ANALYSIS AND DESIGN

OF

METRORAIL - RAILROAD BARRIER SYSTEM

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Metrorail - Railroad Barrier System

Sponsored by

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GLOSSARY

Barrier Height - Height of barrier above the top of the rails.

Barrier Lateral Offset Distance - Distance from centerline of track to face of barrier wall.

C.G. Height - Center of Gravity Height - The height of the center of mass of the railroad car above the top of the rails.

Coefficient of Friction - Ratio of friction force to normal force.

e = Coefficient of Restitution = For direct central impact of two bodies it is the ratio of the relative velocity of separation (after impact) to the relative velocity of approach (before impact).

g - Acceleration due to gravity - 32.2 ft/sec^2.

G - A unit of acceleration in terms of gravity, g's.

Impact Angle - For a longitudinal barrier, it is the angle between a tangent to the face of the barrier and a tangent to the rail car's path at impact.

Kip = 1,000 pounds = A unit of force.

Railroad Barrier - A longitudinal barrier wall whose primary function is to prevent penetration of derailer railroad car or cars into an adjacent railroad travelway or other nearby area.

Ton = 2,000 pounds = 2 kips = A unit of force or weight.

Train Car Stiffness - The force required to crush a corner of a railroad car one foot. Also a spring constant K = kips per ft of crush.

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INTRODUCTION

The Washington Metropolitan Area Transit Authority (WMATA) shares several common corridors with other railroads. In the spring of 1987 there were two derailments where freight cars came to rest on WMATA passenger car tracks. One disrupted traffic on WMATA lines for two days.

In view of the recent developments, two common corridors have been cited by WMATA to undergo special studies with a view to reducing or eliminating the risk of intrusion by the adjacent railroad vehicles into the WMATA right-of-way. These are:

(a) B-6 Corridor

This corridor consists of approximately five to six miles of WMATA operating system on the Red Line situated between Takoma Park, District of Columbia, and Silver Spring, Montgomery County, Maryland. The two WMATA tracks are flanked by the CSX railroad with variable horizontal and vertical separations between the two systems. Among the possible measures to protect WMATA operations, the provision of a barrier system is to be considered as an integral part of this study.

(b) E9-E10 Corridor

This corridor, on the Green Line between College Park and Greenbelt in Prince George's County, Maryland, is in the final design and construction phase. Some latitude exists in redefining the existing interrelationship between the WMATA and the adjacent CSX railroad tracks from the viewpoint of improving the safety of future WMATA operations. The twin tracks of the two systems run side-by-side with variable vertical and horizontal separations. Among other possible safety measures, provision of barrier walls to contain derailed railroad cars is included in the study.

The objective of this report is to present conceptual analysis and design data in order to determine the feasibility of "provision of barrier wall to contain derailed railroad cars" . . . "with a view to reducing or eliminating the risk of intrusion by the adjacent railroad vehicles into the WMATA right-of-way."

RESEARCH APPROACH

The conceptual analysis and design data presented here provides some insight to answer the following questions:

1. Where should the barrier be located?
2. How strong should the barrier be?
3. How high should the barrier be?

These three questions are interrelated since the location of the barrier will influence the required strength.

To answer these questions, the following information is needed.

1. TRACK GEOMETRY

   RIGHT-OF-WAY, WIDTH, CLEARANCES, CURVES (horizontal and vertical), TRACK CHARTS, TRACK INTERSECTIONS, SWITCHES, ETC.

2. OPERATIONAL PRACTICES

   INSTRUCTIONS, AGREEMENTS, SPEEDS, SPECIAL MAINTENANCE, REQUIRED CLEARANCES, ETC.

3. TRAIN DATA

   TYPE TRAINS, TYPE CARS, TRAIN LENGTHS, SPEEDS, LOCOMOTIVES, TRAIN MAKEUP, etc.

   TYPE CARS - LENGTH, WEIGHT, HEIGHT OF CENTER OF GRAVITY (CG HEIGHT), WIDTH

   TYPE COUPLERS, AUTOMATIC BRAKES, ETC.

This report presents some of this data which is used to develop conceptual barrier designs. These conceptual barrier designs possess some of the following characteristics.

1. Can be located in existing right-of-way.
2. Construction should be such as to cause minimum disruption of the Metro and other train operations.
3. Cause little to no change in Metro and other train operational practices (train makeup, speed, etc.).
4. Require minimum maintenance and replacement, if required.
5. Be safe and economical.

The conceptual barrier designs also recommend the structural material and foundation design required to meet most of these requirements.

PRELIMINARY TRACK GEOMETRY

Figure 1 shows an idealized B-6 corridor where the WMATA Metro tracks are flanked on each side by the CSX Railroad.

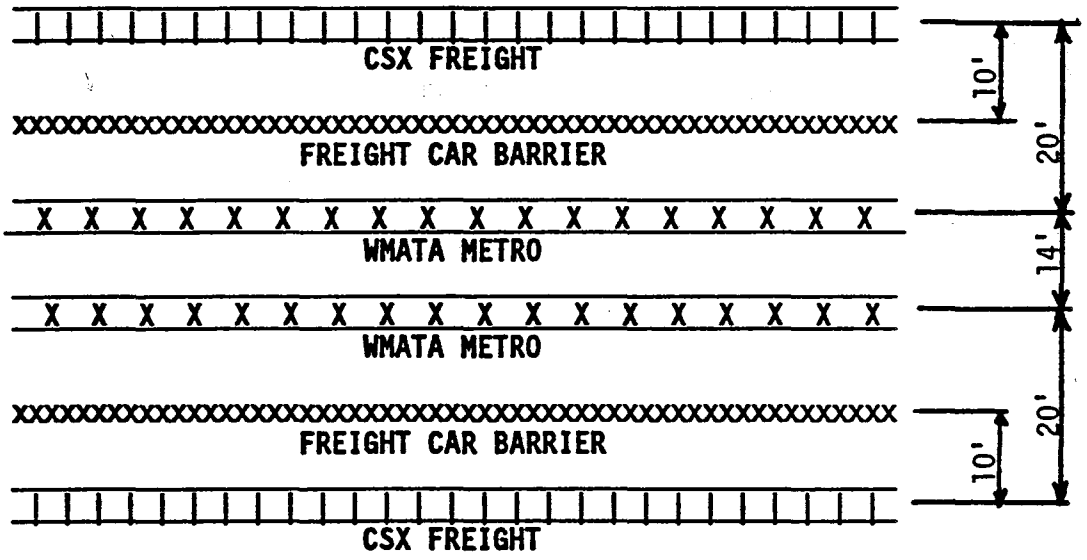


Figure 1. Idealized B-6 Corridor

The distances between the tracks varies, but the 14 ft and 20 ft shown are very common in the B-6 corridor. This indicates the freight car barrier would be located about 10 ft (center to center) from the CSX tracks.

Figure 2 shows an idealized E9-E10 corridor where the twin tracks of the two systems run side by side. The distance between the tracks varies, but the 34 ft minimum between the CSX and Metro tracks is common.

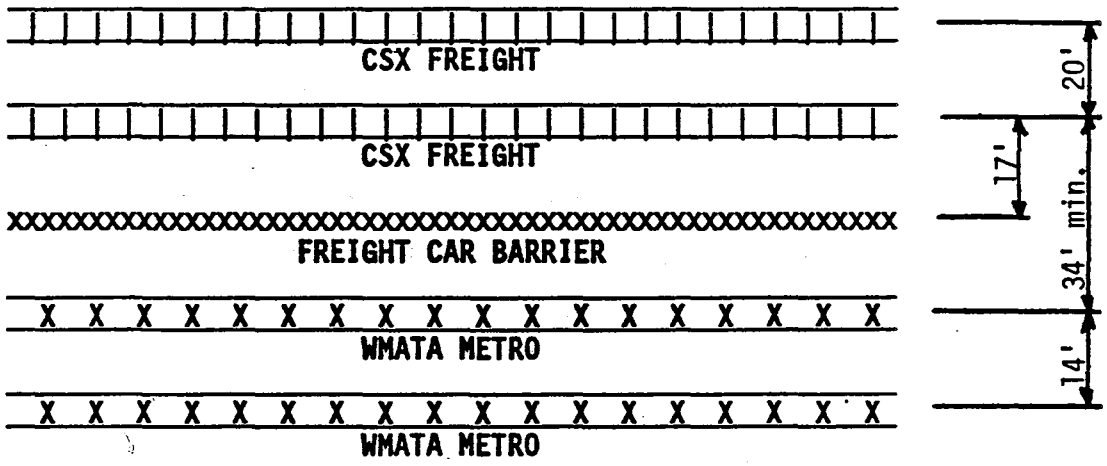


Figure 2A. Idealized E9-E10 Corridor

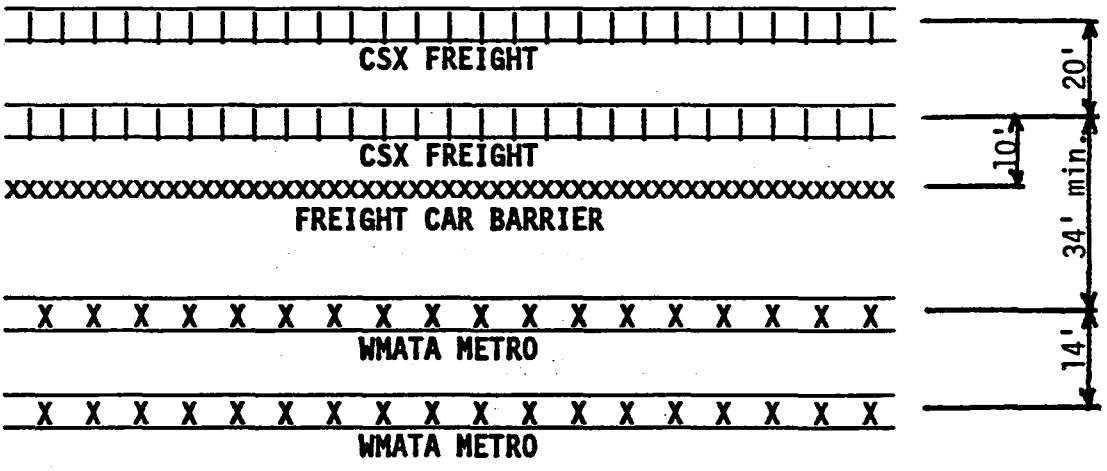


Figure 2B. Idealized E9-E10 Corridor

This idealized track layout would indicate the lateral distance from the CSX track to the proposed barrier in the E9-E10 corridor could be about 17 ft (center to center) as shown in Figure 2A. Later in the report it will be shown that smaller train impact forces would result with the barrier location shown by Figure 2B. The barrier should be about 10 ft from the CSX freight train tracks.

TRAIN DATA

In order to analyze train impact forces on a barrier wall, an idealized design train needs to be selected. TTI researchers have met with CSX officials to determine what types of trains are actually using these two corridors now. Data on eight different trains have been received to date. In addition, data on nation-wide train derailments in 1975 and 1976 were obtained from reference (1).

Train Traffic: In July 1987 the project staff met with representatives of CSX. During that meeting information regarding train traffic characteristics within the study corridor were provided. At that time there were thirty-seven (37) train movements per day. They were distributed as follows:

Train Type	Number Per Day
Passenger	14
Schedule Merchandise (Freight)	15
Extras (Freight)	8
Total	37

The reported train speeds were:

Passenger	60 mph
Freight	30 mph

However, information contained in two (2) accident reports indicated that maximum authorized speeds were 70 and 79 mph for passenger trains and 55 mph for freight trains. These speeds correspond to those reported in the March 1988 study by R.K. Pattison and Parsons Brinckerhoff Quade and Douglas. The tracks in the corridor are FRA Track Class 4 and maximum allowable speeds for this class are: passenger trains (80 mph) and freight trains (60 mph).

According to CSX officials, the twenty-three (23) scheduled and extra freight trains have the following characteristics:

- TDFC Trains (Container Trains)
  Six (6) trains per day
  Average 74 cars per train
  Loaded cars = 67
  Empty cars = 7

- Merchandise Trains
  Nine (9) trains per day
  Average 81 cars per trains
  Loaded cars = 37
  Empty cars = 44
- Extra Trains
  Eight (8) trains per day
  Average 88 cars per train
  Loaded cars = 41
  Empty cars = 47
  Approximately 90% of the cars are coal cars

In addition to the summary information shown above, CSX provided detailed data on eight trains. By using information contained in "The Official Railway Equipment Register," it was possible to obtain dimensions and capacity of the cars. Tare weights of the cars were computed using the AAR Mechanical Designation Code and data from the Car and Locomotive Encyclopedia. Cargo weights were contained in the information provided by CSX. These data were used to select a typical heavily loaded train to be used in the crash barrier computer simulation runs.

Reference (1)* summarizes data on train derailments which occurred during 1975 and 1976. The report published by the Association of American Railroads (AAR) obtained most of the data from Federal Railroad Administration's (FRA) accident data base for 1975 and 1976. References (2, 3, 4, and 5) were also used to supplement these data.

Table 1 summarizes some typical derailed train data for the period of about 1975 and 1976. The average numbers have been used in previous research (2, 3, and 4) to analyze or simulate train derailments. Table 1 indicates the hopper car is the most likely first car to derail about 40% of the time.

Table 2 presents a summary of various train car properties. It is interesting to note that the vulnerable hopper car is the shortest (down to 42 ft) and the heaviest (132 tons or 264,000 lb). The Metro cars are long (75 ft) and fairly light weight (60 tons or 120,000 lb).

The heavy locomotive (184 tons or 368,000 lb) with a length of 65 ft are believed to be not critical since they rarely derail.

*(1) Numbers in parentheses refer to corresponding item in the Reference List.

Table 3 presents a summary of train car data on five CSX trains operating on the common corridors under consideration. It can be seen that the train varied from 44 cars to 110 cars (average 69 cars), car lengths varied from 42 ft to 94 ft (average 62.6 ft), and the car weights from 29 tons empty to 132 tons full (average 75.7 tons). The 63 car grain hopper train No. X40203 with all cars fully loaded to 131.3 tons would appear to be a candidate as a heavy train for barrier design. These cars are 59.3 ft long and should produce the maximum impact forces.

TABLE 1.

TYPICAL DERAILED TRAINS, 1975 and 1976

78% OF ALL TRAIN ACCIDENTS ARE DERAILMENTS

TRAIN LENGTH, 1 TO 150 CARS, AVERAGE 61 CARS

TRAIN SPEED, 1 TO 80 MPH, AVERAGE 40 MPH

CAR WEIGHTS, 31.5 TO 131.5 TONS, AVERAGE 80 TONS

CAR LENGTH, 39 TO 90 FT, AVERAGE 55 FEET

NUMBER OF CARS DERAILED PER ACCIDENT, 1 TO 22 OR MORE, AVERAGE 8 CARS DERAILED

TYPE CAR DERAILED FIRST -	HOPPER	40%
	BOX	34%
	FLAT	9%
	GONDOLA	8%
	TANK	8%
CAUSES OF DERAILMENTS -	TRACK	46%
	EQUIPMENT	34%
	HUMAN	10%
	OTHER	10%

LONGITUDINAL DISPLACEMENT - 150 TO 300 FEET

LATERAL DISPLACEMENT - 35 FT UNIFORM CARS

112 FT MIXED CARS

TABLE 2.

SUMMARY OF TRAIN CAR PROPERTIES - Reference (5) and Others

TYPE CAR	LENGTH (ft)	WEIGHT (tons) EMPTY to LOADED	C.G. HEIGHT (in.) EMPTY to LOADED
HOPPER	42 to 60	29 to 132	47 to 79
TANK	46.5 to 50	29 to 132	
BOX	54 to 69	29 to 132	
GONDOLA	50 to 60	29 to 132	
LOCOMOTIVE	65	184	
FLAT	64 to 95	29 to 132	
METRO	75	35 to 60	

WIDTH OF CARS = 9.0 TO 10.7 FT, AVERAGE 10.2 FT

HEIGHT OF CARS = 12.3 TO 15.3 FT, AVERAGE 14.25 FT

HEIGHT TO CENTER OF GRAVITY = 98 IN. MAX., 8.2 FT

METRO CAR CENTER OF GRAVITY = 64 IN., 5.3 FT (estimated)

TABLE 3.

SUMMARY OF CAR DATA ON FIVE CSX TRAINS ON THE COMMON CORRIDOR

TRAIN NO. X40203 GRAIN TRAIN
East Bound - Michigan to Baltimore

63 grain hopper cars
All fully loaded
Average car length = 59.3 ft (51.6 to 60.1 ft)
Average car weight = 131.3 tons

TRAIN NO. R38106 MIXED MERCHANDISE TRAIN
Jessup Pick-Up

46 cars (42 flat and 4 box)
All empty
Average car length = 93.7 ft (56.8 to 93.8 ft)
Average car weight = 51.8 tons

TRAIN NO. R38106 MIXED MERCHANDISE TRAIN
Baltimore to Chicago

44 cars (17 tank, 9 flat, 6 gondola, 11 special)
(46.5 ft, 88.5 ft, 56 ft, 60 ft)
24 loaded and 20 empty
Average car length = 62.8 ft (46.5 to 88.5 ft)
Average car weight = 72.5 tons

TRAIN NO. X40802 COAL HOPPER TRAIN
West Bound - D.C. to Cumberland

84 coal hopper cars
All cars empty
Average car length = 49.6 ft (43.8 to 53.1 ft)
Average car weight = 29.4 tons

TRAIN NO. R37602 MIXED MERCHANDISE TRAIN
Cincinnati to Baltimore

110 cars
76 loaded and 34 empty (7 other, 22 special, 22 box,)
(- , 42 to 65 ft, 54 to 69 ft)
(4 refrigerator, 24 flat, 5 gondola, 21 tank)
(60.5 to 63.8 ft, 64 to 93.7 ft, 57 to 59 ft, 46.5 ft)
Average car length = 61.6 ft (42 to 93.7 ft)
Average car weight = 90.5 tons (31 to 132 tons)]

ANALYSIS OF TRAIN IMPACT FORCES ON LONGITUDINAL BARRIER WALL

The following description of a typical train derailment was taken from references (2 and 3). This analytical study reveals some interesting and useful characteristics of a derailed train.

"In a train derailment, each car can roll, pitch, yaw and translate in three dimensions, thus having six degrees of freedom. A review of accidents in the past reveals that the pattern is in fact extremely complicated. This study is limited to the most significant motions (those in a horizontal plane) in order to simplify the simulation. The schematic below depicts the problem as analyzed.

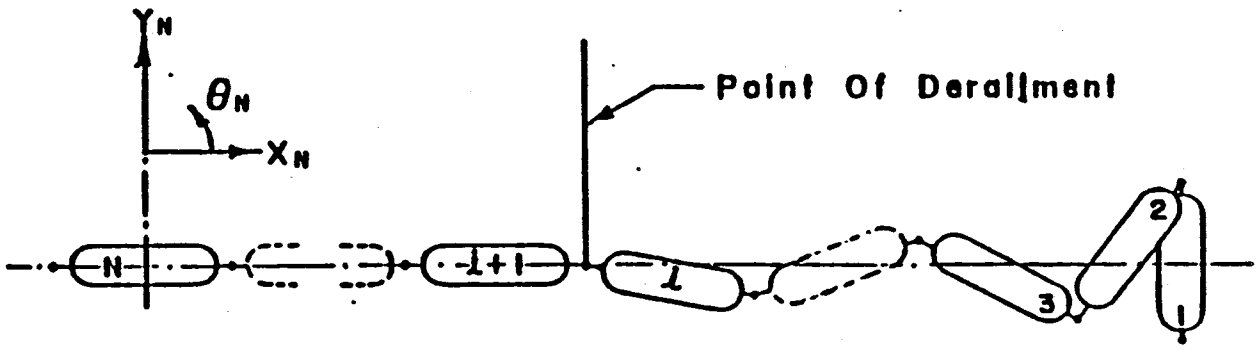


Figure 3. (from References 2 and 3)

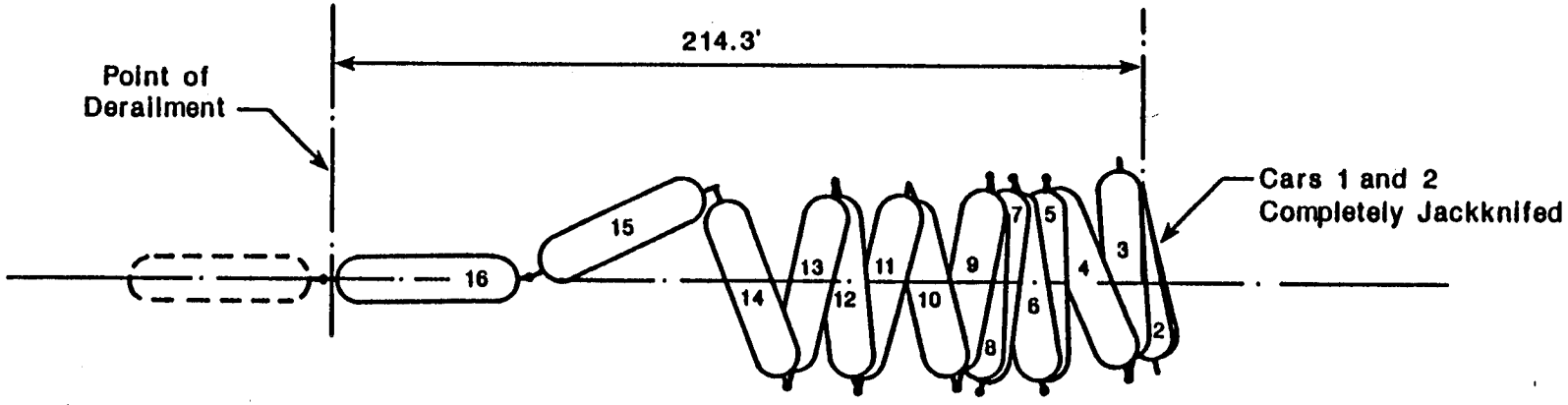
In this simulation it is assumed that a train parting has occurred and only cars from the first derailed car back to the rear of the train are considered. The cars ahead of the parting move on and are not involved in the derailment.

This analysis is based on the following additional assumptions:
(1) The cars are coupled together with resisting moment between cars; (2) there is no width dimension of cars; (3) there is simple ground friction at trucks of derailed cars; (4) emergency braking is applied to non-derailed cars; (5) cars remain coupled; and (6) the rail is interrupted by the first derailed car."

Figure 4 shows the orderly symmetrical behavior of a derailed train where all cars are the same weight and length. Figure 5 shows the erratic behavior when the cars have mixed weights and lengths.

DERAILMENT SIMULATION

Statistical History Case
Case No. 3



Scale: 1"=50'

Number of Derailed Cars - 16.0

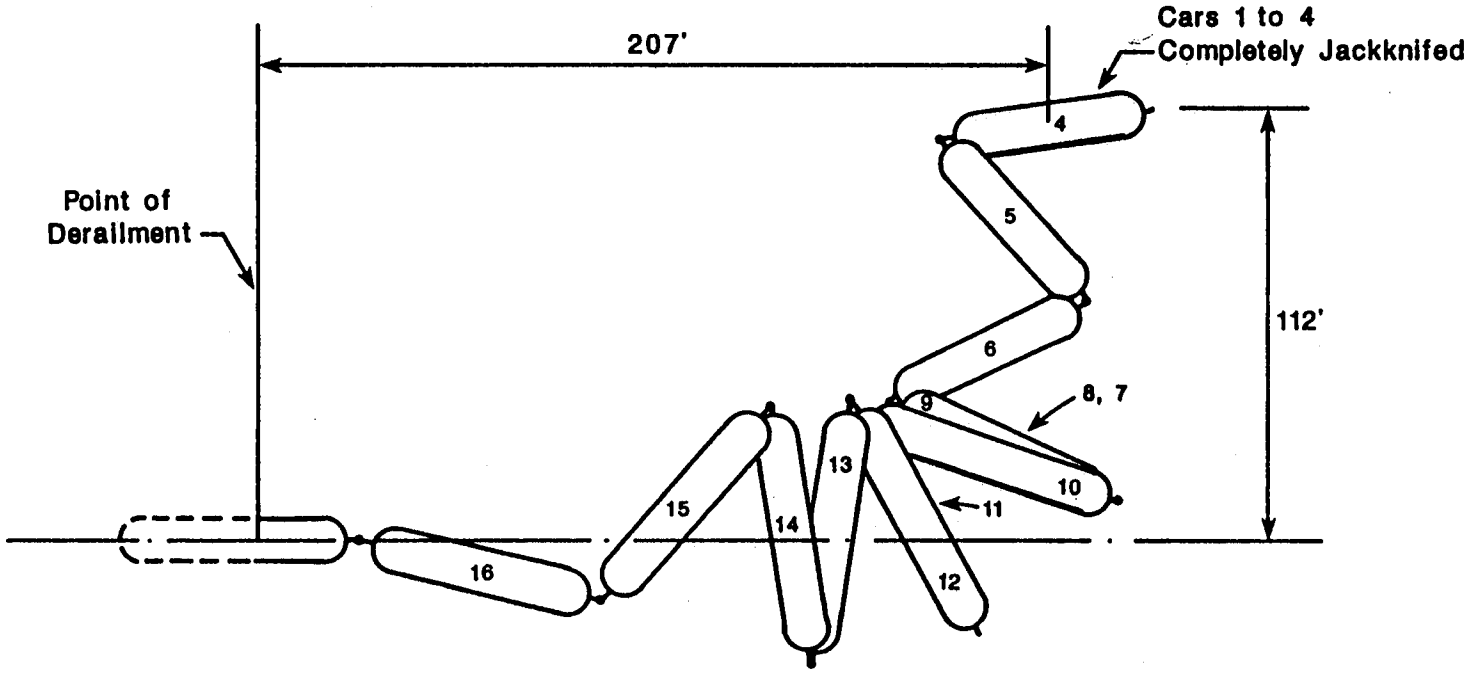
Diagram No. 3

61	Cars in Train
40	Train Velocity, mph
1.0	Ground Friction
124	Car Weight, kips
55	Car Length, ft.
43	Truck Centers, ft.
E-E	Coupler
C.I.	Brake

FIGURE 4. SYMETRICAL BEHAVIOR WHEN ALL CARS ARE SAME WEIGHT AND LENGTH (from Reference 2)

DERAILMENT SIMULATION

Verification Case
(Crescent City)
Case No. 1



Scale: 1"=50'

Number of Derailed Cars - 16.4

Diagram No. 1

90	Cars in Train
45	Train Velocity, mph
1.5	Ground Friction
Mixed	Car Weight
Mixed	Car Length
Mixed	Truck Centers
E-E	Coupler
C.I.	Brake

FIGURE 5. ERRATIC BEHAVIOR WHEN CARS HAVE MIXED WEIGHTS AND LENGTHS (from Reference 2)

This mathematical model of a derailed train was modified by TTI to include a "rigid" longitudinal barrier wall to restrain the lateral displacement of the train as shown by Figure 6. To accomplish this modification:

1. The train cars were given width,
2. The coupling moment between cars was allowed to be increased proportionally to the increasing alignment angle (see Appendix A),
3. A "rigid barrier wall" was constructed parallel to the track at a given lateral offset distance (measured from centerline of track to face of barrier),
4. The crush stiffness of the corners of the cars, K, in kips per ft was provided to generate the normal impact force on the barrier and car (see Appendix B), and
5. Simple barrier friction was provided at car-barrier contact points.

The six previous assumptions (with the exception of #2) are still valid.

The train shown in Figure 6 is typical of CSX Train No. X40203 grain train hopper cars fully loaded to 130 tons. This train contained 63 cars, and it parted between cars 23 and 24 leaving 40 cars behind.

In order to conduct this simulation, the corner stiffness of railroad cars had to be estimated. Figure 6 was computed using K = 80 kips/ft. Appendix B presents an analysis for determining this value. This value was determined by extrapolating TTI's crash test results on rubber tired vehicles from 1,800 lb to 4,500 lb cars, 5,400 lb and 18,000 lb trucks, 20,000 lb school bus, 32,000 intercity bus, 40,000 lb intercity bus and 46,000 lb tractors out to a 64,000 lb empty railroad car. No test results on railroad cars were available to verify this extrapolation.

In the computer simulations printed here the coefficients of friction used were as follows:

Trucks on soil = 1.0
Cars on barrier = 0.4
Trucks on track = 0.2 (when brakes are applied)

Figure 7 (nine figures from 0.0 sec [when brakes are applied] to 19.5 sec) shows time sequence of events of the derailed train cars interacting with the barrier. This train shown is very similar to CSX Train No. X40203 grain hopper cars fully loaded to

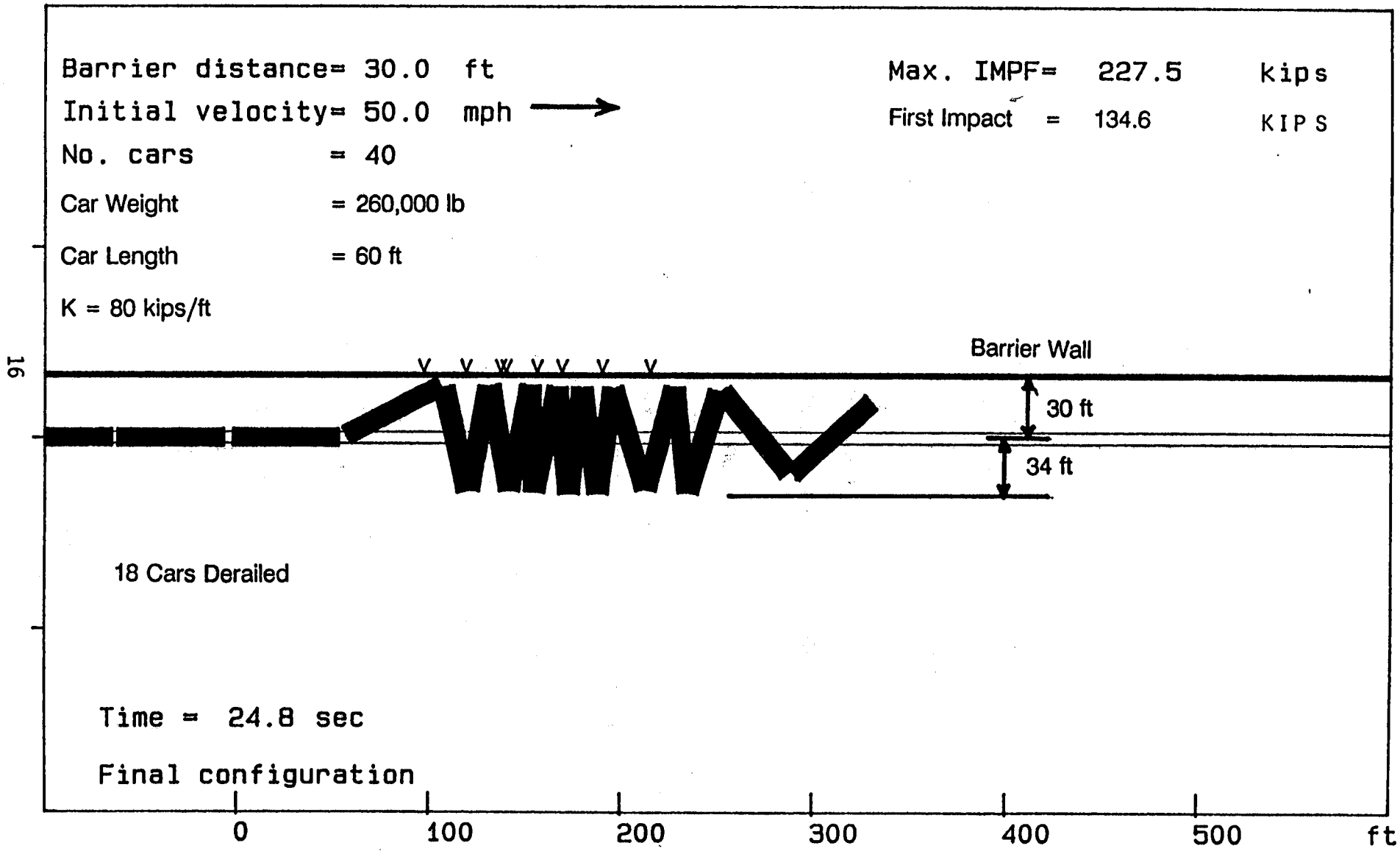


FIGURE 6. TYPICAL TRAIN DERAILMENT WITH BARRIER WALL

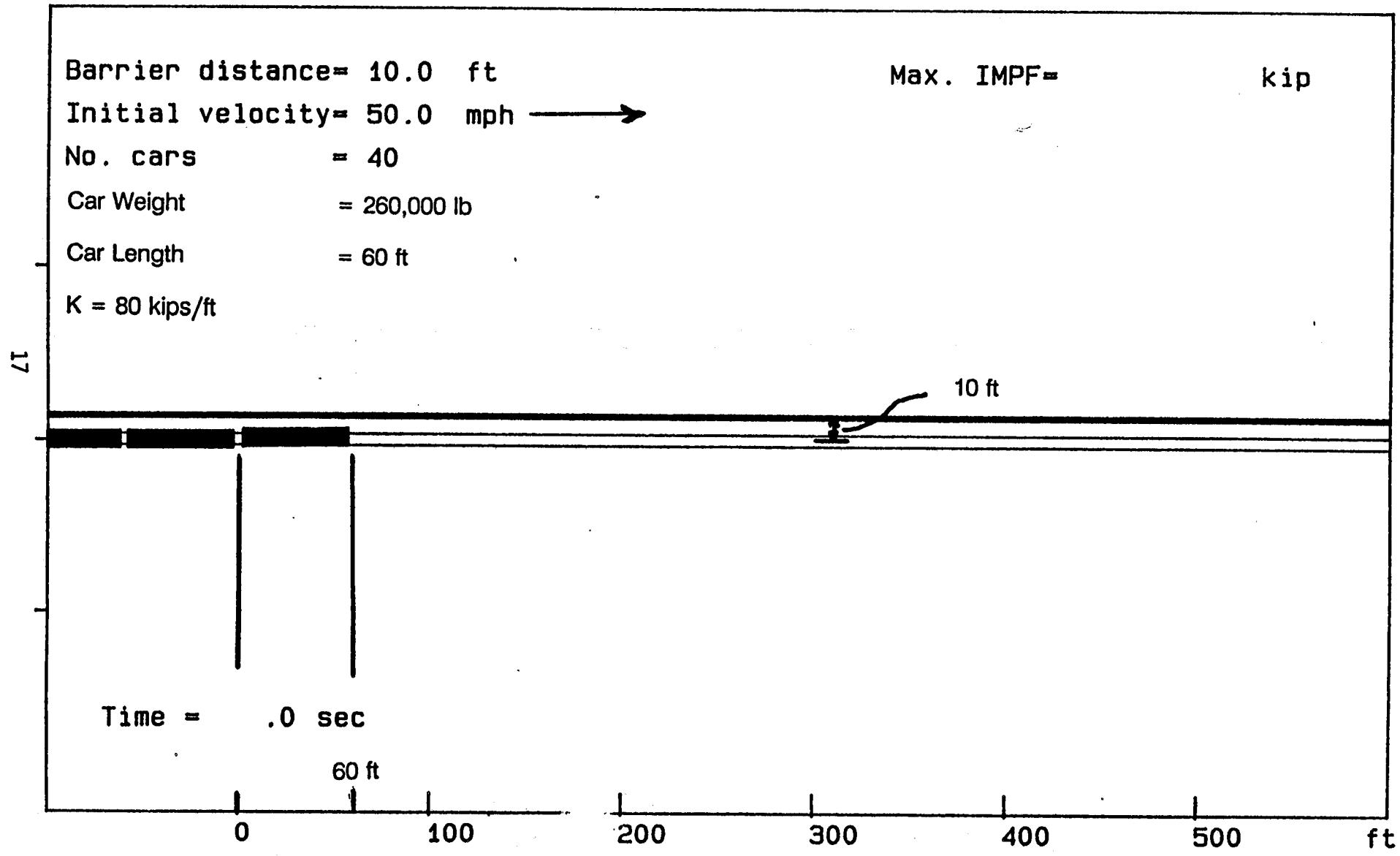


FIGURE 7. TIME SEQUENCE OF TRAIN - BARRIER IMPACTS

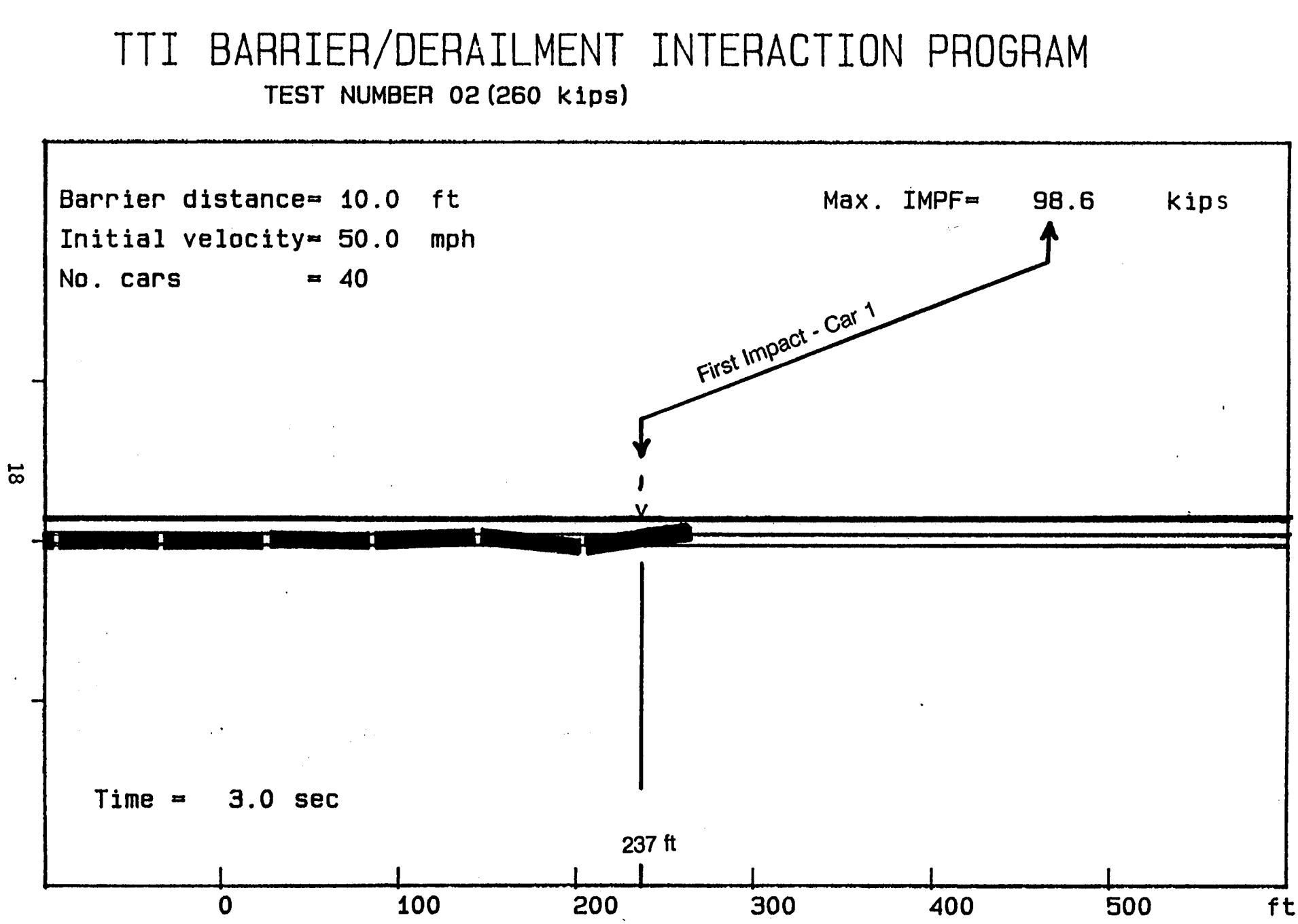


FIGURE 7. (CONTINUED)

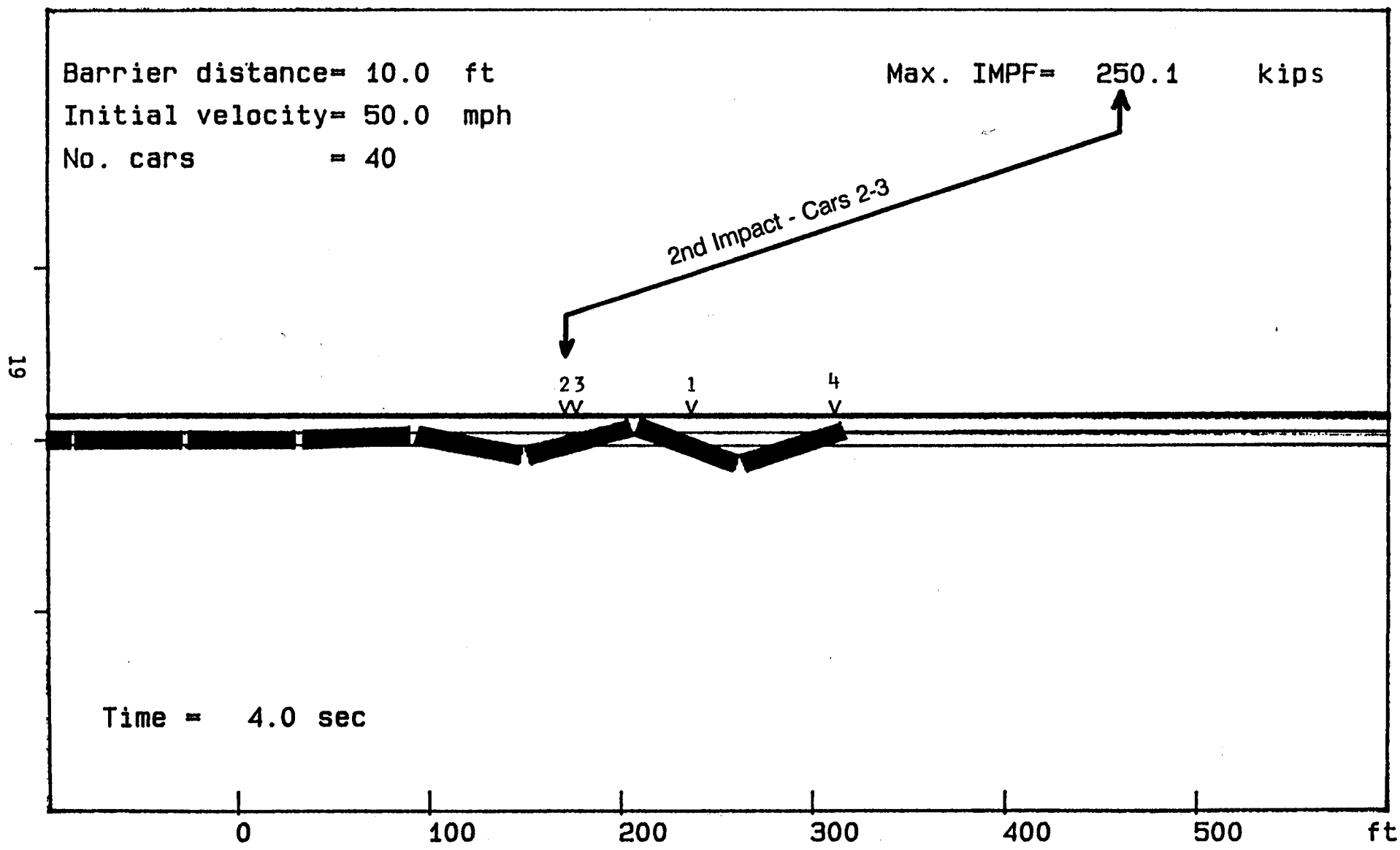


FIGURE 7. (CONTINUED)

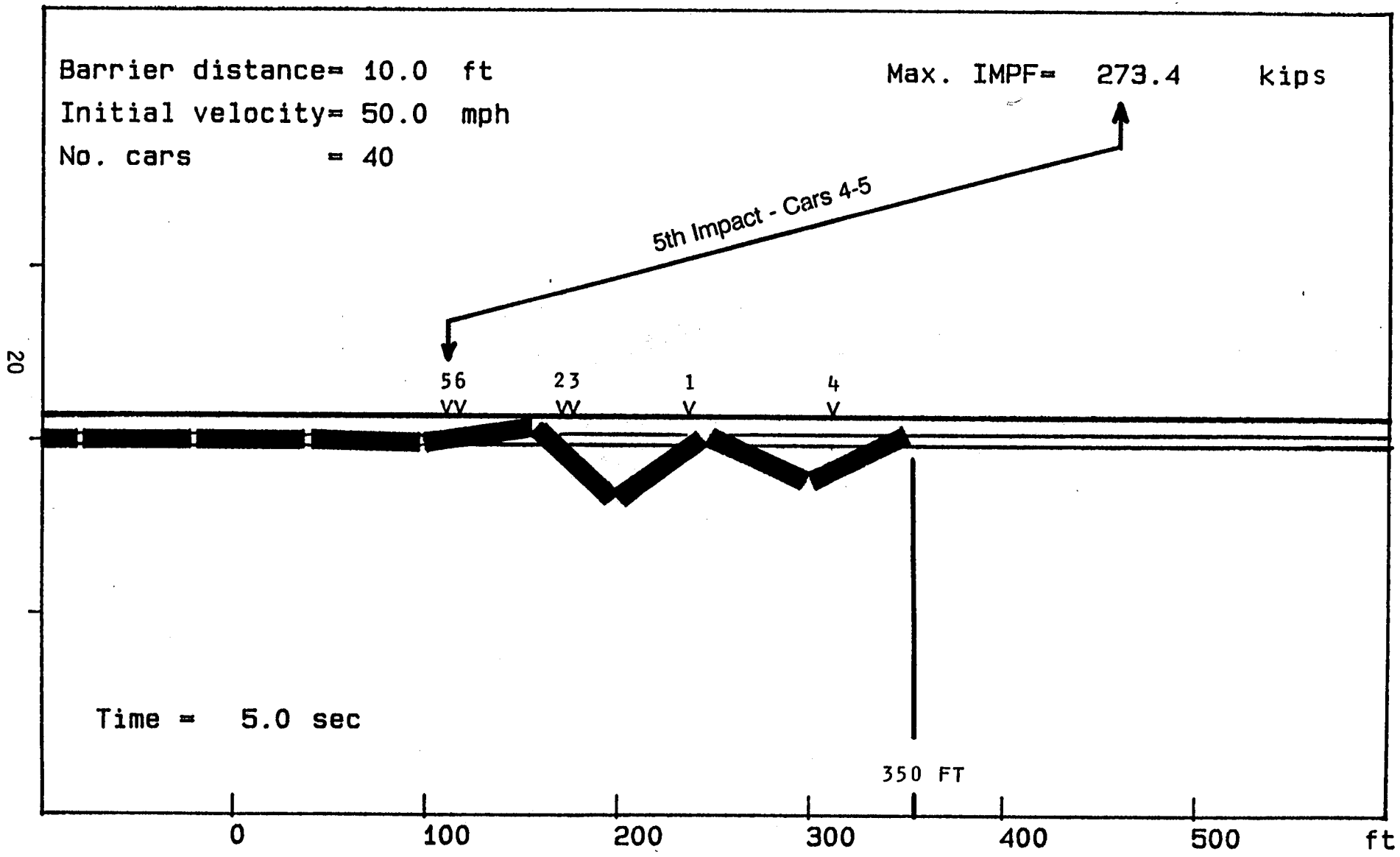


FIGURE 7. (CONTINUED).

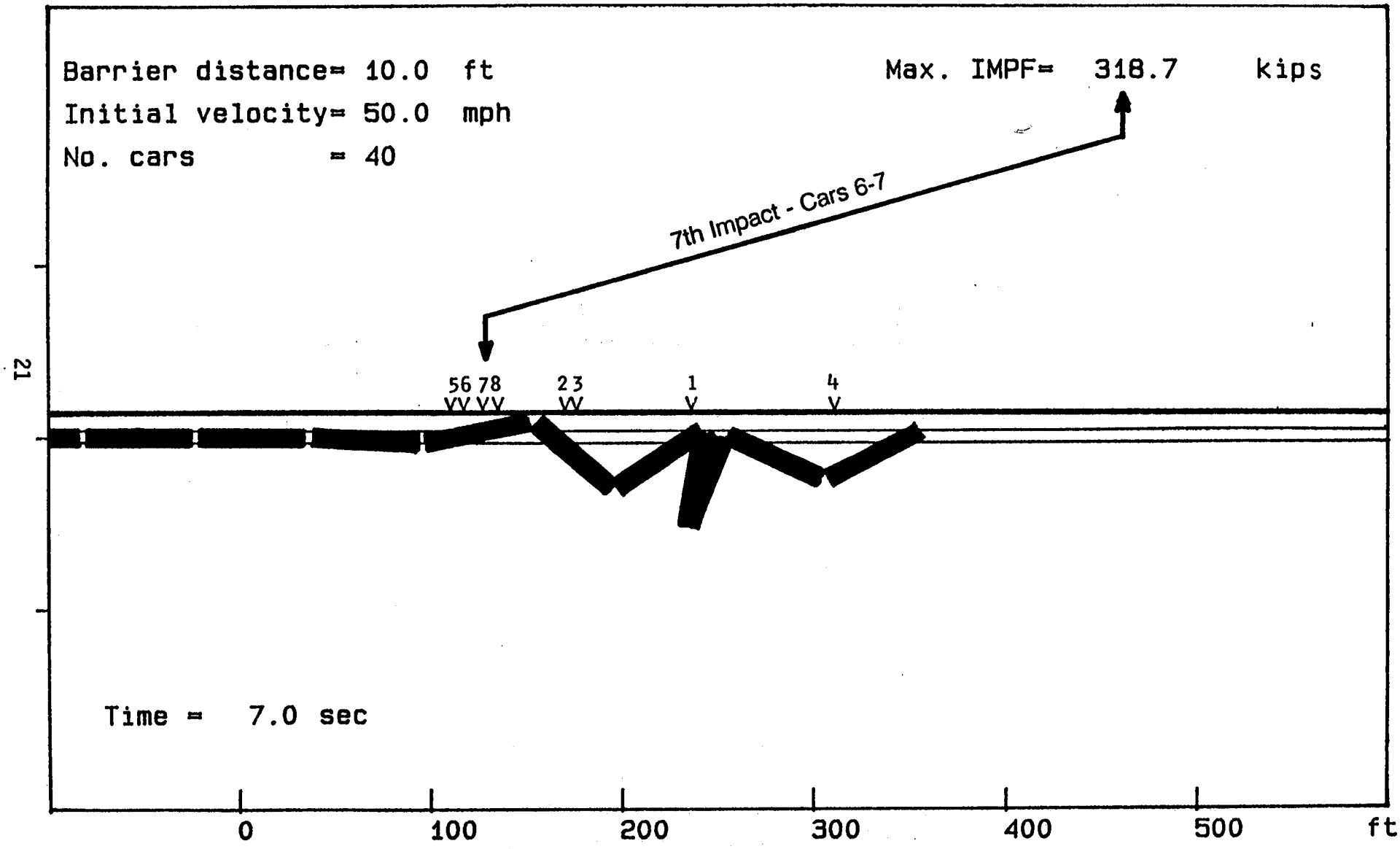


FIGURE 7. (CONTINUED)

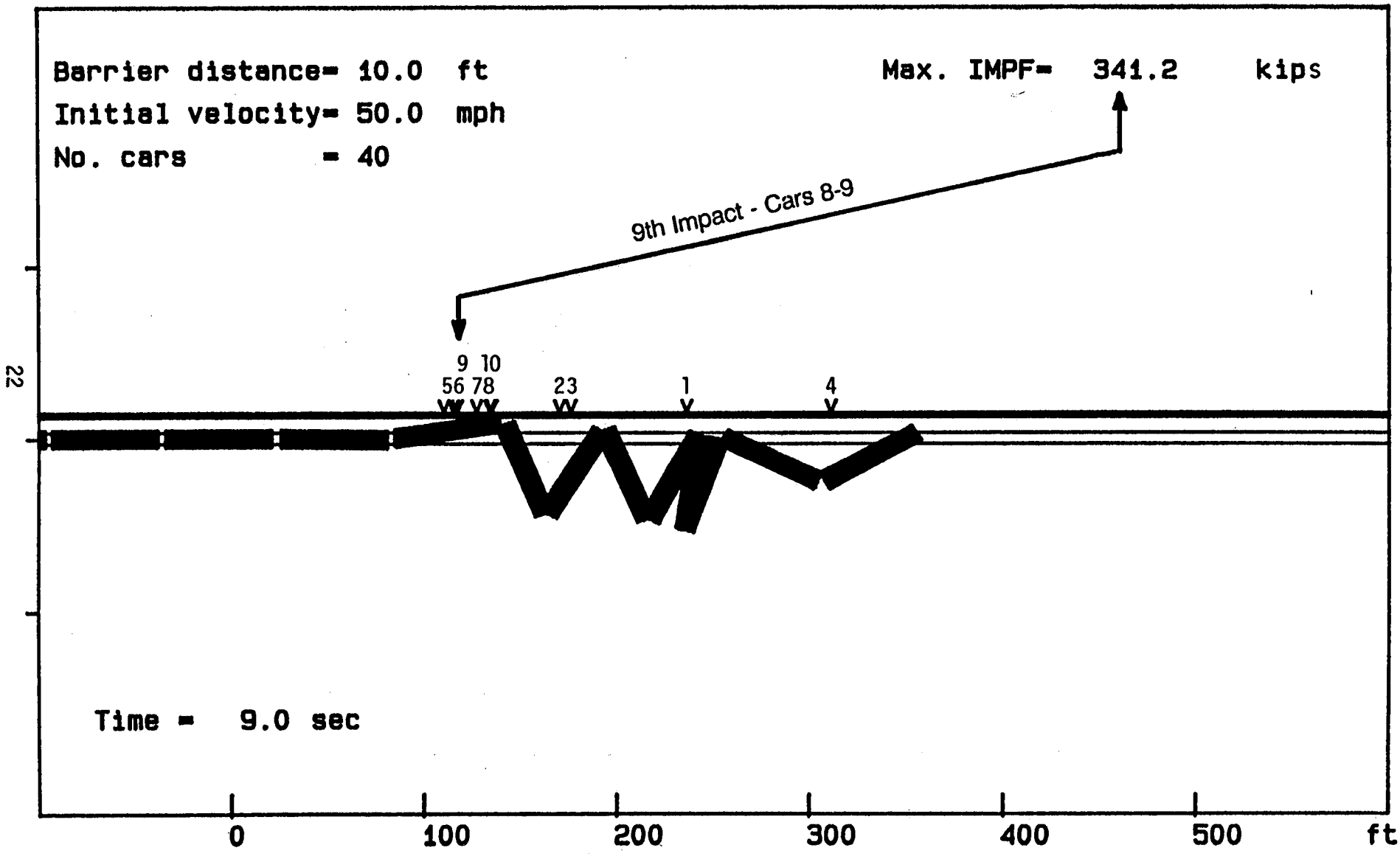


FIGURE 7. (CONTINUED)

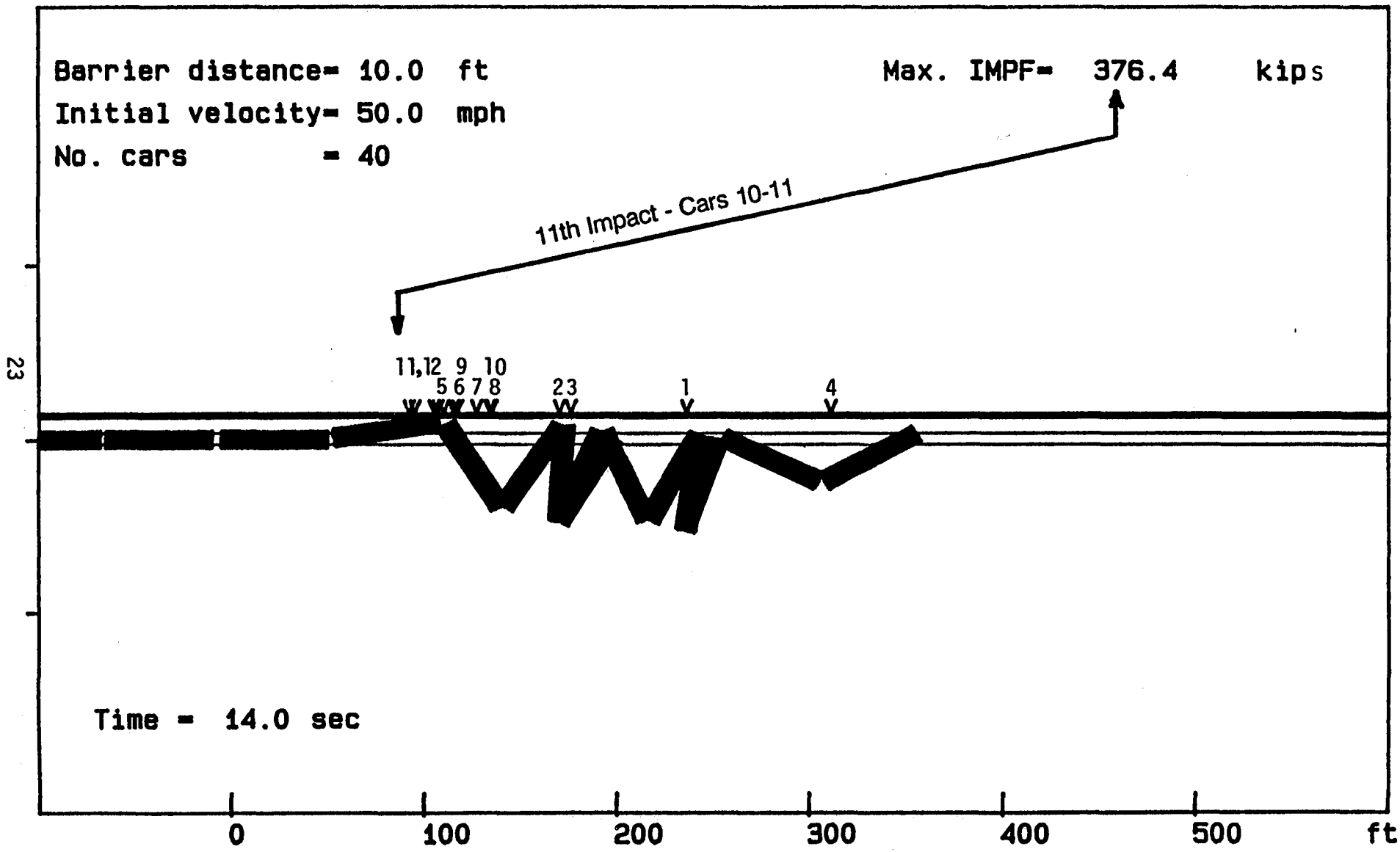


FIGURE 7. (CONTINUED)

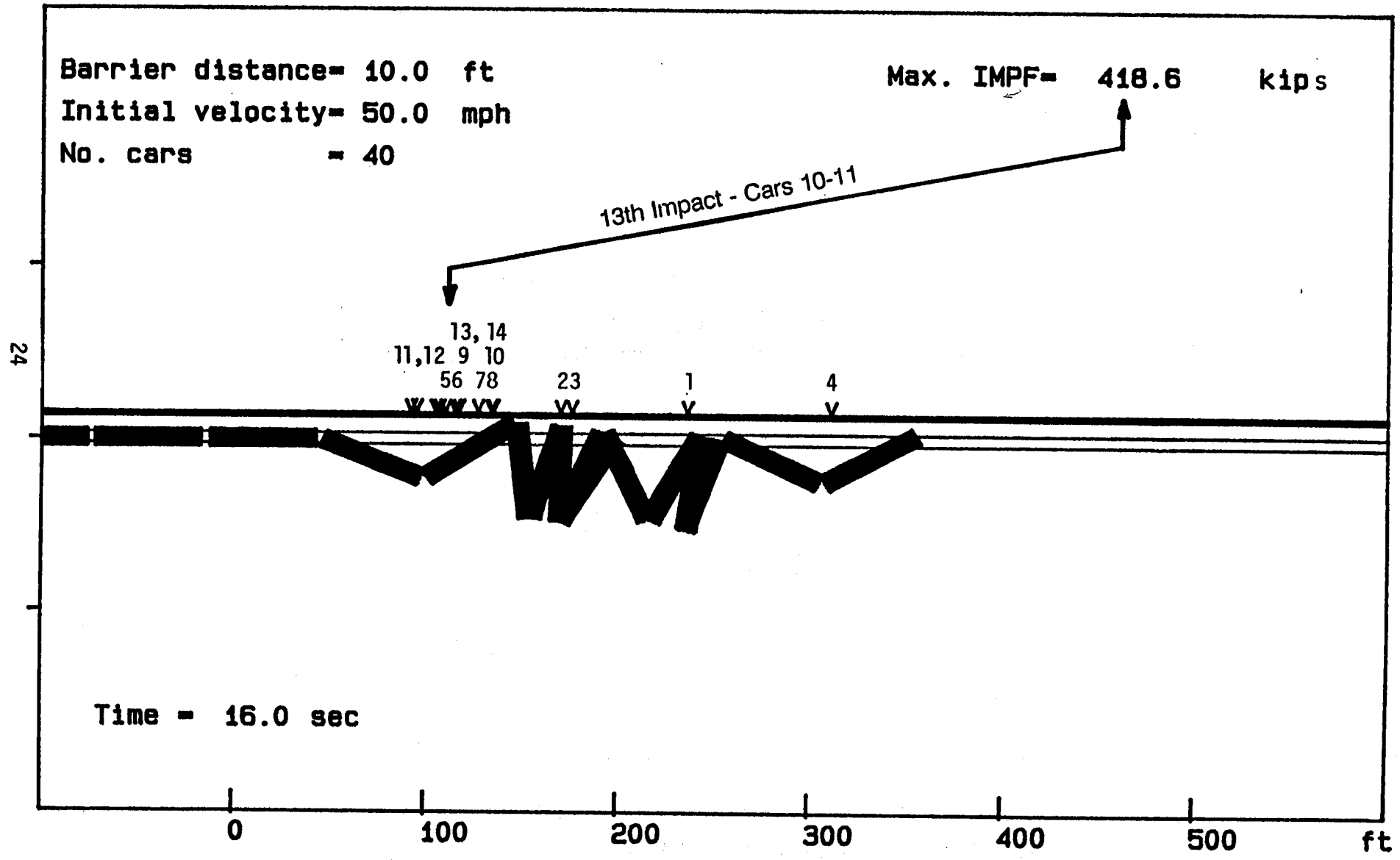


FIGURE 7. (CONTINUED)

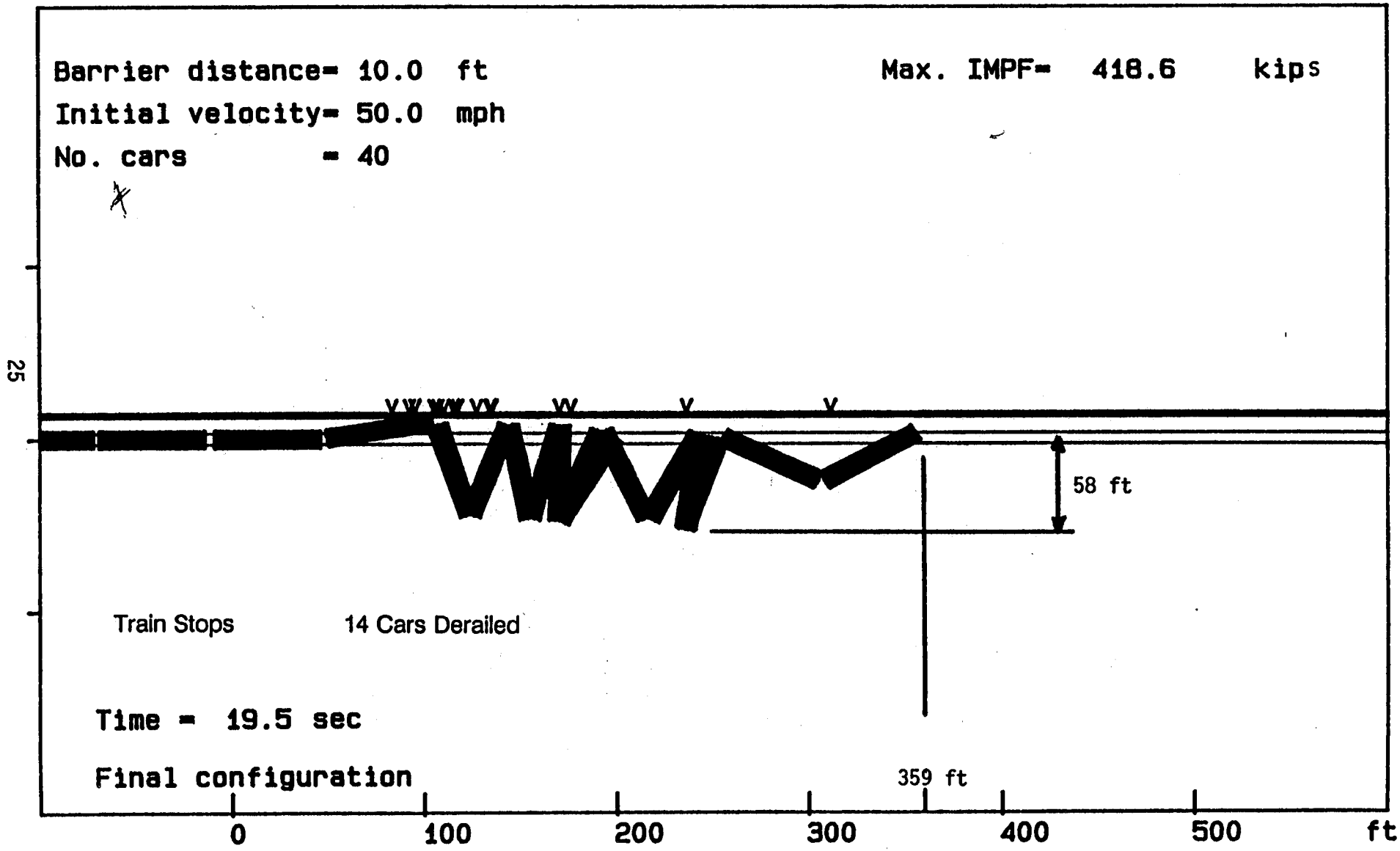


FIGURE 7. (CONTINUED)

260,000 lb. The 40 car rear portion of the derailed train is shown. The face of the barrier is 10 ft from the centerline of the tracks and the initial train speed was 50 mph.

At time 0.0 sec, the first derailed car is 60 ft beyond the derailment point (car length is 60 ft).

At time 3.0 sec, the first car is 237 ft beyond the derailment point and the first barrier impact has occurred producing a 98.6 kips of force.

By time 4.0 sec, the first car has impacted the barrier two times and the second and third cars have hit the barrier twice. The second impact produced a force of 250.1 kips.

By time 5.0 sec, impacts 5 and 6 have occurred and produced an impact force of 273.4 kips magnitude. Car No. 1 has moved 350 ft down the track by this time.

By time 7.0 sec, impacts 7 and 8 have occurred. Impact 7 produced a force of 318.7 kips.

By time 9.0 sec, impacts 9 and 10 have occurred. Impact 9 produced an impact force of 341.2 kips.

By time 14.0 sec, impacts 11 and 12 have occurred. Impact 11 produced a force of 376.4 kips.

By time 16.0 sec, impacts 13 and 14 have occurred. Impact 13 produced the maximum force of 418.6 kips.

By time 19.5 sec, the train has stopped with 14 cars derailed and 15 or more total impacts against the barrier wall. The maximum lateral displacement of the cars was 58 ft and the maximum longitudinal displacement was 359 ft. The maximum impact force was 418.6 kips and the first was 98.6 kips.

Figure 8 shows the force vs. time plot of Car 1 first impact from Figure 7 at 3 sec. The impact rise time is 62 kips in 2 sec or 31 kips/sec. If we divide this by the spring rate of K = 80 kips/ft, we get an impact velocity of 0.39 ft/sec.

Figure 9 shows the force vs. time plot of Car 10 which produced the maximum impact force of 418.6 kips from Figure 7 at 16.0 sec. The impact rise time was about 470 kips/sec. Dividing this value by the spring rate of K = 80 kips/ft yields an impact velocity of 5.9 ft/sec. This impact velocity will be used later in the barrier design to determine a dynamic load factor for driven piles or drilled shaft foundations.

Figure 10 and Table 4 present the results of a parameter study to determine the barrier impact forces versus the lateral offset distance to the face of the barrier. This train

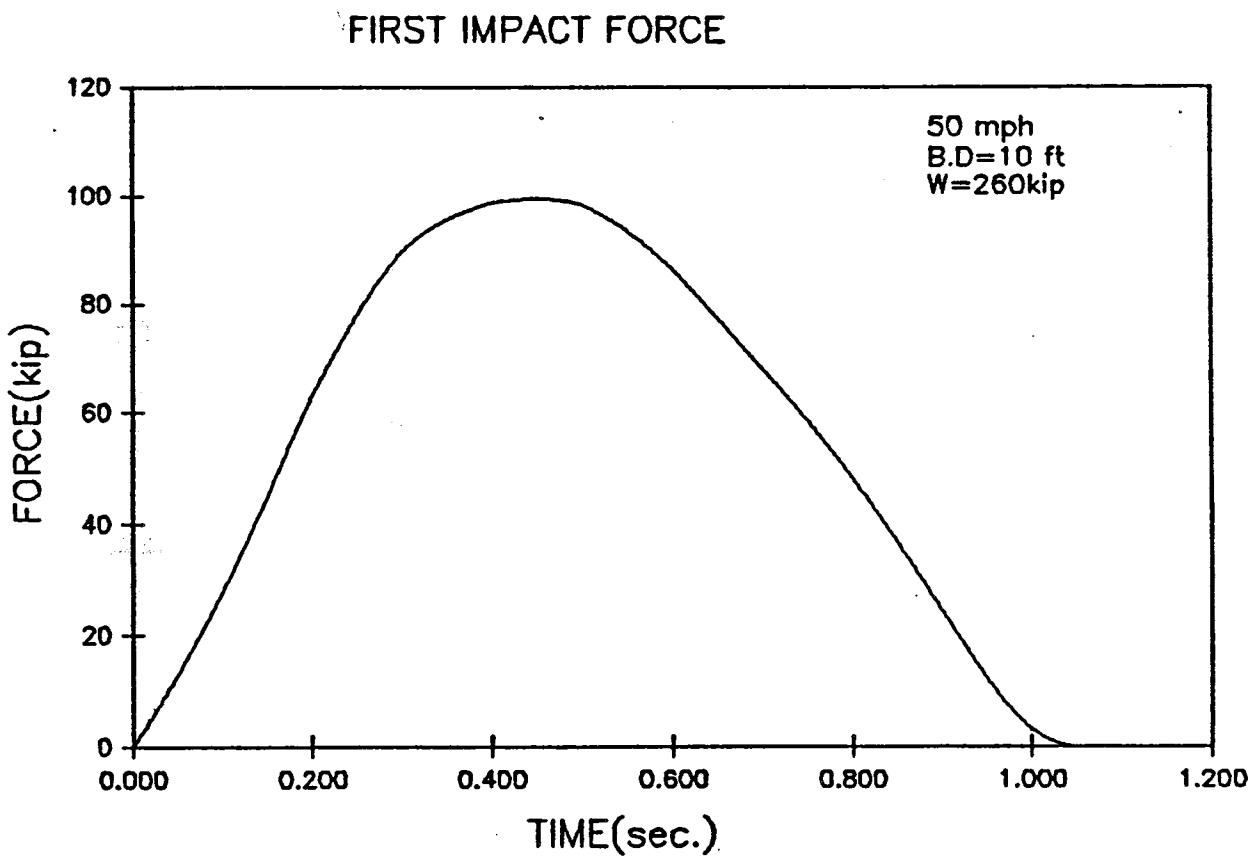


FIGURE 8. FORCE vs TIME PLOT OF CAR 1 FIRST IMPACT (see FIG. 7 at 3 sec)

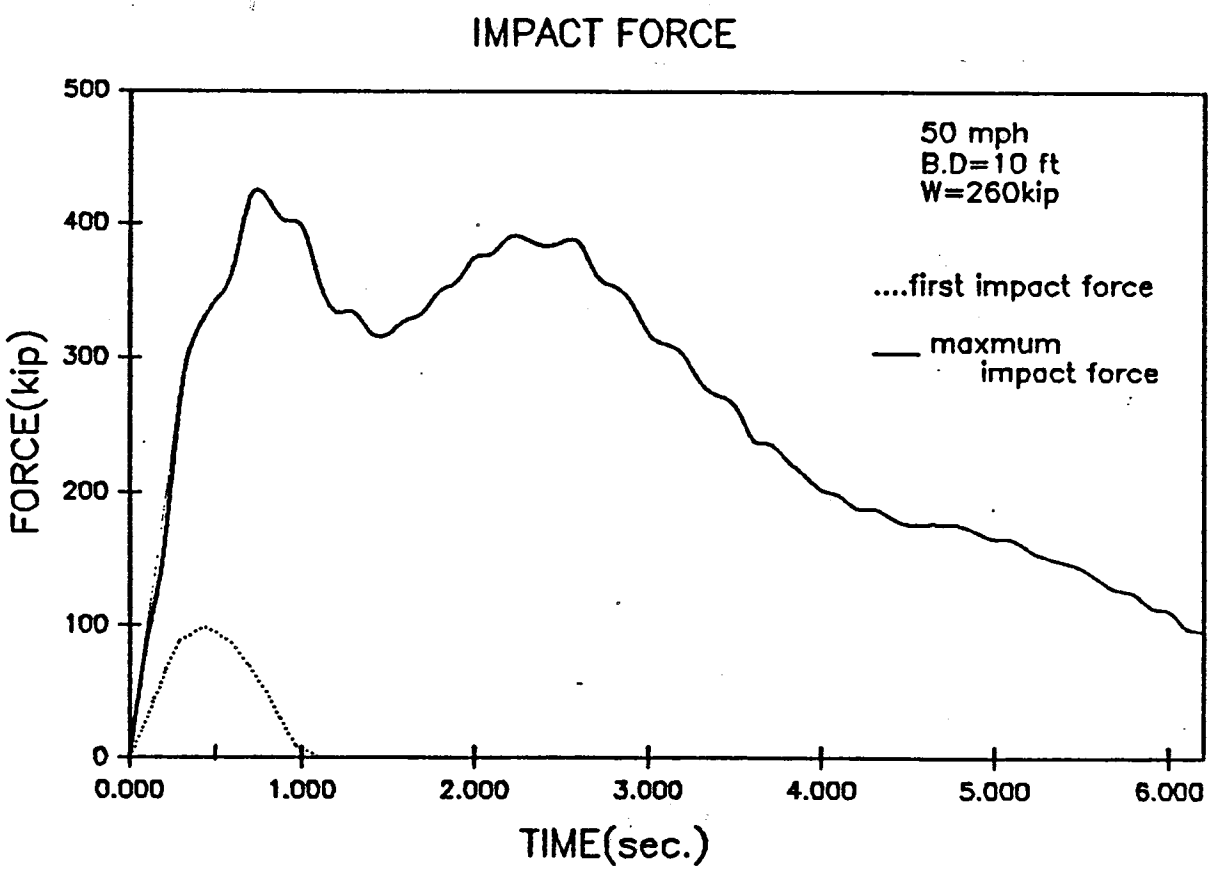


FIGURE 9. FORCE vs TIME PLOT OF CAR 10 THE MAXIMUM IMPACT FORCE (see FIG. 7 at 16.0 sec)

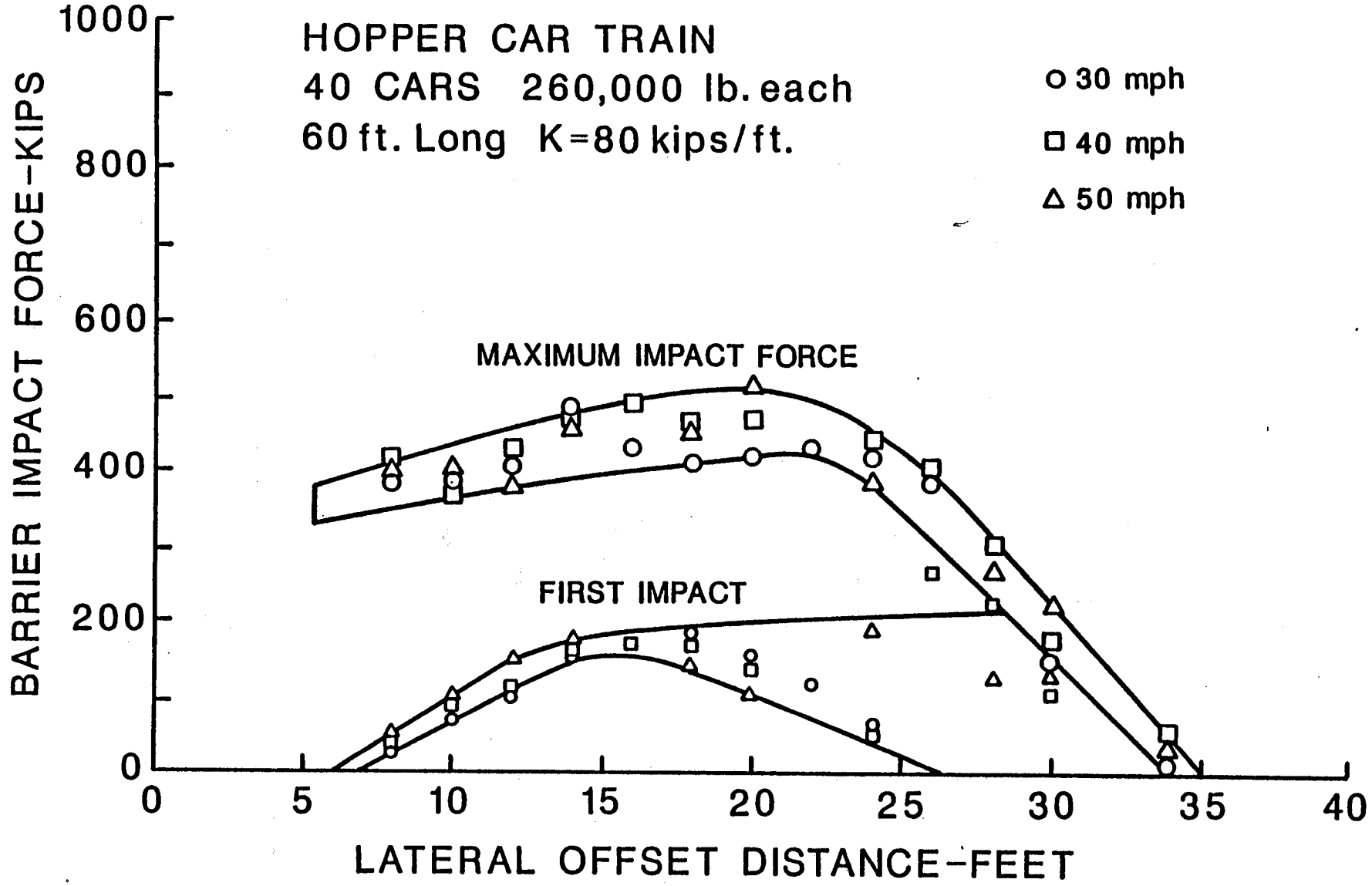


FIGURE 10. IMPACT FORCE vs DISTANCE TO BARRIER.
K = 80 kips/ft

TABLE 4. PARAMETER STUDY OF 40 CAR HOPPER TRAIN-BARRIER IMPACTS
K=80.0 kips/ft, W=260.0 kips, L=60.0 ft

V=30.0 mph

Barrier Dis(ft)	First IMF(kip)	Max IMF(kip)
8	34.867	403.965
10	71.216	386.975
12	115.938	425.709
14	158.022	482.969
16	177.464	434.075
18	182.707	411.413
20	159.231	421.867
22	119.634	434.005
24	64.253	421.497
26	389.903	389.903
28	319.324	319.324
30	239.615	239.615
34	5.728	6.934

V=40.0 mph

Barrier Dis(ft)	First IMF(kip)	Max IMF(kip)
8	41.058	416.208
10	91.130	369.433
14	162.727	474.523
16	171.565	494.748
18	169.326	465.703
20	137.638	474.853
24	53.530	443.446
26	273.403	363.179
28	225.931	302.108
30	111.321	178.646
34	62.003	62.003

V=50.0 mph

Barrier Dis(ft)	First IMF(kip)	Max IMF(kip)
8	46.735	396.735
10	97.439	402.151
12	149.585	382.275
14	165.529	468.908
18	145.861	460.506
20	103.026	509.765
24	183.656	392.058
28	123.432	273.973
30	134.632	224.328
34	39.129	39.129

was the same 40 car grain hopper train fully loaded. The initial train derailment speed was varied from 30, 40, and 50 mph. The dark lines on Figure 10 show the lower and upper bounds of the forces computed. This Figure 10 was produced using a train car stiffness of K = 80 kips/ft.

Figure 11 and Table 5 present the results of a parameter study to determine the barrier impact forces versus the lateral offset distance to the face of the barrier. This is the same train as shown on Figure 8 but the train car stiffness was increased to K = 218.2 kips/ft. The K = 80 kips/ft was based only on the empty weight of the train cars (see Appendix B).

The K = 218 kips/ft used for Figure 11 might be justified if the pay load in the cars contributed to the car stiffness. At this time the value of K = 80 kips/ft is believed to be more realistic and the barrier impact forces on Figure 10 and Table 4 are recommended for design of a railroad car barrier.

Figures 10 and 11 are both presented so the significance of the train car stiffness K can be seen. These figures indicate that the maximum impact force is approximately proportional to the square root of the stiffnesses K.

Example: The max. force from Figure 10 is 490 kips. The max. force from Figure 11 is 820 kips.

$$\frac{490}{820} = 0.60 \qquad \text{while} \qquad \sqrt{\frac{80}{218}} = 0.60$$

The barrier impact forces shown on Figure 10 were developed using trains with cars of uniform length (60 ft) and weight (260,000 lb). Figure 12 shows some results for a train of mixed car lengths (50 ft, 90 ft, and 60 ft) and mixed car weights (260,000 lb, 110,000 lb, and 180,000 lb). The first impact force of 89 kips and the maximum impact force 490.9 kips are very close to those shown on Figure 10.

Occasionally it may be desirable to confine a train between two parallel barriers in order to control the lateral displacement of the derailed train or to reduce the lateral impact force. Figures 14, 15, and 16 show three such simulations with the barrier located 9 ft, 10 ft, and 14 ft, respectively, from the centerline of the track. It can be seen that the left side barrier (barrier first impacted) is always larger than the right side barrier.

Table 6 presents the results of a dual barrier parameter study with various offset distances and train speeds. It can be seen that train speeds of 40 mph and 50 mph

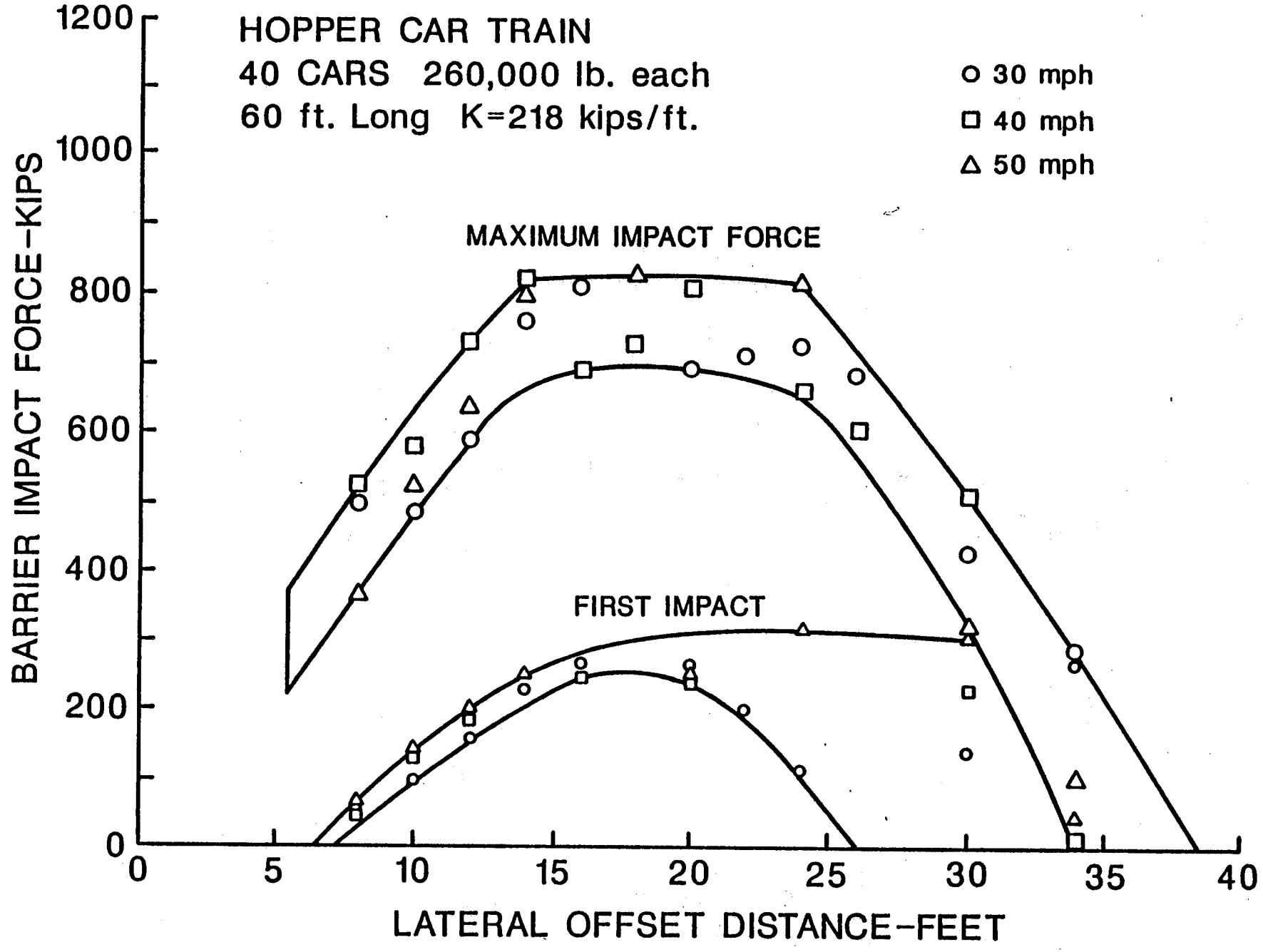


FIGURE 11. IMPACT FORCE vs DISTANCE TO BARRIER.
K = 218 kips/ft

TABLE 5. PARAMETER STUDY OF 40 CAR HOPPER TRAIN-BARRIER IMPACTS
K=218.2 kips/ft, W=260.0 kips, L=60.0 ft

V=30.0 mph

Barrier Dis(ft)	First IMF(kip)	Max IMF(kip)
8	56.711	499.872
10	99.825	488.330
12	158.940	589.105
14	228.541	761.215
16	264.398	810.033
20	262.390	694.575
22	200.117	713.880
24	113.528	725.257
26	668.039	686.067
30	240.774	430.819
34	292.115	292.115

V=40.0 mph

Barrier Dis(ft)	First IMF(kip)	Max IMF(kip)
8	57.935	525.913
10	128.586	578.997
12	187.428	734.192
14	247.074	757.757
16	252.640	693.248
18	265.945	733.947
20	239.495	814.778
24	42.509	658.605
26	468.950	605.043
30	229.490	510.810
34	9.830	9.830

V=50.0 mph

Barrier Dis(ft)	First IMF(kip)	Max IMF(kip)
8	66.282	367.349
10	135.960	521.019
12	200.283	642.525
14	259.471	804.285
18	237.257	828.192
24	311.657	824.503
30	315.288	316.756
34	55.137	98.970

TTI BARRIER/DERAILMENT INTERACTION PROGRAM

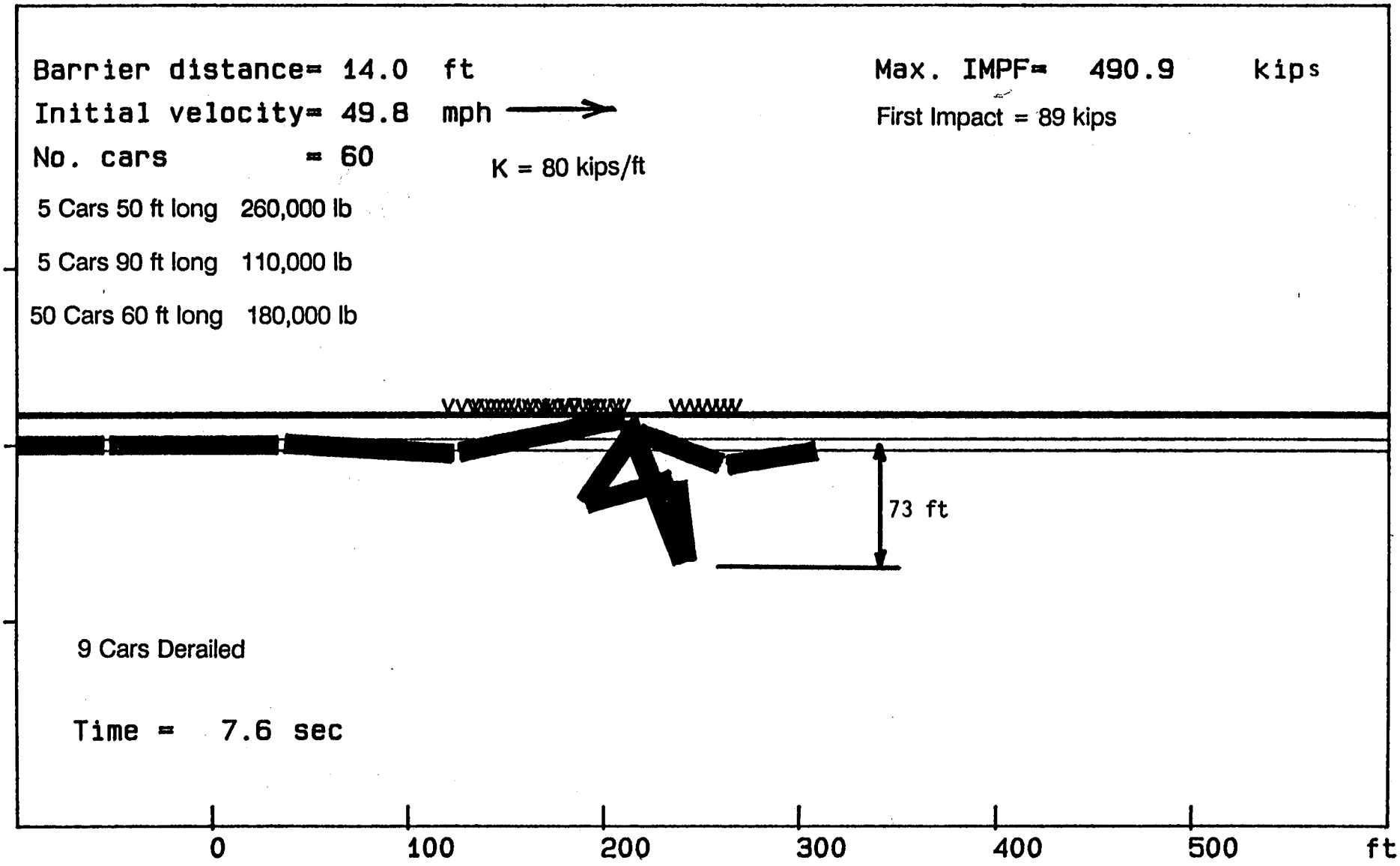


FIGURE 12. MIXED TRAIN CAR LENGTH AND WEIGHT

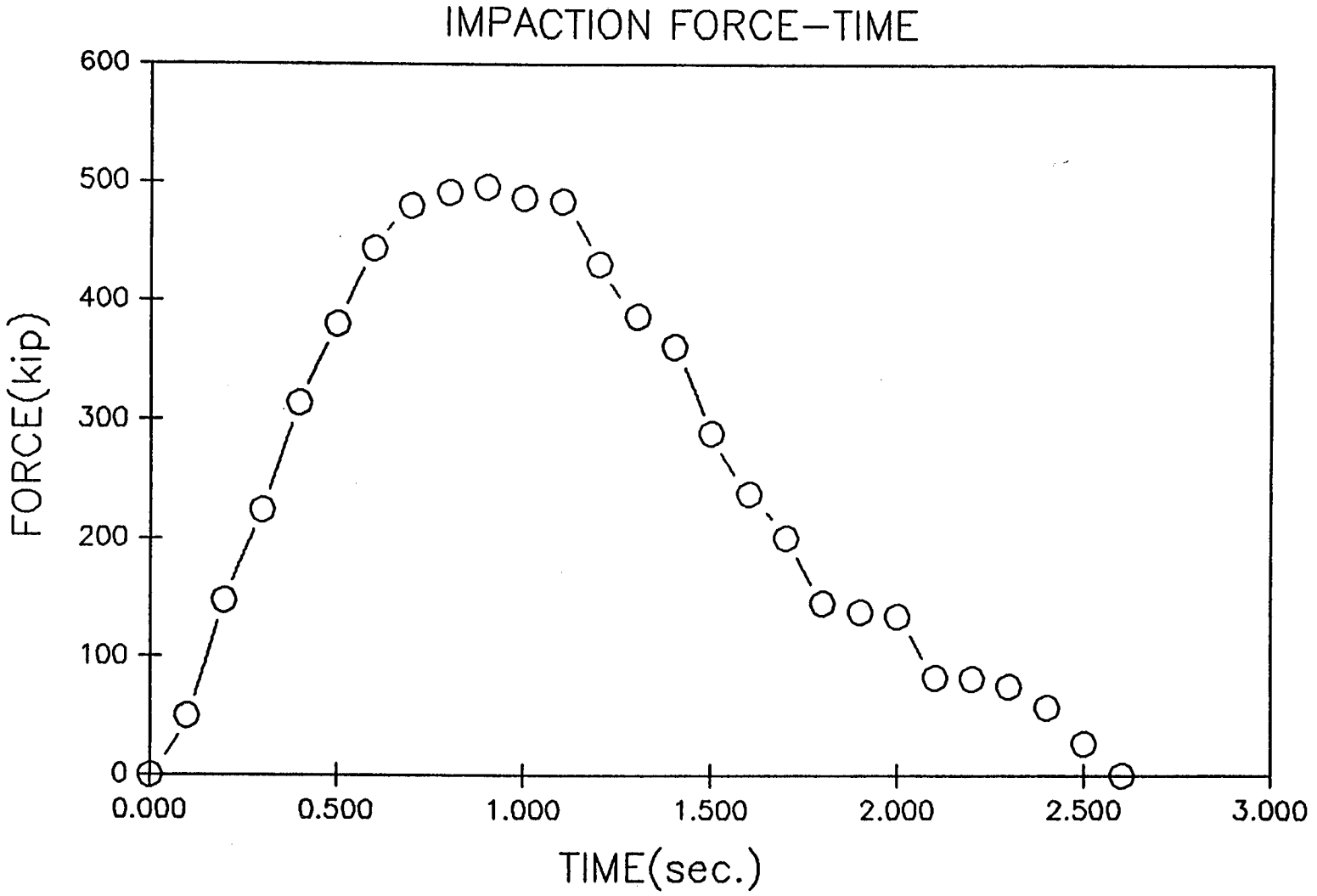


FIGURE 13. FORCE vs TIME FOR MIXED TRAIN OF FIG. 12.

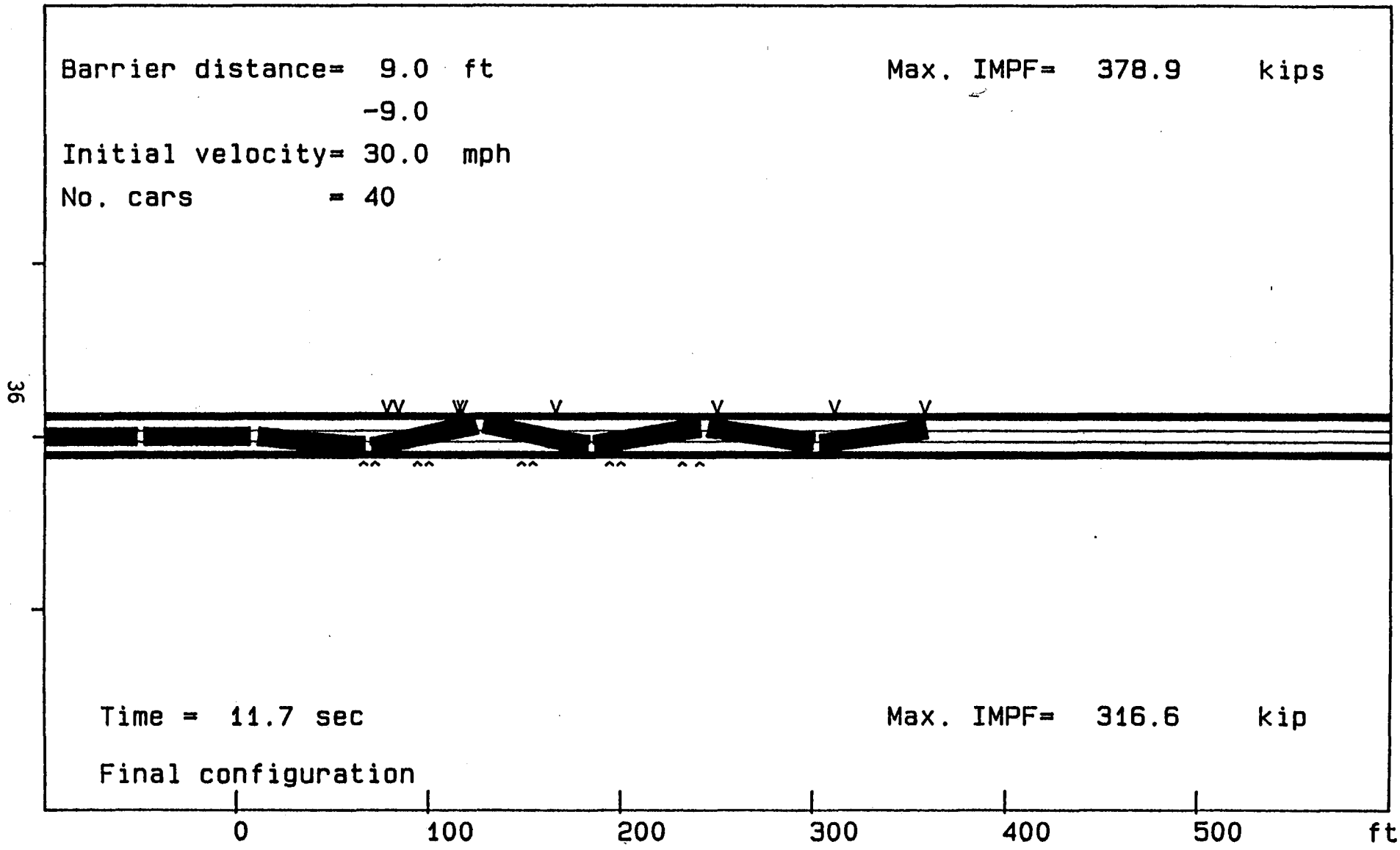


FIGURE 14. DUAL BARRIER WITH 9, FT OFFSET.

TTI BARRIER/DERAILMENT INTERACTION PROGRAM

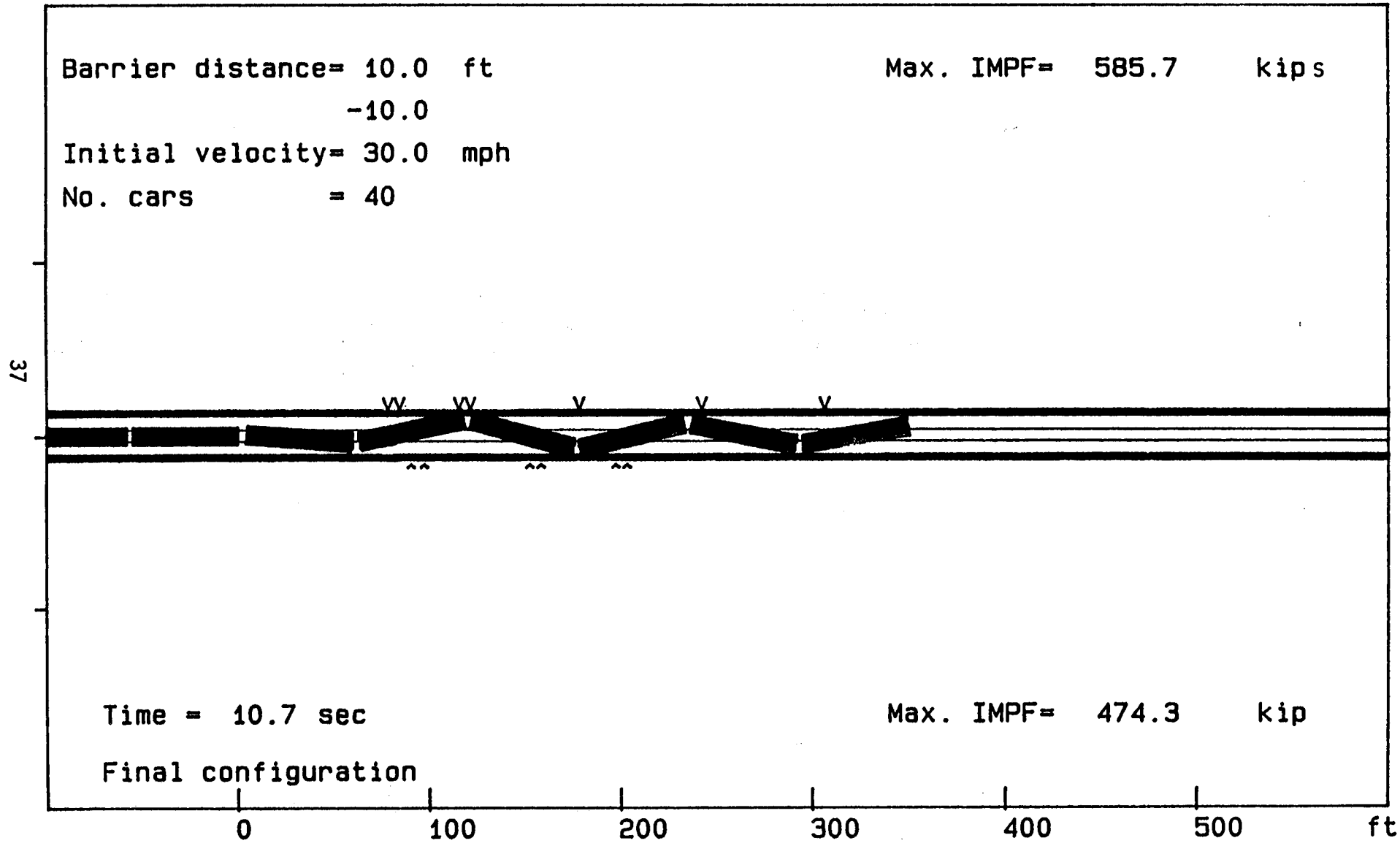


FIGURE 15. DUAL BARRIER WITH 10 FT OFFSET.

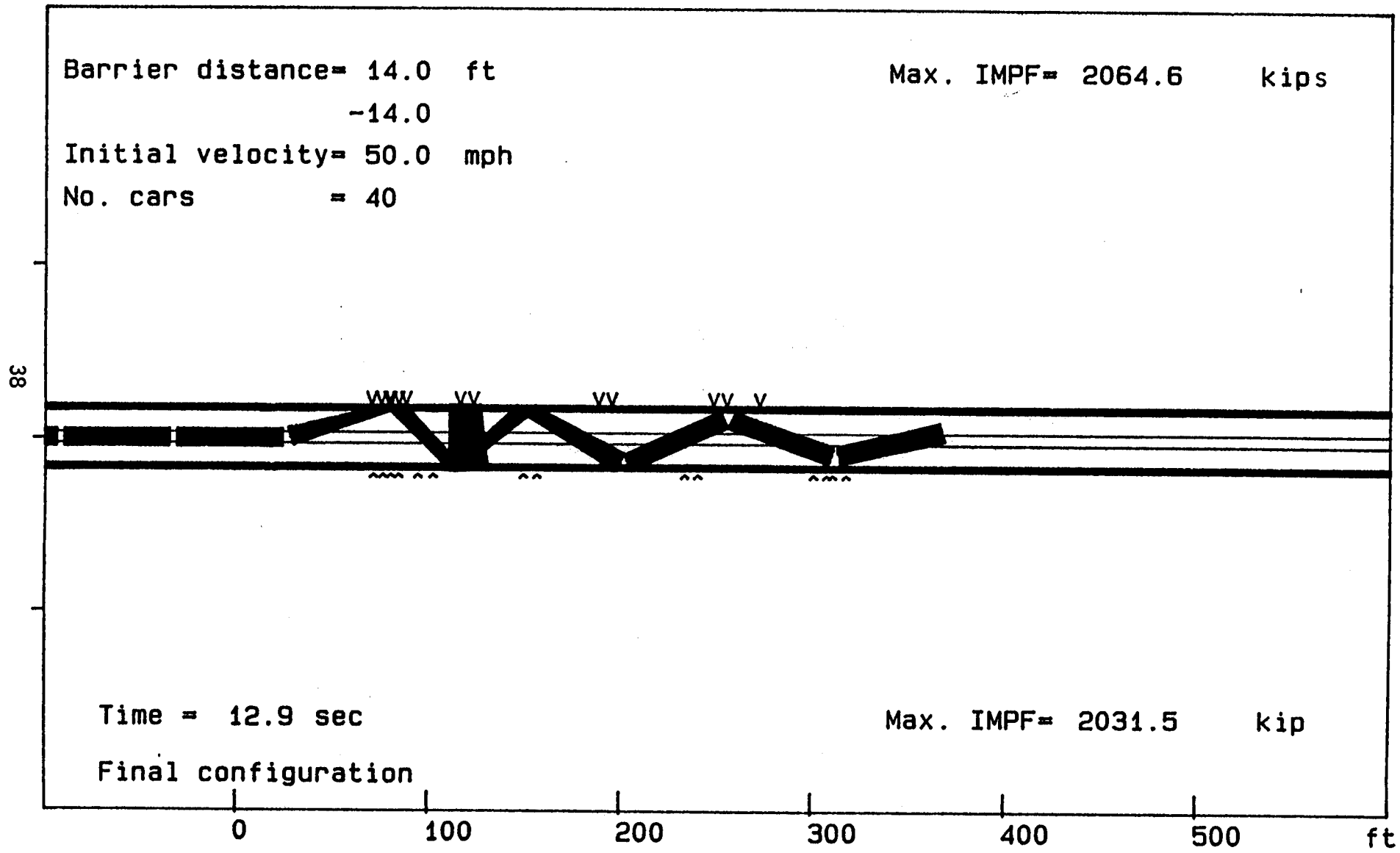


FIGURE 16. DUAL BARRIER WITH 14 FT OFFSET.

TABLE 6. PARAMETER STUDY OF 40 CAR HOPPER TRAIN-BARRIER IMPACTS WITH DUAL BARRIERS
K=80 kips/ft, W=260.0 kips, L=60.0 ft

V=30 mph

BARRIER	DISTANCE	MAX IMPACT FORCE
8	- 8	290.98
9	- 9	378.90
10	-10	585.75
12	-12	1201.00
14	-14	1612.45
16	-16	1561.59
20	-20	1295.06
24	-24	881.85
28	-28	492.59
32	-32	183.50

V=40 mph

BARRIER	DISTANCE	MAX IMPACT FORCE
8	- 8	774.06
10	-10	1721.70
12	-12	2031.05
14	-14	1742.88
16	-16	1748.71
18	-18	1381.93
20	-20	1141.49
24	-24	824.11
28	-28	505.61
32	-32	190.76

V=50 mph

BARRIER	DISTANCE	MAX IMPACT FORCE
8	- 8	2038.79
10	-10	2220.63
12	-12	2446.75
14	-14	2064.59
16	-16	1626.40
18	-18	1337.67
20	-20	1143.36
24	-24	825.51
28	-28	533.37
32	-32	202.96

produce very large impact forces. At a train speed of 30 mph and barrier offset distances of 8 or 9 ft, impact forces are actually smaller than those obtained from a single barrier (see Figure 17).

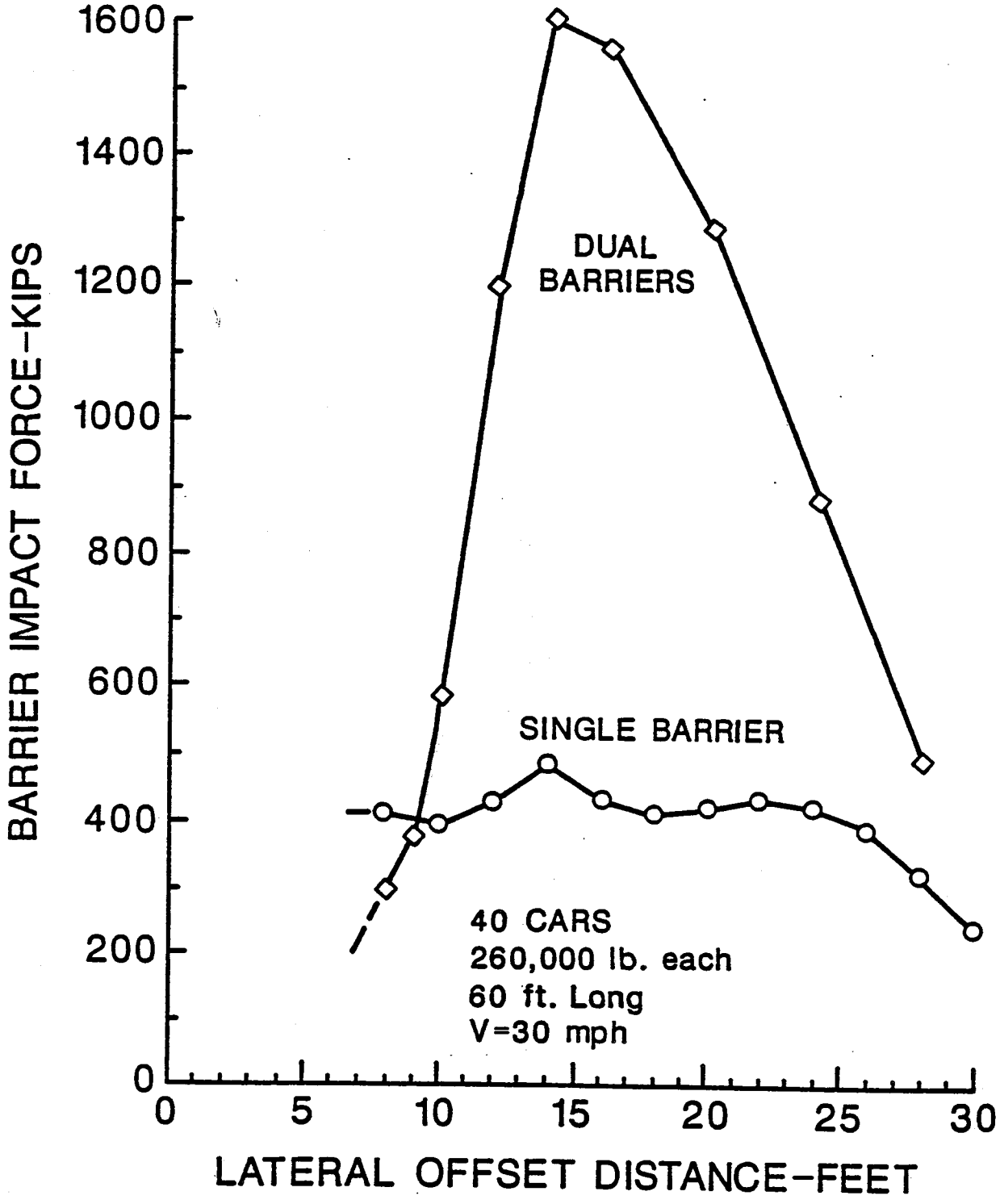


FIGURE 17. TRAIN IMPACT FORCE vs DISTANCE TO BARRIER
K = 80 kips/ft

ANALYSIS OF CSX TRAIN DERAILMENTS OF 6-19-87 AND 9-5-87 IN THE COMMON CORRIDOR

Two train derailments which occurred on 6-19-87 and 9-5-87 in the B-6 common corridor were simulated with the train derailment computer program. The FRA accident reports were used to obtain the following data.

Figure 18 is an aerial photograph of the derailment of 6-19-87 and this was the best documented derailment. The train was 135 cars long and traveling an estimated 42 mph at the time of derailment. Cars 104 thru 115 were derailed blocking both CSX tracks and both Metro tracks (Figure 18).

Figure 19 shows the computer simulation of this train derailment of 6-19-87 with no barrier in place. The final location of Cars No. 104 thru 113 are very similar in Figures 18 and 19. An even closer final location of cars could have been obtained if the ground friction was decreased from 1.0 and the track braking friction increased from 0.2. This form of derailment reconstruction was not done at this time, however.

Figure 20 shows the computer simulation of the train derailment of 6-19-87 with the proposed barrier system in place 10 ft from the CSX track centerline. The maximum barrier impact force was 268.2 kips, well below those recommended for design on Figure 10 (402 kips). The cars would displace laterally off the CSX tracks about 80 ft compared to 44 ft with no barrier in place. This lateral displacement of 80 ft is probably not significant since a mixed train today could displace laterally 112 ft with no barrier present (see Figure 5).

Figure 21 shows the train derailment with Car 104 directed toward the barrier. This is the way the train derailments summarized by Figure 10 were simulated. The maximum barrier impact force of 281.6 kips is more than that obtained on Figure 20 (268.2 kips) which is the more accurate 6-19-87 derailment simulation.

Figure 22 shows the computer simulation of the train derailment of 9-5-87. This train was 90 cars long and all cars were loaded "piggyback" type flat cars. The train was traveling at 55 mph at the time Cars 45 thru 56 were derailed (12 cars derailed). The computer simulation of Figure 18 shows 16 cars derailed which might indicate the train was traveling slower than 55 mph or the brake friction was greater than 0.2 assumed.

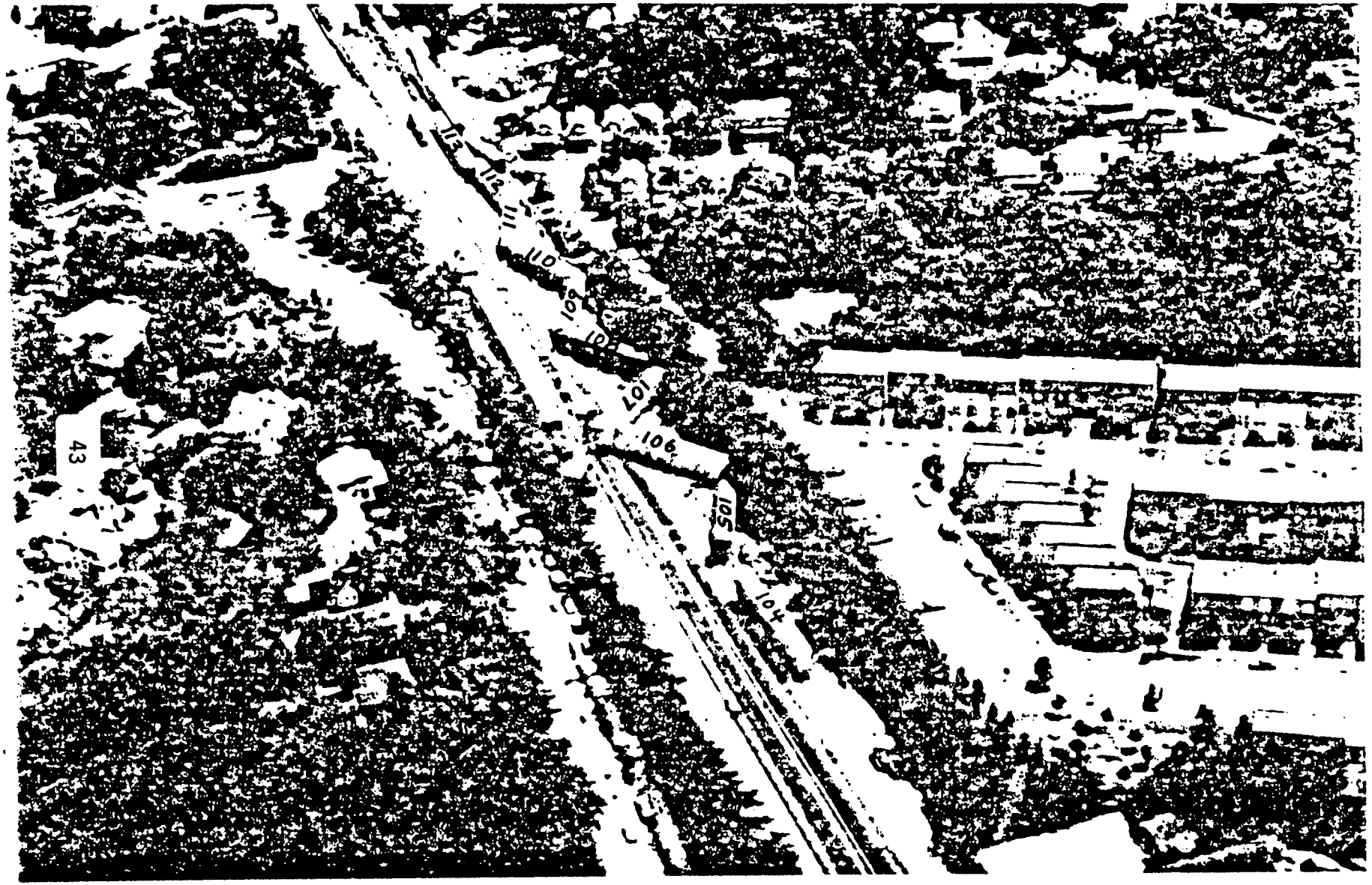


FIGURE 18. AERIAL PHOTOGRAPH OF TRAIN DERAILMENT OF 6-19-87

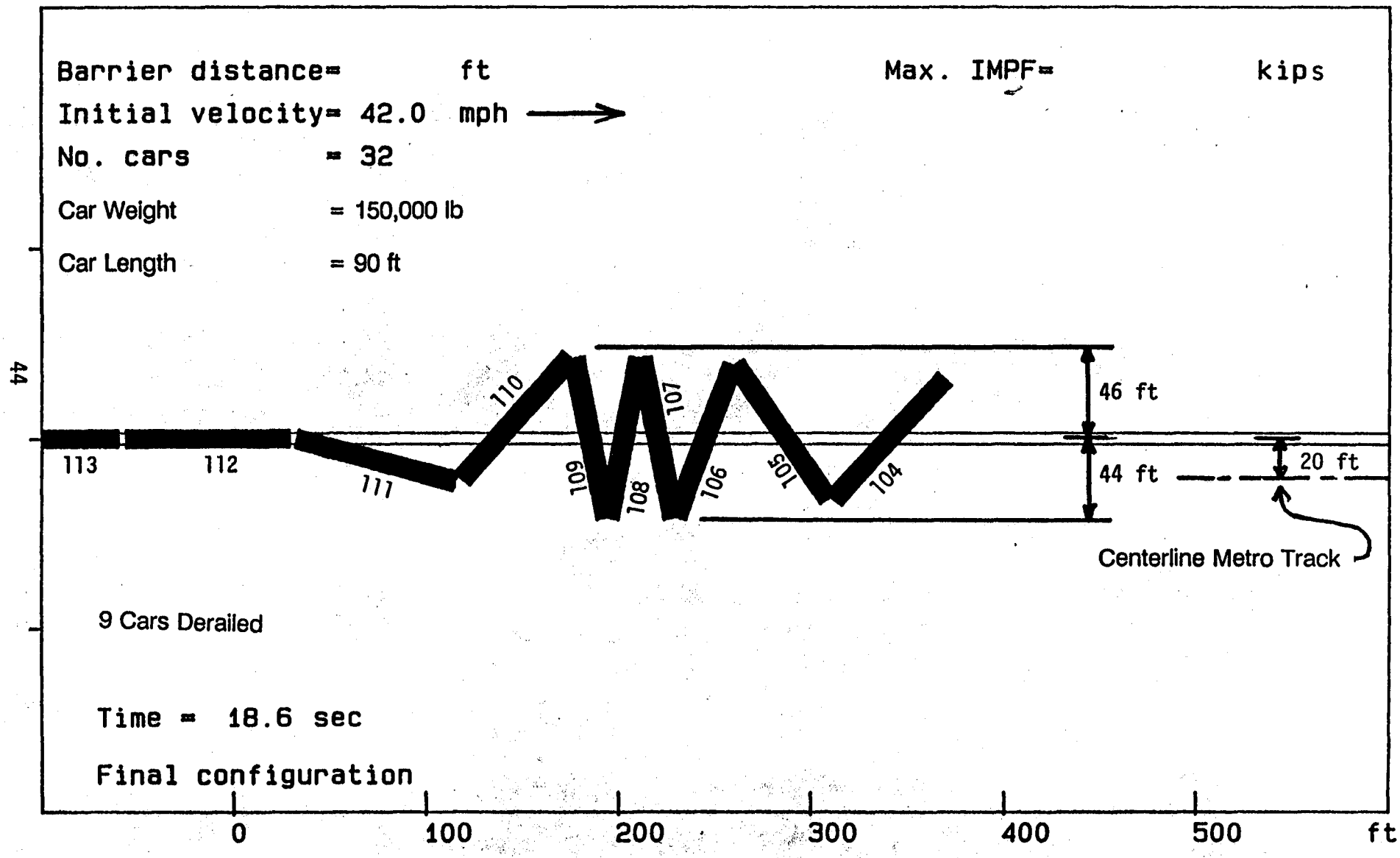


FIGURE 19. COMPUTER SIMULATION OF TRAIN DERAILMENT 6-19-87

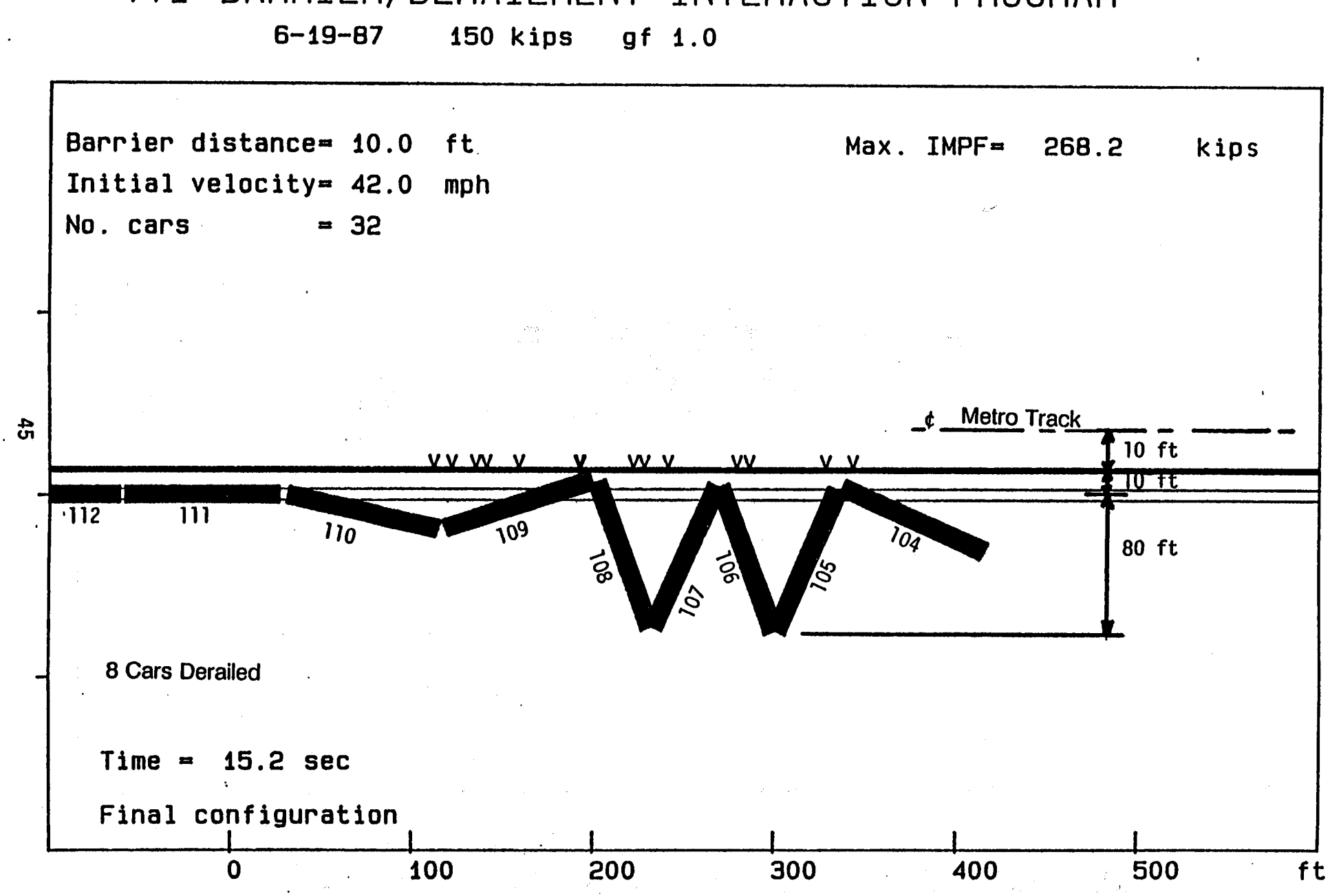


FIGURE 20. SIMULATION OF TRAIN DERAILMENT OF 6-19-87 WITH BARRIER SYSTEM IN PLACE

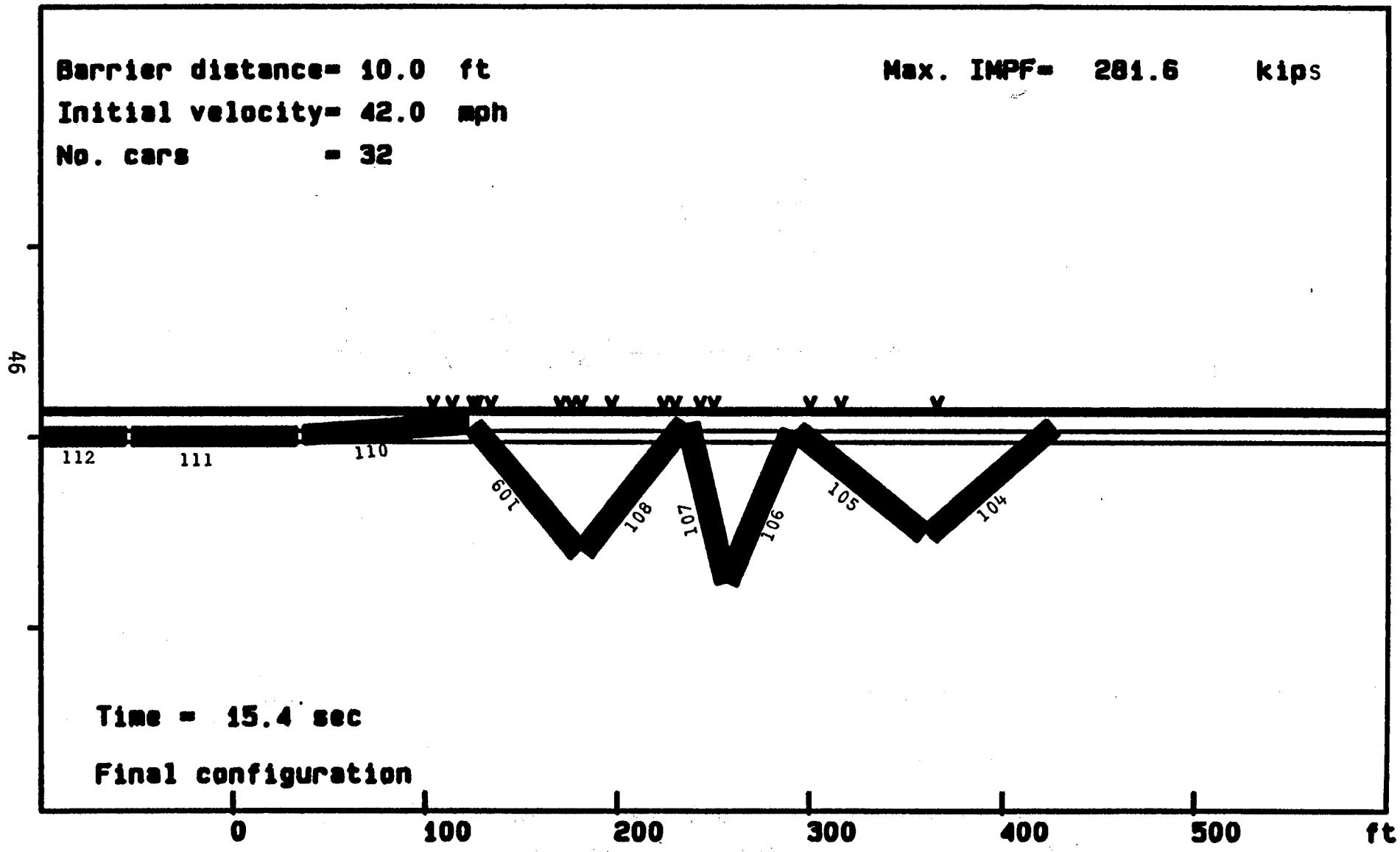


FIGURE 21. SIMULATION OF TRAIN DERAILMENT WITH CAR 104 DIRECTED TOWARD BARRIER

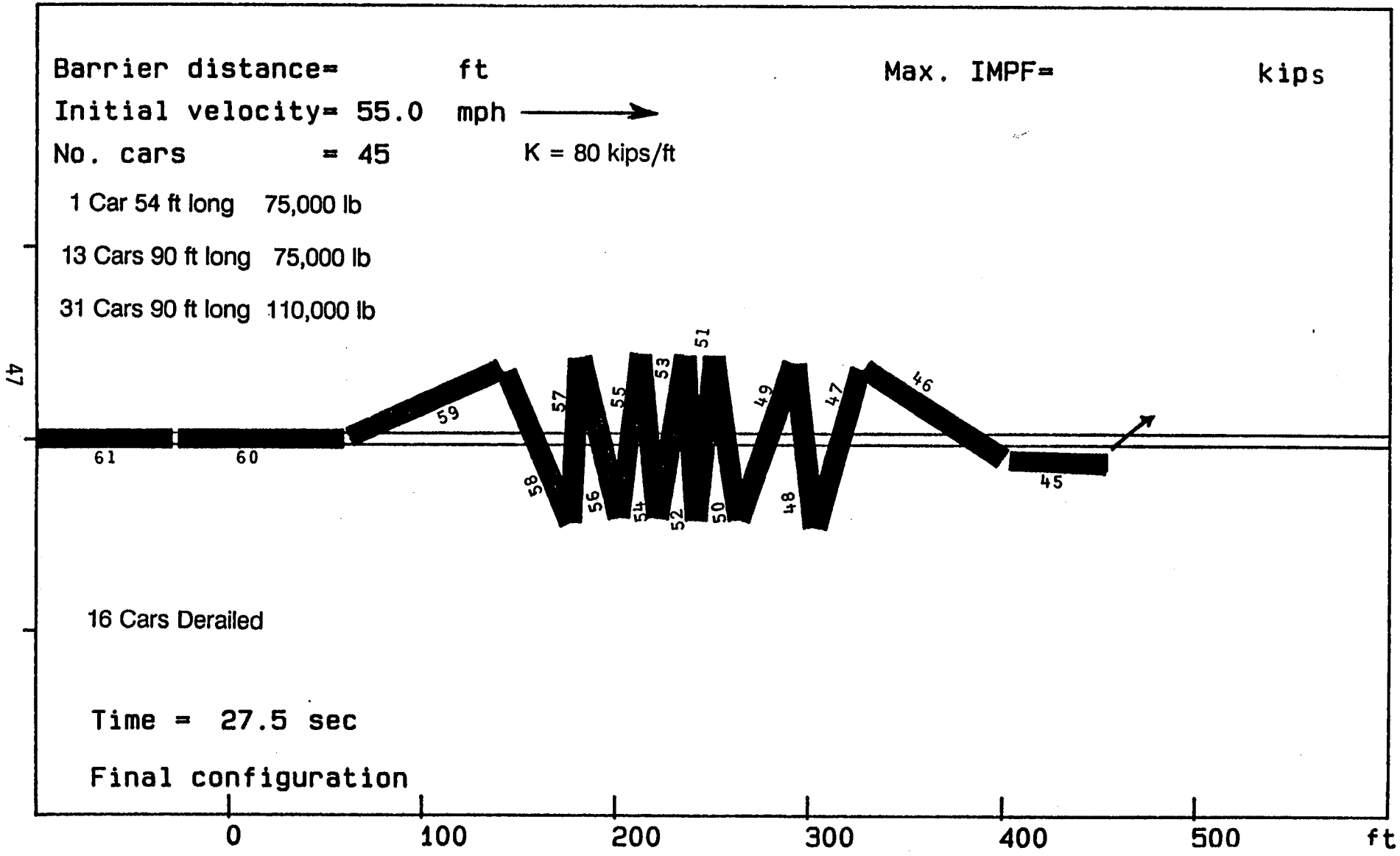


FIGURE 22. COMPUTER SIMULATION OF TRAIN DERAILMENT 9-5-87

Figure 23 shows the simulation of the 9-5-87 train derailment with the proposed barrier system in place. Car 45 was directed toward the barrier and this produced a maximum impact force of 254.5 kips.

Figure 24 shows the simulation with Car 45 directed away from the barrier. This produced a maximum impact force of 197.5 kips. From the data presented in the accident report, it is not known whether Figure 23 or 24 is the more accurate simulation. Without the barrier the maximum lateral train displacement was 48 ft, and with the barrier the maximum lateral displacement was 85 ft. The 85 ft is still less than the 112 ft shown in Figure 5.

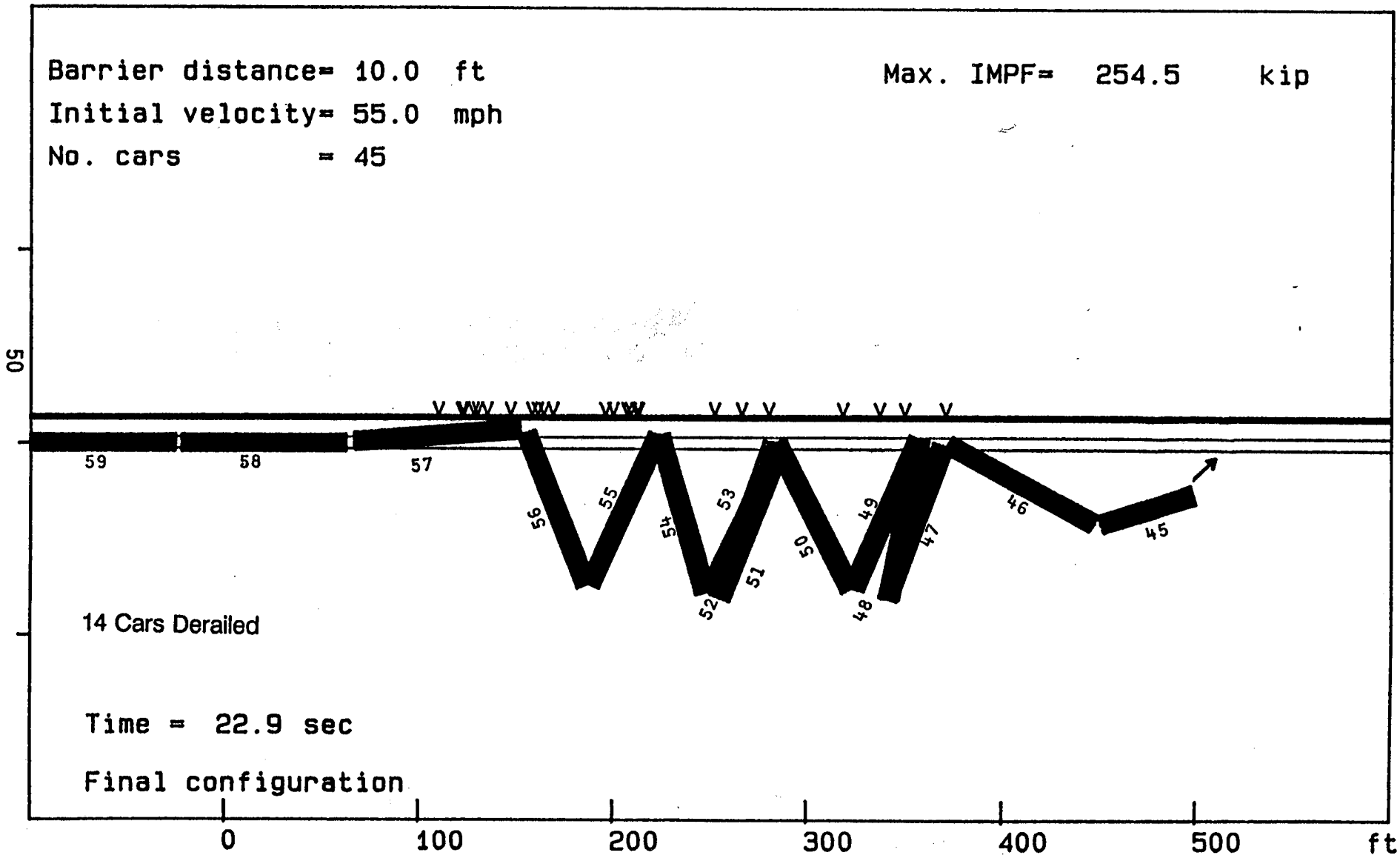


FIGURE 23. SIMULATION OF TRAIN DERAILMENT 9-5-87 WITH BARRIER SYSTEM IN PLACE

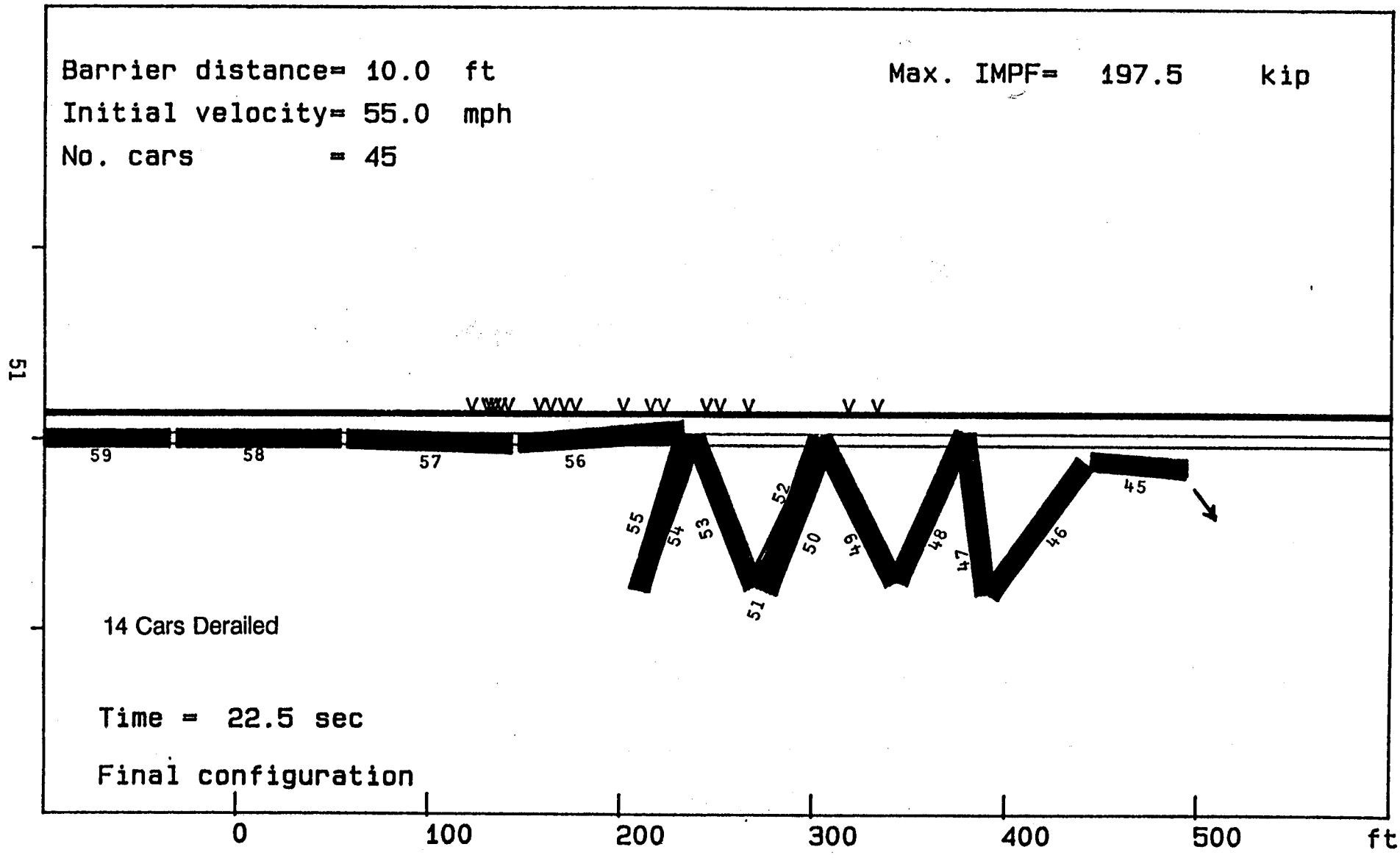


FIGURE 24. SIMULATION OF TRAIN DERAILMENT WITH CAR 45 DIRECTED AWAY FROM BARRIER

ANALYSIS OF METRO TRAIN IMPACT FORCES ON LONGITUDINAL BARRIERS

Figure 25 shows the results of a computer simulation of an eight car Metro train traveling at 70 mph when it derails and impacts a longitudinal barrier wall. The first impact into the longitudinal barrier 14 ft away was 60.1 kips and the maximum impact force was 124.7 kips (see Table 6). Figures 26 and 27 show a plot of the force vs. time for these two impacts. From Figure 27 the maximum rise time is 770 kips/sec which, if divided by K = 80 kips/ft, yields a loading velocity of 9.6 ft/sec.

Figure 28 and Table 7 summarize the impact forces of the Metro train into a single barrier located various distances from the centerline of the track.

Figure 29 shows the results of a computer simulation of the Metro train impacting two parallel barriers (one on each side of the track). With the barrier 16 ft from the centerline of the track, the right side impact force of 196.2 kips was the largest.

Figure 30 shows a simulation when one barrier is 10 ft from the track, the other barrier is 24 ft from the track. This situation would exist in the B-6 corridor (see Figure 1). It can be seen that the impact force is exactly the same as that from the single barrier 10 ft off the track since the Metro does not hit the barrier 24 ft off the track centerline.

Table 8 presents a summary of Metro train-barrier impact force for barriers various distances from the track.

Figure 31 compares dual barrier impact forces with single barrier impact forces. For lateral offset distances of 7, 8, or 9 ft, the dual barrier will have smaller or about equal impact forces.

TTI BARRIER/DERAILMENT INTERACTION PROGRAM

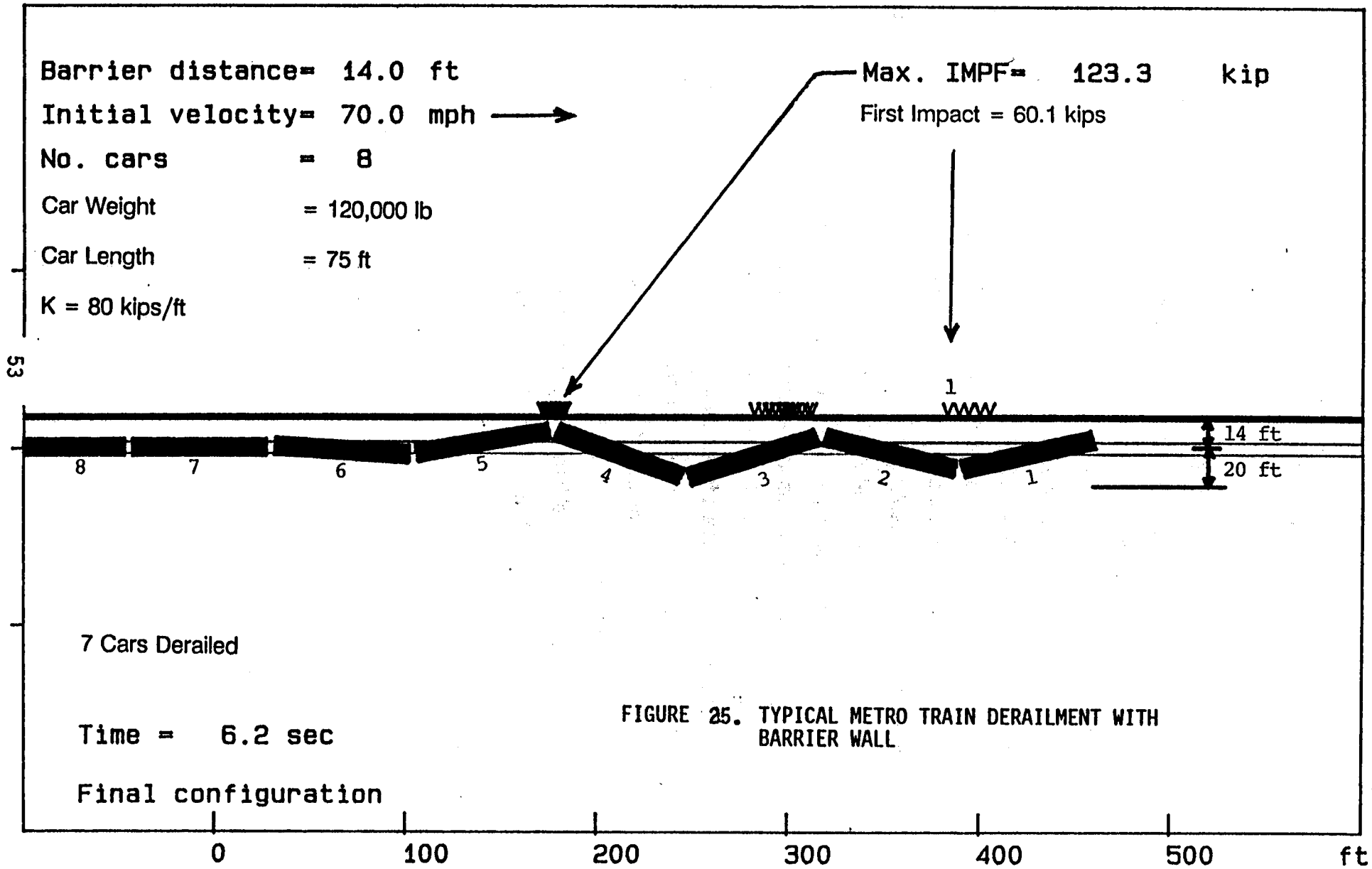


FIGURE 25. TYPICAL METRO TRAIN DERAILMENT WITH BARRIER WALL

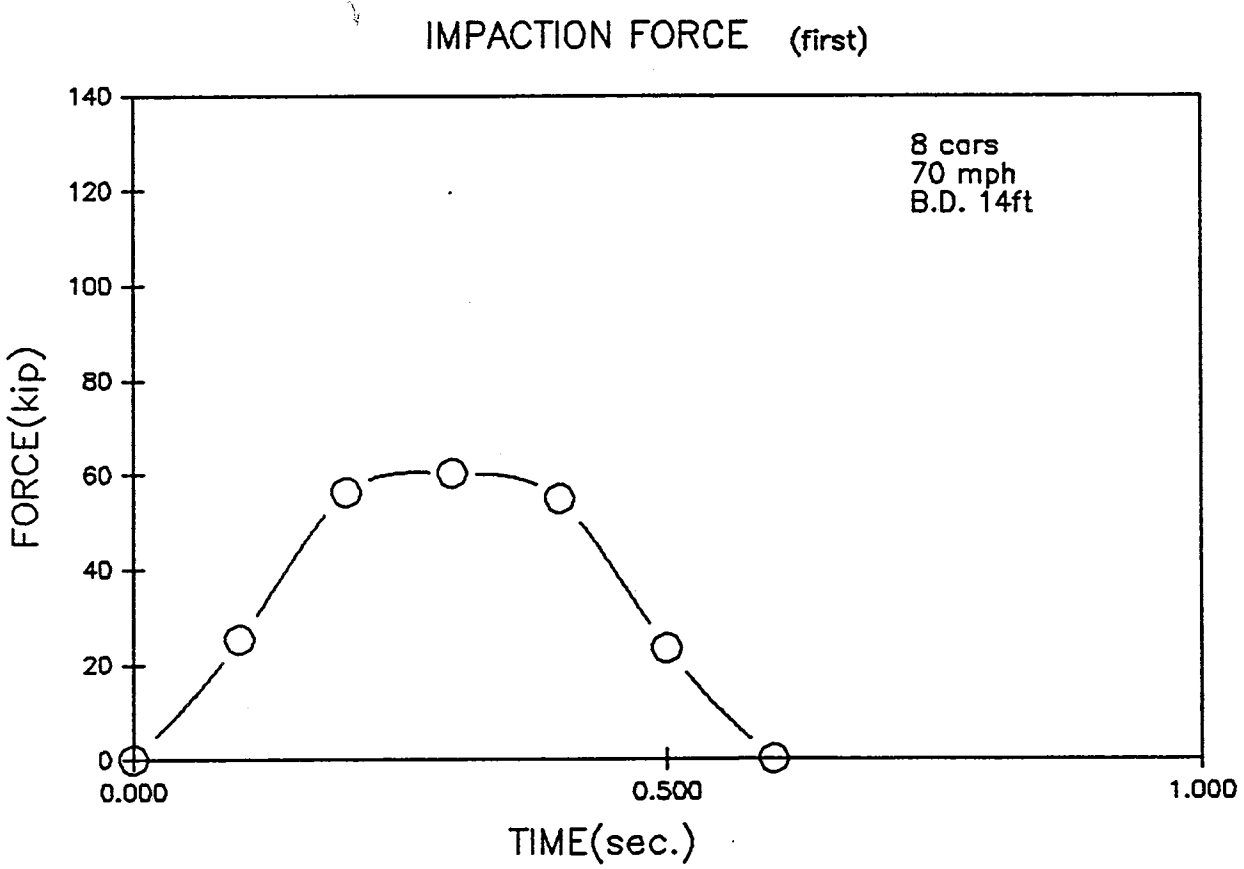


FIGURE 26. FORCE vs TIME PLOT OF CAR 1 FIRST IMPACT

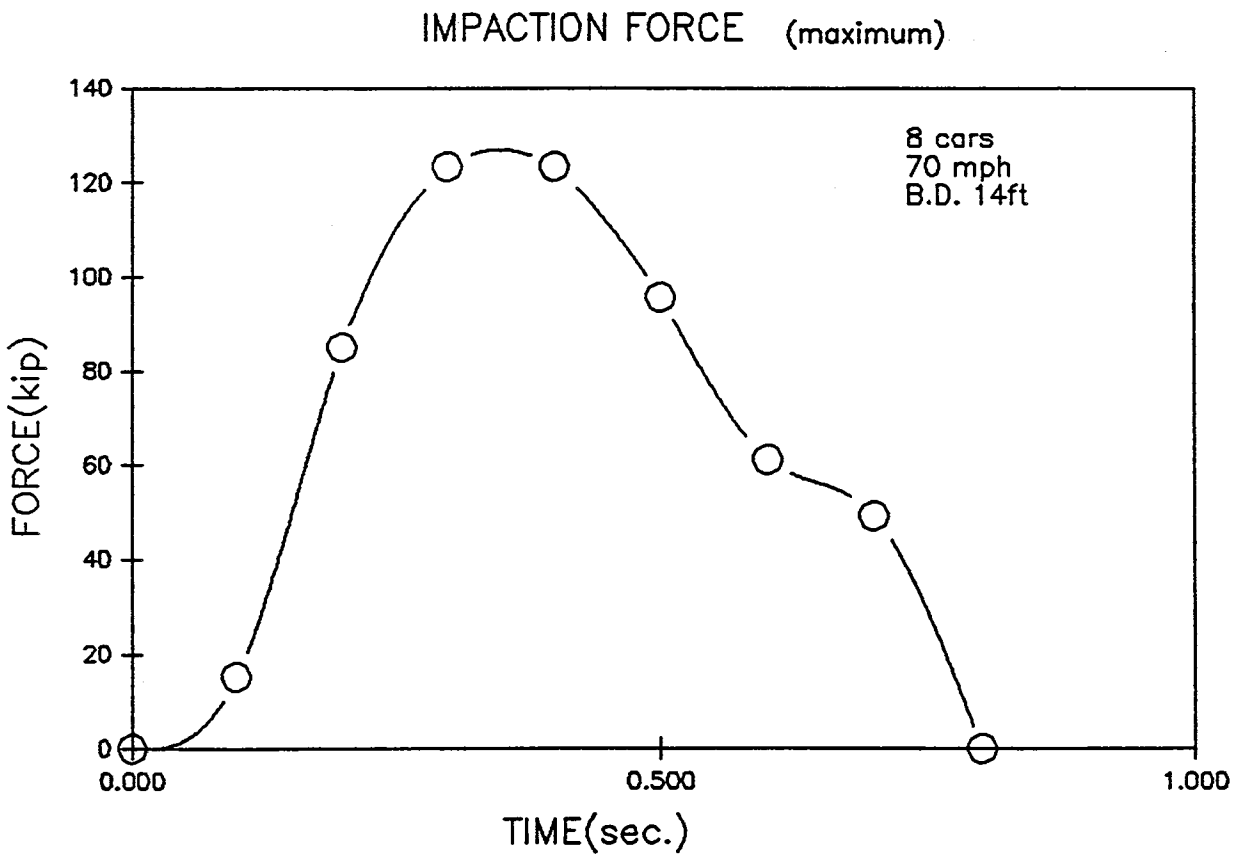


FIGURE 27. FORCE vs TIME PLOT OF CAR 5 THE MAXIMUM IMPACT FORCE

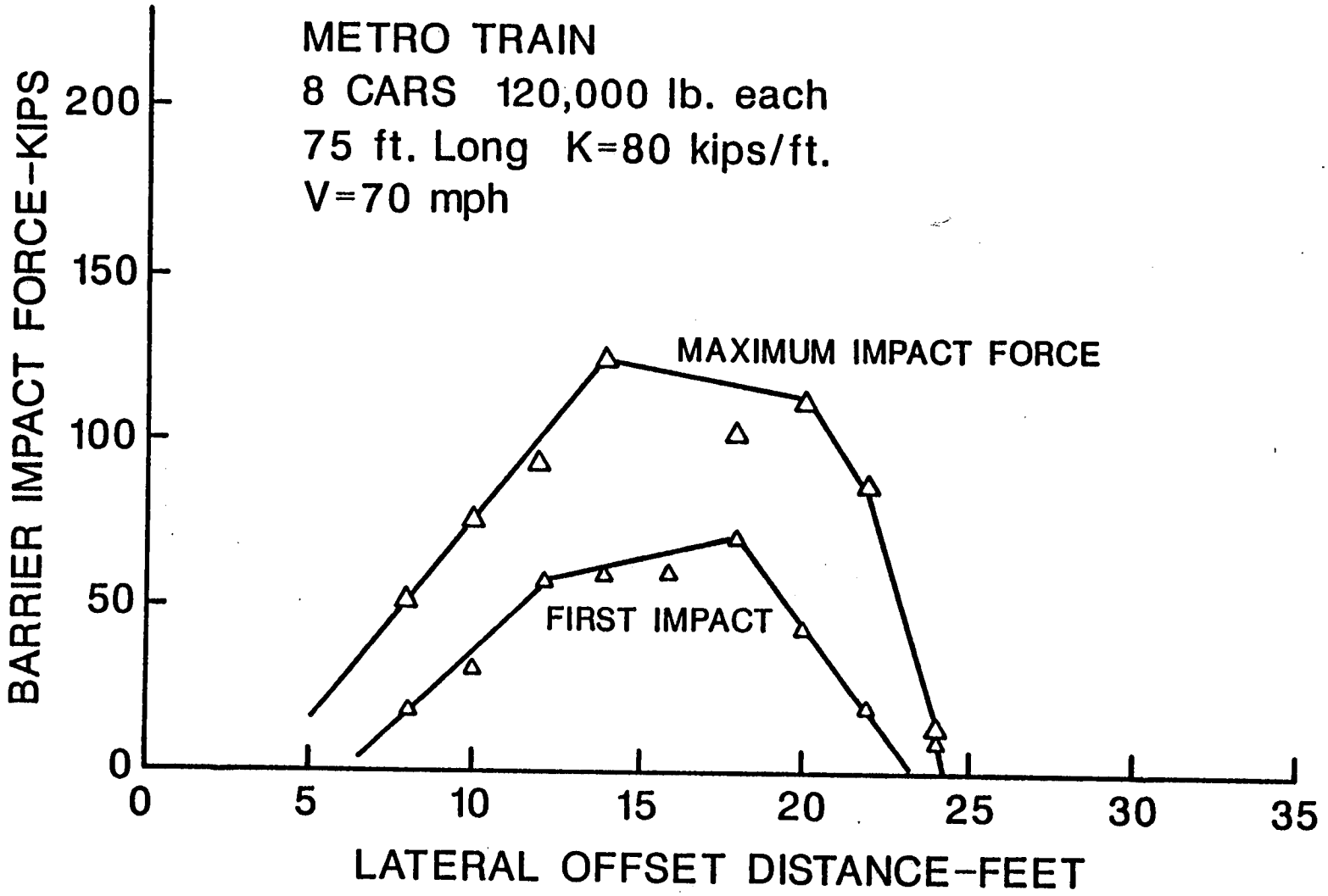


FIGURE 28. METRO IMPACT FORCE vs DISTANCE TO BARRIER.
K = 80 kips/ft

TABLE 7. PARAMETER STUDY OF EIGHT CAR METRO TRAIN-BARRIER IMPACTS
K=80 kips/ft

MATRO 8 CARs
W=120 kip
V=70 mph
L=75 ft

Barrier Dis(ft)	First IMPF(kip)	Max IMPF(kip)
8	19.48	51.40
10	30.80	76.12
12	57.51	94.20
14	60.07	124.71
16	59.55	116.26
18	70.94	101.49
20	44.19	112.24
22	19.81	85.51
24	9.46	13.40

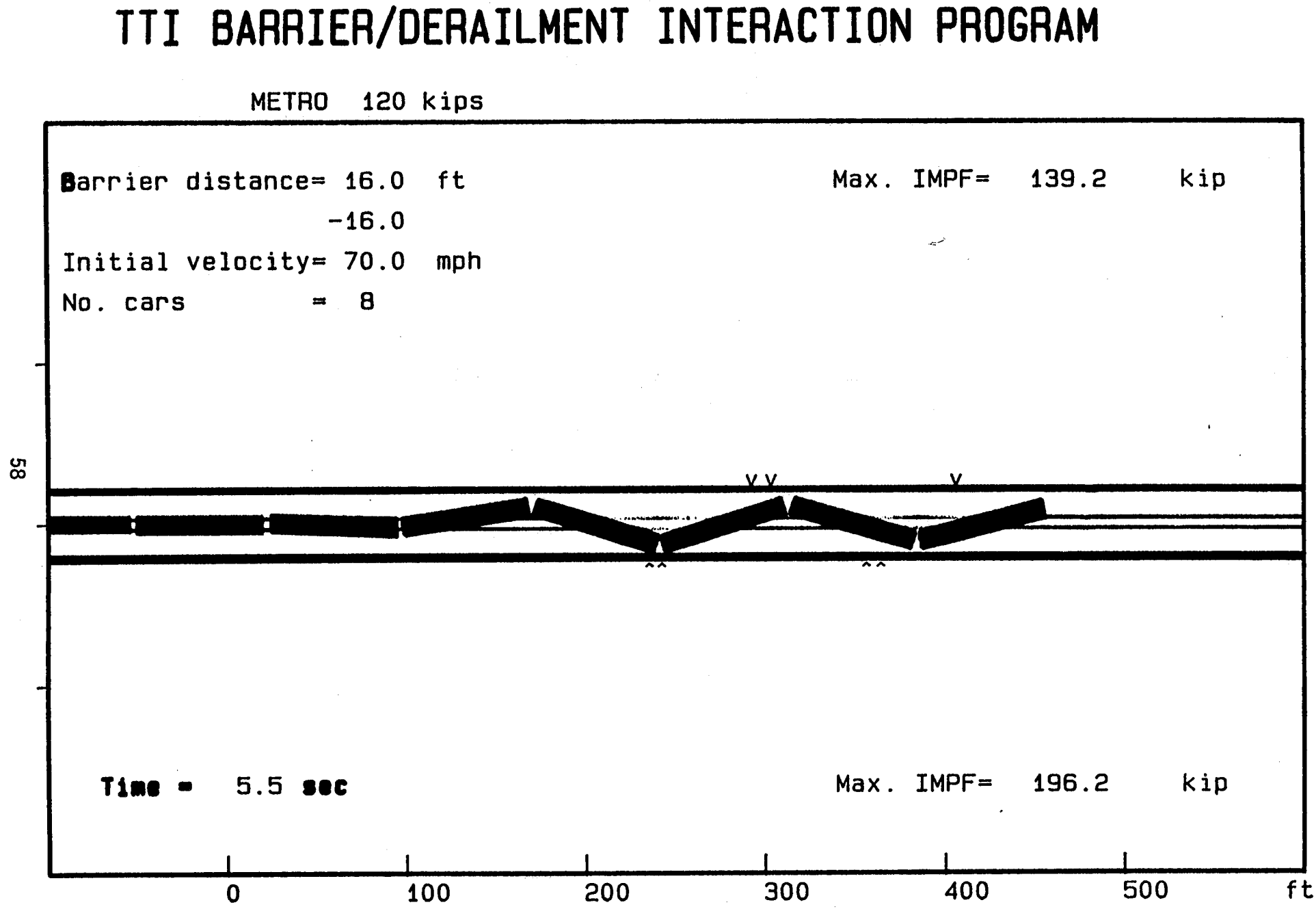


FIGURE 29. DUAL BARRIER WITH 16 FT OFFSET.

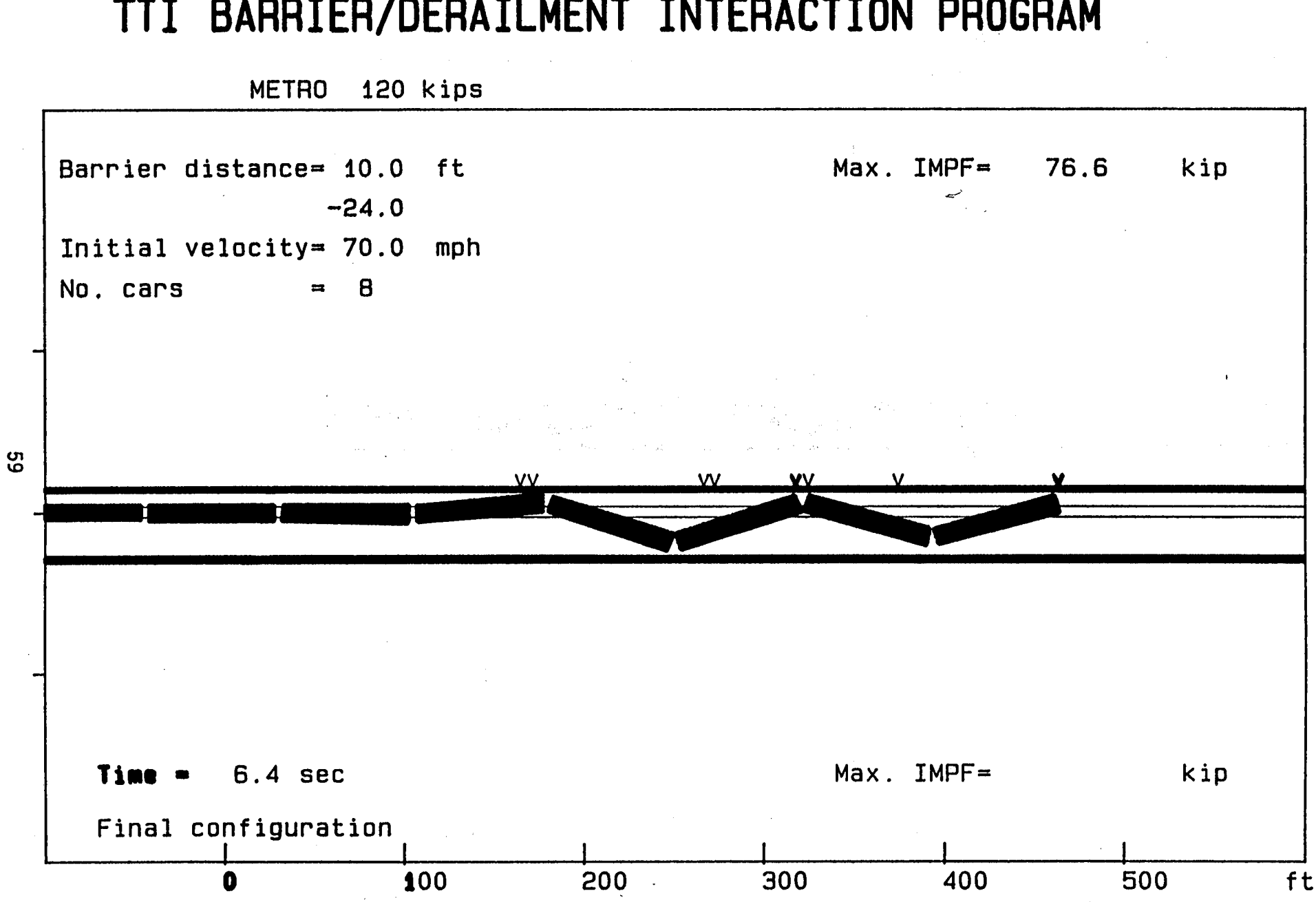


FIGURE 30. DUAL BARRIER WITH 10 FT AND 24 FT OFFSET.

TABLE 8. PARAMETER STUDY OF 8 CAR METRO TRAIN-BARRIER IMPACT WITH DUAL BARRIERS
K = 80 kips/ft

METRO 8 CARS
L=75 ft
W=120 kips
V=70 mph

Barrier	Distance(ft)		MAX IMPF (kip)
8	,	- 8	46.21
10	,	-10	87.68
12	,	-12	132.16
14	,	-14	174.82
16	,	-16	176.20
18	,	-18	209.20
20	,	-20	207.27
22	,	-22	112.28
24	,	-24	14.07
10	,	-14	92.85
10	,	-24	76.62
7	,	-14	37.92
7	,	-20	37.92

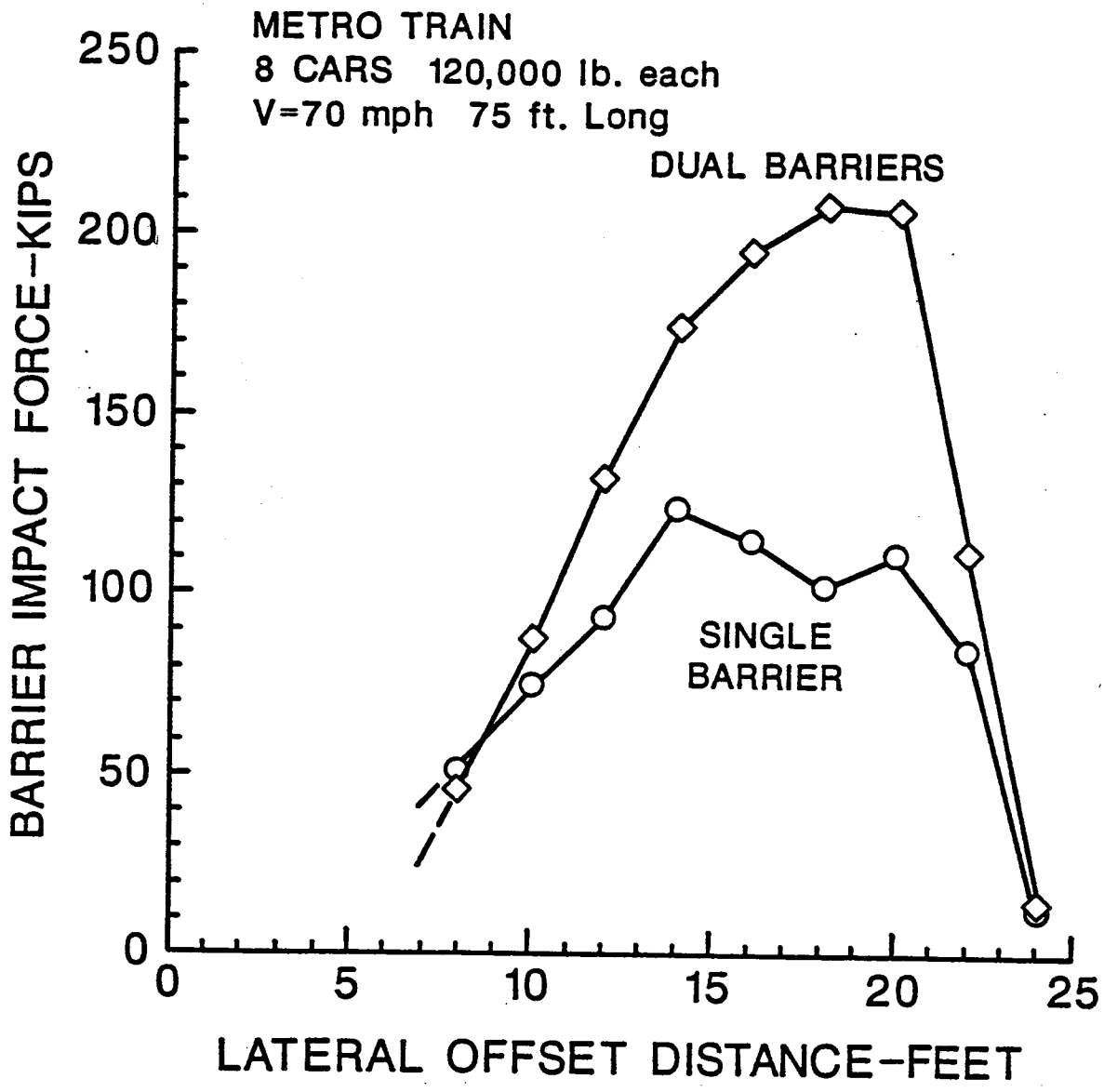


FIGURE 31. METRO IMPACT FORCE vs DISTANCE TO BARRIER
K = 80 kips/ft

ANALYSIS OF BARRIER HEIGHT TO PREVENT OVERTURNING

Figure 32 presents a very conservative simplified analysis of the barrier height required to prevent overturning. A similar analysis has been used for highway motor vehicles (7 and 8). The maximum train acceleration G_{max} is obtained by dividing F_{max} from Figures 10 or 24 by the car weight W.

$$G_{max} = \frac{F_{max}}{W}$$

Example: Metro Car W = 120,000 lb L = 75 ft
V = 70 mph B = 120 in. h = 64 in. C = 14 ft
from Figure 6 F_{max} = 124,700 lb

$$G_{max} = \frac{124,700 \text{ lb}}{120,000} = 1.04 \text{ g's}$$

$$H = \frac{1.04 \times 64 \text{ in.} - 60 \text{ in.}}{1.04} = 6.3 \text{ in.}$$

This calculation indicates the height of the wall to prevent overturning is not critical. The barrier would have to be at least 30 in. above the top of the rail in order to contact substantial strength of the Metro car, however.

Example: Hopper Car W = 260,000 lb
L = 60 ft B = 128 in. h = 98 in. (max.)
C = 20 ft V = 50 mph

from Figure 6 F_{max} = 509,800 lb

$$G_{max} = \frac{509,800}{260,000} = 1.96 \text{ g's}$$

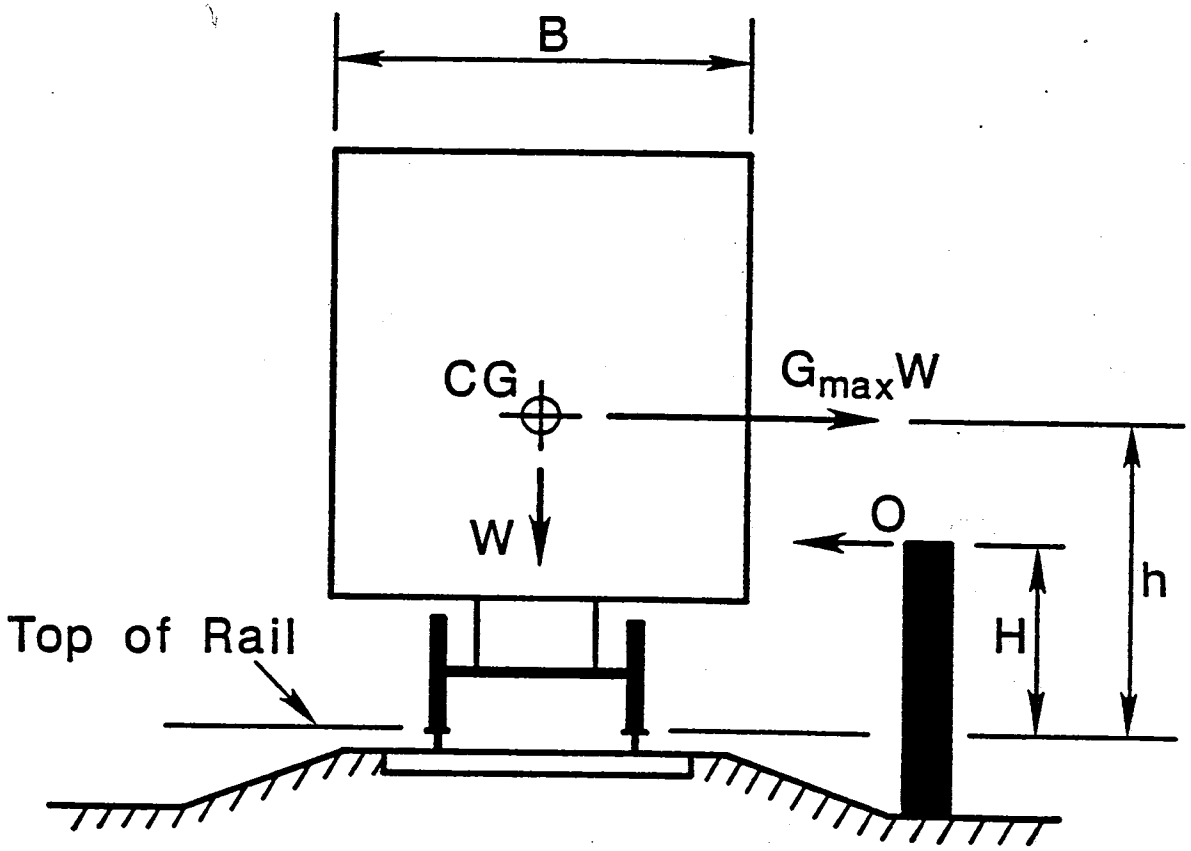
$$H = \frac{1.96 \times 98 \text{ in.} - 64 \text{ in.}}{1.96} = 65.4 \text{ in.}$$

To keep this Hopper car from overturning, the barrier would have to be about 66 in. above the top of the rail. This, or course, is for the maximum center of gravity height permitted by FRA of 98 in. For a more typical center of gravity height of 80 in.

$$H = \frac{1.96 \times 80 \text{ in.} - 64 \text{ in.}}{1.96} = 47.3 \text{ in.}$$

$$\Sigma M_o = G_{max} W(h-H) - W\ B/2 = 0$$

$$H \geq \frac{G_{max}\, h - B/2}{G_{max}}$$



$$G_{max} = F_{max} / W$$

FIGURE 32. BARRIER HEIGHT TO PREVENT OVERTURNING

It can be concluded that a barrier 5 ft-6 in. above the top of the track should be sufficient to prevent the derailed cars from overturning. It is believed these heights are extremely conservative since the train cars are not parallel to the wall when the maximum impact occurs as in the case of highway motor vehicles.

SUMMARY OF ANALYSIS OF BARRIER TRAIN IMPACT FORCES AND REQUIRED HEIGHTS

At this time it is recommended that the maximum impact forces determined using a train car stiffness of K-80 kips/ft train barriers in the common corridor.

Table 9 summarizes the design impact forces and the recommended barrier heights for freight trains. The table contains forces obtained using a single barrier and a maximum train speed of 50 mph. Table 9 also contains forces obtained using dual barriers and a maximum train speed of 30 mph.

When a single barrier is used, the optimum location would be about 10 ft $\pm$ 1 ft from the centerline of the freight train track and the design impact force would be about 408 kips $\pm$ 6 kips. If dual barriers are used, they should be located from 7 ft to 9 ft from the centerline of the freight train track.

Table 10 summarizes the design impact forces and the recommended barrier heights for a Metro train. The table contains forces obtained using a single barrier and a dual barrier. The maximum Metro train speed in both cases was 70 mph. It is interesting to note that barriers located from 7 ft to 9 ft from the centerline of the Metro track would only have to resist impact forces of from 39 kips to 64 kips. These forces are similar to those resisted by modern highway bridge rails (i.e., 56 kips to 60 kips resulting from a 4,500 lb car impacting at 60 mph and 25 degree angle--see Table 1B in Appendix B).

TABLE 9. RECOMMENDED BARRIER DESIGN IMPACT FORCES FOR FREIGHT TRAINS IN COMMON CORRIDORS

SINGLE BARRIER
Maximum Speed = 50 mph

BARRIER DISTANCE ft	DESIGN IMPACT FORCE kips	BARRIER HEIGHT in.
7	402	60*
8	416	60
9	409	60
10	402	60
12	426	60
14	483	64
16	495	65
18	466	65
20	510	66
22	477	63
24	443	61
26	390	60
28	319	60
30	240	60
32	151	60
34	62	60
36	0	0

DUAL BARRIER
Maximum Speed = 30 mph

BARRIER DISTANCE ft - ft		DESIGN IMPACT FORCE kips	BARRIER HEIGHT in.
+7	-7	205	60*
+8	-8	291	60
+9	-9	379	60
+10	-10	586	70
+12	-12	1201	84
+14	-14	1612	88
+16	-16	1562	88
+18	-18	1429	87
+20	-20	1295	86
+22	-22	1089	83
+24	-24	882	80
+26	-26	688	74
+28	-28	493	65
+30	-30	339	60
+32	-32	184	60
+34	-34	92	60
+36	-36	0	0

* 60 in. is minimum height measured from top of rail so barrier can engage substantial freight car structure.

TABLE 10. RECOMMENDED BARRIER DESIGN IMPACT FORCES AND HEIGHT FOR METRO TRAINS

SINGLE BARRIER
Maximum Speed = 70 mph

BARRIER DISTANCE ft	DESIGN IMPACT FORCE kips	BARRIER HEIGHT in.
7	39	30*
8	51	30
9	64	30
10	76	30
12	94	30
14	125	30
16	116	30
18	102	30
20	112	30
22	86	30
24	14	30
26	0	0

DUAL BARRIER
Maximum Speed = 70 mph

BARRIER DISTANCE ft - ft		DESIGN IMPACT FORCE kips	BARRIER HEIGHT in.
+7	-7	27	30*
+8	-8	46	30
+9	-9	67	30
+10	-10	88	30
+12	-12	132	30
+14	-14	175	30
+16	-16	196	30
+18	-18	209	30
+20	-20	207	30
+22	-22	112	30
+24	-24	14	30
+26	-26	0	0
+10	-24	77	30
+7	-20	38	30

* 30 in. is the minimum height measured from top of rail so barrier can engage substantial Metro car structure.

CONCEPTUAL BARRIER DESIGNS

The freight train or hopper car barriers presented here were designed for an impact force of about 408 kips. The longitudinal barriers and bridge rails designed and successfully crash tested by TTI in the past (6, 8, 12, 13, and 16) were designed for impact forces of 200 to 245 kips. When these barriers receive the design impact force, damage and repairs to the rail are anticipated. Consequently, they are designed by failure mode analysis (plastic design of steel and yield line analysis of reinforced concrete).

Figure 33 shows three possible failure modes for a beam and post barrier. An analysis is made of all possible failure modes (one span, two span, three span, four span, and more) until the critical or weakest failure mode is found. This procedure was used to design the metal or steel barriers presented here. All structural steel was assumed to be ASTM A36.

Figure 34 shows a yield line analysis of a concrete wall barrier supported on drilled concrete piers. L is the critical length of wall which resists the smallest lateral load (wl).

These methods of analysis are presented in reference (7). All concrete was assumed to be f_c' = 3,600 psi and reinforcing steel to be f_y = 60,000 psi. The foundation design (depth of embedment) of the steel W post, H piles, and drilled concrete piers were also based on failure mode analysis (14, 22, 24, and 27). In addition, the increased dynamic strength of soil was also used. Test results on guardrail posts (14 and 27) indicate that

$$P_{dynamic} = P_{static} (1+JV)$$

where

V = impact velocity in ft/sec

J = damping constant = 0.14 sec/ft

Three conceptual or possible barrier designs are presented here to demonstrate that a railroad barrier to restrain derailed cars or trains is feasible.

Figure 35 shows a typical situation where the Metro train tracks are 14 ft apart. A longitudinal barrier similar to a typical highway guardrail, bridge rail, or median barrier would be strong enough to restrain a derailed Metro train. Table 10 shows that if a barrier is about 7 ft from the track centerline, the impact force would only be about 39 kips.

Figure 36 presents a summary of the plastic analysis and design calculations (see Appendix D). Each post will resist a load of about 8.6 kips and the beam has a plastic

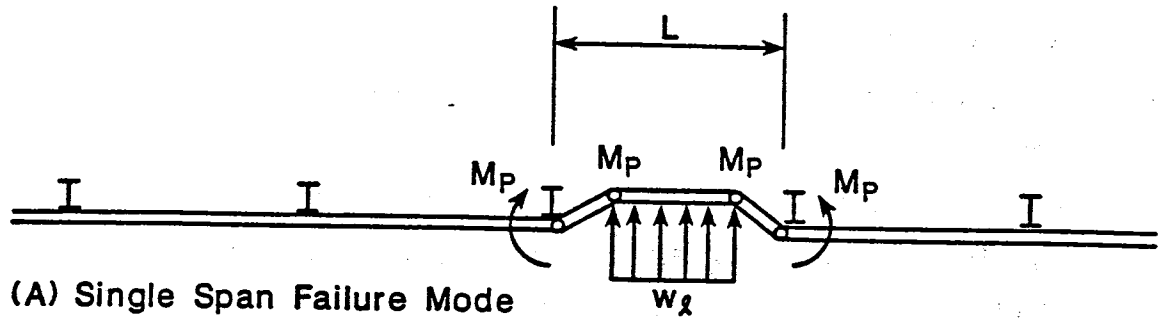
moment capacity of 51 kip-ft. The ultimate load is seen to be 49.2 kips. The failure mode covers 5 spans (30 ft) and includes 4 posts. The three span load of 43.2 kips and four span load of 44.8 kips are not valid because the two end posts (2 x 8.6 = 17.2 kips) cannot support the beam reaction loads of 26 kips and 19 kips respectively (see Appendix D). This Metro train barrier contains about 45 lb of steel per foot of length which indicates it would cost about $45 to $50 per foot of length.

Figure 37 shows a typical situation where the Metro train tracks are 20 ft from a freight train track as in the B-6 corridor. The steel beam and post barrier placed directly between them would be about 9 ft from the centerline of the freight train tracks. Table 9 would indicate the design impact force to be about 408 kips and the barrier about 60 in. above the top of the tracks. The posts are 14 in. steel H piles which can be driven into the soil at 9 ft center-to-center spacing. The steel beam is a 27 in. deep wide flange weighing 84 lb per ft.

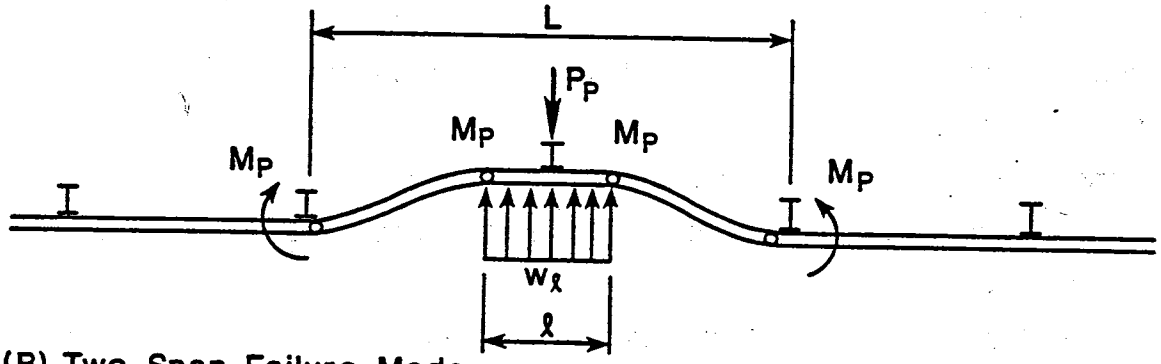
Figure 38 presents a summary of the plastic analysis and design calculations (see Appendix D). Each post can resist a force of 77 kips and the beam has a plastic moment capacity 732 kip-ft. The ultimate load is determined to be 446 kips. The failure mode covers 5 spans (45 ft) and includes 4 posts. The three span load of 393 kips and four span load of 406 kips are not valid because the two end posts (2 x 77 kips = 154 kips) cannot support the beam reaction loads of 239 kips and 175 kips respectively (see Appendix D). This freight train barrier containing about 315 lb of steel per foot of length and would cost about $325 to $350 per foot of length.

Figures 39 and 40 show a concrete barrier wall placed between freight train tracks and Metro train tracks, as in the B-6 corridor. The yield line analysis (Appendix D) indicates an ultimate load of 432 kips. The required strength would be about 408 kips with the face of the barrier about 9.25 ft from the centerline of the tracks.

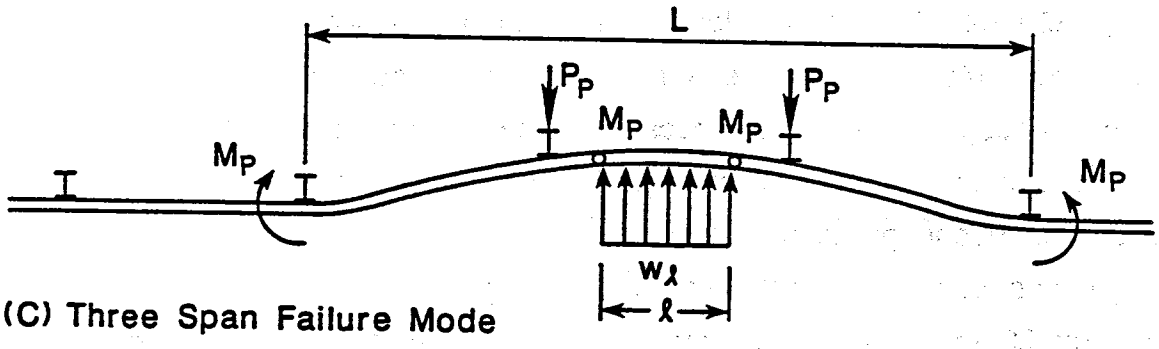
This concrete barrier contains about 22 cubic feet of concrete and 172 lb of reinforcing steel per foot of wall length. The estimated cost would be about $325 to $350 per foot of length.



(A) Single Span Failure Mode



(B) Two Span Failure Mode



(C) Three Span Failure Mode

M_P = plastic moment capacity of rail = Mult.

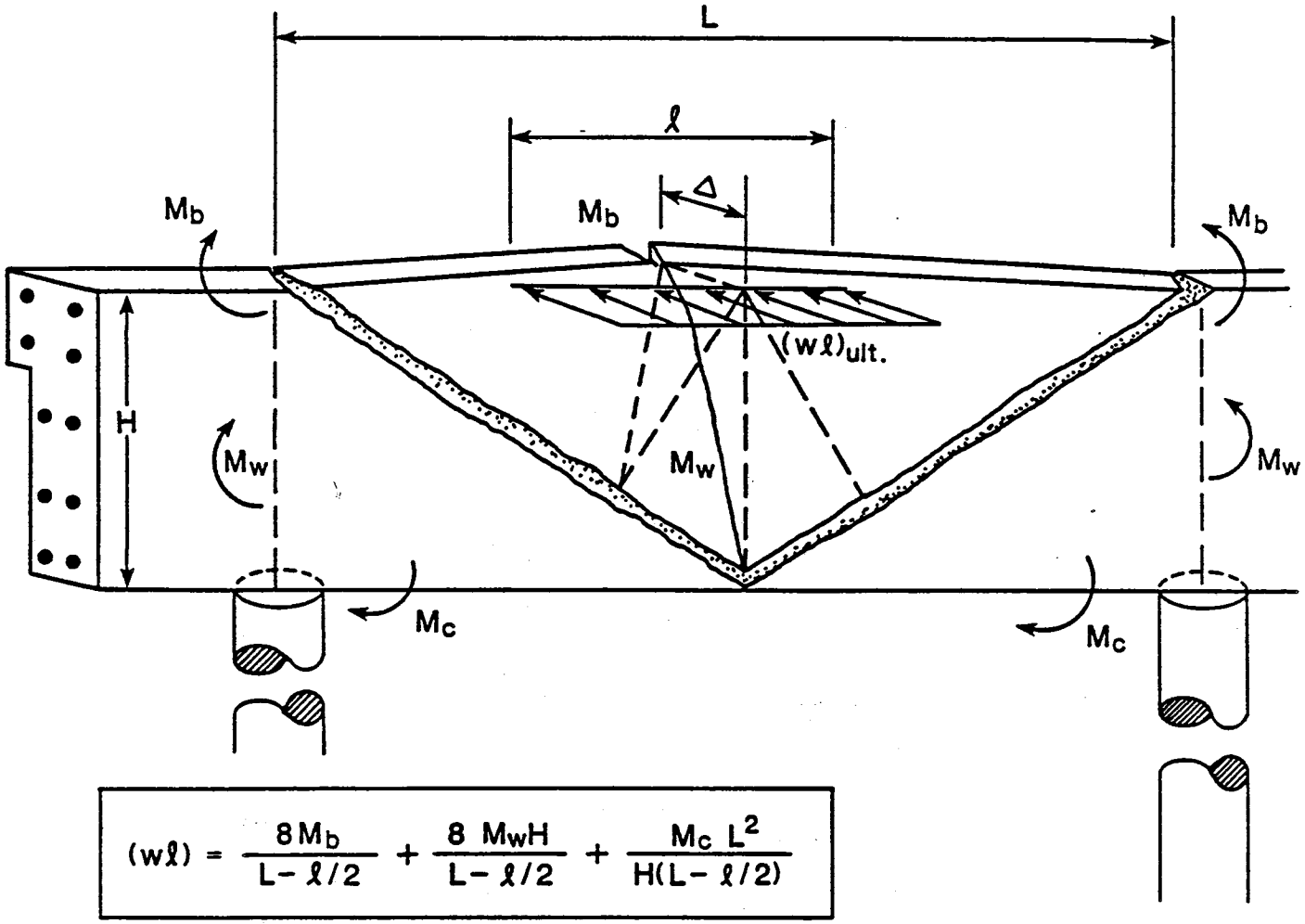
P_P = ultimate load capacity of a single post

w_ℓ = total ultimate vehicle impact load = $\frac{8\,M_P + \Sigma P_P}{L - \ell/2}$

ℓ = 5 ft.

PLAN VIEW

FIGURE 33. POSSIBLE FAILURE MODES FOR BEAM AND POST BARRIER



$$(w\ell) = \frac{8M_b}{L-\ell/2} + \frac{8\,M_wH}{L-\ell/2} + \frac{M_c\,L^2}{H(L-\ell/2)}$$

$$L = \ell/2 + \sqrt{\left(\frac{\ell}{2}\right)^2 + \frac{8\,H\,(M_b + M_wH)}{M_c}}$$

H = height of wall, ft

L = critical length of wall failure, ft

$w\ell$ = total ultimate load capacity of walls, kips

M_b = ultimate moment capacity of beam at top of wall, kip-ft

M_w = ultimate moment capacity of wall per ft of wall height, kip-ft/ft

M_c = ultimate moment capacity of wall cantilever up from drilled piers per ft of length of wall, kip-ft/ft

ℓ = length of distributed impact load, ft

FIGURE 34. YIELD LINE ANALYSIS OF CONCRETE PARAPET WALL

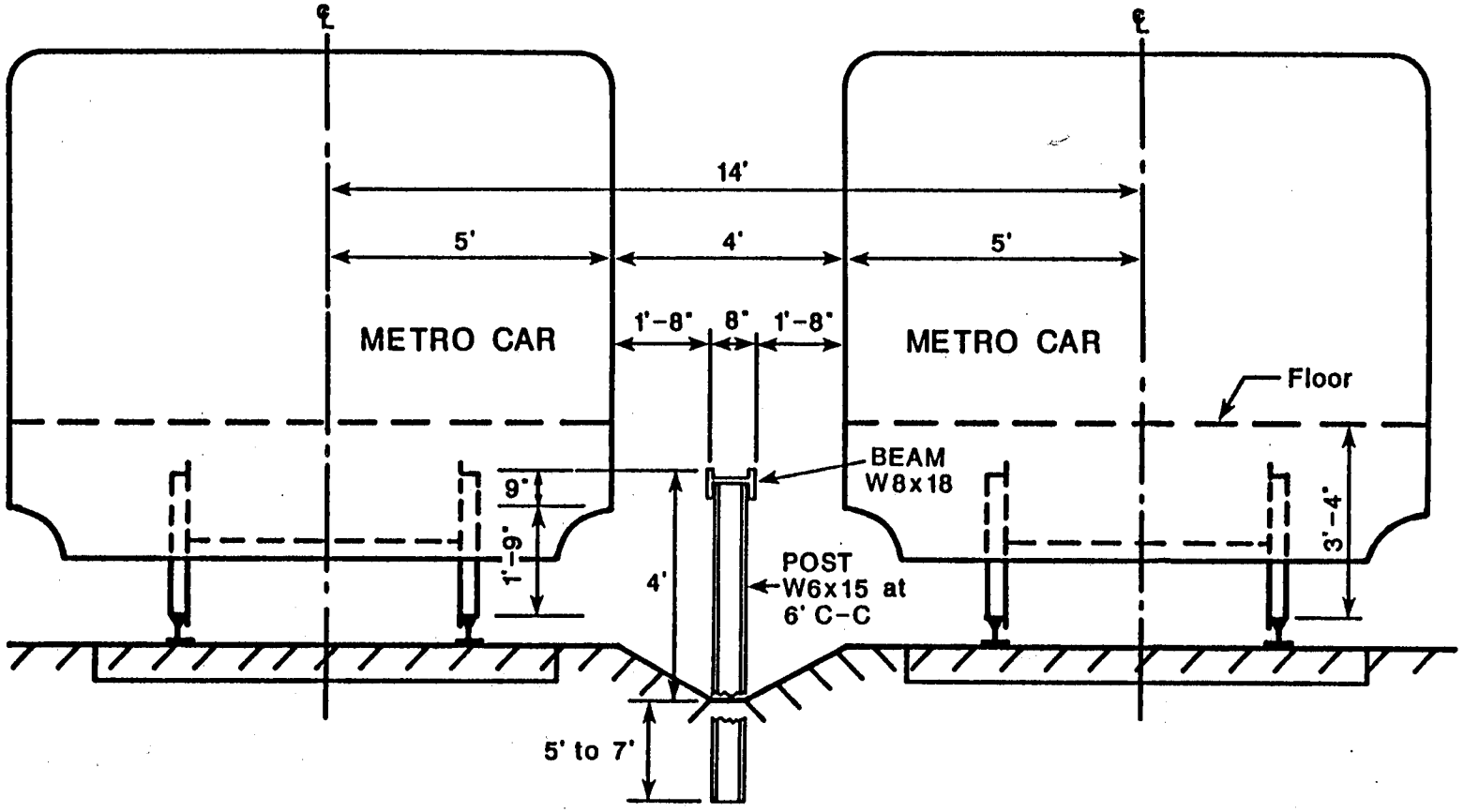


FIGURE 35. METRO TRAIN BARRIER DESIGNS - STEEL

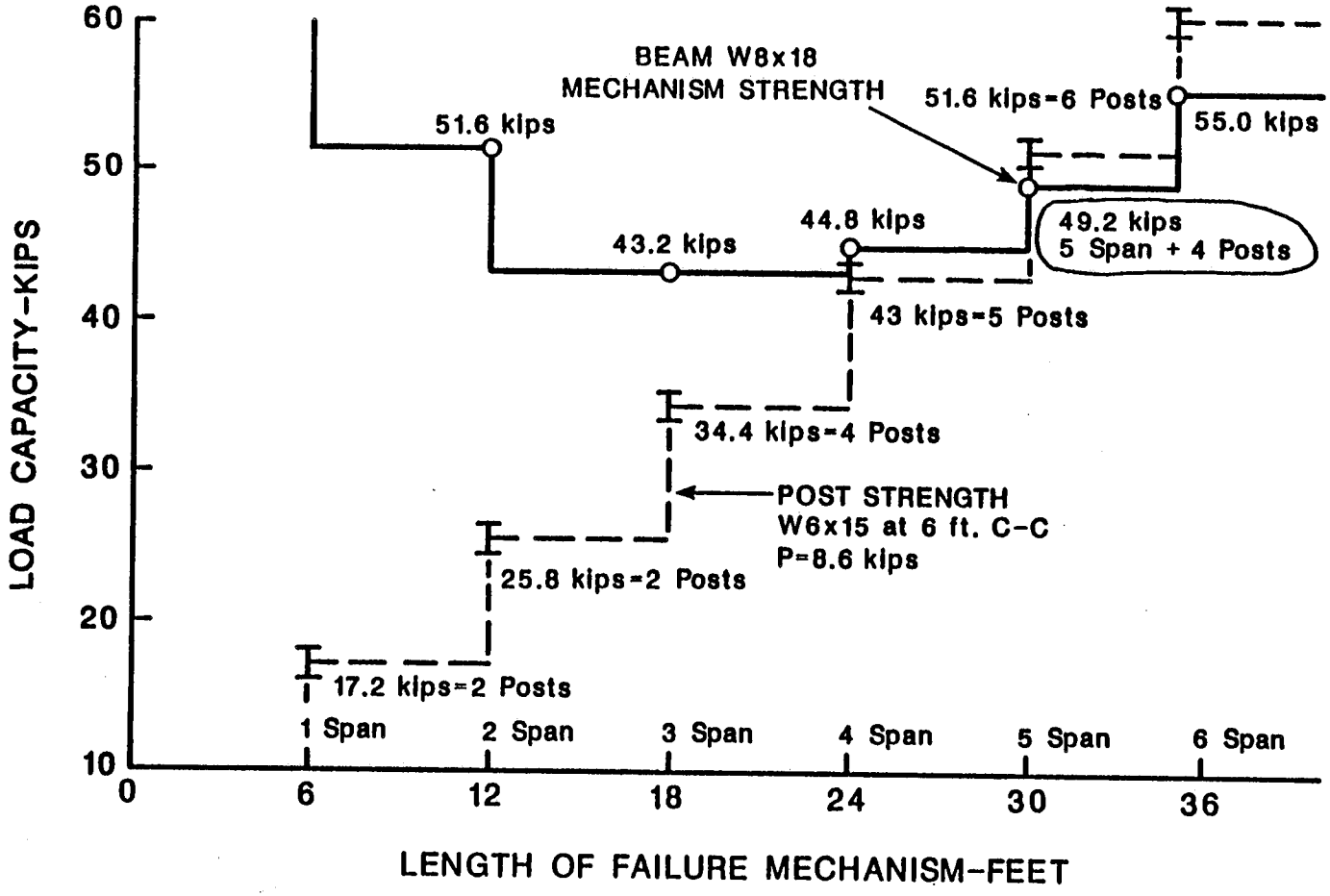


FIGURE 36. FAILURE MECHANISM ANALYSIS OF METRO TRAIN BARRIER

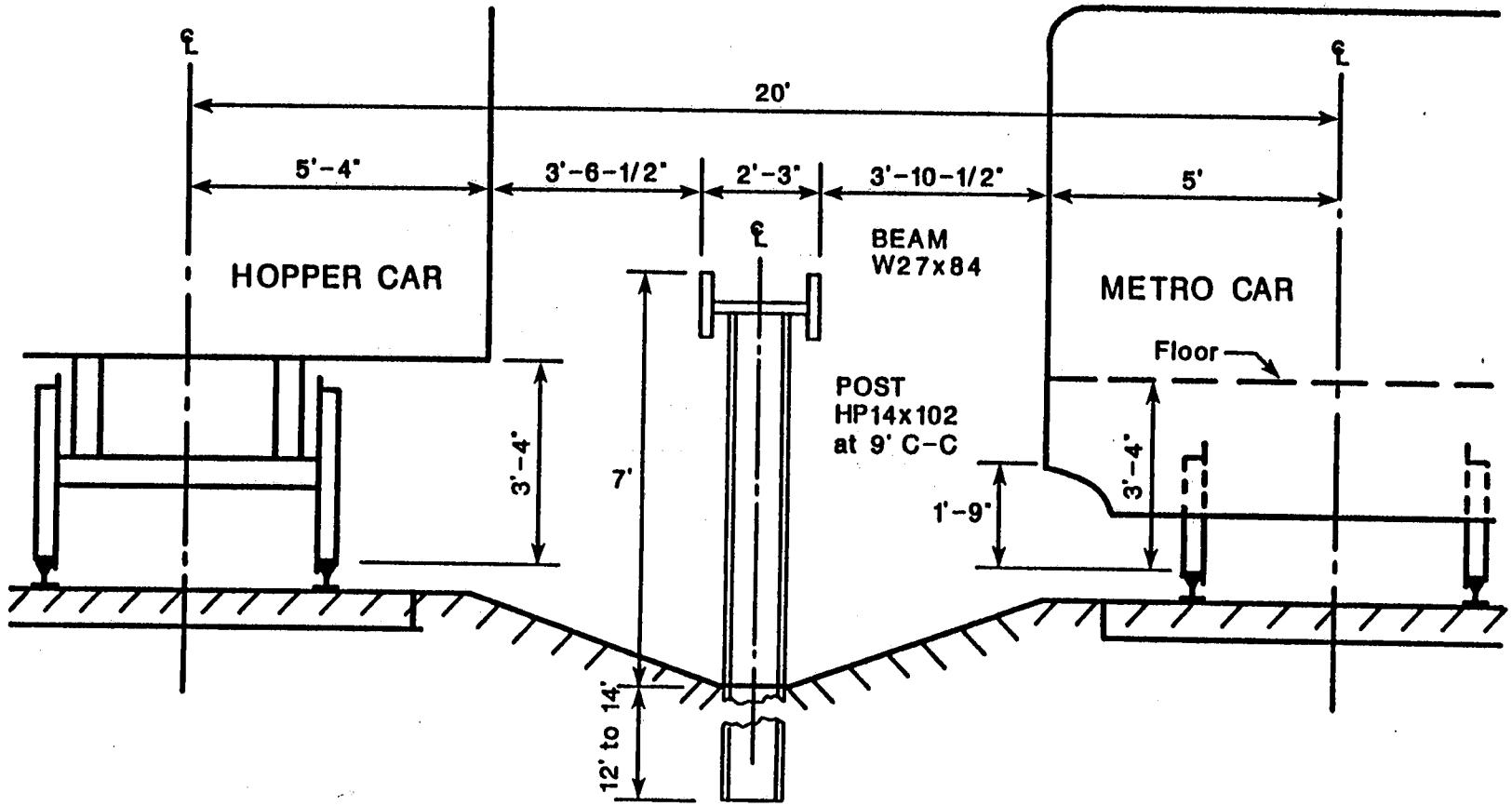


FIGURE 37. HOPPER CAR BARRIER DESIGN - STEEL

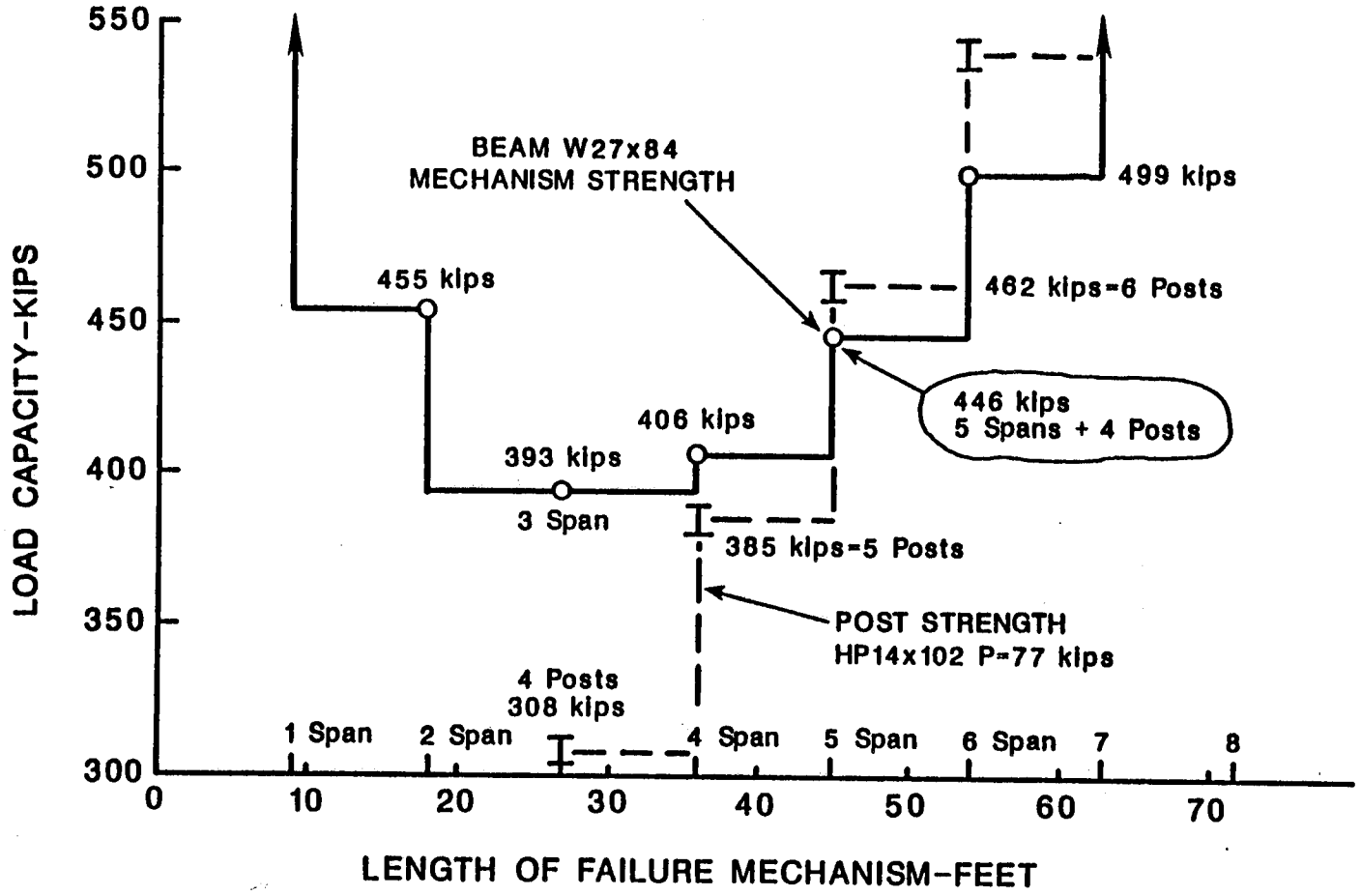


FIGURE 38. FAILURE MECHANISM ANALYSIS OF HOPPER CAR BARRIER

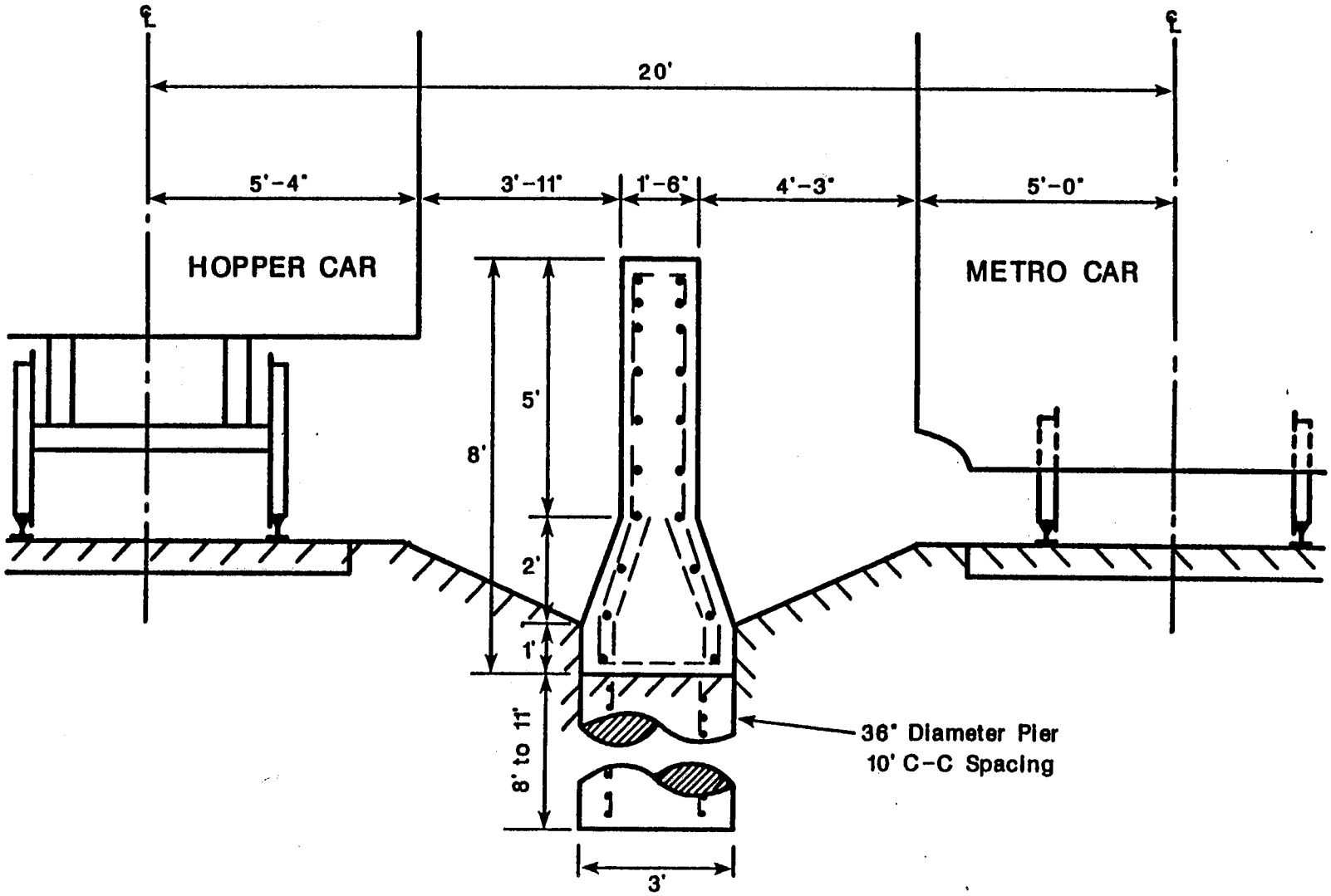


FIGURE 39. HOPPER CAR BARRIER DESIGN - CONCRETE

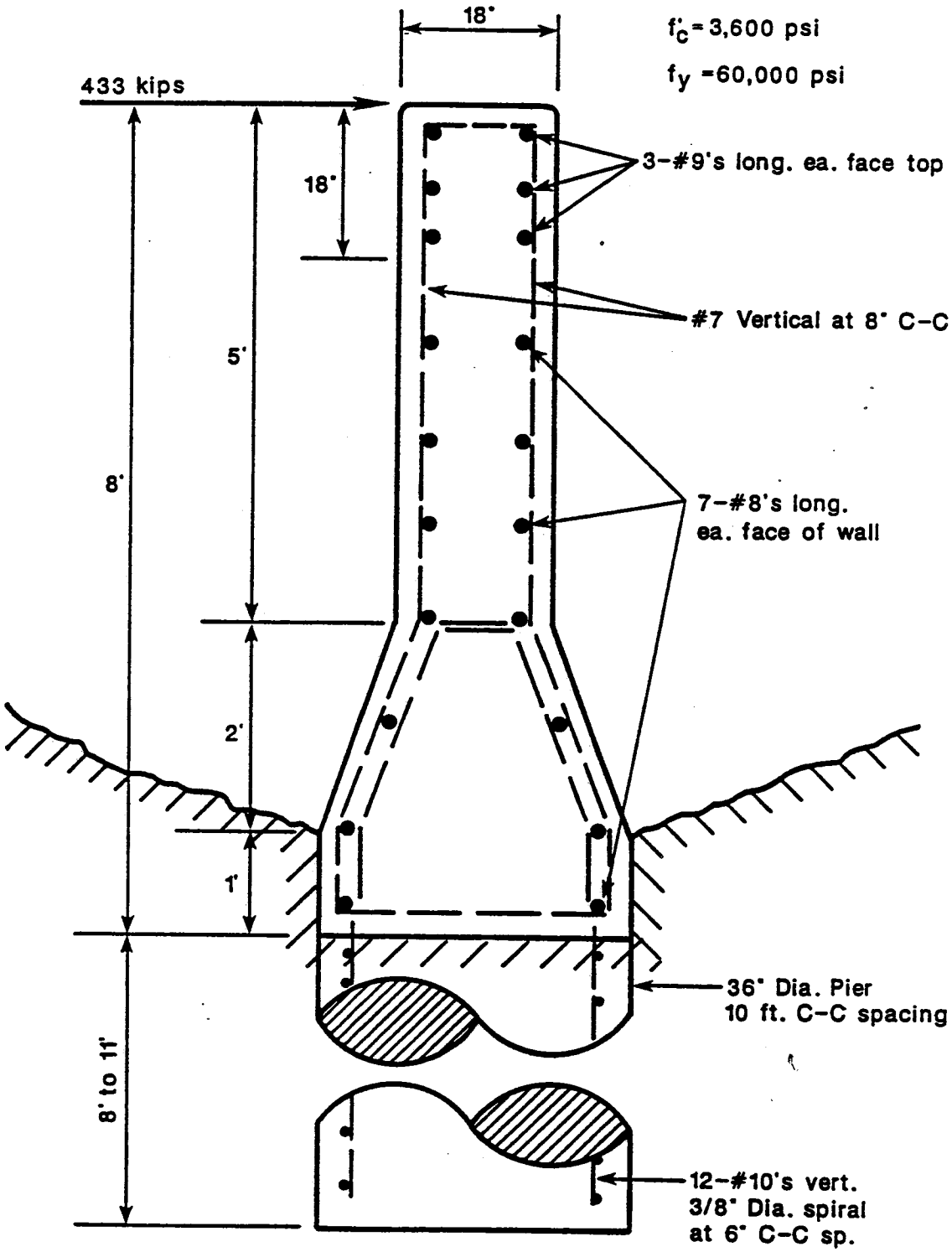


FIGURE 40. DETAILS OF CONCRETE BARRIER

DISCUSSION OF BARRIER DESIGNS

Of the two freight train barriers presented, the concrete wall is preferred. The smooth wall surface provides little to no opportunity for the freight cars or their loads to snag. The openings between the steel posts on the beam and post design provides an opportunity for flat cars and their loads to penetrate between and snag on the posts. Other freight cars may do the same thing.

The snagging problem is very slight with the beam and post Metro barrier, however. The Metro cars have a smooth exterior and are unlikely to penetrate the openings between the posts and snag.

Figure 41 shows the potential effect of the sloping track ballast on the rail car's impact point with the barrier. This figure indicates the impact point is lowered about one foot or more when the cars are off the track. The computer simulation of the derailed train cars indicates that this impact point occurs only on the front or rear corner of the cars, however. The rail cars should not become parallel to the barrier as the single rubber -tired motor vehicles do on impact with highway barriers.

The barrier designs presented show a range of foundation penetrations. In general, the shallower penetration was in a cohesive soil (clay) and the deeper penetration was in a cohesionless soil (sand or gravel). In addition, the foundation design took advantage of the increased impact or dynamic strength of soils. Impact tests on steel and timber guardrail posts at 17 mph (25 ft/sec) showed an increase in strength of four to five times the static strength (14 and 24). The dynamic load factors used here were 1.83 for the freight train barrier (impact velocity - 5.9 ft/sec) and 2.34 for the Metro barrier (impact velocity - 9.6 ft/sec). These calculations are shown in Appendix D. When the more precise foundation conditions are known, these depths of foundation penetration should be recalculated for each site.

The proposed design impact forces presented in Tables 9 and 10 are based on the corner stiffness of a railroad car of 80 kips per ft of crush. This number was arrived at by extrapolating crash test data from rubber-tired highway vehicles. It is desirable to verify this stiffness by crash testing single empty railroad cars into the same instrumented wall located at the TTI Proving Grounds. Computer simulations with a stiffness of 218 kips per

ft of crush increased the impact forces by about 65% equal to about the ratio of the square root of the stiffnesses.

$$\sqrt{\frac{218}{80}} = 1.65$$

Mr. William P. Manos (our railroad consultant) recommended that train car stiffness K be modified to recognize that much of the car crush is plastic. This has been done by adding a coefficient of restriction e to the car stiffness model. The train and barrier impact shown on Figure 7 was simulated using the K = 80 kips/ft and e = 0.4, and the impact forces were increased by about 50%. This further illustrates why the single rail car crash tests into the instrumented wall are desirable to more precisely define the values of stiffness and coefficient of restitution.

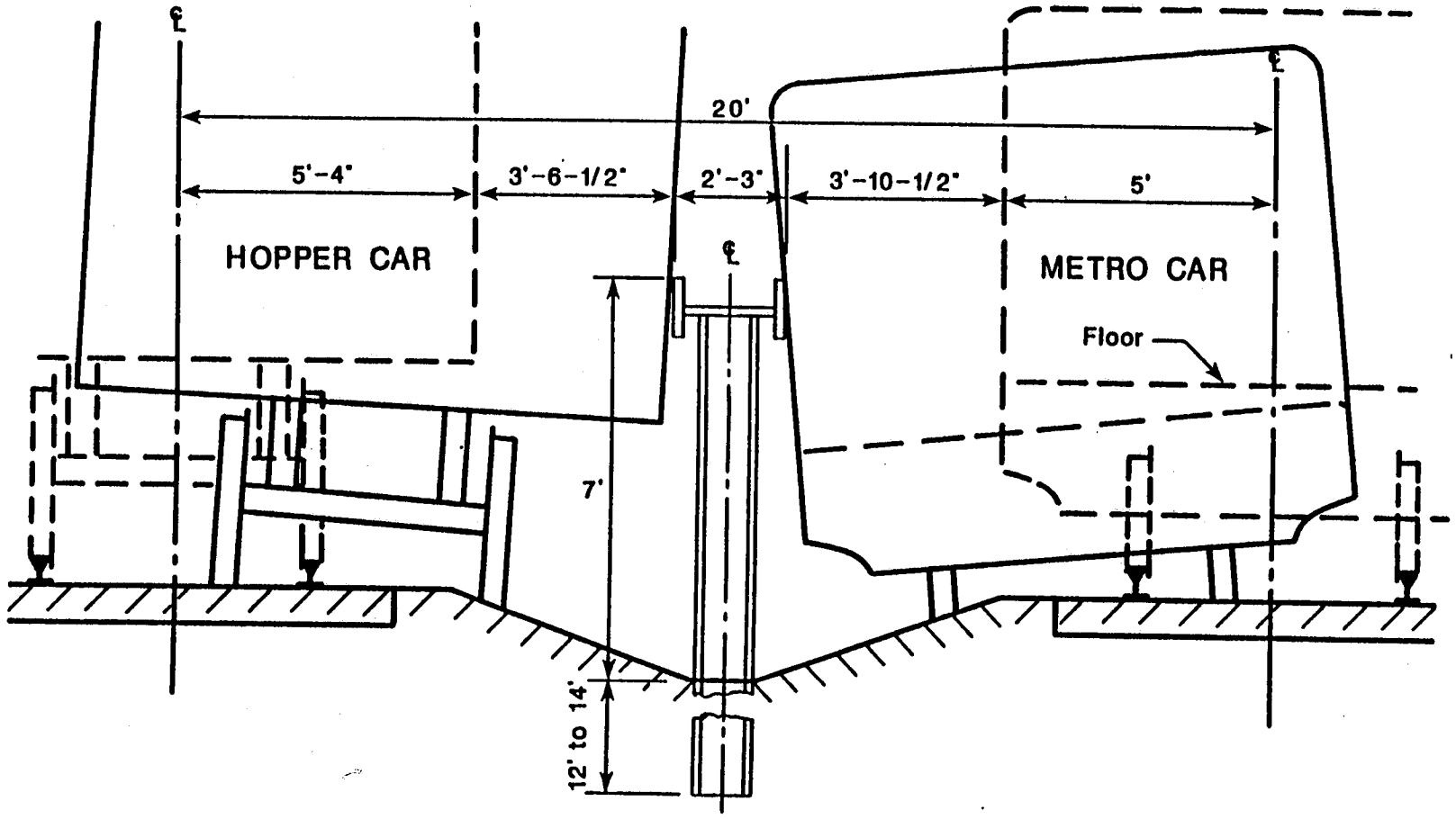


FIGURE 41. POTENTIAL EFFECT OF SLOPING TRACK BALLAST ON RAIL CAR IMPACT POINT

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APPENDIX A.

DESCRIPTION OF TRAIN DERAILMENT COMPUTER PROGRAM

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Dynamic Analysis of Train Derailments

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A Division of Pullman Incorporated,
Hammond, Ind.

In actual train derailments little is known except the end result. An analytical simulation has been developed to determine the influence of various parameters on derailment severity. In this theoretical analysis the equations of motion for each derailed car are derived in general in the horizontal plane. These are then coupled with a system of constraint equations and the equations of motion for the nonderailed cars. The equations are then solved numerically (by digital computer) in their non-linear forms with the first car derailed as the sole initially assumed condition; and with the ground friction, mating coupler moment and brake retarding force in action accordingly. This work was sponsored by the RPI/AAR Tank Car Safety Research and Test Project Committee and represents one phase of the overall RPI/AAR study of means to improve tank car safety in accidents.

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Dynamic Analysis of Train Derailments

T. H. YANG W. P. MANOS B. JOHNSTONE

NOMENCLATURE

- D = truck center distance (ft)
- F_b = brake retarding force (lb)
- F_x, F_y = coupler force in x and y direction (lb)
- H_x, H_y = forces at rear truck in x and y direction (lb)
- ID = number of the last derailed car at each instant
- I = moment of inertia of tank car about its center of gravity (lb-sec^2-ft)
- L = tank car length (ft)
- M = mass of tank car (lb-sec^2/ft)
- m = coupler mating torque (ft-lb)
- N = number of cars involved in derailment
- R_x, R_y = forces at front truck in x and y direction (lb)
- T = time (sec)
- ΔT = increment of time (sec)
- V = initial velocity of the train (fps)
- W = weight of the tank car (lb)
- X = displacement, in X-direction (ft)
- Y = displacement, in Y-direction (ft)
- α_1 = direction angle of front truck velocity (rad)
- α_2 = direction angle of rear truck velocity (rad)
- θ = angular displacement of tank car (rad)
- μ = coefficient of ground friction

INTRODUCTION

The Association of American Railroads and the Railroad carbuilders for many years have applied their best efforts toward the reduction of accidents of all types involving railway locomotives and cars. In addition, literally millions of dollars have been spent in research, in design, in laboratory testing, in field testing, and in specification development to make the nation's railway rolling stock highly resistant to accident damage. Despite this, derailment accidents continue to happen.

Therefore, there is a need for more knowledge in the areas of car behavior and reacting force levels during the derailment cycle. Such information can then be used for evaluating existing car equipment and needed improvements.

The mathematical simulation technique has been developed to study the influence of the following variables on derailment behavior:

1. Ground friction
2. Number of cars in train
3. Train velocity
4. Coupler moment
5. Car length
6. Car weight
7. Braking.

The study was made by analysis of 26 simulations of interest. Results are summarized in this paper.

DESCRIPTION OF SIMULATION

In a train derailment, each car can roll, pitch, yaw, and translate in three dimensions, thus having six degrees of freedom. A review of accidents in the past reveals that the pattern is, in fact, extremely complicated. This study is limited to the most significant motions (those in a horizontal plane) in order to simplify the simulation. The schematic in Fig. 1 depicts the problem as analyzed.

In this simulation, it is assumed that a train parting has occurred, and only cars from the first derailed car back to the rear of the train are considered. The cars ahead of the parting move on and are not involved in the derailment.

This analysis is based on the following additional assumptions: (a) the cars are coupled together with resisting moment between cars, (b) there is no width dimension of cars, (c) there is simple ground friction at trucks of derailed cars, (d) emergency braking is applied to the non-derailed cars, (e) cars remain coupled, and (f)

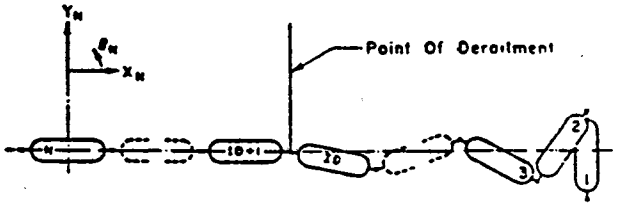

Fig.1 Train schematic

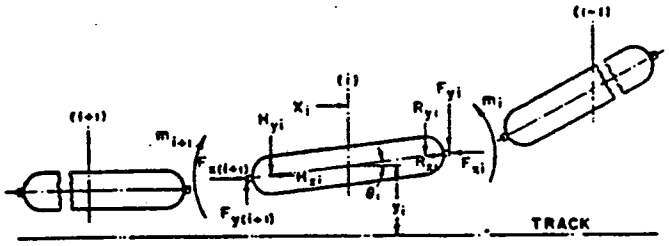

Fig.2 Free body diagram of i-th car

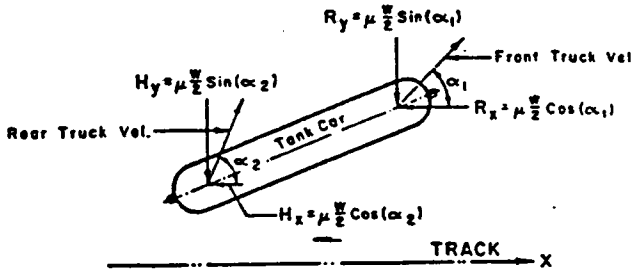

Fig.3 Ground friction force at trucks

the rail is interrupted by the first derailed car. There are no mathematical simplifications; consequently, the equations of motion governing the model are nonlinear.

MATHEMATICAL DEVELOPMENT

In this study, each car has three coordinates (X_i, Y_i, θ_i) as shown in Fig. 2. The equations of motion of each car can be written separately and then related by the constraint equations at the coupler location.

The equation of motion of the i-th derailed car in x-direction:

$$M_i \ddot{X}_i = F_{x(i+1)} - F_{xi} - (R_{xi} + H_{xi}) \quad (1)$$

The equation of motion of the i-th derailed car in y-direction:

$$M_i \ddot{Y}_i = F_{y(i+1)} - F_{yi} - (R_{yi} + H_{yi}) \quad (2)$$

The equation of motion of the i-th derailed car in θ direction:

$$I_i \ddot{\theta}_i = (F_{x(i+1)} + F_{xi})\frac{L_i}{2}\sin\theta_i + (R_{xi} - H_{xi})\frac{D_i}{2}\sin\theta_i - (F_{y(i+1)} + F_{yi})\frac{L_i}{2}\cos\theta_i - (R_{yi} - H_{yi})\frac{D_i}{2}\cos\theta_i - (m_{(i+1)} - m_i)\cdots \quad (3)$$

The equations of constraint in x, y directions are:

$$X_i = X_N + \frac{1}{2}(L_i\cos\theta_i + L_N\cos\theta_N) + \sum_{i+1}^{N-1} L_k\cos\theta_k \quad (4)$$

$$Y_i = \frac{1}{2}(L_i\sin\theta_i + L_N\sin\theta_N) + \sum_{i+1}^{N-1} L_k\sin\theta_k \quad (5)$$

The acceleration constraint equations can be obtained by differentiating equations (4) and (5) twice in respect to time:

$$\ddot{X}_i = \ddot{X}_N - \frac{1}{2}\left[L_i(\ddot{\theta}_i\sin\theta_i + \dot{\theta}_i^2\cos\theta_i) + L_N(\ddot{\theta}_N\sin\theta_N + \dot{\theta}_N^2\cos\theta_N)\right] - \sum_{i+1}^{N-1} L_k(\ddot{\theta}_k\sin\theta_k + \dot{\theta}_k^2\cos\theta_k) \quad (6)$$

$$\ddot{Y}_i = \frac{1}{2}\left[L_i(\ddot{\theta}_i\cos\theta_i - \dot{\theta}_i^2\sin\theta_i) + L_N(\ddot{\theta}_N\cos\theta_N - \dot{\theta}_N^2\sin\theta_N)\right] + \sum_{i+1}^{N-1} L_k(\ddot{\theta}_k\cos\theta_k - \dot{\theta}_k^2\sin\theta_k) \quad (7)$$

There are a total of six unknowns — F_{xi}, F_{yi}, $\ddot{X}_i$, $\ddot{Y}_i$, $\ddot{\theta}_i$, and $\ddot{X}_N$ — and five equations — (1), (2), (3), (6), and (7). Another equation is established by lumping all the non-derailed cars together as a free body and assuming that they move together as shown in Fig. 1.

The equation of motion can be written as:

$$\left(\sum_{ID+1}^{N} M_j\right)\ddot{X}_N = -F_{x(ID+1)} - \sum_{ID+1}^{N} F_{bj} \quad (8)$$

Equation (8) contains the coupler force in the X-direction, F_x (ID + 1), and the summation of the brake retarding forces

$$\sum_{ID+1}^{N}$$

on the non-derailed cars for the particular time and speed.

There are six equations and six unknowns; therefore, the derailment behavior can be solved. In solving the problem, the six unknowns are reduced to three: $\ddot{\theta}$, F_x, and F_y.

The Gauss-Jordan elimination method (2)[1] was employed to solve the simultaneous equations for each time instant, and an Euler-Cauchy integration

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

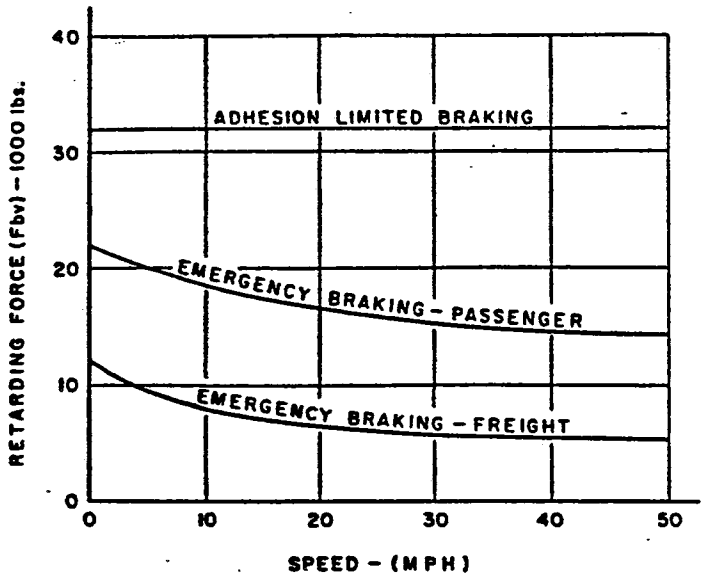

Fig.4 Brake retarding force versus train speed

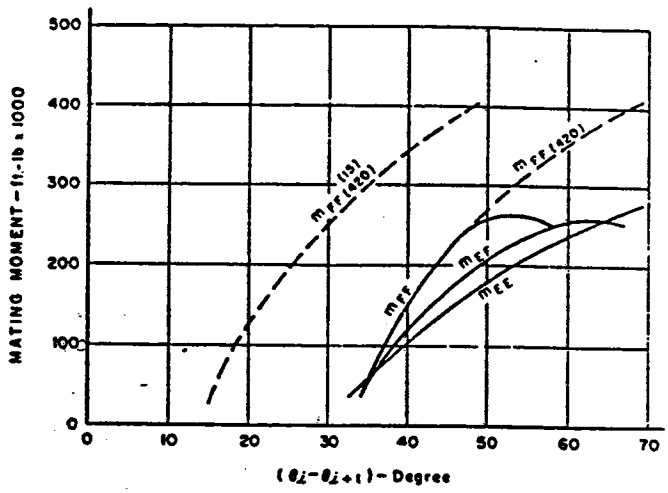

Fig.5 Coupler moment characteristics

was used to proceed with increasing time increments.

Initially, there is one derailed car and three equations; as each succeeding car derails, the number of equations increases by three.

The external force applied to the trucks of each derailed car is the ground friction which is in the direction opposite to the motion at the truck locations as indicated in Fig. 3.

The external force applied to the non-derailed cars is the retarding brake force which is considered time- and velocity-dependent. The brakes are applied to each car with appropriate train-line propagation starting with the instant of derailment. The retarding force is obtained from the instantaneous brake cylinder pressure and the retarding force versus train speed characteristics shown in Fig. 4.

The mating moment between cars is a function of angular differential between adjacent cars. Characteristic curves were provided by the RPI-AAR Railroad Tank Car Safety Research and Test Project Committee from static test of actual components, using various combinations of E and F head couplers. Other curves were used, as shown in Fig. 5, to study the effects of changes in these characteristics.

RESULTS

The computerized solutions of the study are based upon the dimensions and dynamic parameters as tabulated in Table 1.

Table 2 shows the major results of the derailment simulations; it includes the number of derailed cars, time duration, maximum coupler force in the derailing cycle, maximum acceleration or deceleration of the derailed cars in X-direction, and maximum displacements in X- and Y-directions. The maximum X-displacement is defined as the X distance from the point of derailment to the center of the leading derailed car, without consideration of jackknifed card widths. The maximum Y-displacement is defined as the maximum derailed car lateral displacement in the derailing cycle.

The curves in Figs. 6 and 7 show the effects of changes in five parameters on the number of derailed cars and on the maximum coupler force. The parameters are normalized with respect to the base case values on the abscissa.

VERIFICATION

To validate the mathematical model, a simulation of the Crescent City, Illinois, derailment of June 21, 1970, was run. Crescent City was chosen as most closely following the two-dimensional assumption and having the most complete accident records. The train consist and dynamic parameters were taken from field records provided by the RPI-AAR Railroad Tank Car Safety Research and Test Project Committee.

The simulation produced 16.4[2] derailed cars compared to the actual 16, using 1.5 for the coef-

[2] By definition, the number of derailed cars is that number of cars, or fraction thereof, that has passed beyond the "Point of Derailment" which is labeled in all the sketches.

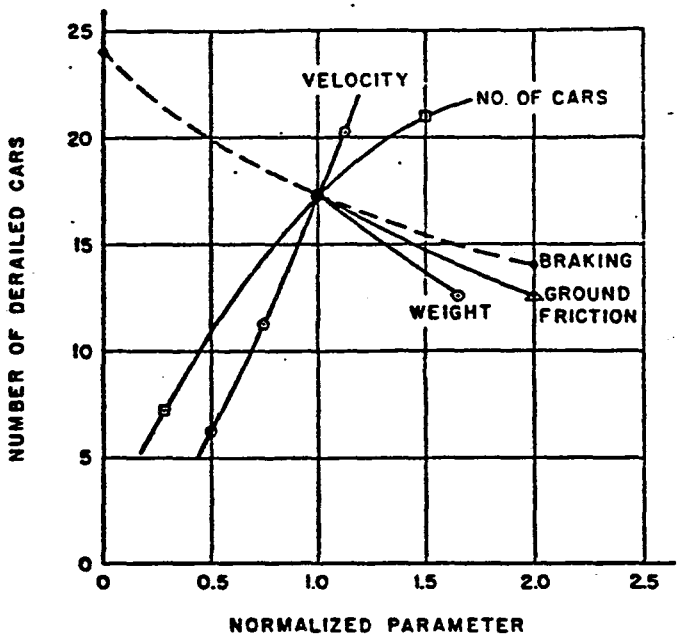

Fig.6 Effect on number of derailed cars of various parameters normalized to base case value

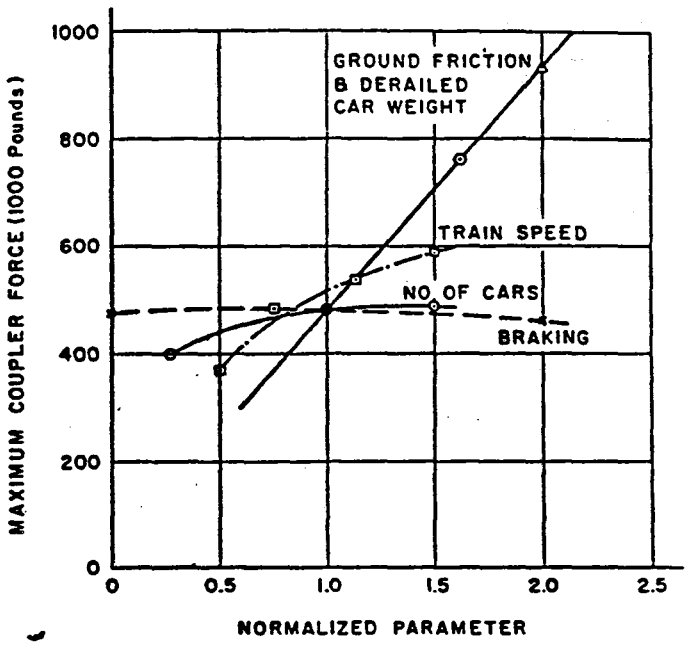

Fig.7 Effect on maximum coupler force of various parameters normalized to base case value

Knowing at between impact, rear wt. could precisely determined

ficient of ground friction. The correlation of the final configuration is shown in Fig. 8. Cars 1 through 4 were completely jackknifed. The fourth car traveled 207 ft from the point of derailment. Correlation of the two configurations is good, considering the conditions.

BASE CASE

This case was carefully chosen to be typical of actual derailment conditions without impairing the clarity of results for variable changes. Based on 87 derailments from AAR files, this case used the average number of cars, 61; the normal coupler moment, E-E; normal braking characteristics, using cast iron brake shoes; the average train speed, 40 mph; and average car dimensions. All cars were the same to produce uniform results, with easily identifiable differences for variable changes. All cars were taken as the average loaded car, 80 tons, to obtain valid coupler forces in the derailing cars.

The number of derailed cars for the base case was 17.1 as shown in Fig. 9. This base case should produce valid comparisons when variables are changed since it produced a typical number of derailed cars.

GROUND FRICTION — 4 above

The coefficient of ground friction was varied to twice the base case value in Case 9. The effect of increased ground friction is to reduce the number of derailed cars (Fig. 6) and increase the coupler forces (Fig. 7). The final configuration is shown in Fig. 9 for comparison to the base case.

BRAKING

Three aspects of braking were studied:

1 Variation of normal braking characteristic
2 Instant brake signal propagation
3 Maximum achievable braking (adhesion limiting).

Normal, twice normal, and no braking results are plotted in Fig. 6 showing the reduction in number of cars derailed with increased braking and in Fig. 7 showing no effect on coupler force.

Instant brake signal propagation shows a reduction of only 0.6 derailed cars (Table 2), but the adhesion limited braking case reduced the number of derailed cars to 10.4, from 17.1 cars for the base case with normal braking. In both cases, the simulations showed no change in other major results.

COUPLER MOMENT

No significant effect was found from changes in coupler moment characteristics. Increasing max-

Table 1 Input Dimensions and Parameters

Case No.	No. of Cars	Speed	Length Ft/IC Ft.	Weight kips	Coupler	Brakes	Ground Friction	Comments
1	90	45	Mixed	Mixed	E-E	C.I.	1.5	Verif.
4	61	40	55/83	160	E-E	C.I.	1.0	BASE
5	90	40	55/83	160	E-E	C.I.	1.0	No. Cars
7	61	40	55/83	160	F-F_{420}	C.I.	1.0	Coupler Moment
8	61	40	55/83	160	E-E	2xC.I.	1.0	Braking
9	61	40	55/83	160	E-E	C.I.	2.0	Ground Friction
10	61	45	55/83	160	E-E	C.I.	1.0	Speed
11	61	30	55/83	160	E-E	C.I.	1.0	Speed
12	16	40	85/60	160	F-F_{260}	Pass.	1.0	Passenger Train
13	16	40	55/83	260	F-F_{260}	C.I.	1.0	No. of Cars
14	61	40	55/83	160	F-F_{420}^{15}	C.I.	1.0	Coupler Moment
15	61	40	55/83	160	E-E	C.I. Instant	1.0	Instant Signal
16	61	40	55/83	160	E-E	Some	1.0	Braking
17	61 $\binom{30}{31}$	40	55/83	$\binom{263}{57}$	E-E	C.I.	1.0	Weight
20	61 $\binom{38}{23}$	40	68/54.5 55/83	$\binom{223}{57}$	E-E	C.I.	1.0	Alternating
21	61	20	55/83	160	E-E	C.I.	1.0	Speed
22	61	20	55/83	160	F-F_{420}^{15}	C.I.	1.0	Coupler Moment
24	61	40	55/83	160	E-E	32,000#	1.0	Braking
26	65	40	85/60	150	F-F_{260}	Pass.	1.0	Long Passenger Train

Precise

Table 2 Simulation Results

Case No.	Description	No. of Derailed Cars	Time Duration (sec.)	Max. F_z (kips)	Max. $\ddot{x}$ (ft/sec2)	Maximum Displacement $X_{(ft)}$	Maximum Displacement $Y_{(ft)}$
1	Crescent City - Verification	16.4	25.4	1,522	-127/71	207	107.3
4	Base Case	17.1	26.2	473	-41.9/39	216.8	13.0
5	No. of Car in Train	20.9	31.8	485	-45.3/40	216.7	13.0
7	Coupler Moment - F-F_{420}	17.1	26.2	471	-43.3/38.5	217.1	13.0
8	Braking - Twice C.I.	14.1	19.8	471	-45.6/44.5	212.3	13.0
9	Ground Friction - Twice	12.6	19.4	938	-81.7/75.3	176.9	14.0
10	Speed - 45 mph	20.3	28.3	535	-44.2/42.6	209.3	12.7
11	Speed - 30 mph	11.2	21.3	480	-41/41.8	172.3	13.8
12	Passenger Train	4.9	9.7	330	-37.4/38.1	249.3	20.5
13	No. of Cars/Coupler	7.3	10.4	406	-41.3/40.4	196.6	12.8
14	Coupler Moment - F $^{15°}_{420}$	16.8	25.9	520	-41.7/39.5	221.7	13.5
15	Braking - Instant	16.5	25.5	470	-43/42.5	217.2	13.0
16	Braking - Some	24.3	44.2	477	-41.4/40.6	148.5	13.0
17	200-ton Loads First	12.6	18.7	759	-43.1/40.8	210.7	13.0
20	Alternating - 2 Loads, 1 Empty	14.2	25.0	683	-72.3/46.1	256	112.0
21	Speed - 20 mph	6.2	15.7	375	-38.8/39.1	140.2	14.9
22	Coupler Moment/Speed	6.2	15.6	376	-35.1/33.9	140.4	15.1
24	Braking - 32,000 lbs.	10.4	13.6	459	-42.8/40.7	212.2	13.0
26	Long Passenger Train	8.5	17.3	452	-35.2/40	271.7	17.4

mum moment capacity from the 280 ft-k E-E test value to 420 ft-k showed no effect, and decreasing the unrestricted coupler swing angle from 30 to 15 deg in combination with the 420 ft-k only reduced the number of derailed cars from 17.1 to 16.8.

NUMBER OF CARS

Increasing the number of cars in the train behind the derailment increases the number of derailed cars, as would be expected. The plot in Fig. 6 shows the effect.

The coupler force is virtually unaffected through the range of 16 to 90 cars, as shown in Fig. 7.

TRAIN VELOCITY

Train velocity shows a marked effect on the number of derailed cars. The plot in Fig. 6 shows the relation between number of derailed cars and train speed; it is related to the kinetic energy of the train.

Coupler forces are affected only slightly by changes in train speed, as shown in Fig. 7.

CAR LENGTH

Car length was not studied directly. However, its effect can be approximated by using Cases 4 and 26.

The results indicated that the number of derailed cars is inversely proportional to car length. There was little effect on the other major results.

CAR WEIGHT

A study of Cases 4 and 17 provides information on the effects of the weight of the derailing cars given a constant total train mass. Case 4 is the base case with 160,000-lb cars, and Case 17 has the 263,000-lb cars in front.

The plot in Fig. 6 shows a strong reduction in the number of derailed cars with increasing derailed car weight. Fig. 7 also shows a strong effect of increasing coupler force with increasing car weight.

TRAIN MAKEUP

In this study, the results indicate that a mixed consist may not form a typical jackknifed pattern. Case 20, a mixed consist, alternating two loads and one empty, produces a major departure from the normal jackknife pattern (Fig. 9) and a great increase in y-displacement which will increase track-side damage.

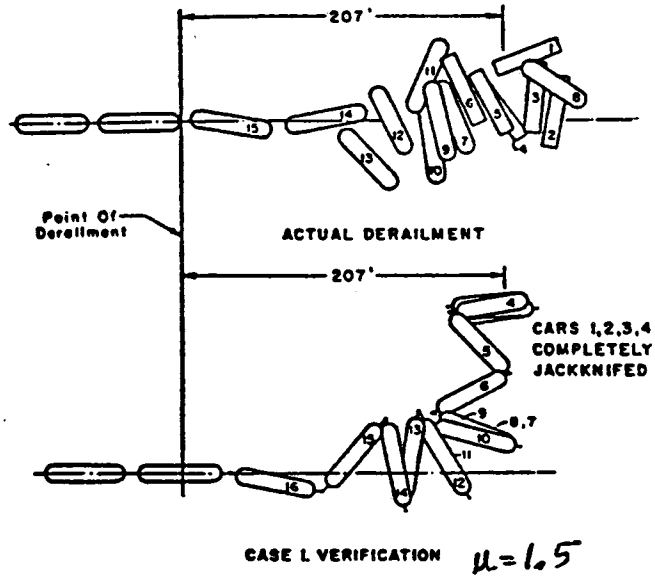

Fig.8 Schematics of actual derailment and verification case

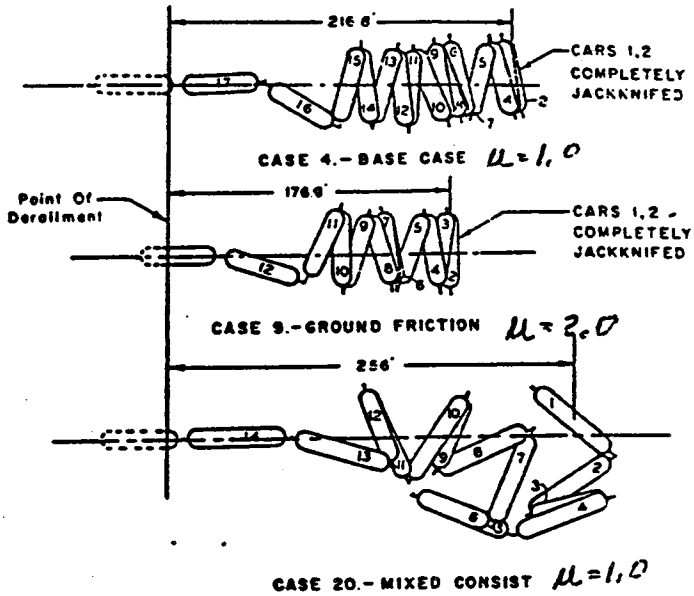

Fig.9 Derailment schematics

PASSENGER TRAINS

In this study, the passenger train also showed jackknifing in derailment accidents. The number of derailed cars is lower than a freight train with the same train mass and speed (Cases 12 and 26 versus Cases 13 and 4). This is because in a passenger train, the derailed cars are longer and the braking is more efficient due to higher brake retarding force, as indicated in Fig. 4, and to faster cylinder pressure buildup.

TRUCK CENTERS

The effect of increased truck centers of the LPG cars was simulated, and no significant effect was noted on the derailment results.

NUMBER OF DERAILED CARS

The number of derailed cars is affected by the train speed, derailed car weight, braking, the number of cars behind the derailment, and ground condition (Fig. 6). The various types of mating coupler or their modifications have negligible effect on the number of derailed cars, either at 40-mph train speed or at 20-mph train speed. This is because the resisting moments between couplers are too small and the contact range is too short to affect the derailed train (Cases 7 and 21 versus Cases 14 and 22).

MAXIMUM COUPLER FORCE

The magnitude of the coupler force in x-direction during the derailing cycle is directly proportional to both the ground friction condition and derailed car weight. Very little effect is caused by the train speed, number of cars in the train, or the different braking (Fig. 7). Again, various types of mating couplers or their modifications have no influence on the coupler force.

The maximum coupler forces shown in Table 2 are rather high and can be considered as an upper bound, because this study did not include the resilient properties of the car body or the cushioning devices.

In all the runs except the Crescent City Verification Case (Case 1), the coupler forces are within the AAR specified buff test load of 1,250,000 lb.

ACCELERATION AND DECELERATION

All the simulations produced almost identical acceleration or deceleration for various train design parameters, except the higher ground friction and the trains with mixed consists.

MAXIMUM DISPLACEMENTS

The maximum displacements in x-direction (along the track) varied from 150 to 300 ft at their final positions of the derailed train. These are primarily influenced by the train speed.

For the displacement in the y-direction (perpendicular to the track), the simulations showed

that, for a symmetrical train, this displacement is essentially constant for the conditions studied; for an asymmetrical train, this displacement is greatly increased.

CONCLUSIONS

The mathematical simulation adequately describes the effect of important variables on derailment severity.

This simulation should be a useful tool to integrate a consideration of derailment severity with other considerations in the total system of railroad operation and car design.

ACKNOWLEDGMENTS

The authors wish to express their thanks to Messrs. Spence of Pullman-Standard, Phillips, Martin, Westin, Olson, and Reedy of the RPI-AAR Railroad Tank Car Safety Research and Test Project Committee for their cooperation and advice throughout this study.

Special appreciation is extended to Westinghouse Air Brake for furnishing the braking performance data from which the brake characteristics used in the study were derived.

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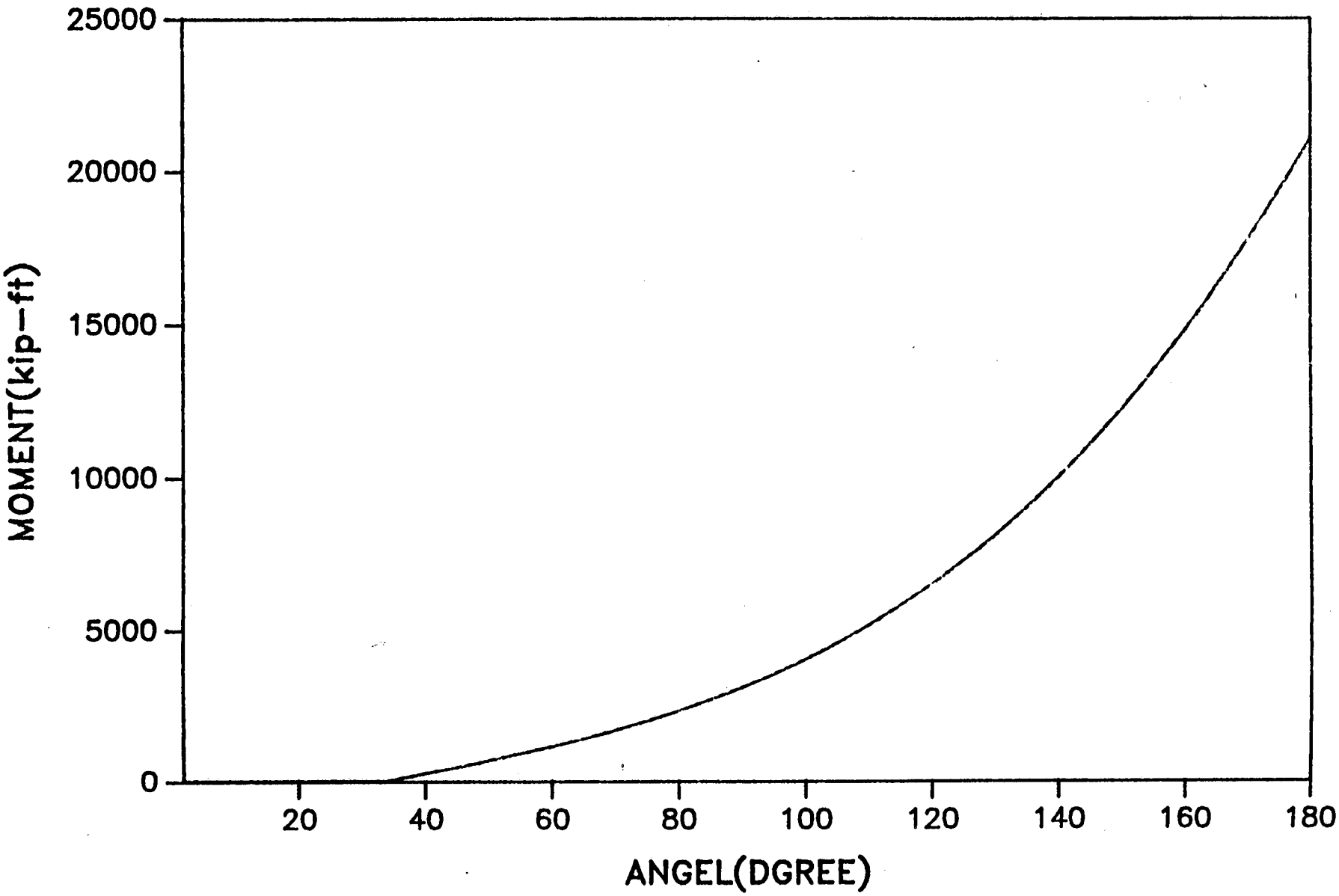
FIGURE A1. COUPLING MOMENTS



January 13,1988

Prof. T. Hirsch, P.E., PhD
Texas A and M University
College Station, Texas

Dear Ted,

This letter is an attempt to summarize the items covered in Our meeting in Texas held on January 9 and 10th. The following items were covered in that meeting and also included are some of my thoughts on them:

1) The enclosed program listing is a workable coding for handling a plastic impact such as that which can occur on a barrier. Although the code is written in Basic, it is readily applicable to be converted to Fortran. The displacement, x, is the motion of the car corner relative to the barrier, the velocity, v, is the velocity of the car corner relative to the barrier, and F is the plastic force assumed to be of the form F=AX^2. The force law can be changed to be any analytical expression that is suitable. Since there is more than one vehicle striking the barrier, all the expressions can be subscripted such as X(I), V(I), F(I), and MODE(I).

2) The attached figure shows my concept of a suitable free body diagram for the impact of a car against the barrier. Although the dynamics is three dimensional, the two dimensional modelling can be applicable if the barrier is high enough to prevent vehicle overturning.

3) Suitable freight car testing can be accomplished at your site by starting with a single empty car that can be accelerated to 40 or so mph, and it can be allowed to run off the track at an angle of incidence of 15 degrees or less. Panel track can be laid on the concrete road bed, and it can be bolted together to a total length of two or three thousand feet as shown in the attached sketch. Later tests can include two or three coupled empty cars that can be accelerated using Jatos or similar propulsion devices. The final tests should include fully loaded

RECD JAN 17 1989

high center of gravity cars and again starting with one car and increasing it to two or three. As you know the carbody cg heights can be as high as 105 inches for body alone; hence it would be important to ultimately run the loaded car tests in order to insure that the cars would not hurdle the barrier.

4) Freight car corner flexibility varies considerably depending on the car type. For example, covered hoppers and open top hoppers have relatively soft corner crushability compared to box cars, gondolas, or flat cars. Box cars and gondolas have heavy side sills and sideplates with heavy corner posts and car ends, and flat cars can have heavy side sills, deck plates, and end sills. I will try to obtain some car drawing on various car types in the near future.

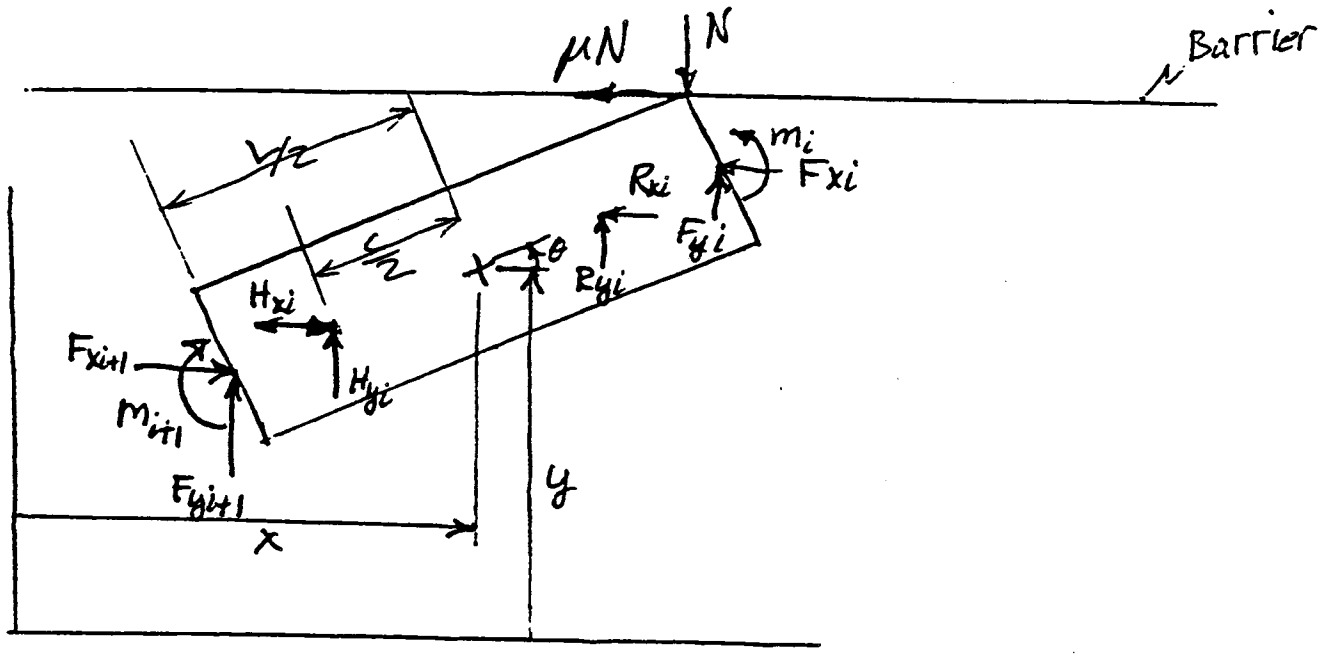
Sincerely,

W. P. Manos

William P. Manos
10350 S. Longwood Dr.
Chicago, IL. 60643

cc Dr. William Harris
Dr. Ray James

```
10 REM PROGRAM FOR TESTING PLASTIC FORCES
20 DATA 0,0,3000,200,1
30 READ XM,X,K,A,MODE
31 PRINT "K=";K, "A=";A
32 PRINT
33 PRINT "  X      V      XO        XM        F"
34 PRINT
40 INPUT "X=";X
50 INPUT "V=";V
60 X1=XM-A*XM^2/K
65 IF (XO<X1) THEN XO=X1
70 IF MODE=1 AND (V<=0 OR X<XM) THEN MODE=2: GOTO 110
80 IF MODE=2 AND X>=XM THEN MODE=1: GOTO 110
90 IF MODE=2 AND X<=XO THEN MODE=3: GOTO 110
100 IF MODE=3 AND X>XO THEN MODE=2: GOTO 110
110 ON MODE GOTO 120, 160, 180
120 IF (X<0) THEN F=0: GOTO 190
130 IF (X>0) THEN F=A*X^2
135 IF (XM<X) THEN XM=X
150 GOTO 190
160 F=K*(X-XO)
170 GOTO 190
180 F=0
190 GG$=" ###     ###     ###.#     ###.#     ######"
192 PRINT USING GG$; X,V,XO,XM,F
197 GOTO 40
200 END
```



APPENDIX B.

EVALUATION OF VEHICLE INITIAL IMPACT FORCE AND STIFFNESS OF FRONT CORNER OF VEHICLE OR RAILROAD CAR

APPENDIX B

EVALUATION OF VEHICLE INITIAL IMPACT FORCE AND STIFFNESS OF FRONT CORNER OF VEHICLE OR RAILROAD CAR

Figure 1B shows Emori's "Analytical Approach to Automobile Collision." Emori showed that the stiffness of a passenger car in a head-on rigid barrier impact was approximately 12.5 times the vehicle weight. This simplifying assumption permits one to easily predict the magnitude of the impact force if one knows the impact speed (V) and weight (W) of the vehicle.

Figure 2B shows how this simple model can be expanded to predict the initial impact force of a vehicle with a longitudinal barrier. The maximum initial impact force data from the instrumented wall permits us to evaluate the stiffness of the front corner of the vehicles and, thus, the initial impact forces with the equations shown.

Table 1B summarizes the stiffness (K) and C factors from the instrumented wall crash test data. Emori found the stiffness of a standard 4,500 lb American passenger car to be 10 to 15 times its empty weight (average 12.5). The instrumented wall found the front corner stiffness to be 9.8 and 13.0 (average 11.4) times the empty weight of the two 4,500 lb cars tested.

This data shows that small cars will have a stiffness of from 18.9 times their empty weight while the heavy truck may have a stiffness from only one to two times their empty weight.

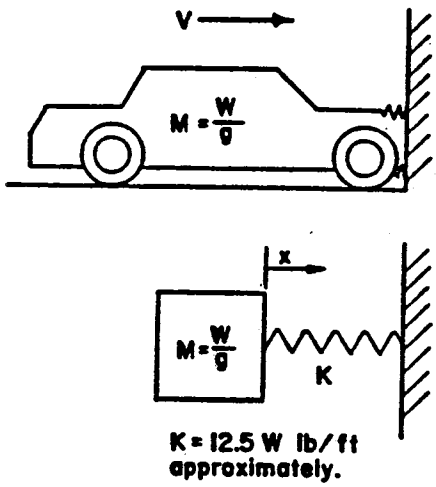
Figures 3B and 4B present this data in graphical form.

From Figure 3B the stiffness of the corner of a railroad car would be about 1.25 times its empty weight. Thus

$$K = 1.25\ W_{empty} = \frac{1.25}{ft} \times 64\ kips = 80\ kips/ft$$

From Figure 4B the initial impact force of a railroad car would be about

$$F_{max} = 0.48\ \frac{W}{3}\ V_{mph}\ \sin\theta$$



$-Kx = M\ddot{x}$ where B.C. are $t=0,\ x=0,\ \dot{x}=V$

$\ddot{x}+\omega^2 x = 0$ where $\omega^2 = K/M$

Solution

$x = (V/\omega)\sin\omega t$

$\dot{x} = V\cos\omega t$

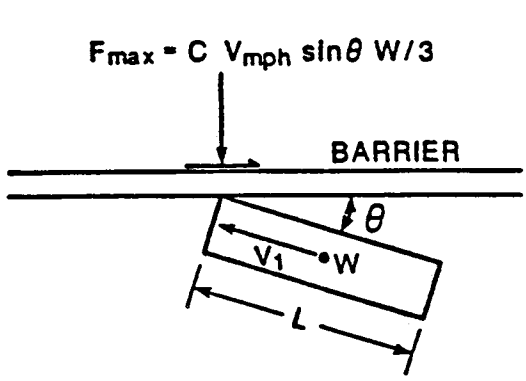
$\ddot{x} = -V\omega\sin\omega t$

$G_{max} = \ddot{x}_{max}/g = -V\omega/g = -.62V_{fps}$ or $-.9V_{mph} = G_{max}$

$G_{avg} = G_{max}(2/\pi) = -.58V_{mph} = G_{avg}$

Emori, R. I. "Analytical approach to automobile collisions." Society of Automotive Engineering Paper 680016, January 1968.

FIGURE B1. SIMULATION OF RIGID-BARRIER COLLISION



$FL = I_0 \ddot{\theta}$

$\sin\ \theta = X/L \quad \theta = \dot{X}/L \quad \ddot{\theta} = \ddot{X}/L$

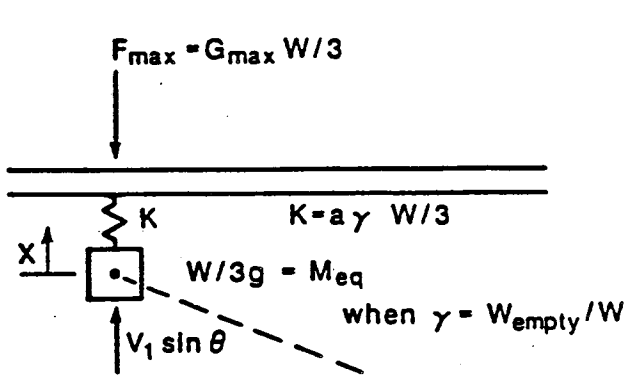
and $I_0 = ML^2/3$ about rear

$FL = ML^2/3\ \ddot{X}/L$

so $F = M/3\ \ddot{X}$

or $M_{eq} = M/3$ or $W_{eq} = W/3$

A.



so $F_{max} = G_{max} W_{eq}$

from Emori's Model

$G_{max} = \ddot{X}_{max}/g = V\ 2\omega/g$

where $V = V_i \sin\theta \quad \omega = \sqrt{K/M_{eq}}$

assume $K = a\ \gamma W/3 = A\ \gamma W$

so $F_{max} = \sqrt{a\ \gamma/g}\ \ V_i \sin\theta\ \ W/3$

or $F_{max} = 1.466\ \sqrt{a\ \gamma/g}\ \ V_{mph} \sin\theta\ \ W/3$

$F_{max} = C\ V_{mph} \sin\theta\ \ W/3$

where $C = 1.466\ \sqrt{a\ \gamma/g}$

$A = a/3$

B.

FIGURE B2. Evaluation of vehicle initial impact force and stiffness of front corner of vehicle.

TABLE B1. SUMMARY OF STIFFNESS AND C FACTORS FROM INSTRUMENTED WALL CRASH TEST DATA

TEST NO.	WEIGHT - lb - VEHICLE TYPE	VELOCITY V mph	ANGLE θ deg	1st FORCE F_{max} kips	LENGTH L ft	$\gamma = \frac{W_{empty}}{W_{gross}}$	C	K kips/ft	A W_e
3451-29	1,970 Honda 74	59.0	15.5	18.4	12.2	.83	1.78	31.0	18.9 W_e
3451-37	2,090 Honda 76	58.5	21.0	21.1	11.9	.84	1.45	21.8	12.4 W_e
3451-30	2,800 Vega 74	58.3	14.8	18.5	15.0	.88	1.33	24.8	10.1 W_e
3451-31	2,830 Vega 75	56.0	20.0	22.0	14.6	.88	1.22	21.0	8.4 W_e
7046-1	4,490 Buick 81	61.8	25.6	56.0	18.4	1.00	1.40	44.0	9.8 W_e
3451-36	4,740 Plymouth 75	59.8	24.0	59.9	18.5	.93	1.56	57.5	13.0 W_e
7046-8	5,350 Suburban 79	44.7	19.5	24.0	18.1	1.00	.90	21.7	4.1 W_e
7046-7	5,400 Suburban 80	64.1	19.7	51.0	18.3	1.00	1.31	46.2	8.6 W_e
7046-5	5,409 Chev C20 P.U.	65.8	19.9	45.0	17.8	1.00	1.11	33.5	6.2 W_e
7046-6	5,432 Chev C20 P.U.	46.8	19.9	20.0	18.0	1.00	.73	14.0	2.6 W_e
7046-10	18,050 GMC 7000 Tr	51.6	16.8	61.0	26.5	.60	.68	41.7	3.9 W_e
3451-34	20,030 Ford Bus 70	57.6	16.5	63.7	34.0	.69	.58	34.0	2.5 W_e
3451-35	32,020 GM Bus 62	56.9	15.8	85.0	35.0	.66	.51	41.9	2.0 W_e
7046-2	40,050 GMC Bus 54	58.6	15.4	101.0	39.8	.72	.49	47.6	1.7 W_e
7046-9	50,000 Int. Tr. & Van 79 (28,300 Tractor Only)	50.4	14.6	39.0	24.6	.53	.33	15.0	1.0 W_e
7046-3	80,080 White Tr & Van 73 (45,820 Tractor Only)	55.0	15.3	66.0	24.2	.33	.30	20.8	1.4 W_e
7046-4	79,900 Peter Tr & Tank 71 (45,410 Tractor Only)	54.8	16.0	91.0	26.2	.36	.40	35.9	2.2 W_e

NOTE: All data from references 10 and 11 of Reference List

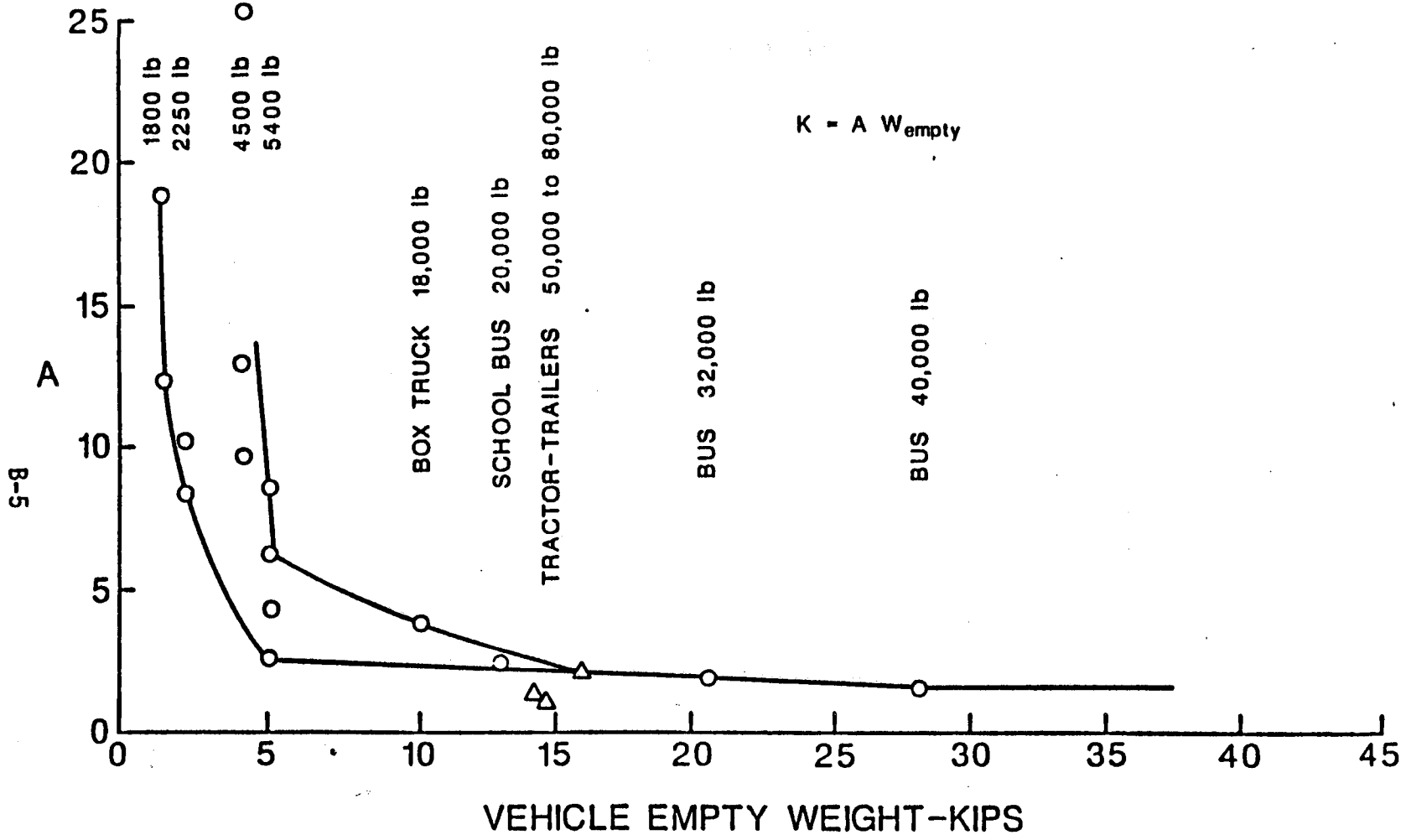


FIGURE B3. Vehicle stiffness vs empty weight.

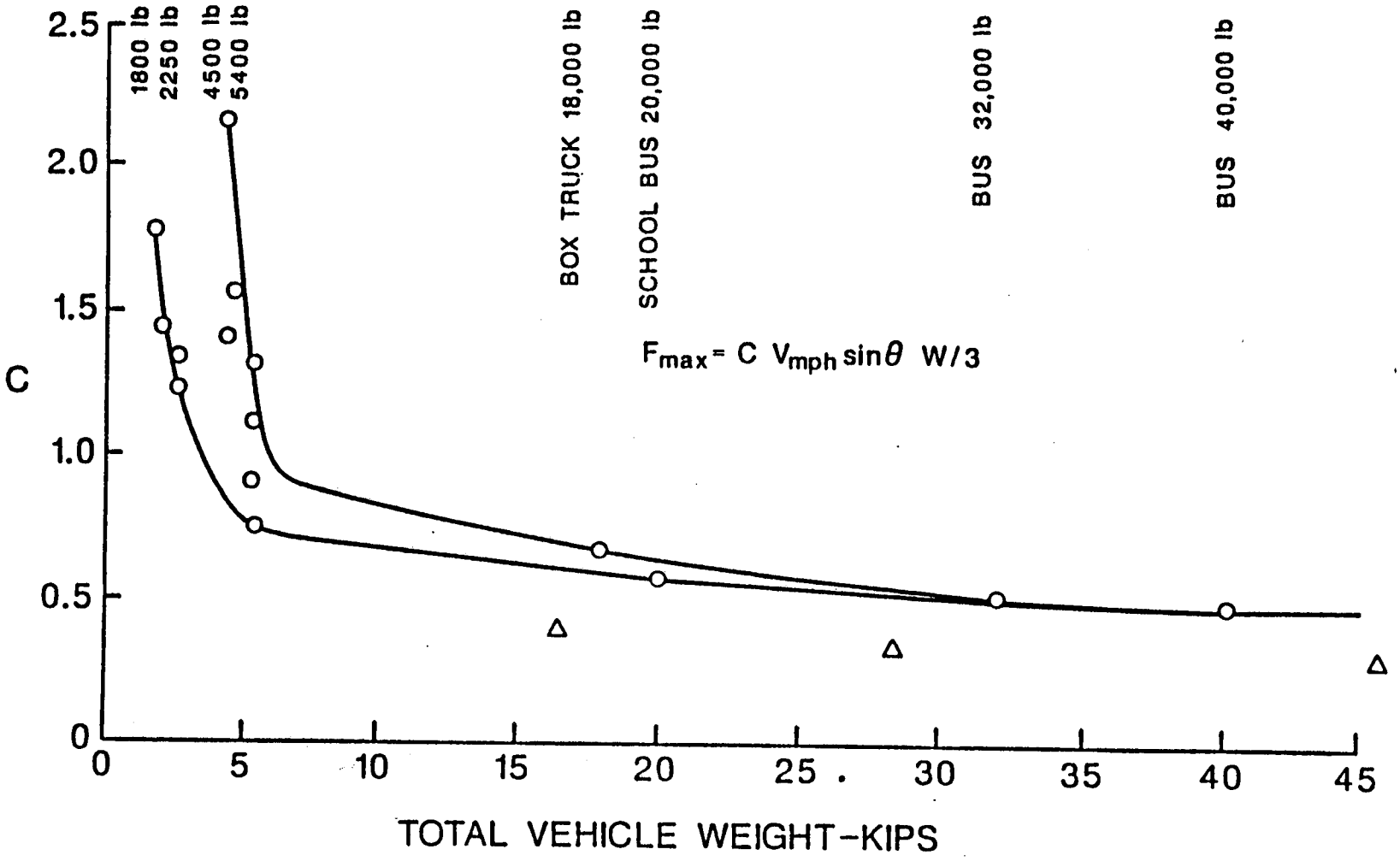


FIGURE B4. C factors vs total vehicle weight.

APPENDIX C.

RAILROAD SAFETY PROGRAMS AND EFFECTIVENESS

RAILROAD SAFETY PROGRAMS AND EFFECTIVENESS

Dr. William J. Harris [1]

INTRODUCTION

Safety in the railroad industry has been a matter of concern to employees, officers, federal agencies and the general public since railroads came into being. Initially, there was concern that the noises associated with the railroad would be very disruptive to horse-drawn vehicles. There was concern that railroads would derail. There were explosions of the boiler in steam locomotives. In short, railroads were looked on with suspicion by many in the community. While railroads themselves had safety codes, public concern demanded governmental intervention. The Interstate Commerce Commission had jurisdiction over railroad pricing and was given jurisdiction over railroad safety. Regulations were not deemed to be sufficient to deal with some concerns of rail labor. Accordingly, legislation was adopted to cover some aspect of safety including safety appliances and brake systems. Despite these regulations and laws, railroad accidents continued to be viewed with concern by the industry, labor, the public, the regulatory agencies, and the Congress.

The nature of safety problems related to the railroads has involved a number of constituencies. Employees who are involved in operation and maintenance of tracks, trains and equipment expect to work in a safe environment. The general public does not expect to be exposed to hazardous conditions as a consequence of railroad operations. Management recognizes its obligation to provide that environment and to protect the interests of its shippers and the general public. Government has assumed an obligation to employees and the general public in regard to railroad safety.

This paper summarizes the recent railroad safety record and includes a discussion of the causes of accidents and measures taken to reduce accident intensity and frequency.

Good data are available for studies of railroad safety. The FRA requires from the industry timely and complete reports on all accidents above a threshold level of cost (currently about $5,000) or involving fatalities or lost time of employees. Major rail accidents are studied railroad personnel to determine a probable cause and by the FRA or the NTSB. Therefore, the accident data base is comprehensive and very useful. It is a basis for organizing research programs and following trends after changes in track, equipment, or operating practices have been introduced. It may suggest changed or new regulations.

THE RAILROAD SAFETY RECORD

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[1] Snead Professor of Transportation Engineering, Department of Civil Engineering, Texas A&M University and Associate Director, Texas Transportation Institute; Formerly Vice President, Research and Test Department, Association of American Railroads.

Railroad safety is often discussed in terms of accidents, particularly derailments. While derailments are spectacular and may involve considerable property damage, they are not a major factor in injuries or fatalities as can be seen from the following discussion of the railroad safety record based on FRA reports and definitions.

Train accidents include mainline derailments, yard derailments, and train to train collisions. They led to only about 2 percent of reported fatalities and 1.4 percent of injuries during the period 1978 to 1986.

Train incidents refer to occurrences involving moving trains in which there was relatively little financial loss but which resulted in death or injury. Non-train incidents involve injuries or deaths not associated with the movement of trains. These two categories accounted for about 40 percent of railroad fatalities, most of whom are trespassers, and 92 percent of railroad injuries during the period 1978 to 1986.

Grade crossing accidents accounted for 58 percent of fatalities and 7 percent of the injuries during the period 1978 to 1986.

Significant improvements in railroad safety occurred in the period 1978 to 1986.

1. Train accidents have decreased from nearly 25 per million train-miles in 1978 to about 8 per million train miles in 1986.

    a. Mainline derailments from 10 to 2.8.

    b. Yard derailments from 7.6 to 2.4.

    c. Collisions from 3.4 to 0.8.

    d. Other from 4 to 2.

2. Fatalities, by type of person, have decreased from 1646 in 1978 to 1091 in 1986. This decrease was observed in all categories except for trespassers.

    a. Employee fatalities from 131 to 63.

    b. Passengers from 13 to 4.

    c. Non-trespassers from 1010 to 505.

    d. Trespassers from 492 to 519. (474 in 1985).

3. Fatalities, by type of accident, have decreased from 1646 in 1978 to 1091 in 1986.

    a. Train accidents from 61 to 13.

    b. Train incidents from 457 to 406.

    c. Non train incidents from 64 to 56.

d. Grade crossings from 1064 to 616.

4. Railroad employee fatalities, by type of accident, have decreased from 131 in 1978 to 63 in 1986.

a. Train Accidents from 28 to 8.

b. Train Incidents from 55 to 21.

c. Non-train incidents from 43 to 32.

d. Grade crossings from 5 to 2.

5. Total injuries by type of accident have decreased from 71,834 in 1978 to 26,707 in 1986.

a. Train accidents from 1065 to 947.

b. Train incidents from 11,095 to 3894.

c. Non-train incidents from 55,321 to 19,318.

d. Grade crossings from 4,353 to 2,458.

6. Injuries to employees by type of accident have decreased from 64,054 in 1978 to 22,172 in 1986.

a. Train accidents from 642 to 215.

b. Train incidents from 9,946 to 3,432.

c. Non train incidents from 53,283 to 18,425.

d. Grade crossings from 183 to 100.

7. Injuries per 100 man-years have decreased from 12 in 1978 to about 7 in 1986.

a. Lost-time injuries from 7.5 to 4.

b. Minor injuries from 4.5 to 3.

REASONS FOR SAFETY IMPROVEMENT

The railroad safety data made available by FRA can be examined in terms of the causes of accidents, fatalities, and injuries. These are as follows:

1. Causes of mainline derailments per million train miles have decreased as follows:

	1978	1980	1982	1984	1986
Track	4.8	3.3	2	1.6	1.1
Equipment	3.1	2	1.4	1.2	0.8
Human Factors	1.1	0.8	0.6	0.5	0.4
Misc.	1.0	0.8	0.6	0.5	0.4

GENERAL DISCUSSION

For competitive reasons, the railroads made a rapid transition from fifty to one hundred ton cars in the decade of the 1960's. In order to pull trains with hundred-ton cars, new classes of six-axle locomotives were ordered and put into service. These locomotives had three-axle trucks in order to distribute the weight adequately and minimize damage to track. It was soon evident that these heavy locomotives and heavy trains were serving as track inspectors. They quickly found cases where the subsoil was marginal and significant damage was done to track alignment and cross level. They quickly found cases where the lateral strength of track was marginal, especially on curves and a number of derailments occurred. At the same time, the profits of the industry fell as a consequence of the completion of the Interstate Highway system, intense competition from trucks that saw the transfer of high-rated commodities from rail to highway, and inability to compete effectively because of a combination of management, labor, technology, and regulatory problems.

The industry organized within itself to deal with this problem.

On the management side, it reorganized, directed much more attention to marketing and operating costs, and sought for changes in labor agreements and regulatory procedures.

On the labor side, it mounted an aggressive program to change work rules, increase safety training, and reduce costs.

On the technical side, it strengthened its research program particularly under the Association of American Railroad Railroads and began to address the dynamic interaction of a train with a track structure, hazardous materials transportation problems, a wide range of track and component failures, employee fatality and injury issues, and grade crossing safety. It proceeded to codify guidelines for proper train handling that had an immediate salutary effect in eliminating these derailments. It improved track maintenance practices.

On the regulatory side, it sought relief from current regulatory practices that required extensive review and delay in regard to rates, encouraged approval of contract rate programs. requested greater freedom to merge and abandon los density lines, and arranged for relief from cross-subsidy of rail passenger service.

The industry enjoyed considerable success in its programs.

Its restructured management did focus on profitable and non-profitable moves and establish procedures to deal with this situation. It did begin to reverse the long period of decline in rail market share.

It did begin to reduce labor costs by bargaining to eliminate inefficient practices, many held over from earlier technologies and practices.

Much progress was made on the technical side. Improved practices and equipment reduced individual causes of accidents and derailments. Attention to injuries identified opportunities for improvement in training and work practices. Attention to grade crossings and new governmental programs made it possible to increase grade crossing protection systems in the most critical sites and to increase public awareness of the need for safer practices.

Significant changes in economic regulation led to increases in disposable income and to major investment in track maintenance and equipment upgrade. At one time, nearly 15 percent of the track was operated under slow orders because of deferred maintenance and nearly 30 percent of the industry was in or facing bankruptcy. After deregulation, in only a few years, bankruptcies were eliminated and massive investments in track maintenance virtually eliminated slow orders.

The Federal Government made major investments in NE corridor railroads organized as Conrail. It provided the difference between costs and the fare box income from passengers in order to make it possible to have a rail passenger system that did not divert freight revenue to passenger services. It created a large research program and supported independent studies as well as programs in which it worked cooperatively with the industry.

The combined consequences of these actions is reflected in the improved safety record of the industry. Some of the details are discussed below.

THE TANK CAR SAFETY PROBLEM

In the decade of the 1960's, railroads began to be a principle transportation element in the movement of liquified petroleum products. These products typified by propane were moved in pressurized cars. The pressurized cars were designed to contain the liquified gas during transport and to control the pressure by allowing over pressure to be released through the pressure release system. The over pressure was based on heating of the gas during transit. Initially small cars of 10,000 gallons or less were used for transport of these products. As 100 ton cars came into service, the size of propane cars quickly rose to 33,000 gallons and some tens of thousands of these units were placed in service to supply many parts of the country. With propane gas required during the heating season, derailments occurred while these cars were being moved. A new phenomena was observed as a consequence of these derailments. An average of about six tank cars ruptured per year. Their violent rupture led to the release of large quantities of flammable products such as LPG under pressure. The combustion of the gas created an intense fire ball that rose to a height of a few hundred feet. The internal pressure in the car at the time that fracture occurred hurled ends and pieces of the car as much as several thousand feet from the scene of the accident. Several small towns were the site of such accidents and public concern about railroad safety grew very rapidly.

Detailed examination of accidents revealed the sequence of events before violent rupture. These started with the puncture during a derailment, of the tank of a car carrying compressed

flammable gases generally by the coupler of an adjacent car. The gases or liquid ignited and began to heat an adjacent undamaged car. The pressure in the tank rose, and the products vented. When the liquid level fell, the flames were against the tank surface that was not kept cool by the liquid in the tank. The steel heated quickly. When it reach a temperature of about 1200 degrees F, its strength was reduced. The internal pressure caused a bulge to form and the steel tore along the thin metal at the top of the bulge. A crack then ran along the tank, turned, and went completely around the tank. At that point, the contents were released in a large fire ball, and the ends of the tank were hurled in opposite directions by the internal pressure in the tank.

Dozens of proposals were put forward to eliminate these actions. A major cooperative research program was started by the tank car owners and the railroad industry that carried out some programs in cooperation with the Federal Railroad Administration. The upshot of the program was the identification of the technical solution which required no change in operating practices and no redesign of the cars. The addition of features including top and bottom shelf couplers to prevent coupler separation during derailment; head shields to minimize the potential for puncture of the end of the car during derailment; and thermal insulation to delay the time of overheating of the unwetted surface of the tank car shell until the contents were completely released through the thermal pressure release system. The research findings were translated into regulations that required installation of the shelf couplers, the head shields and the thermal insulation on all existing and all new cars. The solution has been almost entirely successful. This railroad safety problem has been solved.

THE RAIL PROBLEM

Rails were an early source of significant railroad problems. The manufacturing process introduced hydrogen into rails and did not remove it. This hydrogen could agglomerate and create a situation called shatter cracking. It was difficult to predict and always led to a derailment. Metallurgical studies many decades ago, identified the source of the problem and identified heat treating practices that eliminated hydrogen. Those practices are used by all rail manufacturers, and the problem has disappeared.

With the increased weight of equipment and the increased axle loads, and with increased tonnages, other kinds of rail problems occurred. The holes drilled in rail to accommodate bolts in the plate joining two rails together created an area of weakness in the rail. Joint bar bolt hole cracking was a principal source of rail failure. The batter at the end of a rail caused by wheel impact also created a potential area for rail failure. These two problems were addressed when the industry went to continuously welded rail. Continuous welding eliminated the bolt holes. It also eliminated the gap between rails, and rail end batter occurred only every 1200 feet instead of at 39 foot intervals. A significant reduction was observed in rail failures as a consequence of continuous welding.

For the past forty years, careful attention has been given to means of testing rail to identify cracks or flaws before they grew to a dangerous size. Magnetic and ultrasonic techniques were examined. Ultimately, the ultrasonic systems proved to be the most satisfactory and they are now in extensive use as required by railroad practices and FRA safety regulations. Beyond that, significant improvements have been made in the manufacturing of rail to eliminate sources of flaws in manufacture and to increase the uniformity and quality of the product. Rail failures are rare today and their numbers are decreasing.

THE WHEEL PROBLEM

Wheels were another source of problems for the industry. Two kinds of issues arose: a class of wheels with straight plates was found in which there were premature fatigue failures in the plate; wheels used under the same circumstances, but designed with curved plates, had a significantly lower rate of failure. The required resistance to thermal and mechanical stress in the plate of the wheel was established and new evaluation procedures were adopted. Designs that did not satisfy the requirements were not used. As a consequence, premature plate failures disappeared.

One railroad adopted higher carbon steel for wheels in order to minimize wear. These wheels were somewhat more prone to failure, and the industry quickly recognized their deficiencies. The combination of the lower crack resistance of these plates and high thermal input associated with braking of the heavier cars, led to the complex set of requirements intended to eliminate from service, wheels that had been damaged by thermal input arising from braking. This issue is not fully resolved to this day. However, it is clear that improved train handling practices and improved wheel design have made a major contribution toward reducing wheel failures. Much more careful attention to the removal from service of wheels with cracks in them has also made a significant contribution. Out of twelve million wheels in service, not more than 60 break each year. If all of the wheels were made of what is called a low stress design, that number would be about ten and the industry is rapidly making progress toward the lower stress wheel. In addition, research sponsored by the Federal Railroad Administration and by the industry has recently identified the fact that fully heat treated wheels have even greater resistance to failure. Some railroads have already adopted a requirement for heat treated wheels for all new procurement. Other railroads are considering a move in this direction. With improvement in design and in metallurgical characteristics of the wheel, it can expected that wheel failures will be even more rare. This will make a contribution to safety.

THE BEARING PROBLEM

Another kind of issue relates to bearings. Railroads initially began operations by use of what are called plain journal bearings, in which a half of a cylindrical sleeve of bearing brass is introduced. The axle rotates against this bearing material with a film of oil placed on the axle by the use of cotton waste soaked in lubricating oil, or lubricator pads that cause oil to be wicked onto the axle. However, problems in lack of lubrication led to a significant number of cases of overheated and burned off journals with ensuing derailments. The adoption of roller bearings reduced very significantly the frequency of bearing failure in service. However, some bearing failures did occur. The industry then adopted a practice of using infrared detectors at the side of the rail to report bearings that were overheating. These also reduced the number of bearing failures. Improved lubricants and improved bearing seals have insured the effective lubrication and the retention of lubrication. Improved bearing inspection practices have eliminated from service parts that fell short of the desired high quality. Accordingly, significant reductions in bearing failures are currently being observed.

EMPLOYEE SAFETY

An examination of employee safety indicates that a very small fraction of railroad employee fatalities or injuries are associated with derailments. Other incidents in the work place involving moving equipment or the use of hand tools or lifting are far more serious from the standpoint of employee safety. The Association of American Railroads has recognized this for about ten years

and has organized specific programs directed at dealing with the highest causes of incidence and is making good progress in reducing those kinds of injuries.

A study made jointly with labor and the Federal Railroad Administration examined a locomotive cab. Undesirable protrusions were found in the cab. The floor service was found to be unnecessarily slippery. The door handle was found positioned poorly to permit effective protection of hands while the door was being closed. Many changes have been made in the locomotive cab and the occupants are much safer than they were before the changes were made.

The problem of lifting is a classical problem. Lower back injuries are a matter of profound concern to people in many industries. By careful study of the shop practices of railroad personnel that have to lift products, many opportunities for reducing stress on the lower back have been discovered. Some have to do with lifting posture. Some have to do with the nature of the product being lifted; some with the location of the product being lifted. Progress has been made in reducing the number of back injuries as these more precise, more carefully structured analyses are completed.

Similar studies of other causes of accidents such as use of hand tools and slipping, stumbling and falling are being made to determine what changes in training or job content can be achieved that will maintain or improve productivity and at the same time increase safety.

GRADE CROSSING SAFETY

The grade crossing safety problem has been a matter of grave concern to railroads and the general public for many years. As discussed previously, there has been a very sharp reduction in the number of accidents and in the number of fatalities in accidents. The improvements have not just been a matter of chance. A grade crossing accident involves the interaction of the train and the vehicle operating on a highway. Trains are not in position to stop at grade crossings. Therefore, responsibility is placed on the driver of a vehicle to recognize the presence of a train and to bring his vehicle to a stop. Some crossings are protected by a flashing lights and gates. The vast majority of crossings have only cross bucks.

It was originally expected that at a cross buck, a vehicle would stop, look and listen. But as highway traffic control measures were developed, different kinds of instructions were given to drivers. A classical configuration of a yield sign and a stop sign was developed for highway intersections. Standard practices in regard to lights to control traffic at fully controlled intersections were adopted, but these control systems were not applied to rail crossings. In some states, buses are required to stop at rail grade crossings, but not other motor vehicles.

Incidents occur in which trains strike vehicles as they cross the tracks or in which drivers fail to bring the vehicle under control and strike the side of the train, generally near the head end. Study of these incidents indicated that drivers were not alert to the presence of a train or that they attempted to outrun trains at crossings. The desirability of a more intensive effort to reduce the incidence of grade crossing accidents became evident. While they accounted for only about one to three percent of all highway accidents, they were significantly the highest cause of fatalities in railroad operations.

In order to cope with the situation, the Federal Railroad Administration sponsored a program conducted by the AAR and the Texas Transportation Institute to inventory all grade crossings in the United States. That inventory is being updated. Every crossing is identified and records are

now being kept so as to permit more systematic study of the hazard at each grade crossing. This leads to the ability of the interested parties to identify the most hazardous crossings. In 1972 the Federal Highway Administration included in its program, a fund of about a hundred and seventy million dollars a year to be transferred to the states for selective installation of grade crossing protection systems. Using the inventory and an accident reporting system on accidents, it has been possible to pick out and protect those crossings that are the most hazardous. Additional funding has been provided for this program in recent years.

In addition, the Union Pacific Railroad, followed quickly by others, began a program in the mid-1970's called Operation Lifesaver. The railroads recognized that it would never be possible to protect all grade crossings and even when grade crossings were protected, there were some fatalities. Generally, those involved people deliberately trying to out-race a train; or frustrated by delay, people who went around the end of a grade crossing protection gate assuming the track was clear, only to be struck by a train on a parallel track. It was understood that it was necessary to try to change attitudes toward grade crossings, particularly at protected crossings. Operation Lifesaver involved lectures in schools, discussions with civic groups, and advertising and other measures to increase public awareness of hazards at grade crossings. The combination of Operation Lifesaver, now widely utilized throughout the United States, and the funding made available by the Federal Highway Administration to the several states for protection systems, has made a major contribution to safety as shown above.

THE TRESPASSER PROBLEM

A third general problem in the railroad industry has been the number of trespassers who are killed as shown in an earlier section of the report. These are people who choose to walk on railroad trestles and railroad tracks without recognizing the speed and the quietness with which trains operate today. It involves children who choose to play in rail yards; it may involve some suicides. It is a very difficult problem to cope with. Railroads have fenced large parts of their property. But that has not been enough to keep everyone off the track, and the industry is seeking further means of reducing the number of trespassers who become involved in railroad accidents each year.

A REMAINING MAJOR PROBLEM

In terms of % of derailments, causes are as follows: (1)

1. Track related

Roadbed Defects	3.1
Track Geometry	24.8
Rail and Joint Bar	13.4
Frogs and Switches	4.4
Other Track	0.3
Signal and Com. Fail.	0.1

		46.1
2. Equipment Related		
Brakes	3.0	
Car Body		3.2
Coupler and Draft Gear	3.1	
Truck Components	7.1	
Axles and Bearings	6.9	
Wheels		5.9
Other Equipment	0.9	
		30.1
3. Human Factors		
Misuse of Brakes	1.4	
Signalling	0.2	
Other Rules	0.7	
Use of Switches	1.3	
Speed	1.9	
Misc. Human Factors		6.6
		12.1
4. Miscellaneous	11.7	

GENERAL SUMMARY

It can be seen through examining the three principal causes of accidents and the detailed subsets of these that no one equipment or track cause accounts for more than three or four percent of all accidents. The cost of trying to increase the degree of safety associated with those various systems is extremely high. For example, the installation of ten times as many hot box detectors could require investments of billions of dollars and still would not be expected to eliminate all overheated bearing accidents. The installation of equipment to bring a train to a controlled stop, should there be a violation of the signal, could involve billions of dollars, and yet it would eliminate a very small number of accidents per year, assuming that the equipment was always %100 reliable.

The railroad industry is only about a 35 billion dollar a year enterprise. Its net is less than four percent per year or roughly a billion to a billion and a half dollars. Any one of these measures could take all of that net and with maintenance could reduce the net still further.

The industry is basically a safe industry. Trying to make it an absolutely safe industry may not be possible. Attempts to take specific measures against highly publicized single incidents, can lead to enormous misapplication of funding. While the industry is at the point at which continued vigilance must be maintained to be sure that the accomplishments of the past are reflected in continuing achievement of the current high level of safety in the industry, it must be understood that further improvements in safety will be slow in coming.

Government agencies and the Congress will continue to examine the safety practices of the industry. It is incumbent on the technical community to recognize these social concerns and to find a way to ensure that legislative or regulatory action reflects the more accurate and technically correct approaches. In the absence of an adequate technical input, requirements for changes in or new practices or equipment may fall far short of the desired improvements and increase costs in a disproportionate manner.

REFERENCES

(1) Harvey, Aviva E.; Conlon, Peter C.; Glickman,Theodore S.; Statistical Trends in Railroad Hazardous Materials Transportation Safety, 178 to 1986, AAR Research and Test Department Publication R-640, September, 1987

APPENDIX D.

SUPPORTING CALCULATIONS FOR BARRIER DESIGNS

METRO TRAIN BARRIER - STEEL

POST W 6 x 15 $Z = 10.8$ in.^{3} $F_y = 36$ ksi

$M_p = 32.4$ kip-ft

$P_p = \frac{M_p}{H} = \frac{32.4}{3.75 \text{ ft}} = 8.6$ kips $= P_p$

BEAM W 8 x 18 $Z = 17$ in.^{3} $M_p = 51$ k-ft

FAILURE MODE ANALYSIS

$L = 6$ ft $\quad \ell = 5$ ft

Beam Load $(w\ell) = \frac{8 M_p}{L - \ell/2}$

No. SPANS		BEAM LOAD	+	POST LOAD	=	LOAD Tot. Mech.
1		116 kips	+	0	=	116 kips
2		43	+	8.6	=	51.6
3		26	+	17.2	=	43.2
4		19	+	25.8	=	44.8
5	2 Post = 17 → OK	14.8	+	34.4	=	49.2 kips
6		12.2	+	43	=	55.2

The require capacity is 39 kips for 7 ft offset distance (see Table 10)

FOUNDATION ANALYSIS METRO TRAIN BARRIER

POST W 6x15 $P_p = 8.6$ kips $= P_{dynamic}$

$P_{dynamic} = P_{static}(1 + JV)$

$J = 0.14$ sec./ft V = impact velocity ft/sec
$= 9.6$ ft/sec Metro

$P_{dyn} = P_{st}(1 + .14 \times 9.6) = 2.34\, P_{st}$

$P_{st} = \frac{P_{dyn}}{2.34} = \frac{8.6 \text{ kips}}{2.34} = 3.7 \text{ kips} = P_{ST}$

for shallow foundation assume $q_u = 1$ tsf $B = .5$ ft

$$D = \frac{P}{\sigma B} + \sqrt{2\left(\frac{P}{\sigma B}\right)^2 + 4H\left(\frac{P}{\sigma B}\right)}$$

$C = 1$ ksi/ft^2
$\sigma = 9c$
$H = 3.75$ ft

$$= \left(\frac{3.7 \text{ kips}}{9 \text{ kips/ft}^2}\, .5 \text{ ft}\right) + \sqrt{2(.82)^2 + 4 \times 3.75(.82)}$$

$= .82$ ft $+ 3.70$ ft $= 4.52$ ft $= D$

Use $D = 5$ ft

COHESIVE SOIL

$K_p = 4.6$

$K_p = \tan^2(45° + \phi/2)$

assume $\gamma = 120$ pcf $\phi = 40°$

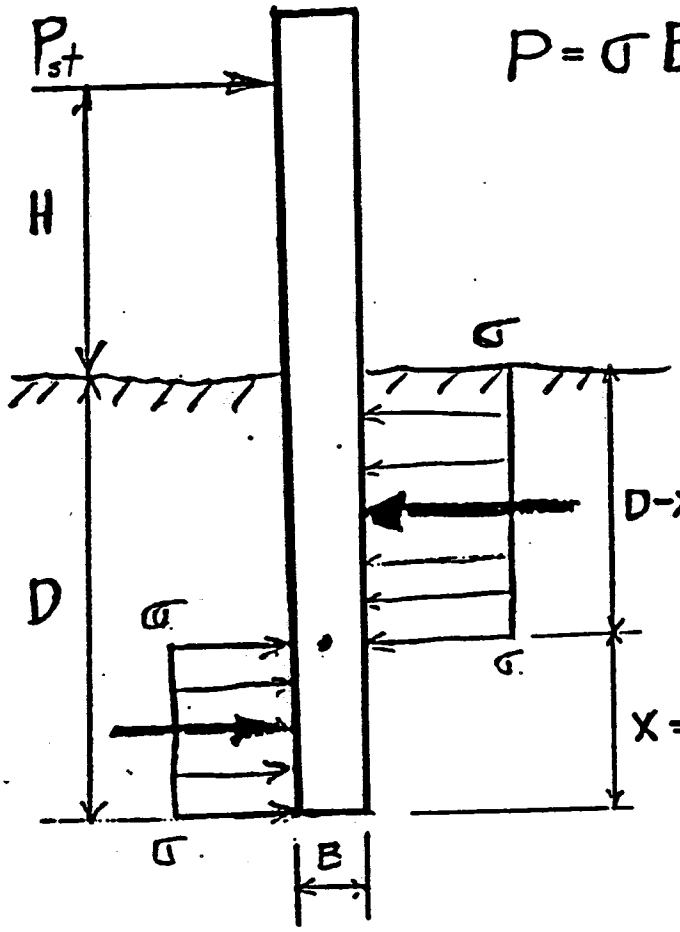
$$D^3 = \frac{2P(H + D)}{\gamma B K_p}$$

$D = 6.5$ ft Use $D = 7$ ft

COHESIONLESS SOIL

COHESIVE SOIL

$\Sigma M = 0$



$$P = \sigma B\left[\sqrt{4\left(\frac{D}{2}+H\right)^2 + D^2} - 2\left(\frac{D}{2}+H\right)\right]$$

$\Sigma F_x = 0$

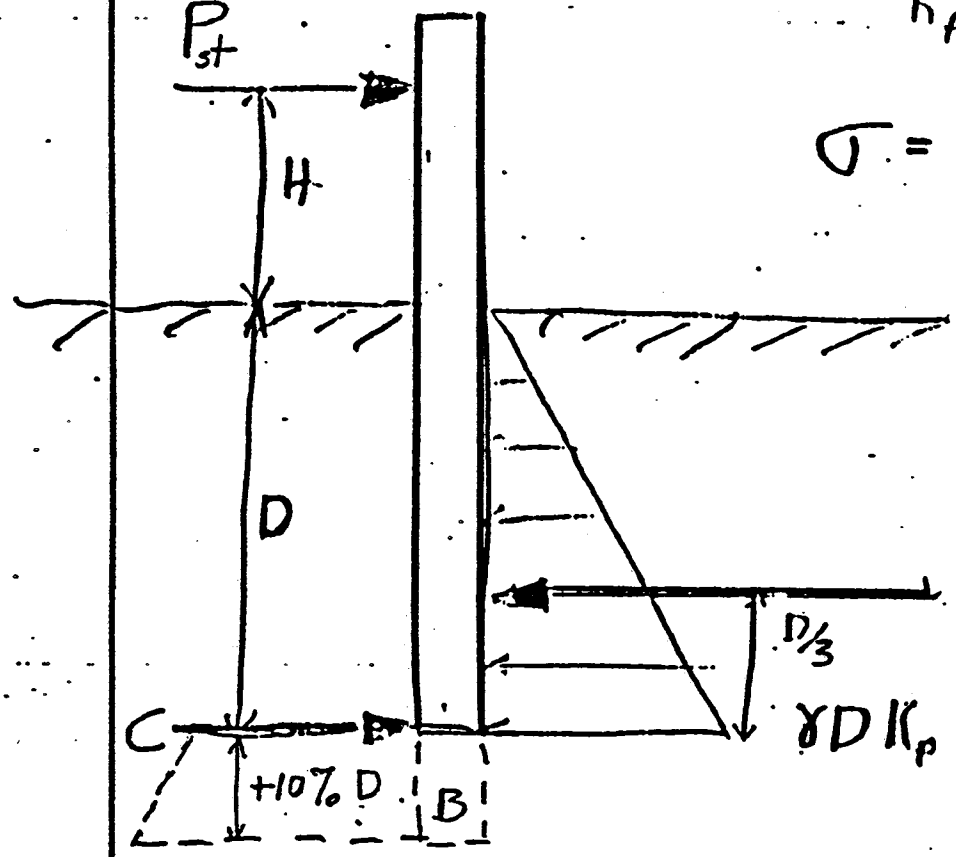
$$X = \frac{1}{2}\left(D - \frac{P}{\sigma B}\right)$$

$$D = \frac{P}{\sigma B} + \sqrt{2\left(\frac{P}{\sigma B}\right)^2 + 4H\frac{P}{\sigma B}}$$

$\sigma = 9c$

FIGURE D1. FOUNDATION DESIGN OF POST IN COHESIVE SOIL (Ref. 27)

COHESIONLESS SOIL



$$K_p = \tan^2(45° + \phi/2)$$

$$\sigma = \gamma z K_p \times 3$$

$$\Sigma M_c = 0 = P(H+D) - \frac{\gamma D K_p}{2} BD \times \frac{D}{3} \times 3$$

$$P = \frac{\gamma B K_p D^3}{6(H+D)} \times 3$$

$$P = \frac{\gamma B K_p D^3}{2(H+D)}$$

$$D^3 = \frac{2P(H+D)}{\gamma B K_p}$$

FIGURE D2. FOUNDATION DESIGN OF POST IN COHESIONLESS SOIL (Ref. 27)

FREIGHT TRAIN BARRIER - STEEL

POST HP 14 x 102 $Z = 169\ in.^3$ $b = 14.8\ in$

$F_y = 36\ ksi$ $d = 14\ in.$

$M_p = 507\ kip\text{-}ft$

$H = 6.58\ ft$ $P = \frac{507}{6.58} = 77\ kips = P_p$

BEAM W 27 x 84 $Z = 244\ in.^3$ $b = 10\ in$

$M_p = 732\ kip\text{-}ft$ $F_y = 36\ ksi$ $d = 27\ in$

FAILURE MODE ANALYSIS

$L = 9\ ft$ $\ell = 5\ ft$ $M_p = 732\ kip\text{-}ft$

Beam Load $(w\ell) = \frac{8M_p}{L - \ell/2}$

NO. SPANS	BEAM LOAD	+	POST LOAD	=	TOTAL MECH LOAD
1	901 kips	+	0 kips	=	901 kips
2	378	+	77	=	455
3	239	+	154	=	393
4	175	+	231	=	406
5	138 (2 POST → 154)	+	308	=	446
6	114	+	385	=	499

FOUNDATION ANALYSIS HP 14 x 102 POSTS

$P_{dynamic} = 77$ kips $V = 5.9$ ft/sec

$P_{dy} = P_{st}(1 + JV)$ $J = 0.14$ sec/ft

$= P_{st}(1 + .14 \times 5.9) = 1.83\, P_{st}$

$P_{st} = \frac{P_{dy}}{1.83} = \frac{77 \text{ kips}}{1.83} = 42.1$ kips

COHESIVE SOIL assume $q_u = 1.5$ tons/ft^2

$B = 15$ in. $= 1.25$ ft $C = 1.5$ kips/ft^2

$H = 6.58$ ft $\sigma = 9c$

$$D = \frac{P_{st}}{\sigma B} + \sqrt{2\left(\frac{P_{st}}{\sigma B}\right)^2 + 4H\left(\frac{P_{st}}{\sigma B}\right)}$$

$$D = \frac{42.1}{13.5 \times 1.25} + \sqrt{2\left(2.5\right)^2 + 4 \times 6.58 \times 2.5}$$

Cohesive Soil

$D = 2.5 + 8.9 = 11.35 \approx$ **12 ft = D**

COHESIONLESS SOIL $\gamma = 120$ pcf $\Phi = 40°$

$K_p = \tan^2(45° + \Phi/2)$

$K_p = 4.6$

$$D^3 = \frac{2P(H+D)}{\gamma B K_p}$$

$$D^3 = \frac{2 \times 42.1 (6.58 + D)}{.12 \times 1.25 \times 4.6}$$

assume $D = 12$'

$D^3 = 2267$

$D = 13.1$ ft

D = 14 ft Cohesionless Soil

FREIGHT TRAIN BARRIER – CONCRETE

YIELD LINE ANALYSIS

$F_y = 60$ ksi

$f_c' = 3.6$ ksi

TOP BEAM M_b

$b = 18$ in. $d = 14.5$ in.

$A_s = 3\text{-}\#9 = 3$ in.^{2}

$$a = \frac{f_y A_s}{.85 f_c' b} = \frac{60 \times 3}{.85 \times 3.6 \times 18} = 3.27 \quad a/2 = 1.63 \text{ in.}$$

$$M_b = .9 \times 60 \times 3 \left(14.5 - 1.63\right) = 2085 \text{ k-in}$$

$$M_b = 173.7 \text{ kip-ft}$$

WALL M_w

$b = 78$ in. $d = 14.5$ in.

$A_s = 7\text{-}\#8 = 5.59$ in.^{2} $\rho = .049$

$$a = \frac{60 \times 5.59}{.85 \times 3.6 \times 78} = 1.4 \text{ in.} \quad a/2 = 0.7 \text{ in.}$$

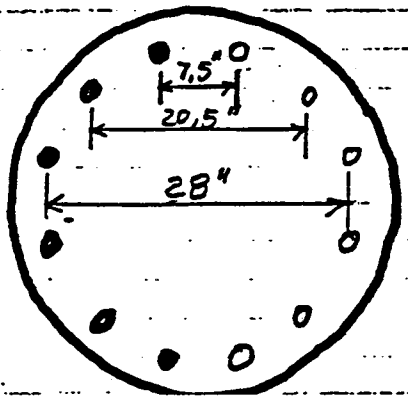
$$M_w = .9 \times 60 \times 5.59 \left(14.5 - .7\right) = 4166 \text{ k-in}$$

$$M_w = \frac{347 \text{ k-ft}}{6.5 \text{ ft}} = 53.4 \frac{\text{k-ft}}{\text{ft}} = M_w$$

DRILLED PIER M_c

36 in. Diam.

12 = #10 bars $A_s = 1.27$ in.^{2}



$M_c = 2 \times 1.27 \times 60 \times 7.5$ in. +
$2 \times 1.27 \times 60 \times 20.5$ in. +
$2 \times 1.27 \times 60 \times 28$ in. $= 711.4$ k-ft

$$M_c = 71.2 \frac{\text{kip-ft}}{\text{ft}}$$

LOAD CAPACITY – CONCRETE WALL

$\ell = 5$ ft $H = 8$ ft $\ell/2 = 2.5$ ft

$$L = 2.5 \text{ ft} + \sqrt{(2.5)^2 + 8 \times 8\left(\frac{173.7 + 347}{71.2}\right)}$$

$$L = 2.5 + 21.8 = 24.3 \text{ ft}$$

$$(w\ell) = \frac{8 \times 173.7}{24.3 - 2.5} + \frac{8 \times 347}{21.8} + \frac{71.2 \times 24.3^2}{8 \times 21.8}$$

$$= 63.7 + 127.3 + 241.0$$

$$(w\ell) = 432 \text{ KIPS}$$

DESIGN OF SPIRAL STEEL IN 36" DIAM. PIER

$M_c = 711.4$ k-ft

$P_p = \frac{M_c}{H} = \frac{711.4 \text{ k-ft}}{8 \text{ ft}} = 89$ kips

Shear per Pier $V_{tot.} = 89$ kips $= V_c + V_s$

$V_c = \phi \sqrt{f_c'}$ Area $= .85 \times 60 \times 36^2 \times .785 = 52$ kips

$V_s = 37$ kips

Stirrup Design $V_s = A_s f_y \frac{d}{s}$

$s = \frac{A_s f_y \, d}{V_s}$ Spiral Steel as stirrups

try (3/8" diam.) $A = .11$ in.^{2} $A_v = .22$ in.^{2}

$s = \frac{.22 \times 60 \times 30}{37} = 10.75"$ (use 6" c-c)

$f_c' = 3{,}600$

Development Length of Bars $f_y = 60{,}000$

$\ell_d = .0004 \, d_y \, f_y = 16 d$

$\ell_d = .04 A_b \frac{f_y}{\sqrt{f_c'}} = 29.2 A_b$

		use	
#7	17.5 in	18"	#7
#8	23.1 in.	24"	#8
#9	29.2 in	30"	#9
#10	37.1 in	38"	#10

FOUNDATION DESIGN

$P_{dy} = P_{st}(1 + JV)$ $\quad J = 0.14 \frac{sec}{ft}$ $\quad V = 5.9$ ft/sec

$P_{dy} = P_{st}(1 + .14 \times 5.9)$

$P_{dy} = 1.83\ P_{st}$

$P_{st} = \frac{P_u}{1.83} = \frac{89 \text{ kips}}{1.83} =$ 48 kips $= P_{st}$

COHESIVE SOIL $\quad q_u = 1.5$ tsf $\quad C = 1.5$ ksf

$H = 8$ ft $\quad B = 3$ ft $\quad \sigma = 13.5$ k/ft^2

$$D = \left(\frac{P_{st}}{\sigma B}\right) + \sqrt{2\left(\frac{P_{st}}{\sigma B}\right)^2 + 4H\left(\frac{P_{st}}{\sigma B}\right)}$$

$$= \frac{48}{13.5 \times 3} + \sqrt{2\left(\frac{48}{13.5 \times 3}\right)^2 + 4 \times 8\left(\frac{48}{13.5 \times 3}\right)}$$

$= 1.2 + 6.5 = 7.7$ ft $\quad$ use 8 ft Cohesive Soil

COHESIONLESS SOIL $\quad \gamma = 120$ pcf $\quad \phi = 40°$

$D^3 = \frac{2P(H+D)}{\gamma B K_p}$ $\quad K_p = \tan^2(45° + \phi/2)$

$K_p = 4.6$

$D^3 = \frac{2 \times 48(8 + D)}{.120 \times 3 \times 4.6}$ $\quad$ try $D = 10'$

$D^3 = 1044$

$D = 10.14'$ $\quad$ use 11 ft Cohesionless Soil