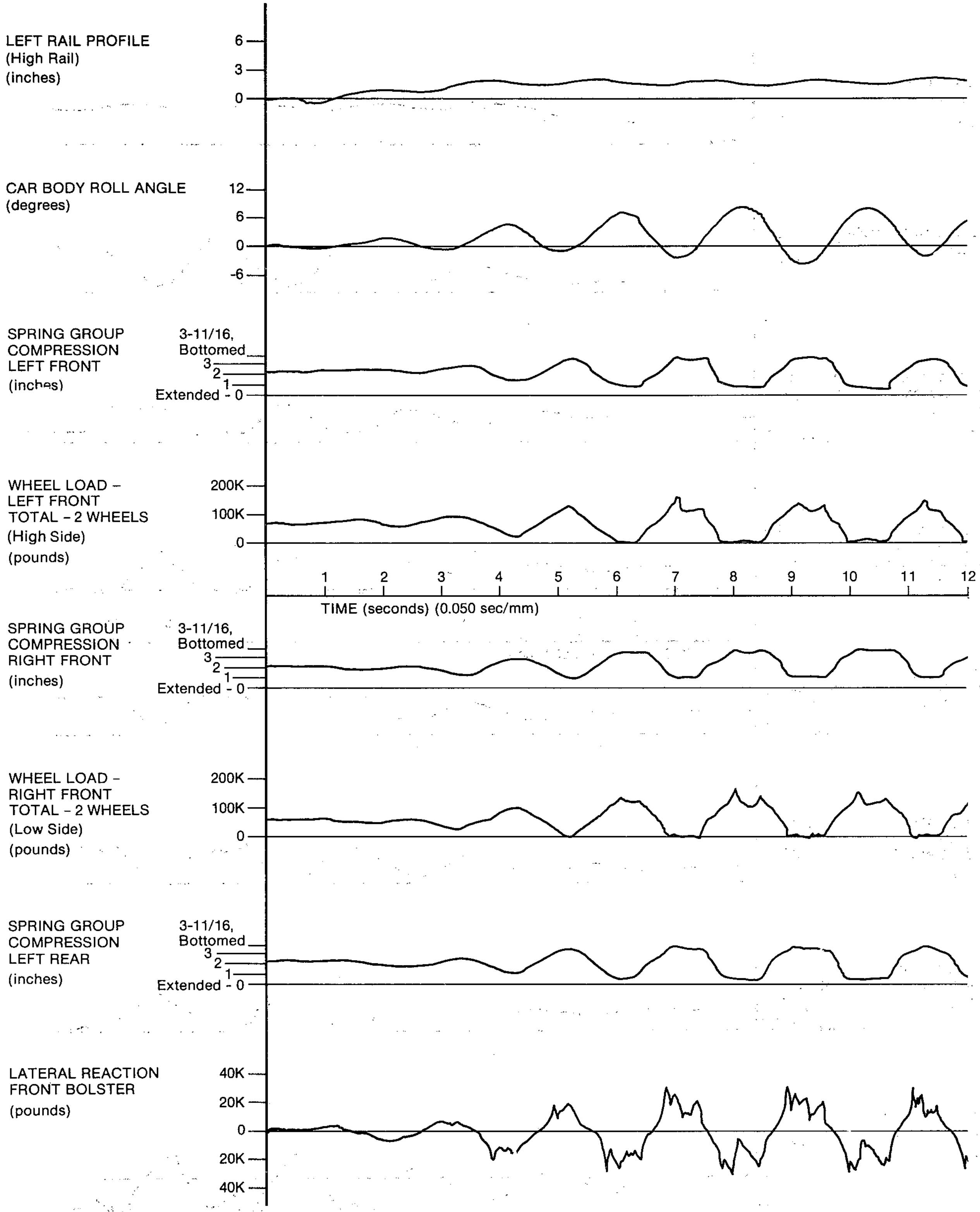
Figure 46
100 Ton Box Car
Symmetric Load
3° Curve
4" Superelevation
13 mph



RUN NO. 2079

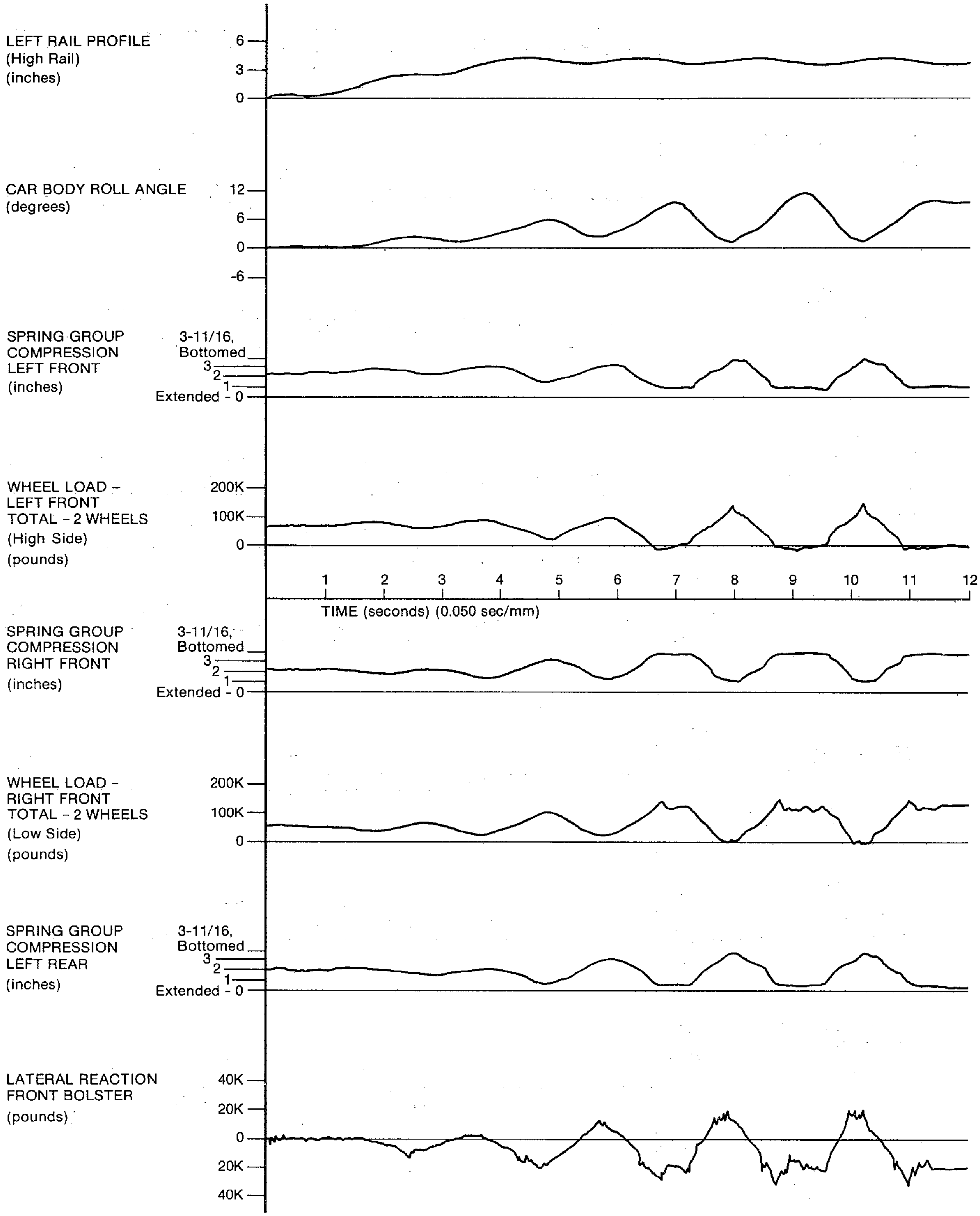
Figure 47

100 Ton Box Car
Symmetric Load
3° Curve
6″ Superelevation
13.5 mph



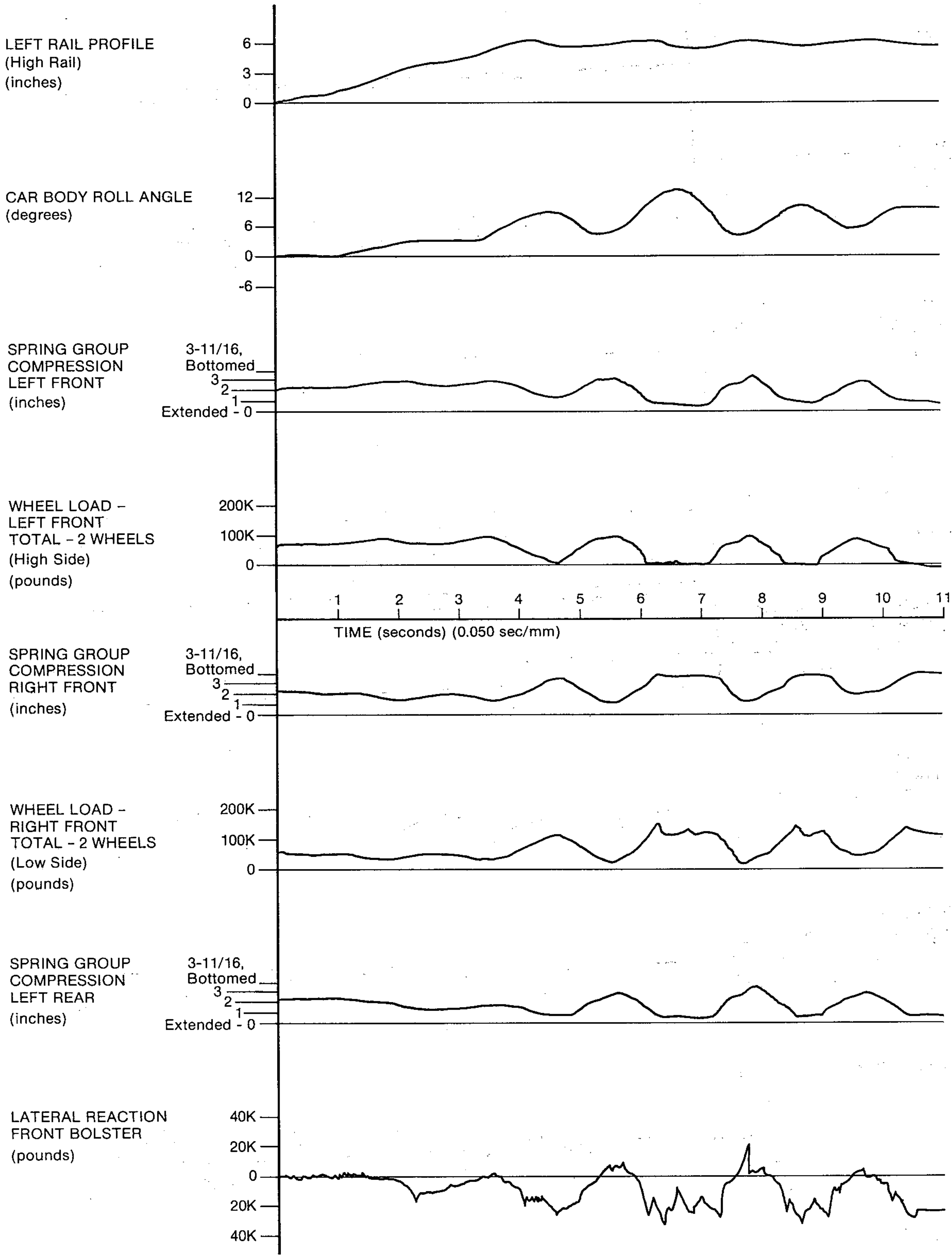
RUN NO. 2073

Figure 48
100 Ton Box Car
Symmetric Load
9° Curve
2″ Superelevation
14 mph



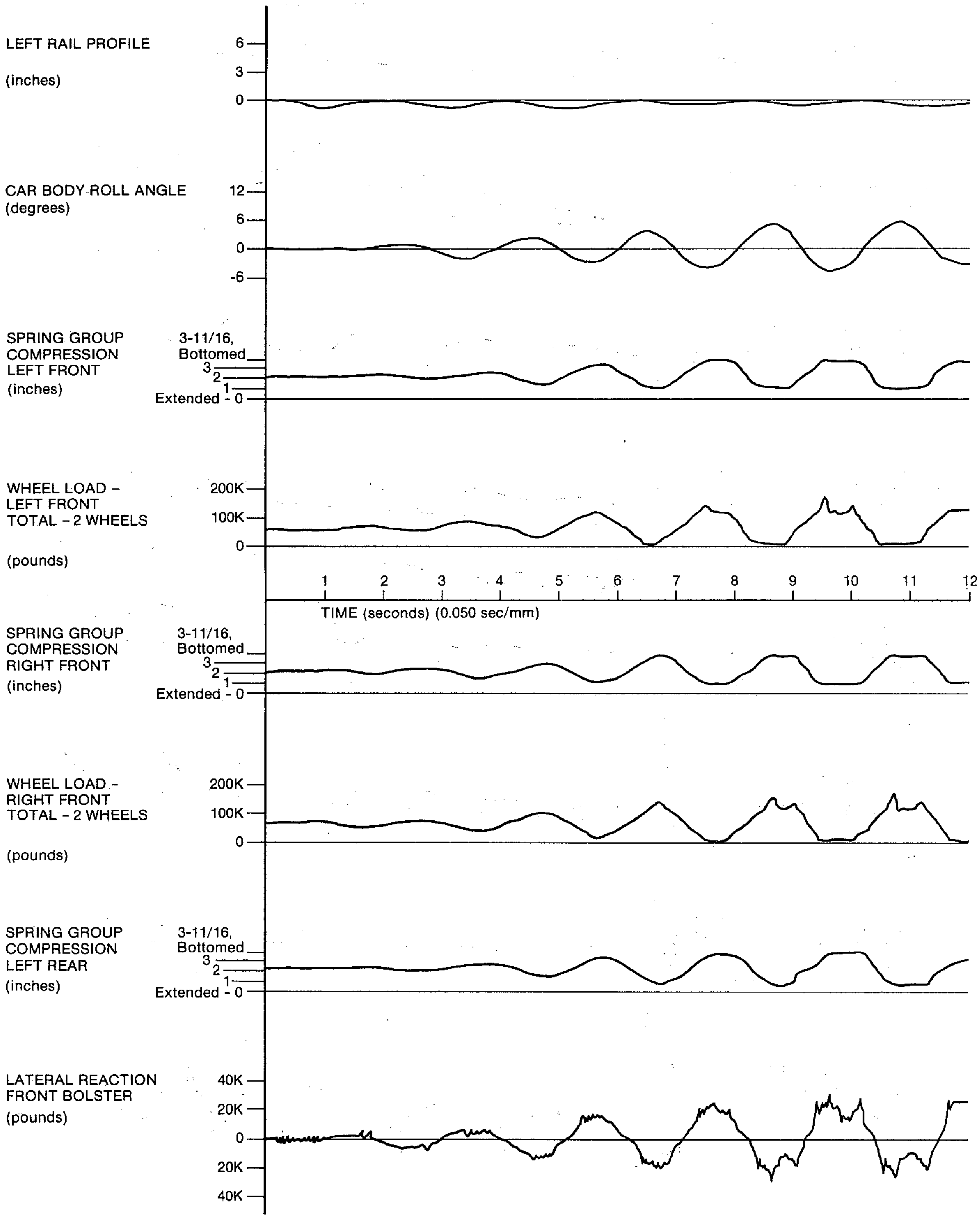
RUN NO. 2077

Figure 49
100 Ton Box Car
Symmetric Load
9° Curve
4" Superelevation
13 mph



RUN NO. 2076

Figure 50
100 Ton Box Car
Symmetric Load
9° Curve
6" Superelevation
14 mph



RUN NO. 2083

Figure 51
100 Ton Box Car
3" Eccentric Load
Tangent Track
13 mph

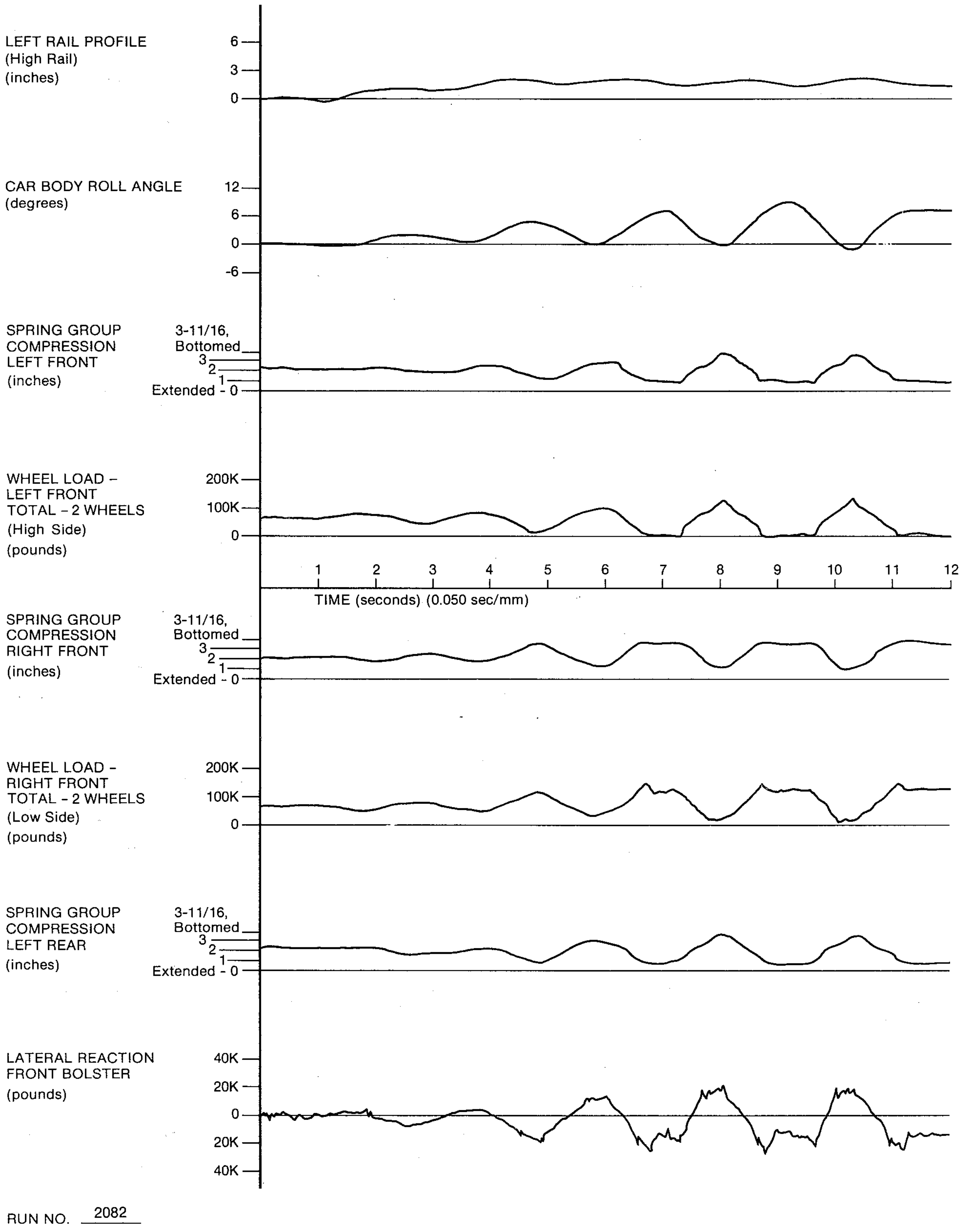
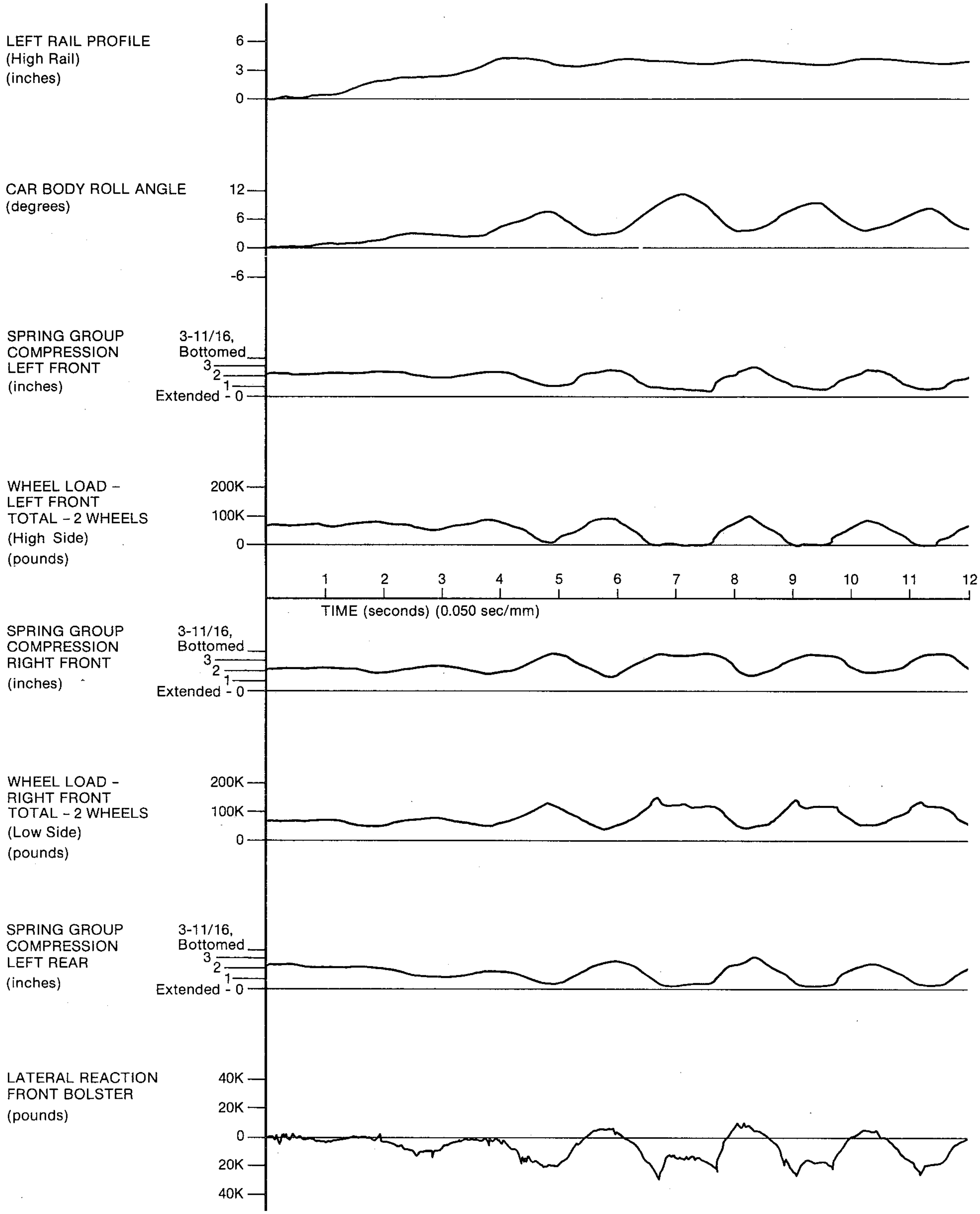


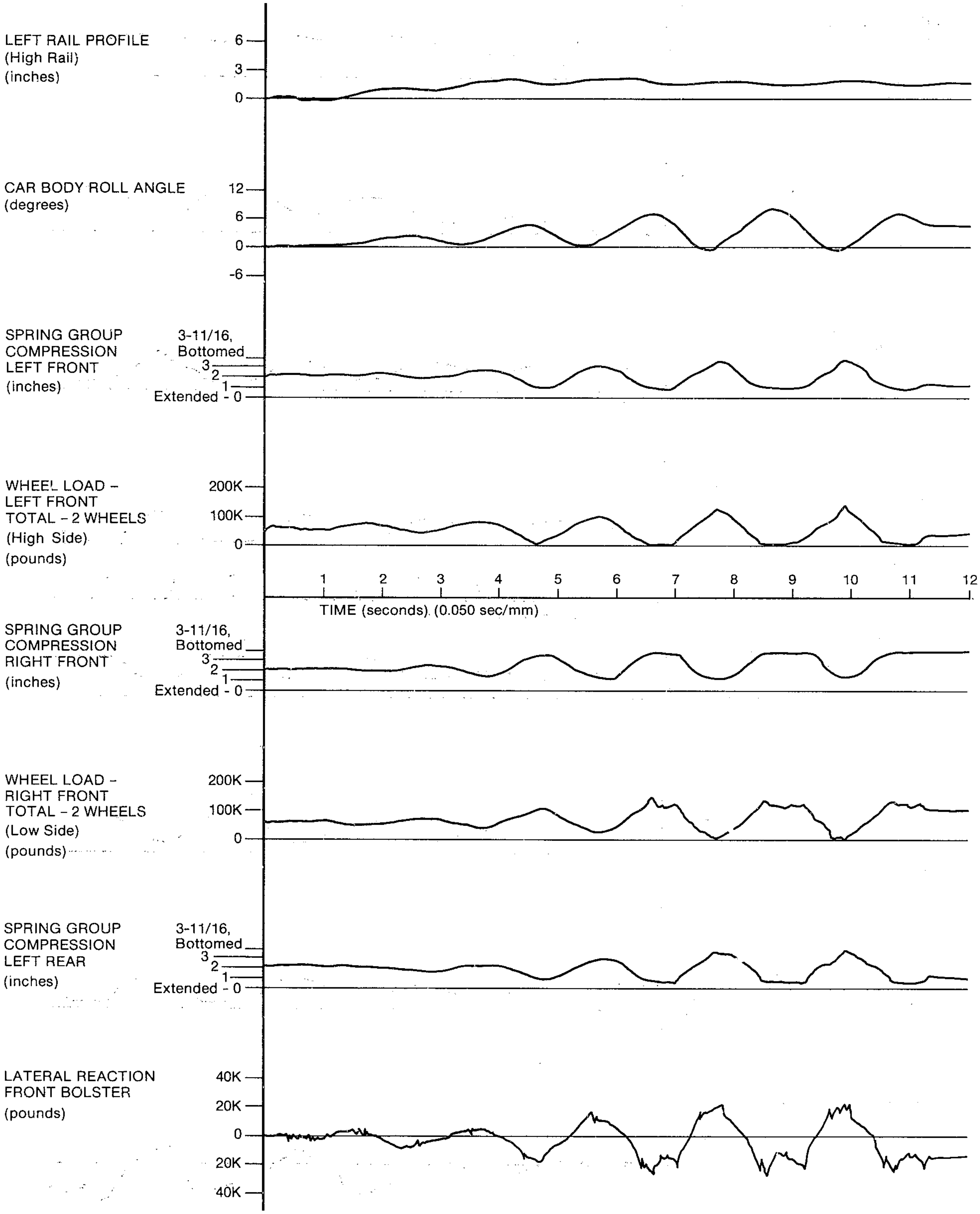
Figure 52
100 Ton Box Car
3″ Eccentric Load
3° Curve
2″ Superelevation
13 mph



RUN NO. 2081

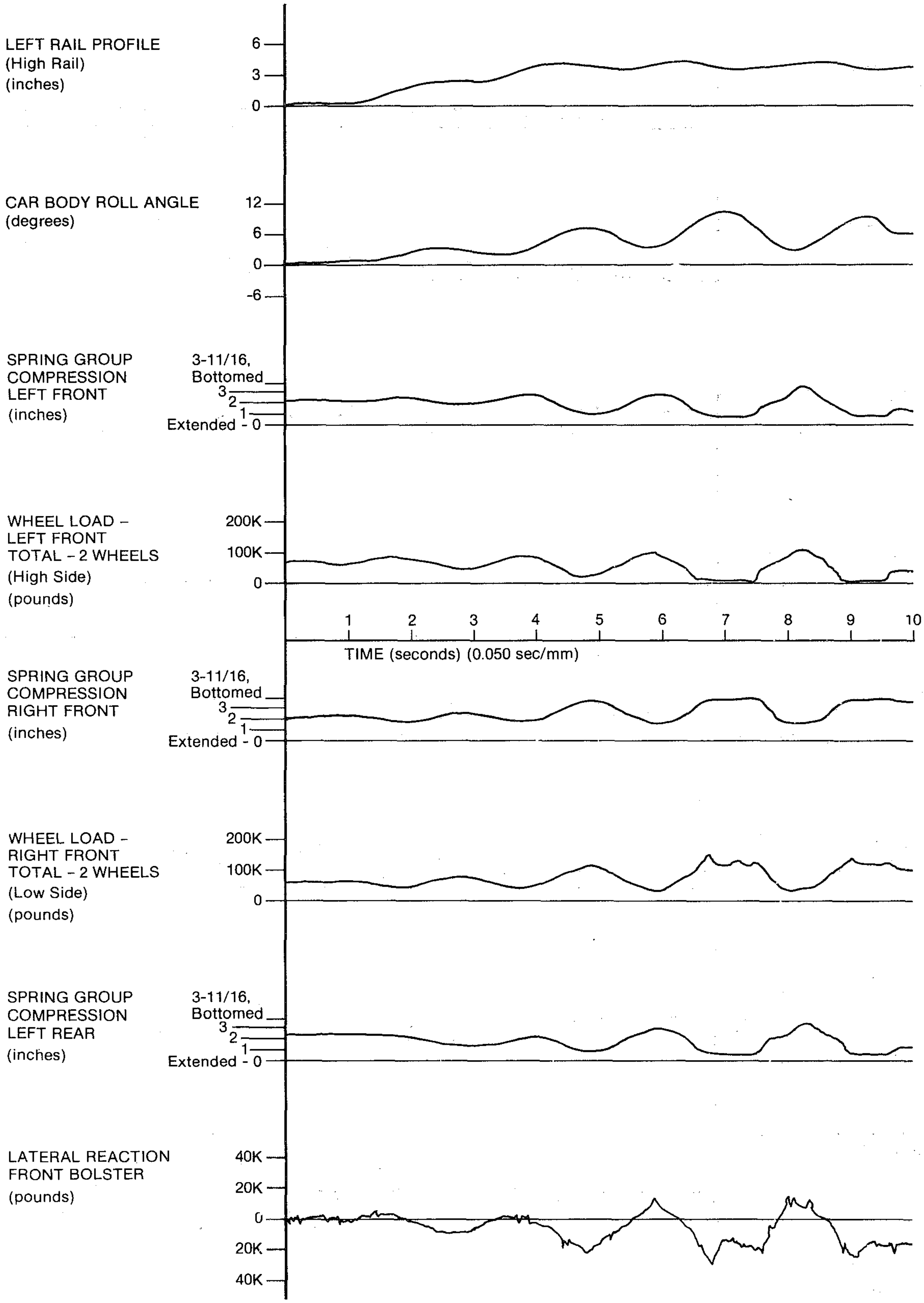
Figure 53

100 Ton Box Car
3" Eccentric Load
3° Curve
4" Superelevation
13 mph



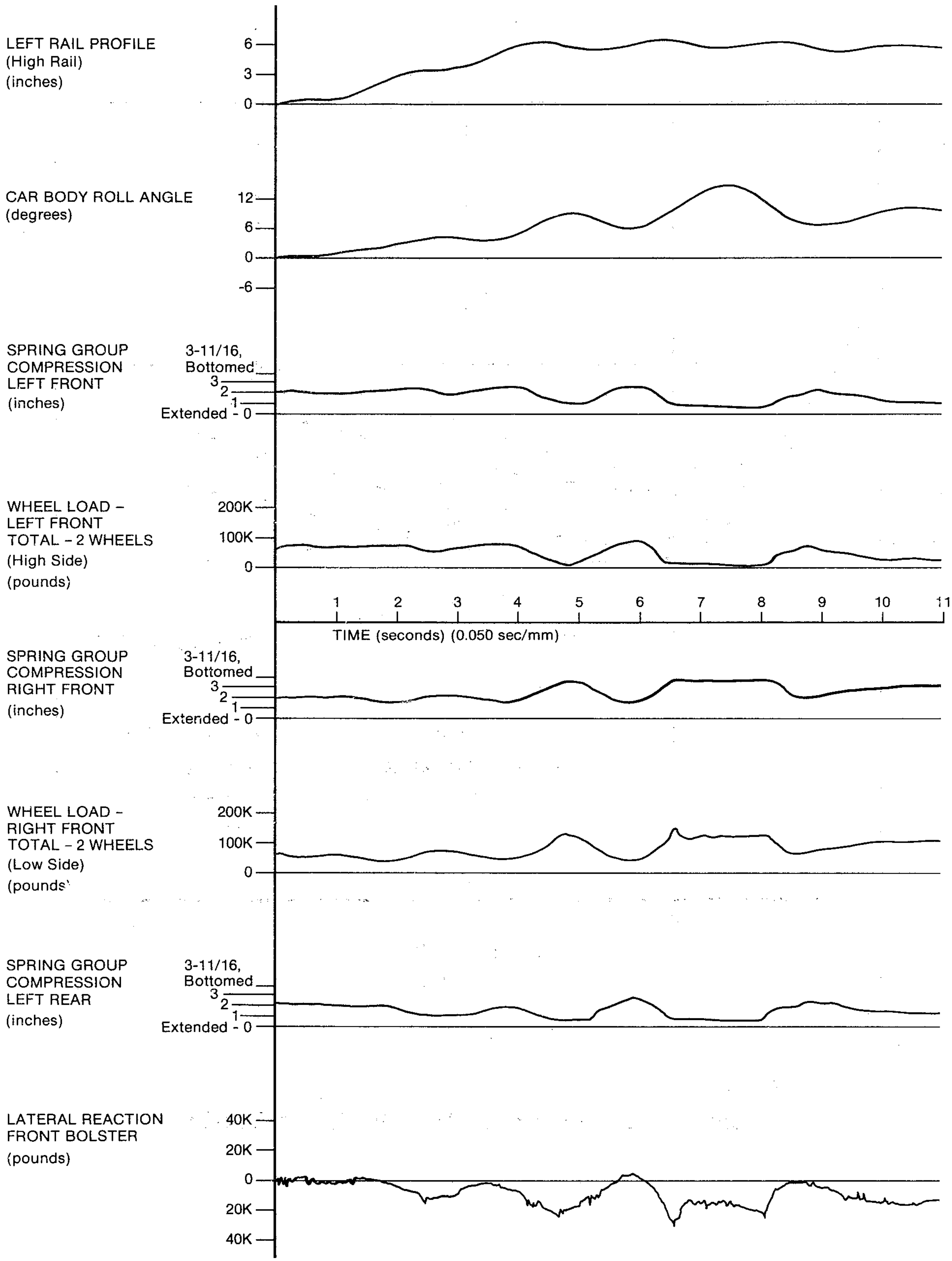
RUN NO. 2086

Figure 54
100 Ton Box Car
3" Eccentric Load
9° Curve
2" Superelevation
13.5 mph



RUN NO. 2084

Figure 55
100 Ton Box Car
3" Eccentric Load
9° Curve
4" Superelevation
13.5 mph



RUN NO. 2085

Figure 56
100 Ton Box Car
3" Eccentric Load
9° Curve
4" Superelevation
13.5 mph

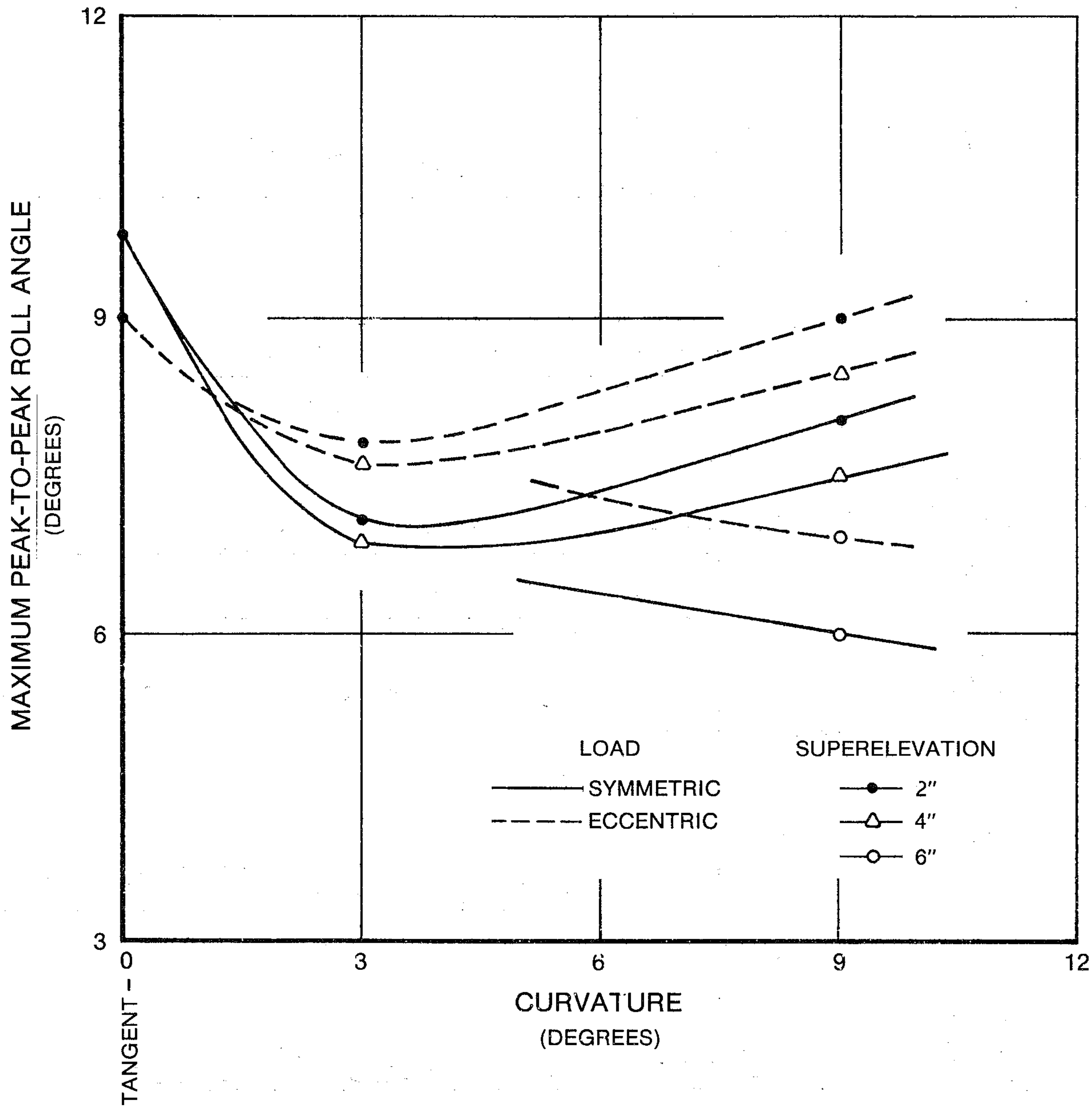


Figure 57
Maximum Peak-to-Peak Roll Angle for 70 Ton Box Car

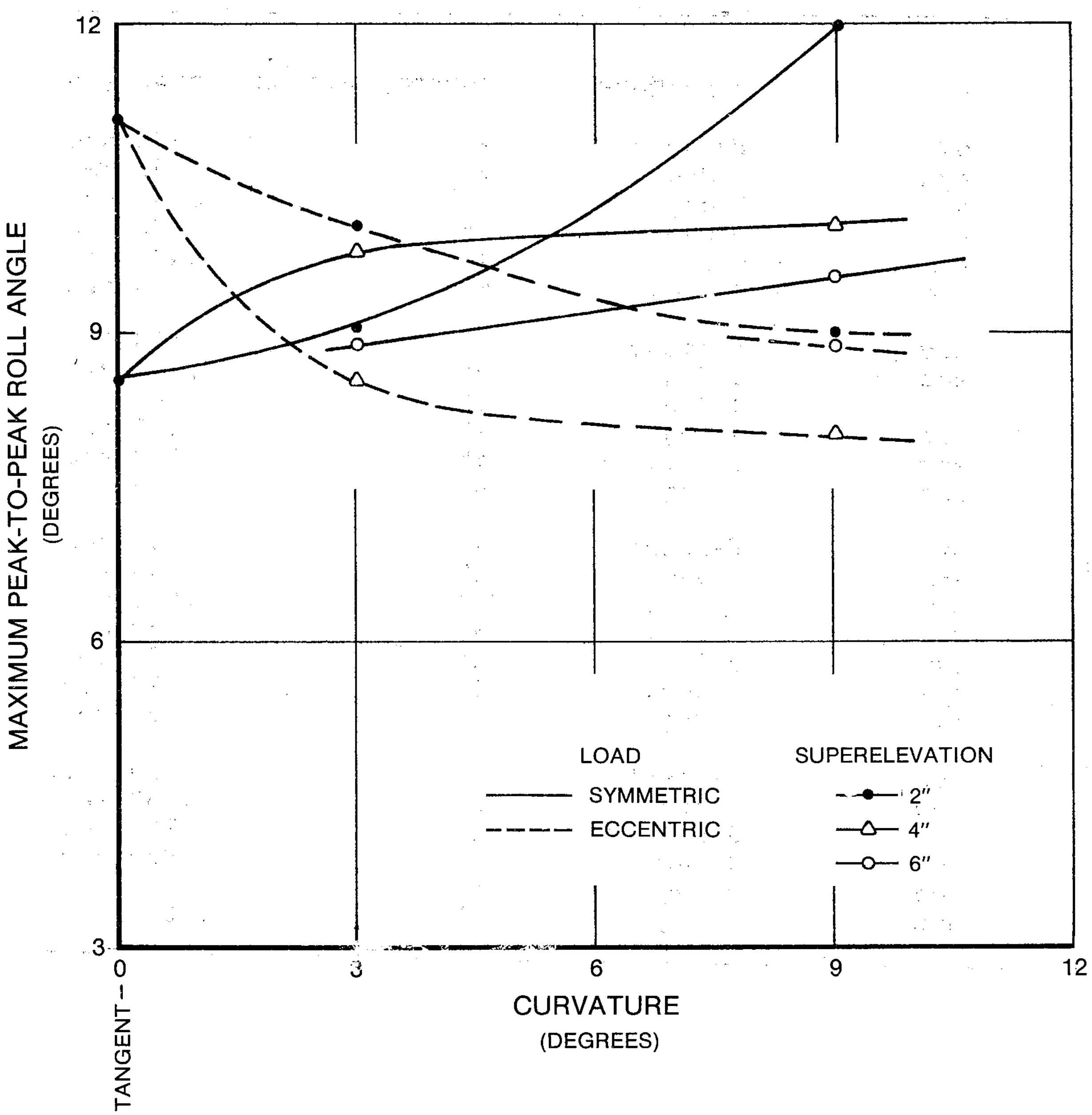


Figure 58
Maximum Peak-to-Peak Roll Angle for 100 Ton Box Car

Table 8. Duration of Zero Wheel Loads for Freight Car Simulation Series (At Critical Speed)

CAR	LOAD	CURVATURE	SUPERELEVATION	DURATION
70 T	sym	Tangent	-	.450 sec.
70 T	sym	3 deg.	2"	.475 sec.
70 T	sym	3 deg.	4"	.500 sec
70 T	sym	9 deg.	2"	.350 sec.
70 T	sym	9 deg.	4"	.435 sec.
70 T	sym	9 deg.	6"	.550 sec.
70 T	ecc	Tangent	-	.475 sec.
70 T	ecc	3 deg.	2"	.590 sec.
70 T	ecc	3 deg.	4"	.700 sec.
70 T	ecc	9 deg.	2"	.600 sec.
70 T	ecc	9 deg.	4"	.689 sec.
70 T	ecc	9 deg.	6"	.750 sec.
100 T	sym	Tangent	-	.600 sec.
100 T	sym	3 deg.	2"	.700 sec.
100 T	sym	3 deg.	4"	1.000 sec.
100 T	sym	3 deg.	6"	1.150 sec
100 T	sym	9 deg.	2"	.825 sec.
100 T	sym	9 deg.	4"	.975 sec.
100 T	sym	9 deg.	6"	1.100 sec
100 T	ecc	Tangent	-	.700 sec.
100 T	ecc	3 deg.	2"	.950 sec.
100 T	ecc	3 deg.	4"	1.150 sec.
100 T	ecc	9 deg.	2"	.800 sec.
100 T	ecc	9 deg.	4"	.950 sec.
100 T	ecc	9 deg.	6"	1.600 sec.

PROJECT SUMMARY AND RECOMMENDATIONS

Throughout the text of this manual, general statements have been made about the results of the simulations and the implications of these results on current practices within the industry. It is not within the scope of this manual to propose new track maintenance standards and car design specifications, but to demonstrate the need for review of these items with the intention of performing additional research to develop detailed standards. This task is left for Phase II of the Track-Train Dynamics Program. It should be beneficial to individual railroads to study the contents of this manual and apply the results to particular problem areas.

A good example of the application of these results would be with respect to Rule 88 of the Office Manual of AAR Interchange Rules which governs the mechanical requirements for acceptance of cars in interchange. In particular would be the application to the section governing the selection of springs, item 2, which defines the need for supplemental snubbing of certain cars. This section applies only to cars having 6½ x 12 journals or larger (nominally 100 ton capacity or larger). The Stucki simulation series proves there is a need for extension to lower capacity cars. According to the rule, a car is a candidate for snubbing only when it has a truck spacing between 28 and 45 ft. and the loaded center of gravity exceeds 84". Case study 2 shows a need for extension of this truck spacing. Case study 1 shows a need for closer investigation of the center of gravity limits. Also, the parameter study and the simulation series shows a need for studying closer the list of approved devices for controlling car stability since the cars modeled were equipped with trucks that are approved under the list in item 4. Close inspection of the simulation results could be beneficial to a railroad in establishing their own "interim specifications" and criteria for selecting new car designs until sufficient studies are completed by the industry and new industry-wide specifications are written.

Many other areas may be investigated by individual railroads by utilizing the results herein and by making use of the flexible car body model. As pointed out in the text further work should be done in the following areas:

1. Setting track maintenance standards for mainlines, branch lines, and industrial spurs according to operating speeds, track configuration, type of cars, etc.

2. Investigation of suspected "bad actors" in existing and new freight car designs.

3. Investigation of derailments.

4. Establishing freight car maintenance standards in areas such as spring changeouts, side bearing clearance, etc. and predicting fatigue life.

5. Study loading practices by existing and new shippers to determine any detrimental dynamic response effects.

6. Rewrite specifications for testing rocking control devices and develop specifications for dynamically qualifying new car designs.

Therefore, as Volume III concludes the Harmonic Roll Series, a challenge is issued to the various railroads and industry committees to make full utilization of the tools and experience presented in the entire series. The end result will be the rapid definition of problems, manageable cost effective solutions, and the overall improvement of the railroads' ability to provide efficient service.

APPENDICES

I. Constants Used For Parameter Study

II. Constants Used For Stucki Simulation Series

APPENDIX I.

CONSTANTS USED FOR PARAMETER STUDY

The data used in the simulation for the 70 ton box car and the 100 ton covered hopper are: (unless otherwise specified)

	70 ton box car	100 ton covered hopper
The pitching moment of inertia for the carbodies	239844 lb-ft-sec^2	750000 lb-ft-sec^2
The rolling moment of inertia for the carbodies	53700 lb-ft-sec^2	76000 lb-ft-sec^2
The yawing moment of inertia for the carbodies	230131 lb-ft-sec^2	750000 lb-ft-sec^2
The rolling moment of inertia for the bolsters	180 lb-ft-sec^2	183.5 lb-ft-sec^2
The rolling moment of inertia for the wheelset/side frames combined	1363.2 lb-ft-sec^2	1534.3 lb-ft-sec^2
The mass of the carbodies	2644.88 slugs	3784.63 slugs
The mass of the bolsters	36.0 slugs	42.39 slugs
The mass of the wheelsets/side frames combined	198.0 slugs	256.83 slugs
Diameter of the centerplate	14.0 in	14.0 in
Side bearing spacing from car centerline	25.0 in	25.0 in
Spring group spacing from car centerline	3.29 ft	3.293 ft
Rail gage	4.70 ft	4.70 ft
Center-of-gravity of wheelsets below the lateral springs at equilibrium	0.17 ft	-0.5 ft
Distance between the center-of-gravity of the carbodies and the center-of-gravity of the bolsters in the vert. direction	5.83 ft	6.567 ft

Distance between the center-of-gravity of the carbodies and the center-of-gravity of the bolsters in the longitudinal direction	6.56 ft	9.0 ft
Gib clearances	0.03125 ft	0.03125 ft
Flange clearances	0.0339 ft	0.0339 ft
Rail length	39.0 ft	39.0 ft
Maximum cross level diff.	0.75 in.	0.75 in.
Truck center distance	39.5 ft	45.0 ft
Wheel base	5.67 ft	5.8333 ft
Centerplate stiffness	25440000 lb/ft	25440000 lb/ft
Side bearing stiffness	42960000 lb/ft	42960000 lb/ft
Side bearing viscous damping coefficient	500 lb/ft/sec	500 lb/ft/sec
Suspension group stiffness	243960 lb/ft	298800 lb/ft
Lateral spring stiffness	111600 lb/ft	150000 lb/ft
Vertical track stiffness (two wheels combined)	3000000 lb/ft	2520000 lb/ft
Lateral track stiffness (two wheels combined)	2000000 lb/ft	2000000 lb/ft
Coulomb's friction coefficient at the column	2000 lb	4000 lb
Torsional stiffness between the carbodies	6000000 ft-lb/rad	8.0×10^7 ft-lb/rad
Bending stiffness about verical axis between the carbodies	2.4×10^8 ft-lb/rad	2.4×10^8 ft-lb/rad
Bending stiffness about lateral axis between the carbodies	2.4×10^8 ft-lb/rad	2.4×10^8 ft-lb/rad
Shearing stiffness of the vertical axis between carbodies	2.4×10^8 ft-lb/rad	2.4×10^8 ft-lb/rad

Length of the spring travel	0.3073 ft	0.3073 ft
Lateral side frame stiffness	1000000 lb/ft	1000000 lb/ft
Vertical side frame stiffness	38520000 lb/ft	38520000 lb/ft
Coefficient of friction at the gib	0.3	0.3
Time step	0.00025 sec	0.00025 sec

APPENDIX II.

CONSTANTS USED FOR STUCKI SIMULATION SERIES

Car Description A. ride control spring group. 70 Ton Box Car, 5576 cu. ft., standard 70 ton trucks, 7 D-5 outer, 5 D-5 inner coils per

Car Description B. ride control spring group. 100 Ton Box Car, 6100 cu. ft., standard 100 ton trucks, 7 D-5 outer, 9 D-5 inner coils per

Constant	70 Ton Car	100 Ton Car
Gross weight on rails	185,400 lb.	252,150 lb.
Truck centers	39 ft. 6 in.	40 ft. 10 in.
Center gravity height above rails (loaded-including trucks)	86.5 in.	99.9 in.
Center gravity above center plate (car body and lading only)	5.7 ft.	6.8 ft.
Moment of inertia about rolling axis through center gravity	107,400 lb.ft.sec.^{2}	176,600 lb.ft.sec.^{2}
Moment of inertia about pitching axis through center gravity	1,387,465 lb.ft. sec.^{2}	1,604,000 lb.ft. sec.^{2}
Moment of inertia about yawing axis through center gravity	1,368,040 lb.ft. sec.^{2}	1,541,200 lb.ft.sec.^{2}
Car body torsional stiffness over truck center length	6×10^6 lb.ft./rad.	6×10^6 lb.ft./rad.
Car body vertical bending stiffness over truck center length	240×10^6 lb.ft./ rad.	240×10^6 lb.ft./rad.
Truck spring rate-each group	20,330 lb./in.	24,610 lb./in.
Truck spring solid capacity-each group	75,302 lb.	91,354 lb.

Side bearing spacing from center line	25 in.	25 in.
Side bearing clearance	1/4 in.	1/4 in.
Bolster gib clearance	3/8 in.	3/8 in.
Center plate diameter	14 in.	14 in.
Journal centers	5 ft. 8 in.	5 ft. 10 in.