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TANK CAR HEAD PUNCTURE MECHANISMS

D. A. PETERS

B. A. SZABO

W. B. DIBOLL

SCHOOL OF ENGINEERING AND APPLIED SCIENCE

WASHINGTON UNIVERSITY

ST. LOUIS, MISSOURI 63130

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16. Abstract

In a number of railroad accidents the shell heads of hazardous material tank cars have been punctured. This report is concerned with the description and analysis of head puncture mechanisms. Three classification yard accidents and one main line accident were studied in detail, train-to-train collision tests were analyzed and the results of impact experiments were evaluated. The main conclusion of the report is that head puncture in classification yards is invariably due to overspeed impact. Such accidents can be prevented either by providing some fail-safe control for keeping impact speeds below 8 mph or, if impact speeds cannot be kept below 8 mph, not humping or flat switching more than one hazardous material tank car, or cars following it, onto any one track and requiring tank car heads to be designed or retrofitted to absorb a minimum amount of impact energy to be specified on the basis of further experiments and analytical studies recommended in this report. At the time of writing, experimental information is not available on the effectiveness of head shields and shelf couplers in preventing head puncture in main line accidents.

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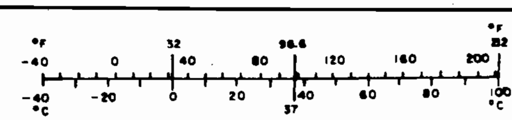
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

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

Approximate Conversions from Metric Measures

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



EXECUTIVE SUMMARY

Introduction

A relatively large number of railroad accidents involving hazardous material tank cars have been aggravated by loss of lading due to penetration of the tank car head by the coupler of an adjacent car. Washington University was awarded a contract under the Hazardous Material Tank Car Safety Program of the Federal Railroad Administration of March 1, 1974 to study the effectiveness of coupling systems, particularly those involving shelf couplers, in reducing the likelihood of head puncture in railroad accidents. Mr. Don Levine, acting chief, Rail Vehicle Safety Division, Federal Railroad Administration, has served as the contracting officer's technical representative. Dr. B. A. Szabo, A. P. Greensfelder Professor of Civil Engineering, served as the project director for Washington University.

Problem studied

The most important tasks under this contract were to acquire an understanding of the mechanisms of coupler override in accident situations and to determine the minimum amount of energy required for head puncture to occur. Obviously, these are necessary prerequisites to evaluating the effectiveness of alternative protective devices and operational procedures. The problem has been approached from two directions: A mathematical model was constructed to represent the dynamics of impact between railroad cars; and in-depth studies of four major railroad accidents, (those of East St. Louis [1972], Decatur [1974], Houston [1974] and Des Moines [1975]) were conducted. In addition, controlled non-destructive impact experiments were carried out early in 1975; destructive train-to-train impact experiments, executed under the auspices of the Transportation Systems Center in 1975, were evaluated and simulated switchyard collision experiments were carried out in collaboration with the RPI/AAR Railroad Tank Car Safety Research and Test Project.* The collision experiments, commonly referred to by their RPI/AAR designation as "Phase 15 Tests" are continuing beyond the termination date of this contract.

*RPI: Railway Progress Institute; AAR: Association of American Railroads. The test project was established in 1969.

Results achieved

Four different coupler override mechanisms have been identified and the mechanism which requires the minimum amount of impact energy for head puncture to occur was thoroughly investigated. In addition, a mathematical model, suitable for simulating train action in the vertical plane, was developed and validated. A number of static and dynamic parameters of railroad cars were measured directly or identified through a parameter identification scheme, described in this report.

Utilization of results

Certain operational guidelines and structural requirements can be established on the basis of findings presented in this report. These guidelines and requirements would serve to minimize the chances of head puncture due to coupler override in classification yards.

Conclusions

Head puncture occurrences in classification yards are usually caused by overspeed impact. If impact speeds could be reliably controlled to sufficiently slow speeds then other operational restrictions or special structural requirements would not be necessary.

Given that impact speeds are difficult to control with great reliability, head puncture occurrences in classification yards could be significantly reduced by imposing an operational restriction and a complementary structural performance requirement in lieu of a speed restriction. The proposed operational restriction is that not more than one hazardous material tank car, or cars following it, should be humped or flat switched onto any one track. The structural requirement is based on the observation that, with the single car switching restriction, although coupler override can occur above approximately 12 mph, the energy available to puncture the tank car head is limited by the weight and maximum velocity of runaway cars, and by energy losses in override mechanisms. Consequently, it is possible to state a performance criterion to govern the design or retrofitting of tank car heads in terms of a minimum required energy absorption capacity.

This report contains recommendations concerning the considerations and methodology that should be followed in determining the value of the minimum required energy absorption capacity and in developing standard acceptance test procedures.

Head puncture occurrences in main line accidents are usually caused by buff forces sufficiently high to induce plastic buckling in the underframe. The energy levels in such accidents can be so high that no protective device of any kind is fully effective.

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1. INTRODUCTION

The results of a 26-month research project, concerned with the determination of causes of tank car head puncture in railroad accidents, and evaluation of measures proposed to prevent such accidents, are presented in this report. The project began on March 1, 1974 when contract DOT-OS-40106 was awarded to Washington University by the Department of Transportation to study criteria and technology for the development of guidelines for shelf couplers. The contract constituted part of the Hazardous Material Tank Car Safety Program of the Federal Railroad Administration, and was specifically concerned with the prevention of tank car head puncture caused by coupler override in high speed impact, derailment, and other emergency situations. At that time the available knowledge on the dynamics of high speed impact between railroad cars was very meager. The only previous work available to the investigators was a report by J.B. Raidt of Pullman-Standard(1) in which the results of a preliminary investigation of the horizontal-vertical motions of railroad cars in moderately high speed impact were reported. Reference 1 was subsequently published in abbreviated form(2).

The first goal of the present project was to acquire an understanding of coupler override mechanisms and to develop means to simulate the dynamics of coupler override mathematically. The mathematical simulation was deemed necessary to allow for generalization of test results and to enable investigators to simulate the motion of cars in specific impact situations.

Studies of four major accidents which involved penetration or tear of one or more tank car heads by the coupler of an adjacent car, observations of train-to-train impact experiments, and simulated switchyard impact tests led to the definition of four basic coupler override mechanisms. These are discussed in detail in this report. Of obvious and particular interest has been the override mechanism which requires the least amount of energy for causing puncture or tear in the head of a tank car. In fact, an important measure for the effectiveness of a protective device is the amount by which the minimum value of kinetic energy, necessary to induce loss of lading through destructive damage to the head, increases as a result of installation of the protective device.

Mathematical simulation of high speed impact between railroad cars does not present difficult or unusual problems. Relatively simple yet powerful mathematical models can be developed to represent the motion of the car body and the trucks in impact. It is quite difficult, however, to measure or estimate with sufficient accuracy the parameters that characterize the dynamic behavior of railroad cars. For this reason, a large amount of attention was devoted to the measurement of parameters, preliminary testing and correlation studies. Part of this work was presented in an earlier report[3], the results of a parameter identification study are presented in this report.

At the present it is possible to accurately simulate the motion of railroad cars in impact up to the point where inelastic buckling begins to dominate. Very little is known however on the post-buckling behavior of railroad cars; consequently, simulation in the post-buckling range is beyond the present state of the art. Thus, the mathematical model is useful for determining whether conditions favorable for coupler override to occur exist in a given situation. Engineering judgement must then be applied to evaluate possible modes of structural response to dynamic loads in the inelastic buckling range. The mathematical model, together with the results of full scale collision experiments, provides for a reasonably complete although definitely not deterministic understanding of head puncture mechanisms.

2. MATHEMATICAL MODEL

A major portion of the research effort in the tank car safety program was the development of an analytical model of train longitudinal and vertical dynamics. Such a model is necessary for determining the horizontal loads, vertical coupler loads, and coupler height offsets that occur in impact situations. These three quantities greatly influence the probability of coupler override in a given impact situation. Therefore, a quantitative estimate of their magnitude is the first step in determining where conditions favorable for coupler override exist.

2.1 Description of model

The mathematical train dynamics model used in the tank car program is based largely on the earlier model by Raidt[(1)]. However, as the work in the tank car program progressed and the computer predictions were compared with experimental data, several features, not included by Raidt, were incorporated into the model. These features provide for realistic simulation of certain dynamic effects, of importance in predicting certain types of override. A brief outline of the unique aspects of the Washington University model is given below.

The first feature to be discussed is the "longitudinal dynamics only" option of the program. In this option, the program can be run without the vertical or pitching degrees of freedom. The main advantage of this option is that, for impacts involving long strings of cars, the magnitude of horizontal coupler loads throughout the train can be calculated at a low cost. Then, if any particular segment of the train shows signs of possible override, the complete analysis can be applied to the portion of the train that is of interest.

A second change with respect to Raidt's program is the addition of lading dynamics. This feature allows the lading to interact with car body motions through springs, viscous dampers, and Coulomb friction. The model is also applicable to liquid lading for which sloshing dynamics can be an important consideration. The simulation of sloshing is discussed in Appendix A.

Another feature of the program is an accounting of truck horizontal motions separately from car body motions. In high speed impact the car body often moves independently of the trucks because the center plate may slide out of the bolster dish or because the car body may separate from the trucks at high pitch angles. The present computer model allows each truck to have a separate horizontal degree of freedom so that this motion can be modeled.

The present program also includes vertical friction forces in calculating truck vertical forces. The possibility of viscous and Coulomb friction is considered. In addition, car couplers are analyzed including vertical and horizontal coupler slack. The vertical coupler slack has an important effect on the pitching motion and the possibility of override.

Another important new feature of the analytical model is the capability to model sliding sill units and hydraulic end-of-car cushion units. This feature allows realistic modeling of the force transmitted through cars equipped with other than conventional draft gears. A description of the mathematical model for sliding sill motions and a comparison with experimental data is given in Appendix B.

A more complete description of the program is given in References 3, 4 and 5. Included in these references are the equations of motion, assumptions of the model and flow charts.

2.2 Comparisons with experimental data

The mathematical model described above has been verified experimentally through a number of correlations with actual switchyard impacts. This work motivated many of the improvements of the model. However, the correlation studies also point up certain deficiencies in the present version of the model.

The first practical application of the Washington University train action model was a a simulation of three documented switchyard impacts each of which resulted in puncture or tear in a tank car head. The three accidents were: East St. Louis 1972, Decatur 1974, and Houston 1974. The detailed findings are presented in Reference 4. A major conclusion was that coupler misalignments due to pitching motions caused by collisions between heavy cars and loose light cars can contribute to coupler override. Figure 1, taken from Reference 4, illustrates how the pitching of a loose car could result in override.

The second application of the model was a detailed correlation of the test data from controlled impact experiments conducted at the ramp facilities of Miner Enterprises in Chicago early in 1975. These tests were designed specifically to check the validity of the analytic model. A detailed account of the correlations can be found in Reference 3. A major conclusion of this study was that the truck degrees of freedom are very important in predicting car motions during high energy impacts. Figure 2 gives one of the typical comparisons of theory and test data for horizontal accelerations.

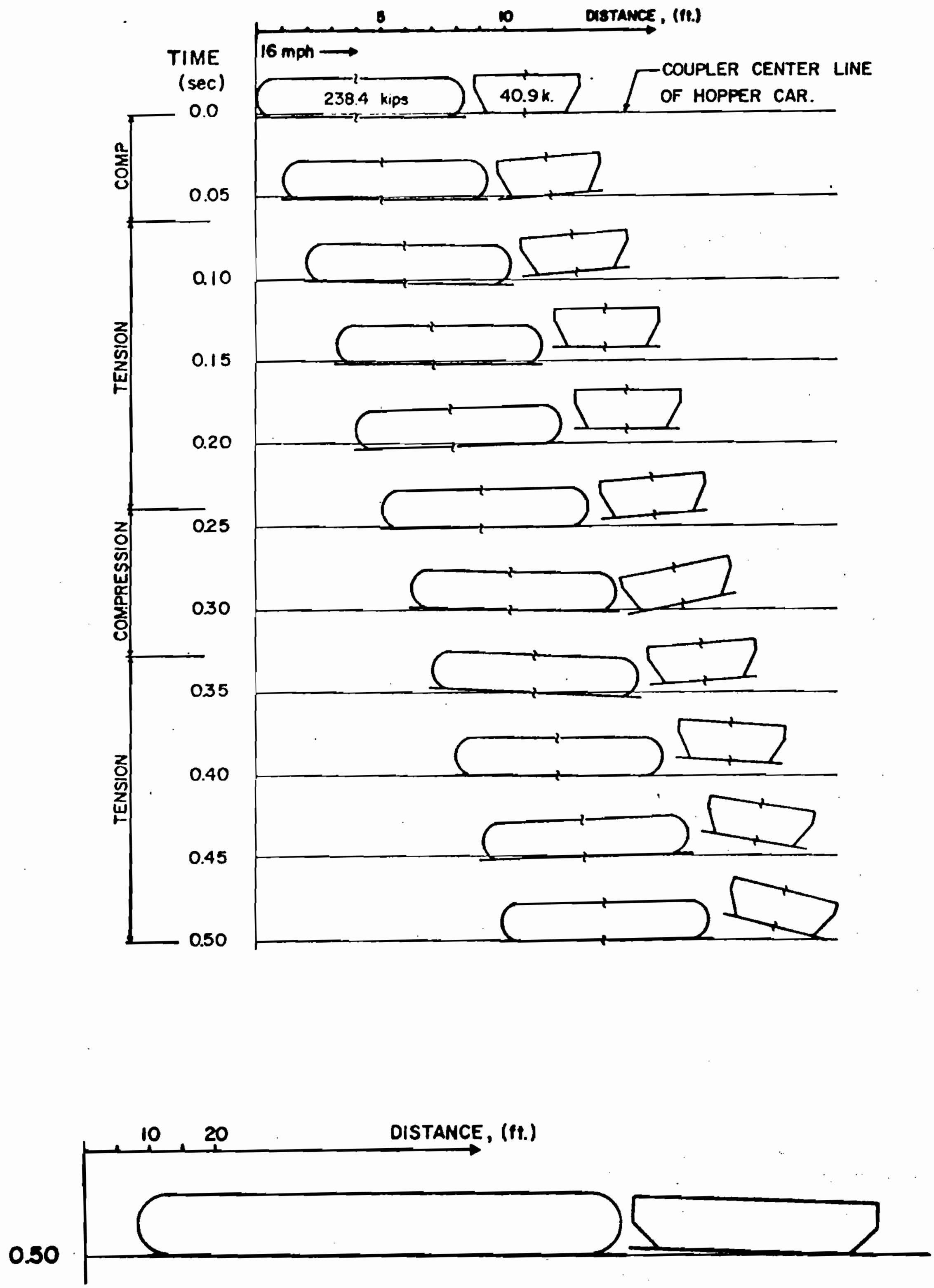

Figure 1 Impact of heavy tank car into free-standing hopper car

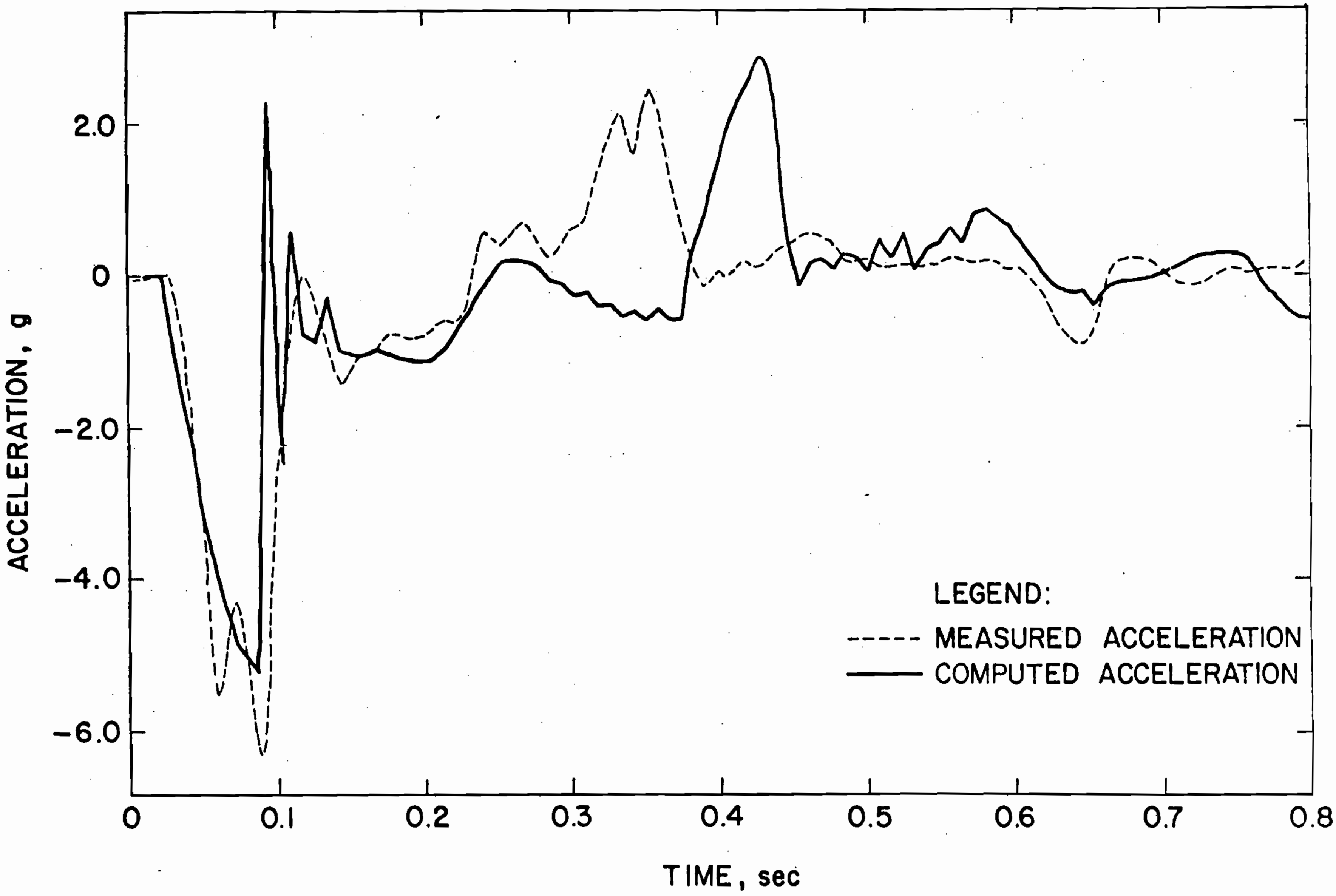

Figure 2

Correlation between experimentally measured and analytically predicted horizontal accelerations of a 31,500 pound hopper car, impacted at 6 mph and a loaded tank car, weighing 199,700 pounds.

The third comparison with test data is found in Reference 6. In that reference, computer predictions were compared with data obtained by the National Transportation Safety Board from the reenactment of a switchyard impact. The particular configuration in question consisted of 5 striking cars and 15 standing cars including many sliding sill cars and one end-of-cushion car. The comparison of theory and experimental data shows good correlation of maximum coupler forces.

2.3 Choice of system parameters

The mathematical model used in the rail car impact simulations is a lumped parameter system shown in Figure 3. The masses are those measured separately for car body and for trucks, and the spring rates are chosen to reproduce forces and motions in tests of full scale cars. An important parameter is the longitudinal underframe spring rate because after bottoming of the draft gears, the stiffer longitudinal spring of the center sill (underframe spring) controls the magnitude of the impact force.

In every test analyzed, the comparison of theory and experiment revealed that not enough was known about the representative values of the lumped parameters that should be used in the analysis. Very little data was available and even that did not agree with the experimental dynamics. Therefore, each test analysis began with an attempt to identify the numerical values of the most important physical parameters.

2.3.1 Low speed impact tests

The need to know the spring rates described above plus the need to have controlled impact tests to improve the simulation necessitated impact testing of a loaded tank car into a small empty hopper car at low speeds. These were conducted at the test facility of Miner Enterprises, Inc. The details of these tests and their results were published in Reference 3. The maximum impact velocity was limited to 8 mph because at this speed the truck bolster lifted off of the truck springs. Any higher speed could have resulted in detrucking.

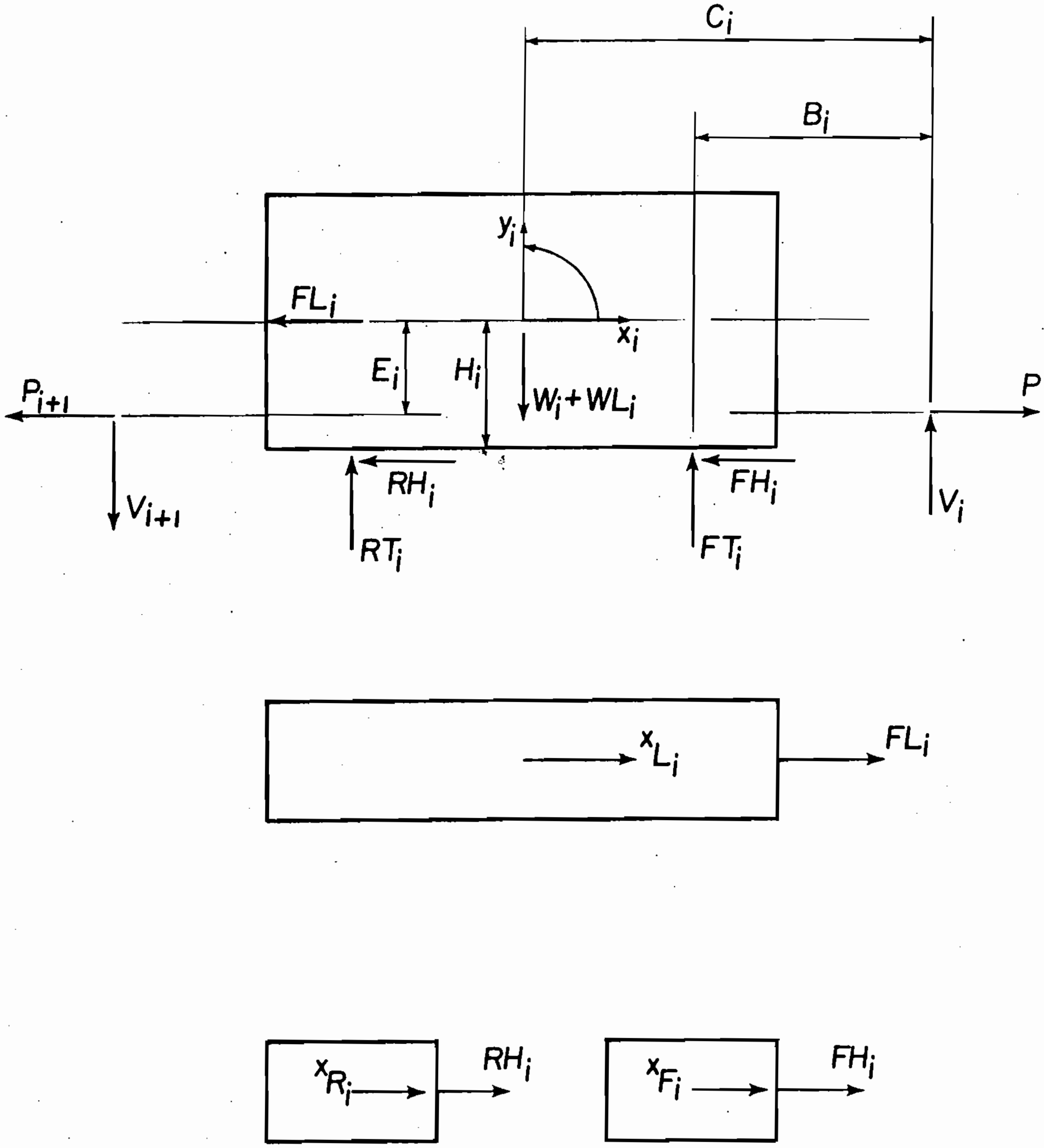

Figure 3 Mathematical model lumped parameters

The test results have shown that the static longitudinal underframe spring rate was approximately 2 times as great as the rate under dynamic conditions. These dynamic spring rates were determined by a parameter identification.

2.3.2 High Speed Impacts

As a preliminary sequence to the tests described in Reference 3, identical hopper cars were impacted. It was felt that the symmetry of the situation would allow for good estimation of the hopper car parameters. Table 1 gives the known hopper car parameters going into the test. The simulation results, presented in Table 2, show the rough trial and error method used to obtain representative values of the underframe spring rate of 3-6 mph. This same underframe spring was then used to simulate the same cars in a 6.5 mph impact. The results showed that the 1200 kips/in spring rate was valid over a range of velocities, Table 3.

A similar set of tests was performed with identical tank cars. Table 4 gives the known tank car parameters and Table 5 gives the trial and error identification method. The identified underframe spring rates were then used to simulate an impact between a tank car and hopper car. Table 6 gives a summary of the correlation. The identified parameter provided satisfactory correlation although fine tuning of parameters was not possible.

Table 1. Hopper to hopper tests (December 3, 1975)

Car data

1) Car weight 22.8 kips
2) Lading weight 0 kips
3) Truck weight 7.0 kips
4) Half distance C.P.F. 231 in.
5) Half distance truck centers 156 in.
6) Car center to center plate 50 in.
7) Car center to coupler 40 in.
8) Mass moment of inertia,900 kip-in-sec^2
9) Truck spring constant 48.6 kip/in
10) Bolster spring constant 750 kip/in
11) Truck spring travel 4.70 in.
12) Vertical coupler slack 1.2 in.
13) Vertical coupler spring constant 50 kip/in

Note: Couplers were prevented from engaging, draft gears were blocked

Table 2. Simulation results

	Simulation (V_0 = 3.6 mph)				Experiment
	1	2	3	4	
Max. Coupler Force (kip)	457	361	300	247	250
Max. Acceleration* (g)	13.5	17.5	14.5	11.4	10.0

*Longitudinal acceleration at underframe car center

Simulation 1: underframe spring constant = 2500 kips/in
trucks are assumed rigidly attached to the car body longitudinally.

Simulation 2: underframe spring constant = 2500 kips/in
no truck mass longitudinally

Simulation 3: underframe spring constant = 2000 kips/in
no truck mass longitudinally

Simulation 4: underframe spring constant = 1200 kips/in
no truck mass longitudinally

Table 3. Comparison, 6.5 mph

underframe spring constant = 1200 kips/in
no truck mass longitudinally

	Simulation	Test
Max. Coupler Force (kips)	447	400
Max. Acceleration (g)	20.6	Not clear from measured data

Table 4. <u>Tank car to tank car tests</u> (November 3, 1975)

<u>Car data</u>

1)	Car weight	85.3 kips
2)	Lading weight	177.0 kips
3)	Truck weight	10.0 kips
4)	Half distance C.P.F.	400.0 in.
5)	Half distance truck centers	314.0 in.
6)	Car center to center plate	66.0 in.
7)	Car center to coupler	56.0 in.
8)	Mass moment of inertia	31,000 kip-in-sec^2
9)	Truck spring constant	90.6 kip/in
10)	Bolster spring constant	1400.0 kip/in
11)	Truck spring travel	5.44 in.
12)	Vertical coupler slack	1.2 in.
13)	Vertical coupler spring constant	100 kip/in

Note: <u>Couplers were prevented from engaging, draft gears were blocked</u>.

Table 5. Simulation results

$V_0 = 5.1$ mph

Simulation Conditions

	1	2	3	4
Underframe spring constant (kip/in)	1200	1200	800	600
Truck Mass Included Longitudinally	No	Yes	Yes	Yes
Lading Sloshing Considered	No	Yes	Yes	Yes

Comparisons

$V_0 = 5.1$ mph

Simulation No.	1	2	3	4	Test
Max. Coupler Force (kip)	1675	1238	1036	908	900
Max. Acceleration (g)	6.8	8.8	7.1	6.4	6.0

Table 6. Tank car to hopper car tests (December 5, 1975

(same car data were used)

V_0 = 5.3 mph, coupler not coupled

sloshing effect included
Truck masses are included longitudinally
The draft gears are installed.

	Tank	Hopper
Draft gear spring stiffness	86.0 kip/in	86.0 kip/in
Hysteresis load	40 kip	40 kip
Draft gear travel	2.5 in.	2.5 in.
Underframe spring stiffness	600 kip/in	1200 kip/in

	Simulation	Test
Max. Coupler Force (kip)	157	180
Max. Acceleration (g)	4.2	4.0

Same simulation condition for V_0 = 6.7 mph

	Simulation	Test
Max. Coupler Force (kip)	200	320
Max. Acceleration (g)	5.3	9.0

Comment. The sloshing effect which changes the shape, the velocity of the lading at the impact is very important.

2.3.3 Analytic parameter identification

The laborious job of choosing parameters by trial and error led to the application of formal parameter identification techniques.(7) These were applied to a series of thought experiments to determine if the methods were feasible for railcar impacts. In other words, typical values for railcar parameters were chosen and computer results generated. These were then treated as "experimental" data and fed into the parameter identification program. The parameter identification program was given only the "experimental" data and form of the math model. The model was not fed any information about the parameters themselves other than an initial guess.

The analysis showed that when the first guess was within 10% of the actual value convergence was rapid. When the guess was chosen 50% away from actual convergence was greatly slowed. For 100% discrepancies in guessing, no convergence was obtained. Thus, it may be possible to use parameter identification for railcar problems provided that sound engineering judgement is used to obtain fairly accurate initial estimates.

2.4 Limitations of model

Extensive experimental verification of the train action model has provided an understanding of the limitations as well as the strengths of the computer simulation. The most serious limitation of the model is that it includes only elastic deformations. Plastic deformation and, in particular, the development of plastic hinges, is not explicitly included. Thus, the program can predict when loads will reach the yield level, but it can not directly predict the plastic mechanisms that may lead to override. On the other hand, if the location of plastic hinges can be determined by using the loads from the program, then the program can be modified to include post buckling behavior, Appendix C.

A second limitation of the model is that all car body mass is assumed to be concentrated at one point. It follows that the analytical underframe spring rate is the same for both static and dynamic squeeze. In reality, however, test data show that the dynamic spring is generally much softer than the static one due to elastic deformations in the car.

Thus, the train action model must be adjusted for dynamic squeeze. A more accurate modeling procedure would be to include an additional elastic longitudinal degree of freedom. This additional degree of freedom, chosen on the basis of structural modeling, could account for the relationship between velocity and spring rate.

A third limitation of the model is the uncertainty with which the vertical spring rate is known. The vertical spring rate is an idealization of the vertical flexibility between couplers and the car center of mass which results from coupler and sill bending deformations. A direct physical measurement of this quantity is very difficult and depends upon the coupler position in buff or draft. Further, the most important effect of vertical spring rate is its influence on coupler vertical forces. These vertical forces are also difficult to measure; in fact, no satisfactory experimental procedure has yet been devised for their measurement. Thus, this is an unknown parameter in the model.

A fourth area of potential improvement is the draft gear model. The behavior of draft gears in high speed impact is very complicated and not completely understood. Of special interest are the transition from static to sliding friction and the hysteresis behavior of the system. For example, experimental data in Reference 3 show that two adjacent draft gears often trade off energy dissipation from one to the other, which results in discrete steps in the draft gear response.

Finally, when lateral misalignments are present, the lateral or yaw degrees of freedom are necessary for predicting derailment or jackknifing. The program as it now exists can only treat symmetric impact in which only pitch, roll, horizontal, and vertical motions are allowed.

Improvement of the existing program in each of the above areas could be accomplished with present technology. Three steps would be required to do this. First, the mathematical model would have to be expanded to include additional degrees of freedom. In some areas, such as draft gear dynamics, this would require an independent research effort. Second, experimental data would have to be analyzed for dynamic situations that involve each of the added phenomena. Third, parameter identification studies would have to be performed to obtain the most likely parameter values from experimental data.

3. OVERRIDE MECHANISMS

The discussion of coupler override events is greatly simplified if we define four different override mechanisms. The mechanisms are characterized by the initial configuration of cars and, where appropriate, by the resulting override event. Actual override occurrences may be combinations of the four basic mechanisms described in the following.

3.1 The dynamic squeeze mechanism, illustrated in Figure 4, is caused by sustained buff forces of about one million pounds. The buff forces and bending moments cause the formation of one or more plastic hinges in the coupler shanks or the underframe structure, and cause the structure to buckle inelastically. The location of the plastic hinges and consequently the buckling mode are governed by the structural characteristics of the underframe and the distribution and magnitude of bending moments. The bending moments result from dynamic action and eccentricities in the horizontal load path. Two sources of eccentricity are vertical coupler misalignment and pitching oscillations. A further potential source of eccentricities in the horizontal load path is that, under heavy buff forces, the draft gears of mated couplers may bottom out such that one coupler transmits a large percentage of the horizontal force directly to the center sill through its horn in contact with striker plate, while the other coupler head does not contact the striker plate and the entire horizontal force is transmitted through the coupler shank to the draft gear lugs. This is schematically illustrated in Figure 5.

The dynamic squeeze override mechanism was directly observed in the train-to-train collision tests conducted at the Transportation Test Center, Pueblo, Colorado under the direction of the Transportation Systems Center in April and May of 1975. In an 18 mph collision between a locomotive, backed by three loaded hopper cars and a standing caboose, backed by four loaded hopper cars, a plastic hinge formed in the coupler of the caboose opposite to the impact end. The horn of this coupler did not hit the striker plate until after the onset of the plasticity. Hence the load path was through the coupler shank. The coupler was subjected to bending moments due to pitching of the caboose and the frictional

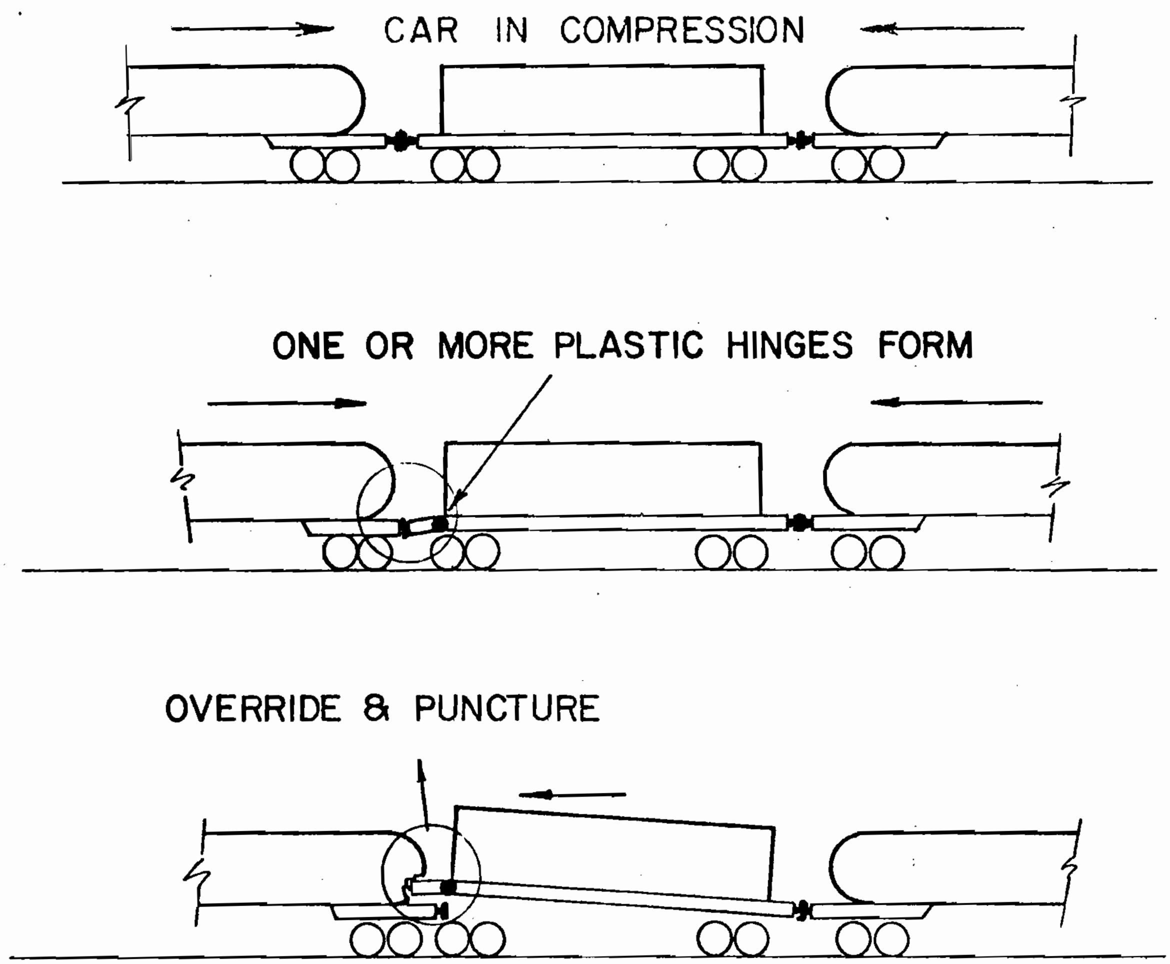

Figure 4 Dynamic squeeze mechanism

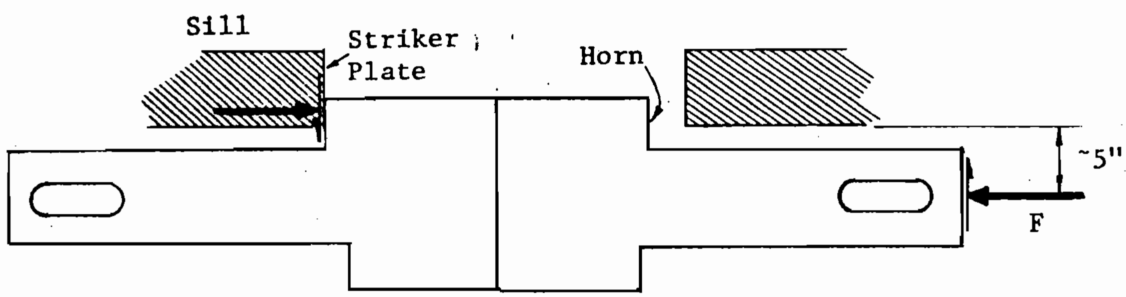

Figure 5

Eccentricity in the horizontal load path
due to "soft-stiff" draft gear combination

restraint imposed by the mated coupler. The plastic hinge allowed the coupler head to rotate and disengage from the mated coupler by sliding. In a 30 mph impact between a standing and moving consist, carried out under the same test program, both couplers of the caboose and their mating couplers hit horn to striker plate and a plastic hinge formed near the middle of the center sill of the caboose. An analysis of these events is given in Reference 8.

Coupler override by the dynamic squeeze mechanism also occurred in the Des Moines, Iowa accident in 1975. The most remarkable observation resulting from Washington University's investigation of this accident is that buckling of the stub sill of one of the loaded tank cars resulted in head puncture by another fully loaded tank car near the top of the tank shell. A detailed analysis of this accident is given in Appendix B.

3.2 The multiple impact override mechanism, illustrated in Figure 6, involves at least three impacts. The first impact is a collision between a light loose car and a heavy tank car. On this impact coupling does not occur and the light car is propelled away from the impacting car due to the elasticity of impact. It then impacts standing "back-up" cars and, assuming that the center of gravity of the light car is above the point of impact, it pitches such that the trailing coupler comes to an elevated position. If the conditions are just right, the incoming (tank) car collides with the light car again while the coupler of the light car is elevated. Override and head puncture can occur at this point.

This override mechanism was first identified in connection with the 1974 Houston accident[4]. The mechanism was successfully induced and photographically recorded in the Phase 15 test program.

It should be remarked that, for an override to occur by the triple impact mechanism just described, a rather narrowly defined relationship must exist among the masses of the impacting tank car and the (loose) light car, the natural pitching frequency of the light car, the impact velocity and the spacing between the light car and the back-up cars.

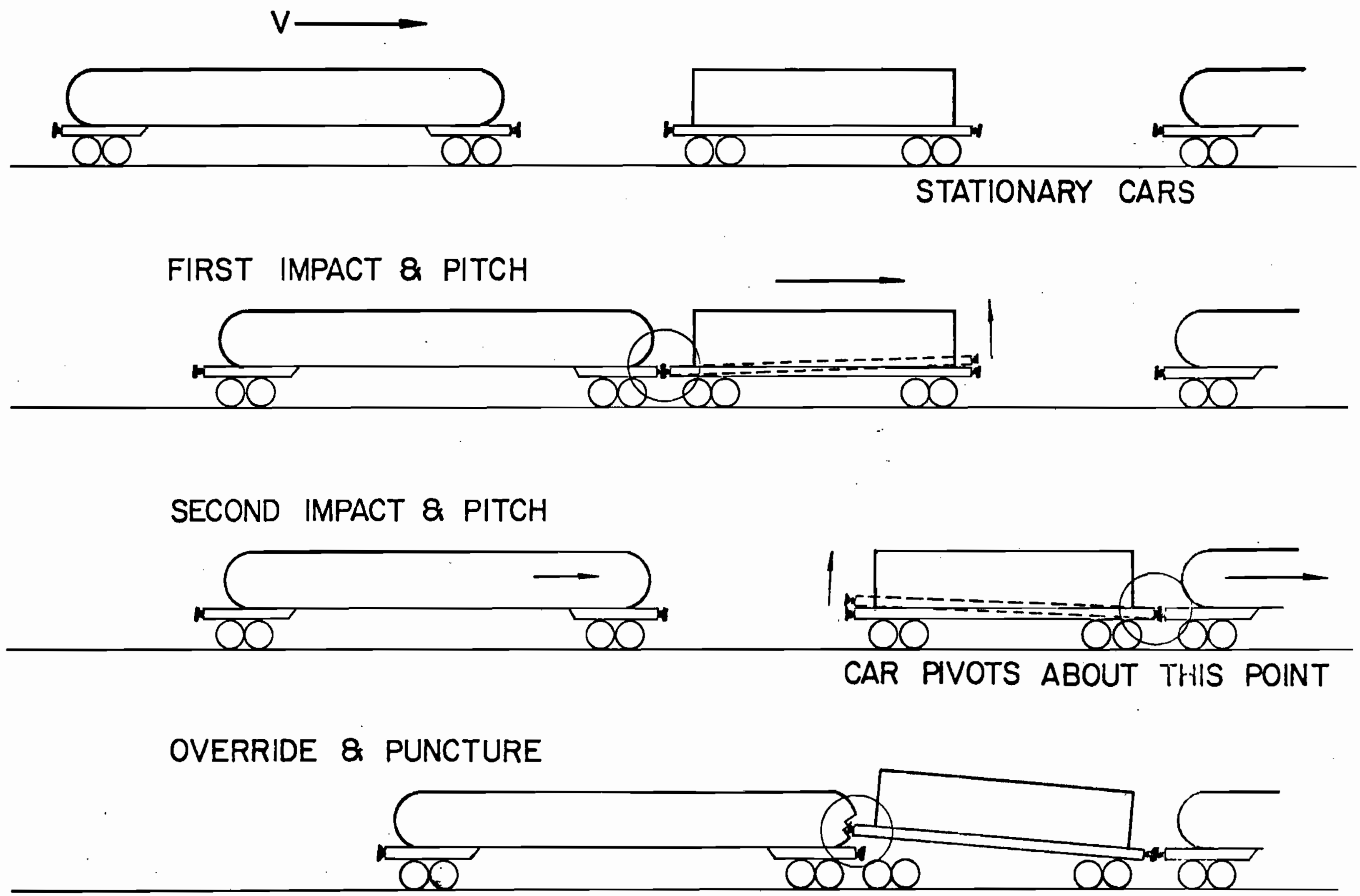

Figure 6 Triple impact override mechanism

The probability of critial combinations randomly occurring in classification yards is extremely low. A more probable event is that more than three impacts occur between the light car, the impacting car and the back-up cars. The longitudinal forces excite the pitching oscillation of the light car about its center of gravity, as explained in Reference 4, causing large vertical coupler misalignments. Override is then caused by dynamic squeeze, (a combination of buff forces and bending moments), when the light car is squeezed between the back-up cars and the impacting car(s).

3.3 The double impact override mechanism, illustrated in Figure 7, occurs when a loose light car is impacted by a heavy car, it pitches as shown in Figure 7, and its leading coupler overrides the coupler of a car situated a short distance away. This mechanism was induced and studied in great detail in the course of Phase 15 tests.

3.4 Coupler override involving detrucking is illustrated in Figure 8a. In this mechanism collision between a loose light car and a heavy (tank) car propels the car body of the light car forward and causes it to fall off its trucks as indicated. The trailing coupler of the light car is in an elevated position when the tank car impacts for the second time. A variant of this mechanism was first identified in connection with the Decatur accident of 1974[4]. Figures 8b and 8c show the derailment and resulting puncture at Decatur. In the Phase 15 tests this mechanism was not realized. It is the opinion of Washington University that the hoppers prevented the body of the light hopper cars from falling off their trucks.

4. THE PHASE 15 PROGRAM

A testing program designed to develop guidelines for various countermeasures to reduce the probability of punctures was initiated in late 1975. This program, designated as the "Phase 15 Full Scale Switchyard Impact Test Program," involves collision experiments between loaded tank cars and light hopper cars at various impact speeds.

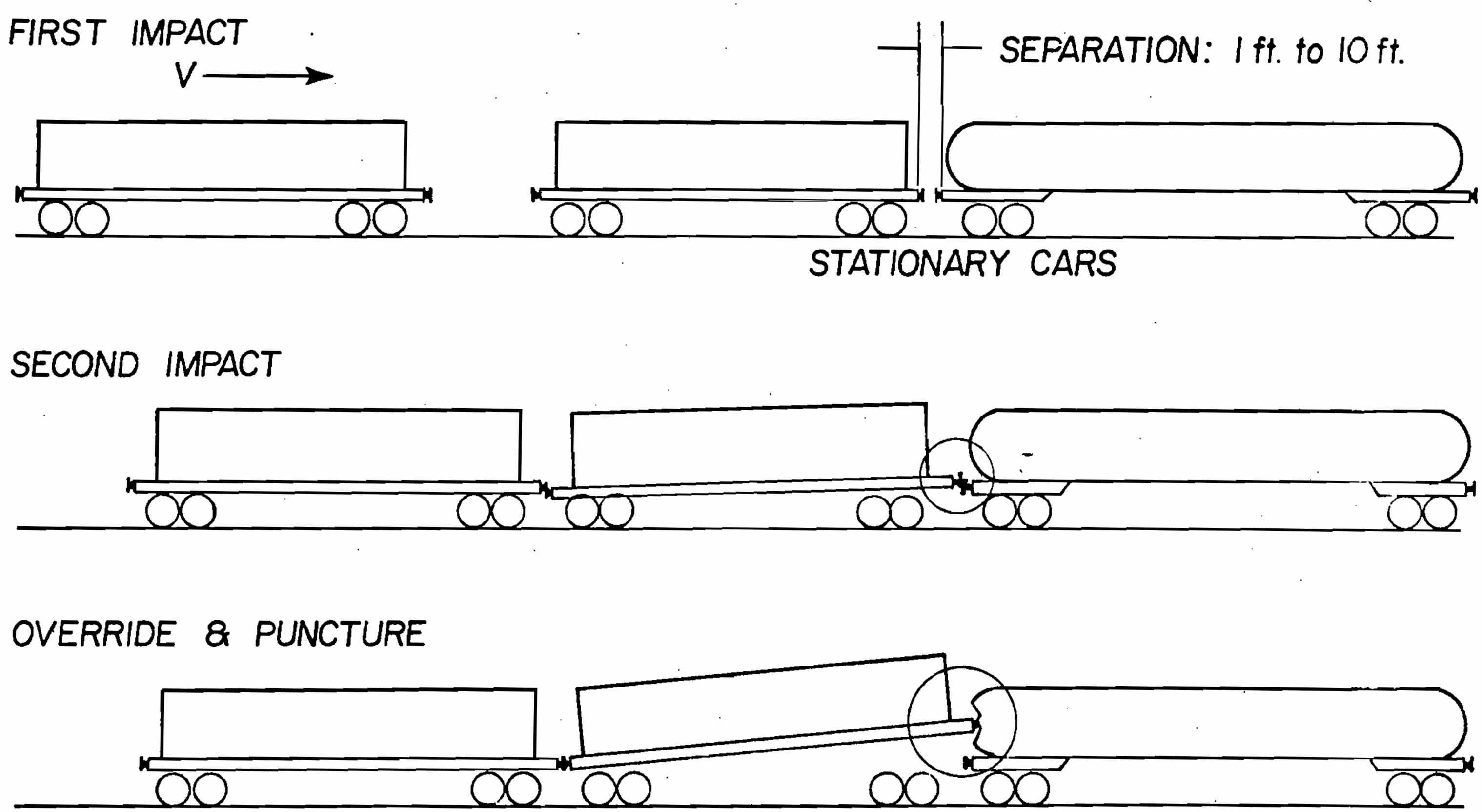

Figure 7 Double impact override mechanism

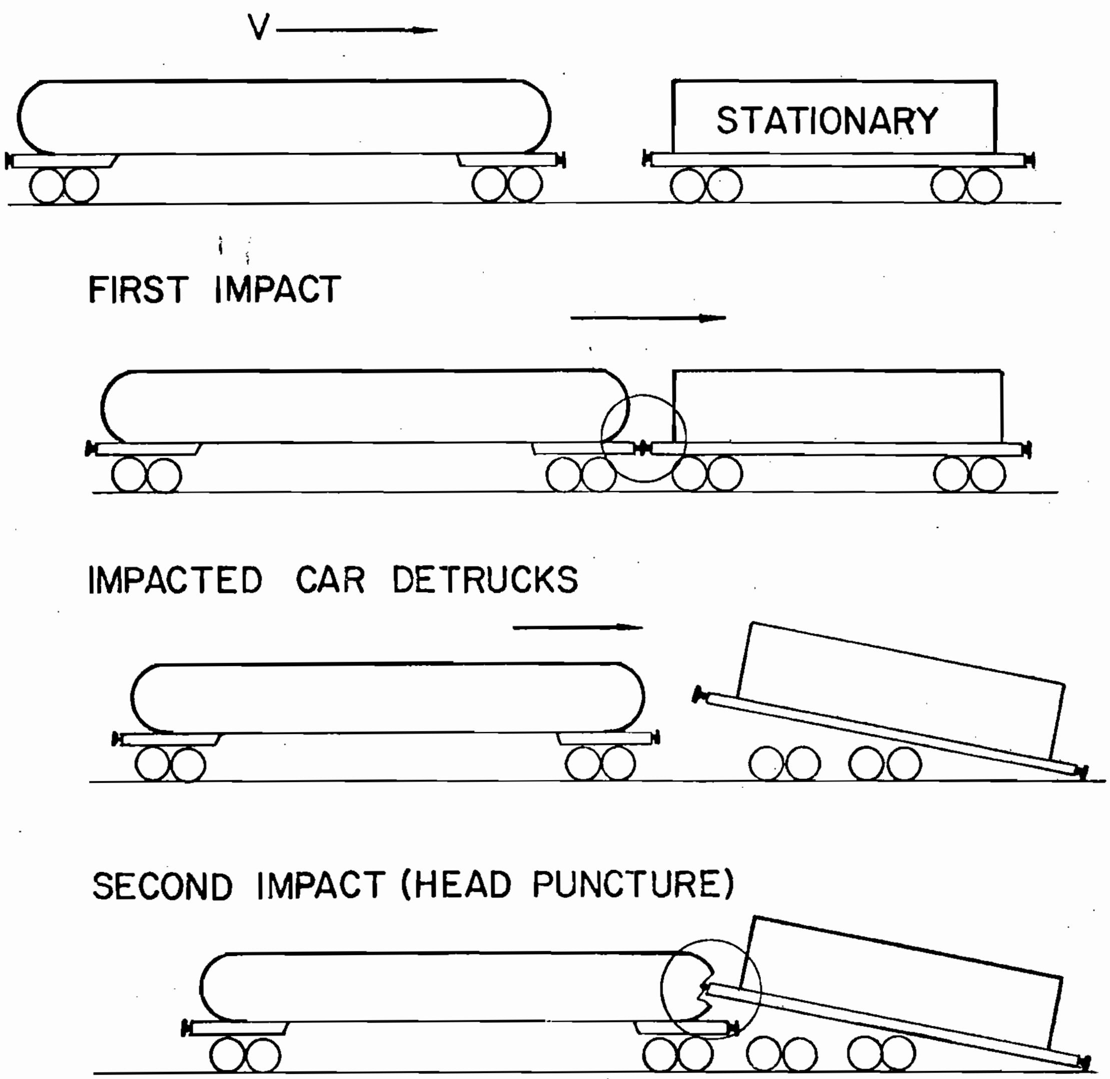

Figure 8a Detruck or derail override mechanism: theory

Figure 8b Derail mechanism: Decatur accident

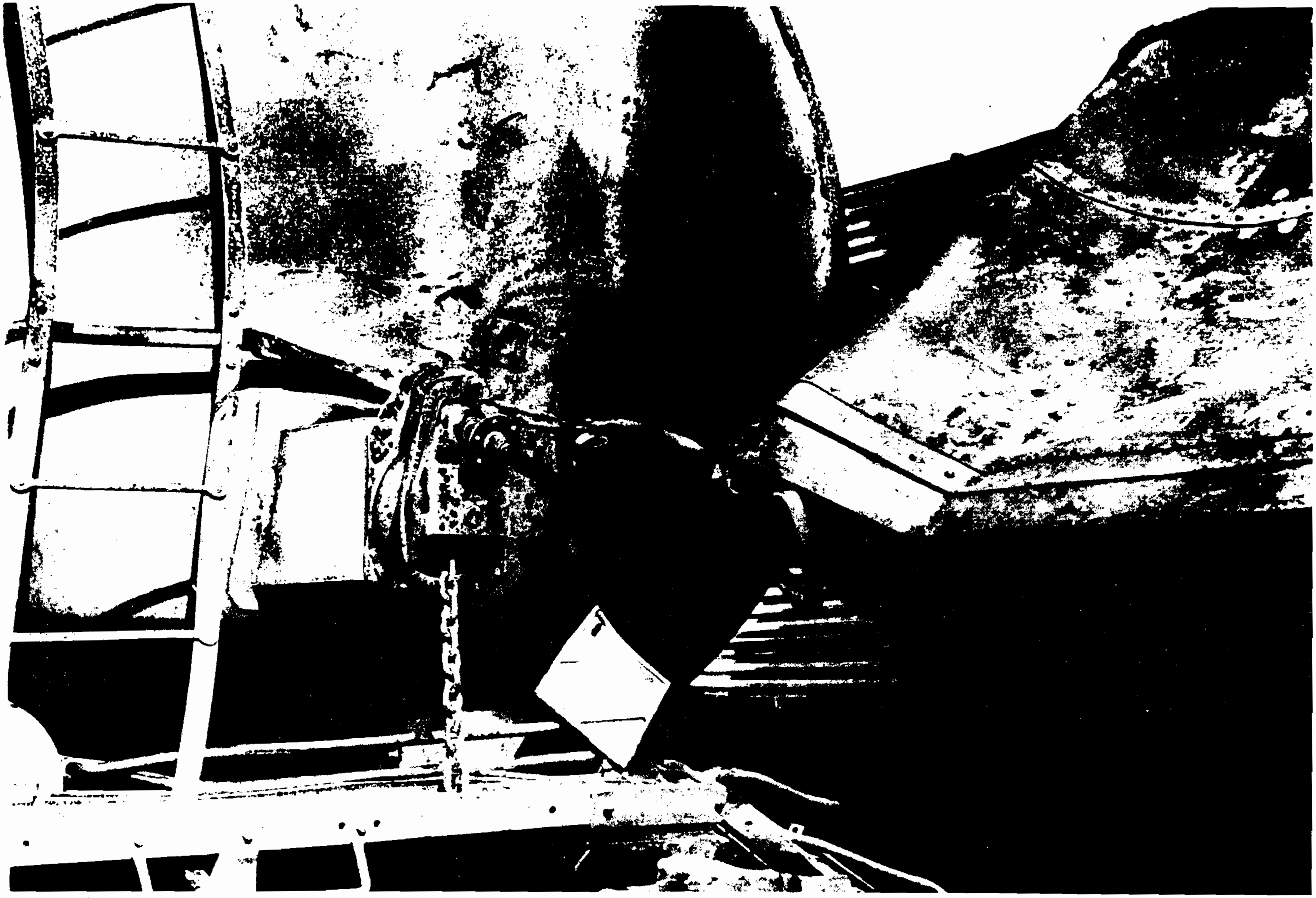

Figure 8c Puncture from derail mechanism

Washington University and the Transportation Systems Center, have provided technical direction for the tests, which were executed by Kentron Hawaii Ltd. at the Transportation Test Center (TTC) in Pueblo, Colorado. The RPI/AAR Tank Car Safety Committee made available all rolling stock for the tests.

4.1 Description of tests through May, 1976

A list of the collision experiments performed through May 1976 is given in Table 7. The tests have been designed to provide information about various override mechanisms and the minimum amount of energy required by each.

A major concern in these tests has been the proper modeling of the tank car lading, liquified petroleum gas products (LPG). No safe, noncombustible material having the same mass density as LPG has been found. Therefore, a difficulty exists if the mass and sloshing characteristics of a fully loaded tank car are to be simulated simultaneously. Although various schemes were proposed to fulfill both requirements in one system, none was found to be technically feasible.

Therefore, in Series 1 and 2, a compromise was made and the tank car mass was modeled by filling the cars to the maximum rail weight by water. This implied that the cars were only 62.7% full by volume. The Series 1 and 2 tests, however, uncovered several undesirable features of having only partially full cars. First of all, the low frequency of the liquid sloshing (period of 8 sec.) made it extremely difficult to control the impact speed. Often the tank cars would noticeably accelerate or decelerate from the time of release to the time of impact. Second, the analysis of sloshing dynamics in Appendix A shows that the entire mass of the water does not take part in the first impact. Thus, the tank car mass is not correctly simulated by a 62.7% full tank. Third, the tank car sloshing is significant in the timing of multiple impacts.

Thus, after the series 2 impacts were completed, some tests were repeated with a tank car 94.6% full of water. Although the car mass is considerably larger for this case than for an actual switchyard situation,

Table 7. Impacts through May 1976

Series	Impact	Date	Configuration	Speed (mph)	Remarks
1	1	Dec. 16	*Sloshing, no back-up	5.4	No coupling
1	2	Dec. 16	Sloshing, no back-up	5.9	No coupling
1	3	Dec. 16	Sloshing, no back-up	5.8	Coupling
1	4	Dec. 16	Sloshing, no back-up	7.0	No coupling
1	5	Dec. 16	Sloshing, no back-up	8.7	No coupling
1	6	Dec. 17	Sloshing, no back-up	10.4	No coupling
1	7	Dec. 17	Sloshing, no back-up	11.9	Coupling (minor damage)
1	8	Dec. 19	Sloshing, no back-up	11.8	No coupling (brakes set)
1	9	Dec. 20	Sloshing, no back-up	15.0	No coupling, detruck
1	10	Dec. 20	Sloshing, no back-up	16.8	No coupling, derail
2	1	Jan. 26	Sloshing, 3 ft. back-up	7.9	No coupling
2	2	Jan. 26	Sloshing, 4 ft. back-up	8.2	No coupling
2	3	Jan. 26	Sloshing, 5 ft. back-up	9.9	No coupling
2	4	Jan. 30	Sloshing, 7 ft. back-up	11.8	No coupling, override
1b	1	Feb. 26	†No sloshing, no back-up	9.7	No coupling
1b	2	Feb. 26	No sloshing, no back-up	11.6	No coupling
2b	1	March 2	No sloshing, 8.5 ft. back-up	11.5	Coupling
2b	2	March 3	No sloshing, 8.5 ft. back-up	11.8	No coupling
2b	3	March 4	No sloshing, 16 ft. back-up	13.7	No coupling, derail, back-up override
3	1	April 15	No sloshing, 2.5 ft. back-up	12.7	Back-up override
3	2	April 29	No sloshing, 3.5 ft. back-up	14.9	Back-up override
3	3	May 6	No sloshing, 3.5 ft. back-up	16.5	Override at each end
3	4	May 20	No sloshing, 2.5 ft. back-up	15.4	Override at each end
3	5	May 27	No sloshing, 2.5 ft. back-up	16.9	Puncture of back-up

*62.7% innage, †94.6% innage

the reduction of liquid sloshing provides for more repeatable tests. Furthermore, the ratio of tank car mass to hopper car mass is so large (in this case 5 to 1) that a change in car weight of 30% results in only a 10% change in collision momentum transfer. All subsequent tests are planned with the tank cars 94.6% full of water.

The test series listed in Table 7 represents a systematic search for the override mechanism in which the smallest fraction of the impact energy is dissipated. Prior the Phase 15 tests, only one override had been achieved under controlled conditions, Reference 8. The tests in Phase 15 explored the known types of override and provided added insight into override dynamics. Test series 3 resulted in a head puncture at 16.9 mph.

4.2 Series 1 and 2 tests

The Series 1 tests were performed with a two-thirds full tank car (maximum rail weight, 273,000 lb.) impacting an empty 55 ton hopper car (40,000 lb.). No back-up cars were used in these tests, and the brakes on both cars were applied by remote control after impact. There were several objectives in these tests. The first objective was to determine whether a detrucking or derailment type override and puncture could be obtained. The tests showed, however, that even at high speeds the hoppers interfered with the trucks; and this interference served to prevent detrucking. This can be seen in Figure 9. Were it not for the hoppers, the car body might have gone over the trucks and pitched up at the impact end, a condition conducive to puncture. Thus a light box car or light tank car which does not have hoppers might be a better choice for the loose car in simulating a derailment type puncture.

The second objective of series 1 was to obtain information on car motions following impact. This information could then be used in setting up the cars for multiple impacts. Three important observations were made. First, it was found that at impact speeds above 5 mph, the hammer and anvil cars often do not couple. The same phenomenon had been observed in the Miner tests[3], but at that time it was not known with

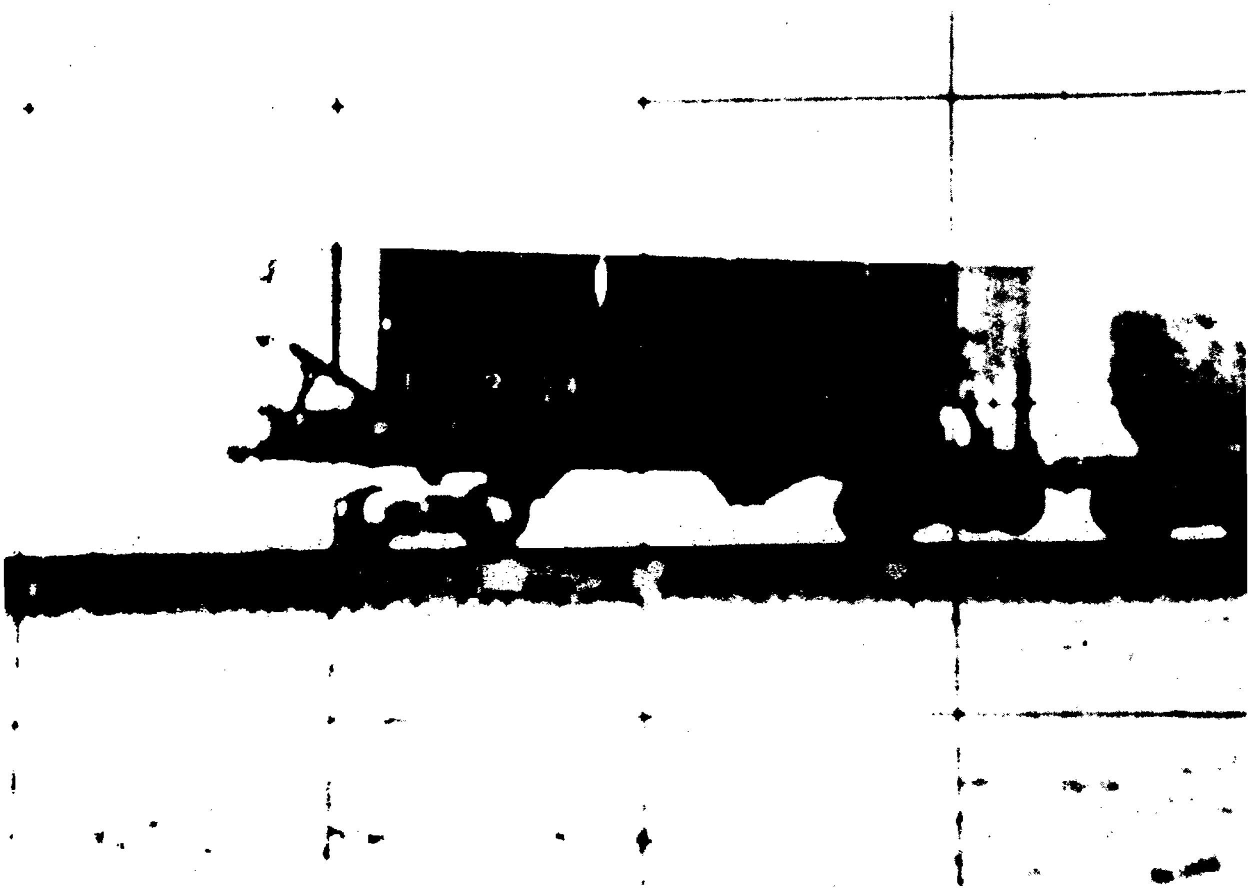

Figure 9 Hopper intereference and derailing, 16.8 mph

certainty that this is a common occurrence. Table 7 shows that only two of the 10 tests in Series 1 resulted in coupling. The general results from all Phase 15 tests indicate that failure to couple is a common occurrence when a loose car is present.

The second observation is also illustrated in Figure 9. At impact speeds above 15 mph the truck on the impact end of the standing car has a tendency to pitch forward. The pitching is a direct result of the interaction between bolster, bolster plate, and bolster pin. In some cases, the pitching can cause the rear wheels to leave the track. Sometimes they come down misaligned causing derailment. Of course, derailment occurrences cause considerable changes in override mechanisms as will be discussed later.

The third observation was that, at high speeds, the motion of the light car is qualitatively different from that at low speeds. At low speeds (less than 15 mph), the end opposite to impact pitches up and the impact end moves slightly down. This is followed by one or two pitching oscillations after which vertical motions are negligible. In the case of high speed impact (greater than 15 mph), however, both ends of the car may rise together and the car body remain relatively horizontal. The phenomenon of both ends rising off the trucks is a direct result of the bolster dynamics. When the bolster on the impact end pitches due to impact, it wedges in the side frame and cannot slide. Therefore the impact end, which originally is forced down by the impact, immediately rebounds up and joins the opposite end in an upward motion. The resulting moment impulse can cause the end opposite of impact to rise not as high as it would if the track bolster at the impact end did not wedge against the side frame.

In the Series 2 tests, the configuration of Series 1 was altered by the addition of back up cars several feet behind the loose hopper car. The back-up distance was chosen to maximize the possibility of the triple impact override. In other words, the distance was set so that the end of the hopper car opposite to impact would go up and then down, and strike the back-up car at the lowest position. This sequence results in the impact end of the hopper pitching up again just as the tank car strikes for the second time. Thus, the elevated hopper car coupler is in a good position to override the incoming tank car. The back-up distance was chosen on the basis of a careful evaluation of the Series 1 films in order to obtain the distance desired. Figure 10 shows the test data and the experimental curve giving the best spacing as a function of impact speed. It should be noted, however, that the spacing is only speculative for speeds above 15 mph where detrucking or derailment may occur. Detrucking or derailment can slow the cars and alter the pitch angle.

The first collision to result in coupler override occurred on January 30, 1976. The sequence of events is illustrated in Figures 11-14. Figure 11 shows the tank car impacting from the left, which causes the right end of the light hopper car to pitch up. Figure 12 shows the right end of the hopper car at the low position as it impacts the first back-up car. This second impact then caused the left end of the hopper to pitch up. Figure 13 shows the subsequent third impact (second tank car impact). Dust and metal can be seen coming from the coupler faces as they slide with respect to each other. Figure 14 shows the final position after the hopper car coupler has disengaged and overridden the tank car coupler. This was the first time that a predicted coupler override was experimentally verified. There was not enough energy remaining, however, to cause a puncture or tear in the tank.

The effects of sloshing in Series 1 and 2 led to an analytic study of the phenomenon. Appendix A provides a derivation of the sloshing model used in the analysis. The results show that even though the first frequency of sloshing is low (period - 8 sec.), the sloshing dynamics is critical because of the large mass of water involved. On the basis of this analysis and the test results, test series 1 and 2 were repeated using a fully loaded tank car (94.6% by volume, 344 kips) in which sloshing would be minimal. The tests without sloshing were designated 1b and 2b.

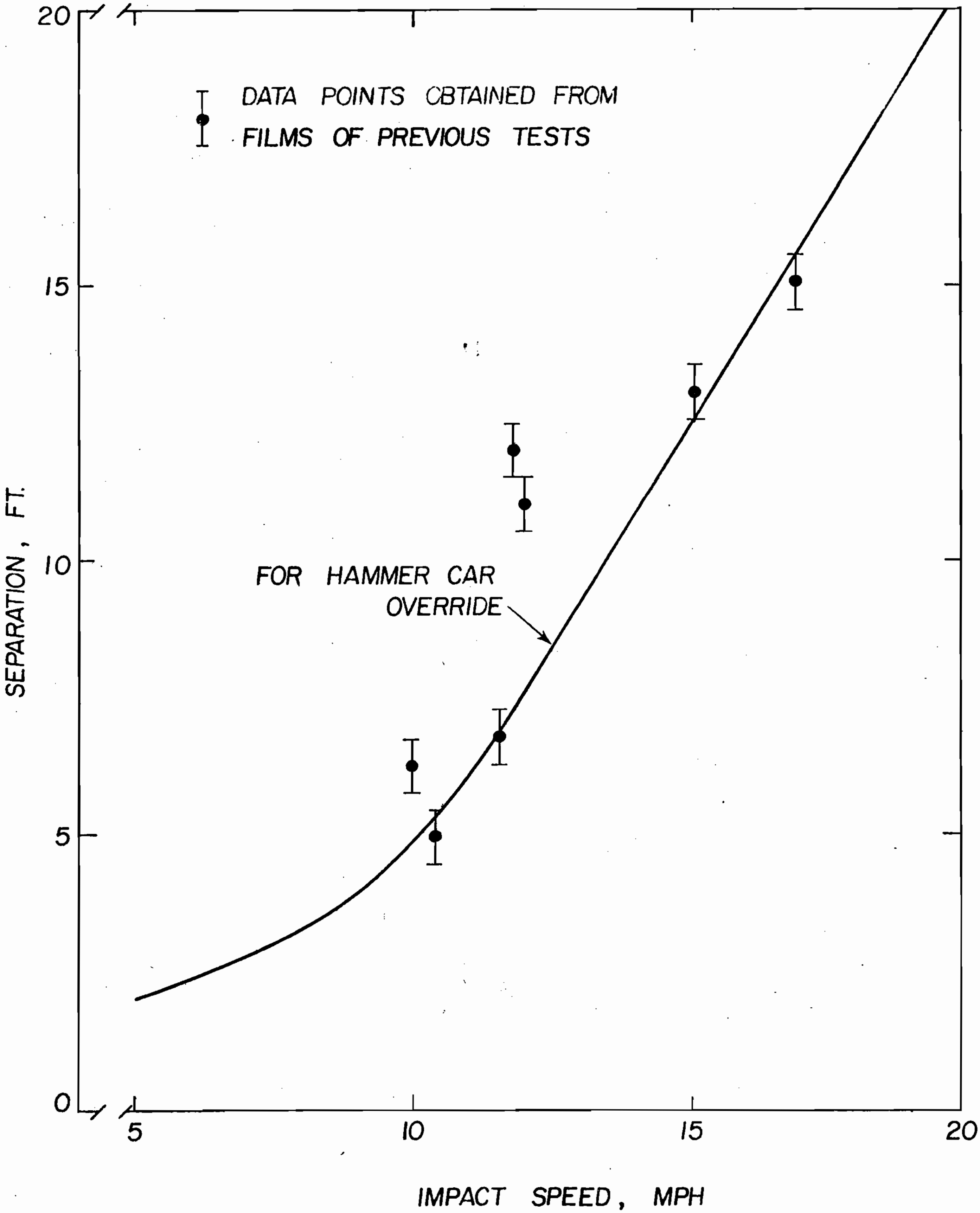

Figure 10 Car spacing for triple impact override

Figure 11 Triple impact mechanism: 11.8 mph first impact

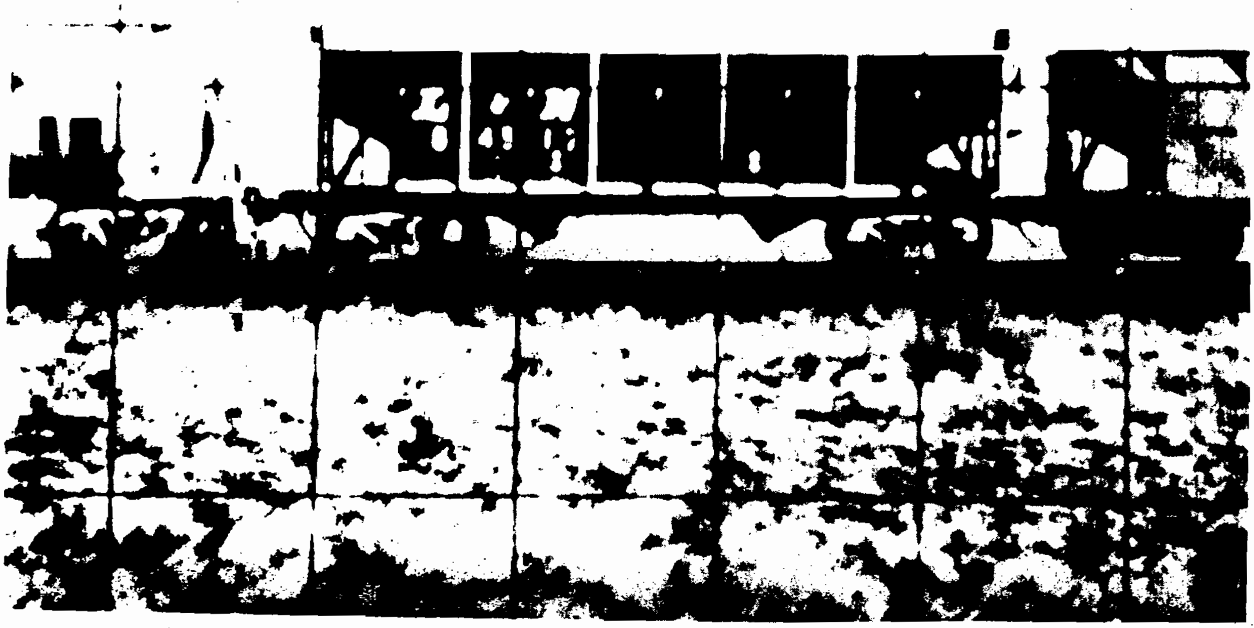

Figure 12 Triple impact mechanism: 11.8 mph second impact

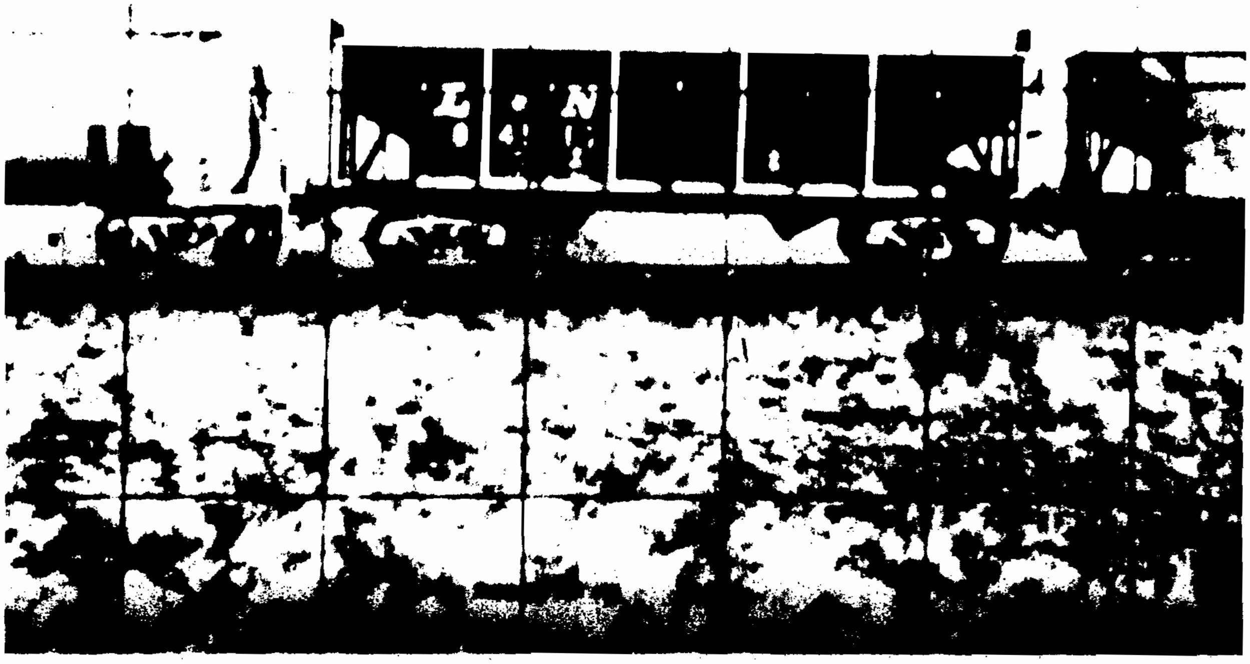

Figure 13 Triple impact mechanism: 11.8 mph third impact

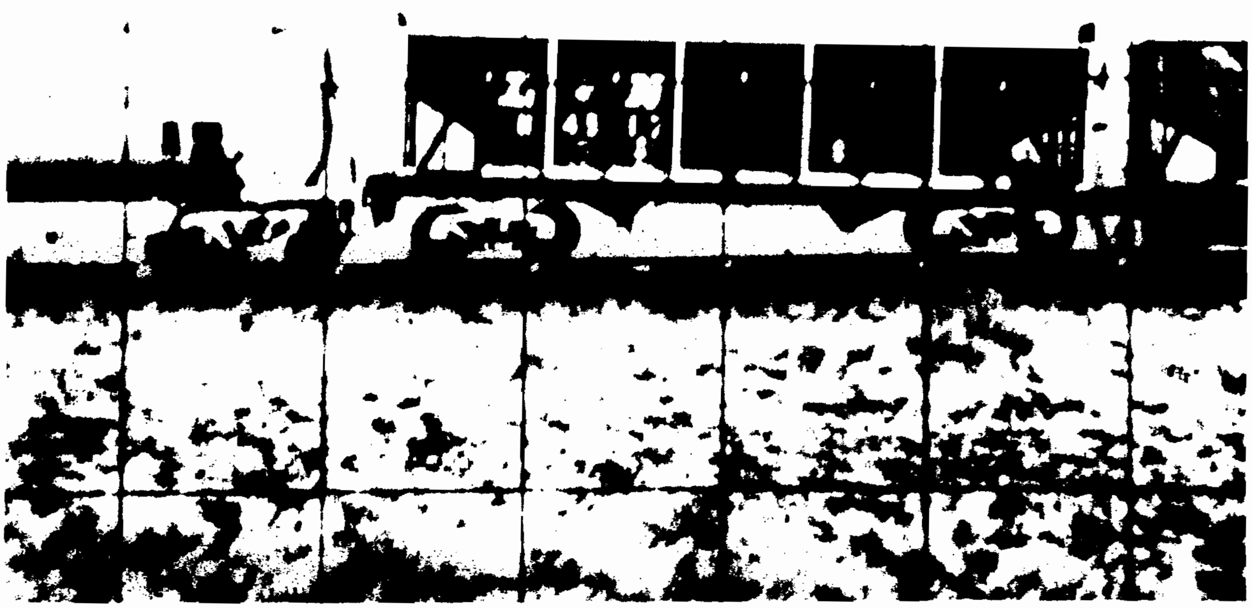

Figure 14 Triple impact mechanism: 11.8 mph override

Series 1b was performed without back-up cars in order to generate a curve, similar to Figure 10, that could be used to determine spacing. The results showed that the hopper car exit speed was 10% higher for a full tank than for the sloshing tank. Consequently, the most critical spacing for series 2b was taken to be approximately 10% longer than it was for series 2. The first two impacts in Series 2b, however, showed that the predicted spacings would not result in override. The reason for this is that in tests where sloshing was present the tank car was sufficiently slowed by the first impact for the hopper car to separate from it by about one foot before the second impact. This allowed sufficient time for the hopper coupler to rise before the tank car hit again. When sloshing was eliminated by filling the tank car, however, the tank car was not significantly slowed by the first impact. Therefore, it remained close to the outgoing hopper after impact. The slow separation between tank and hopper resulted in two phenomena. First, the chance of coupling was increased over the sloshing case. Second, at the moment of second impact (hopper into back-up) the cars were so close that the hopper did not have time to rise before the third impact (tank into hopper again). Therefore override was not obtained.

On the basis of these results, Washington University formulated recommendations to increase the possibility of override. Figure 15 shows curves of car rise time as a function of spacing at 11.7 mph for three of the experimental tests, one with sloshing (January 30) and two without. The rise time is the amount of time between the second and third impacts. The time is directly proportional to car separation speed and is necessary to obtain coupler eccentricity. For the tests with sloshing present, the tank and hopper cars separated rapidly allowing a longer rise time between the second and third impact. For example, the 7 ft. distance resulted in 0.1 sec. rise time. The tests without sloshing would have required a 16 ft. back-up distance to obtain the same rise time. Therefore, Washington University suggested that a longer spacing be used. Two major constraints exists in choosing this spacing. First, care must be taken to insure that the end opposite to the impact is not at a high point in its oscillation when it strikes the

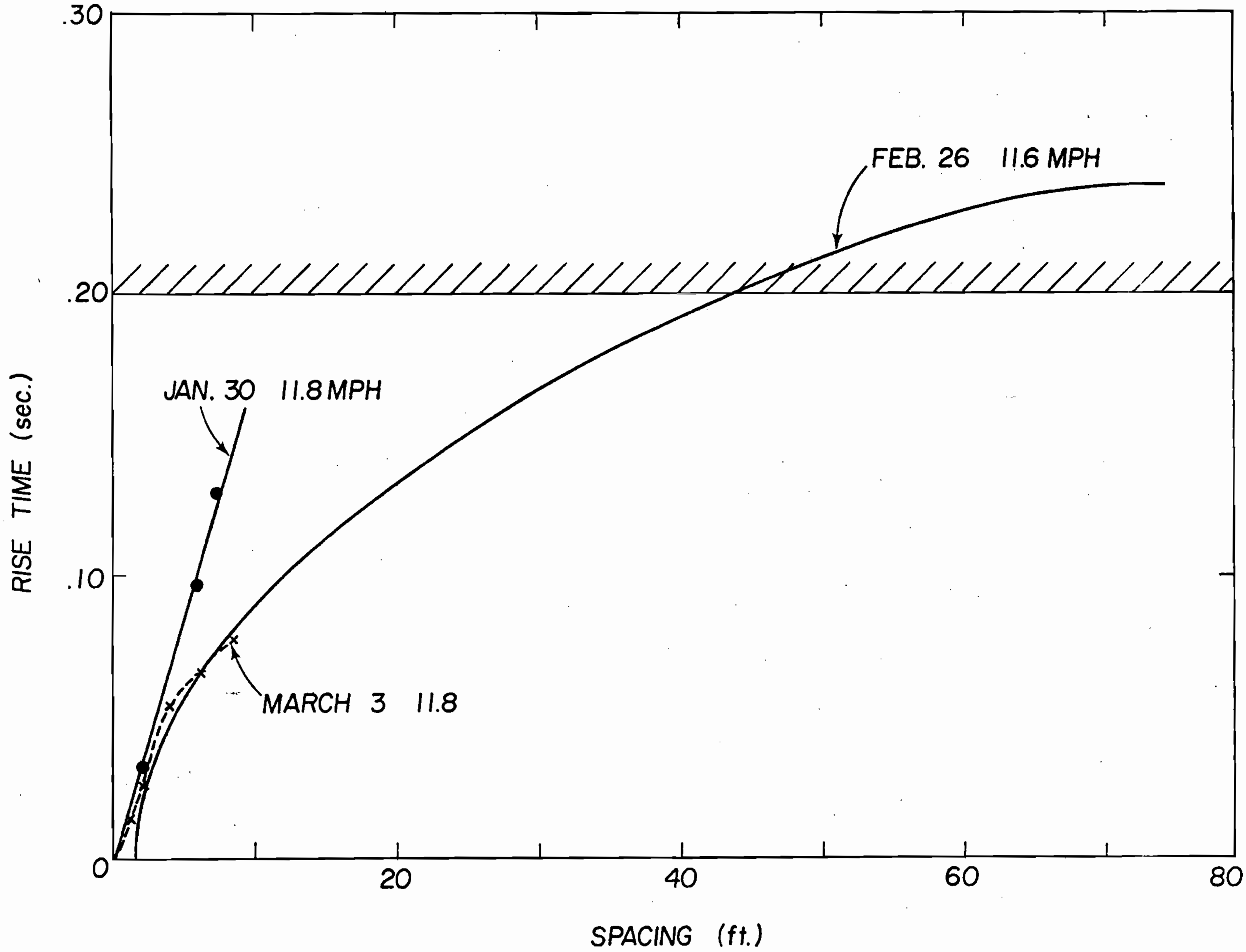

Figure 15 Rise time versus separation, 11.7 mph

back-up car. The critical distance for such an occurrence is 12 ft. for the 11.7 mph impact. Second, there is a maximum obtainable rise time for which the coupler is at its maximum elevation. If more "rise" time is given, the coupler will begin to drop. For 11.7 mph, the maximum possible rise time is .20 sec., which coorresponds to a 40 ft. back-up distance. Thus, the spacing could be chosen to be between 15 ft and 40 ft. Some intermediate spacing is probably best in order to allow for experimental variations and to give some vertical coupler velocity at the third impact. Although the Series 2b tests were terminated without trying the larger spacings, the occurrence of large spacing in switchyards has been discussed in Reference 4.

The final test of series 2b was run with the spacing for a single bounce. In this case, however, the hopper car derailed. The derailment slowed the car so that it experienced a complete pitching oscillation before the second impact. Figures 16-20 show the sequence of events. In Figure 16 the hopper car is pitched up due to the impact from the left, in Figure 17, it comes down, and in Figure 18 it bounces up again. Figure 19 shows that the second impact occurred when the coupler was still elevated from the first bounce. The resultant eccentricity caused the coupler to break, as shown in the figure. After the coupler broke, the light hopper override the back-up car, Figure 20. This was the first photographed override involving a broken coupler. A photo of the broken coupler is shown in Figure 21. The coupler was made of grade C steel.

4.3 Series 3 tests

The tests in Series 1-2 showed that the triple impact override mechanism is not very repeatable for impact speeds above 12 mph, because detrucking or derailment can occur. Therefore, the Series 3 tests were designed to investigate the more repeatable double impact mechanism.

Figure 22 presents the postulated spacing schedule, based on observations of earlier tests with E couplers, that most probably results in maximum coupler height at the second impact. As seen from the figure, a speed of 14 mph is probably required to completely clear a standard E coupler, and a speed of 16 mph is required to clear an E shelf coupler.

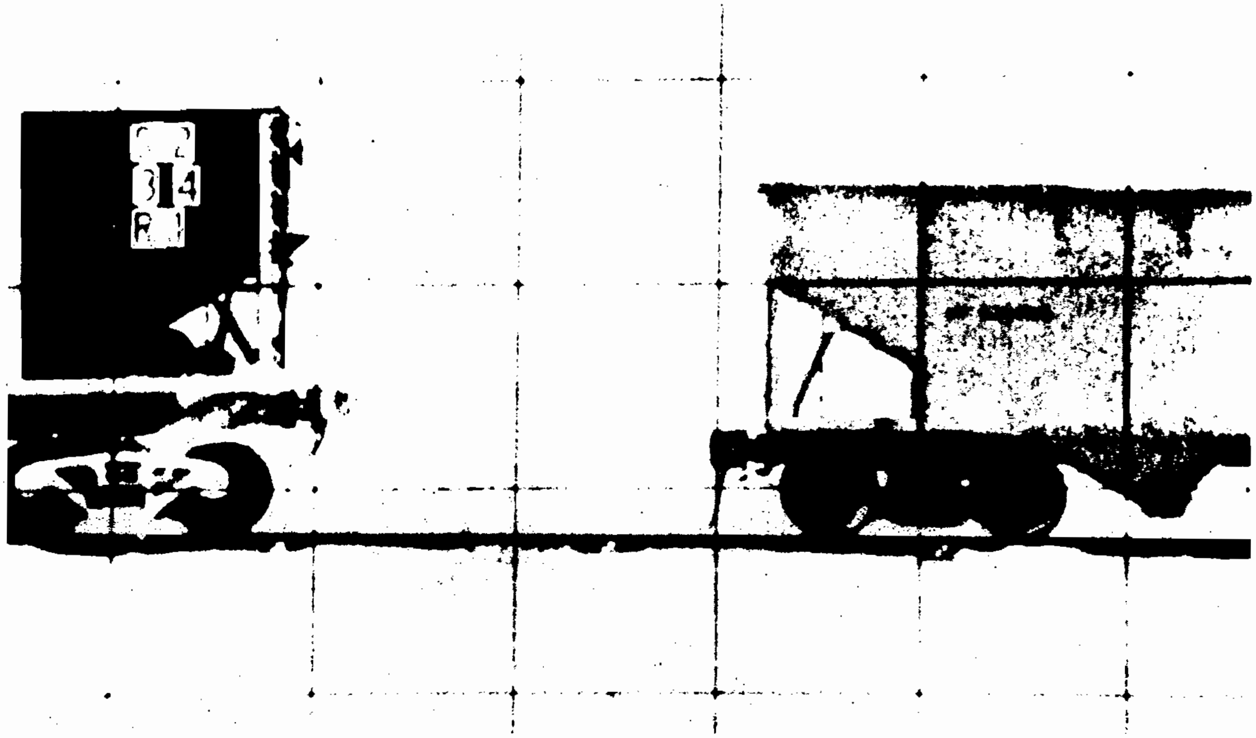

Figure 16 First rise, 13.7 mph

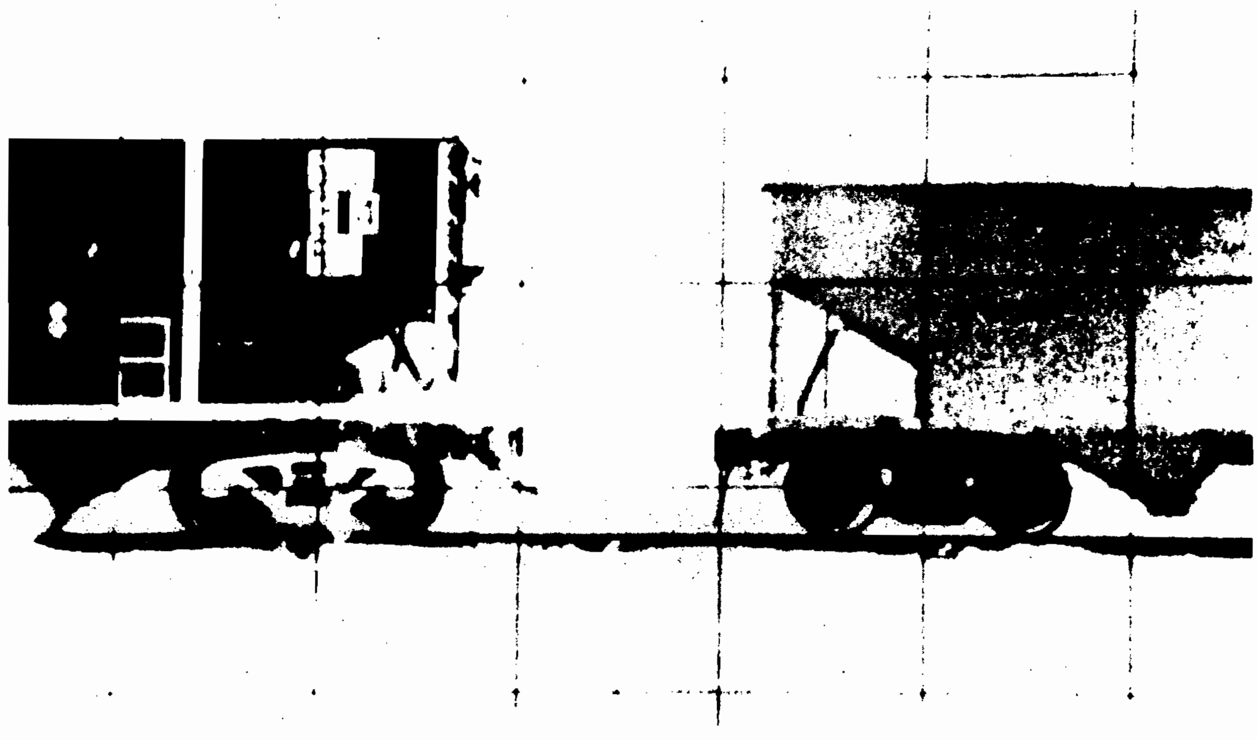

Figure 17 First fall, 13.7 mph

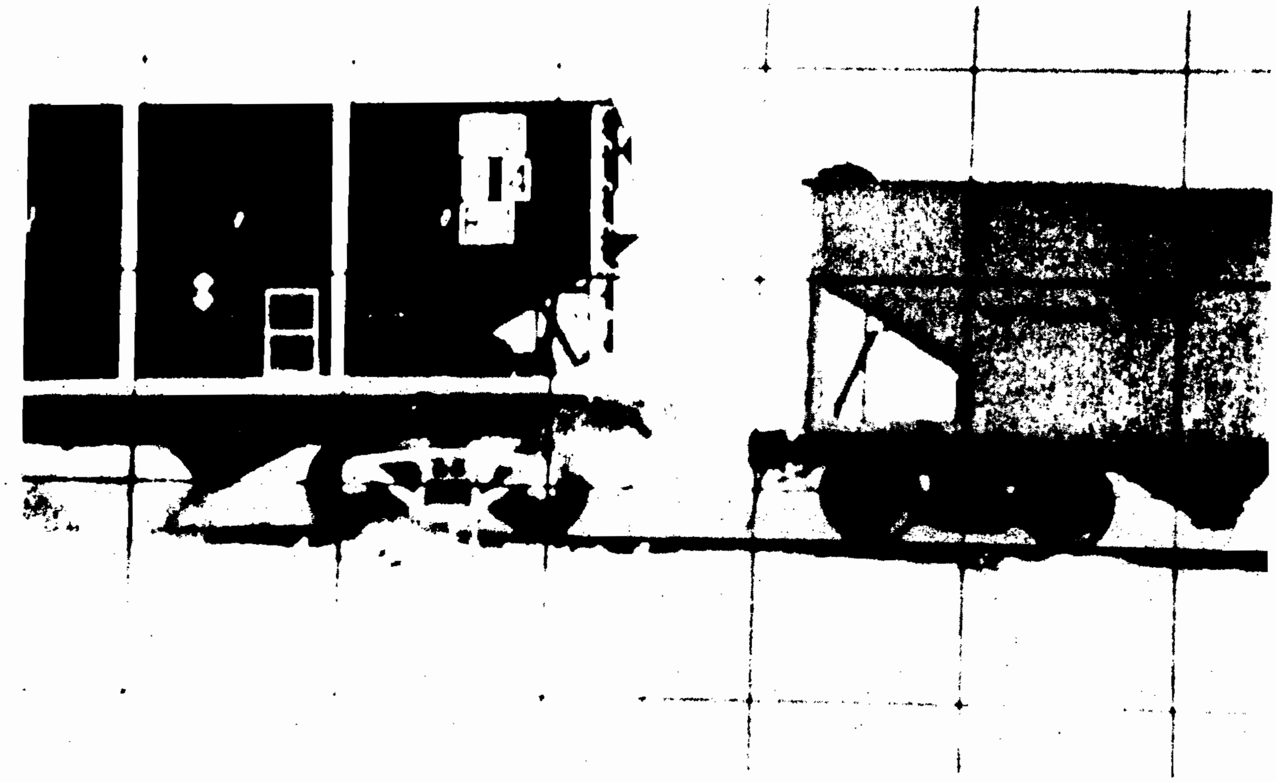

Figure 18 Second rise, 13.7 mph

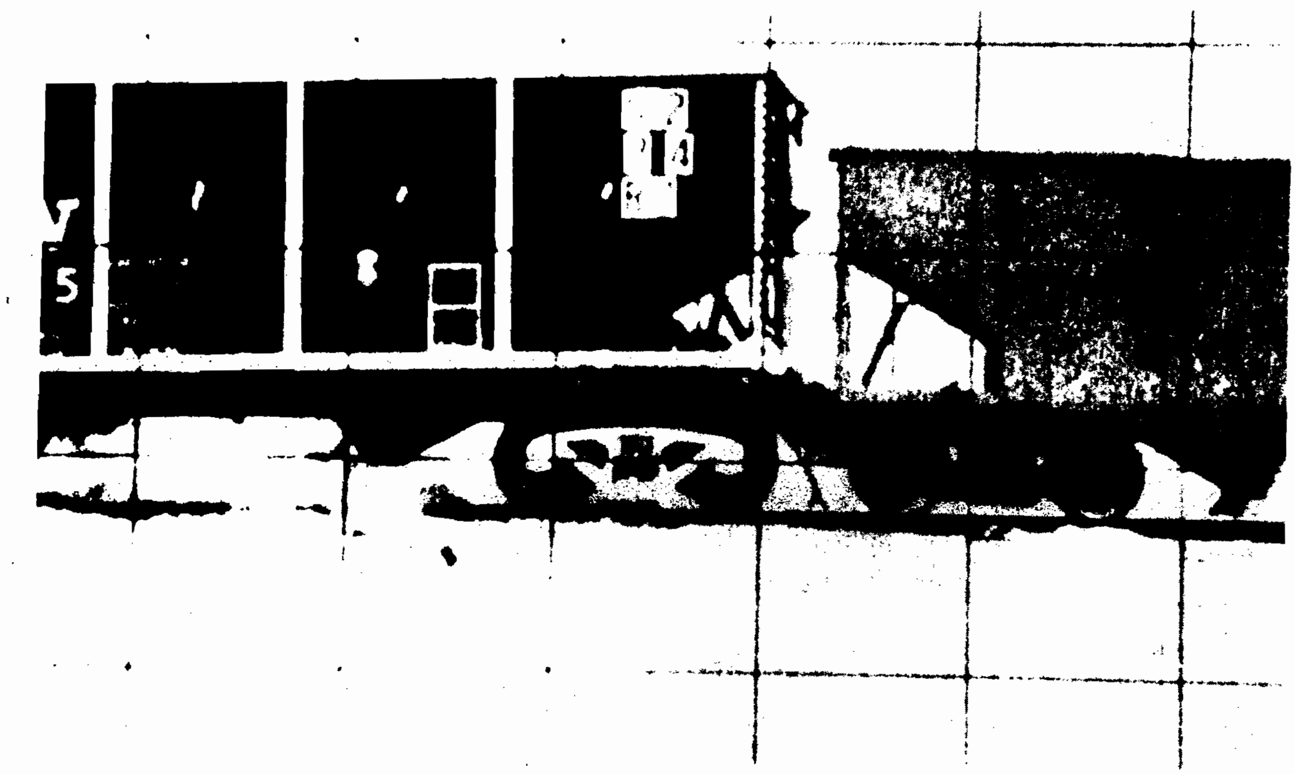

Figure 19 Second impact, 13.7 mph

Figure 20 Override, 13.7 mph

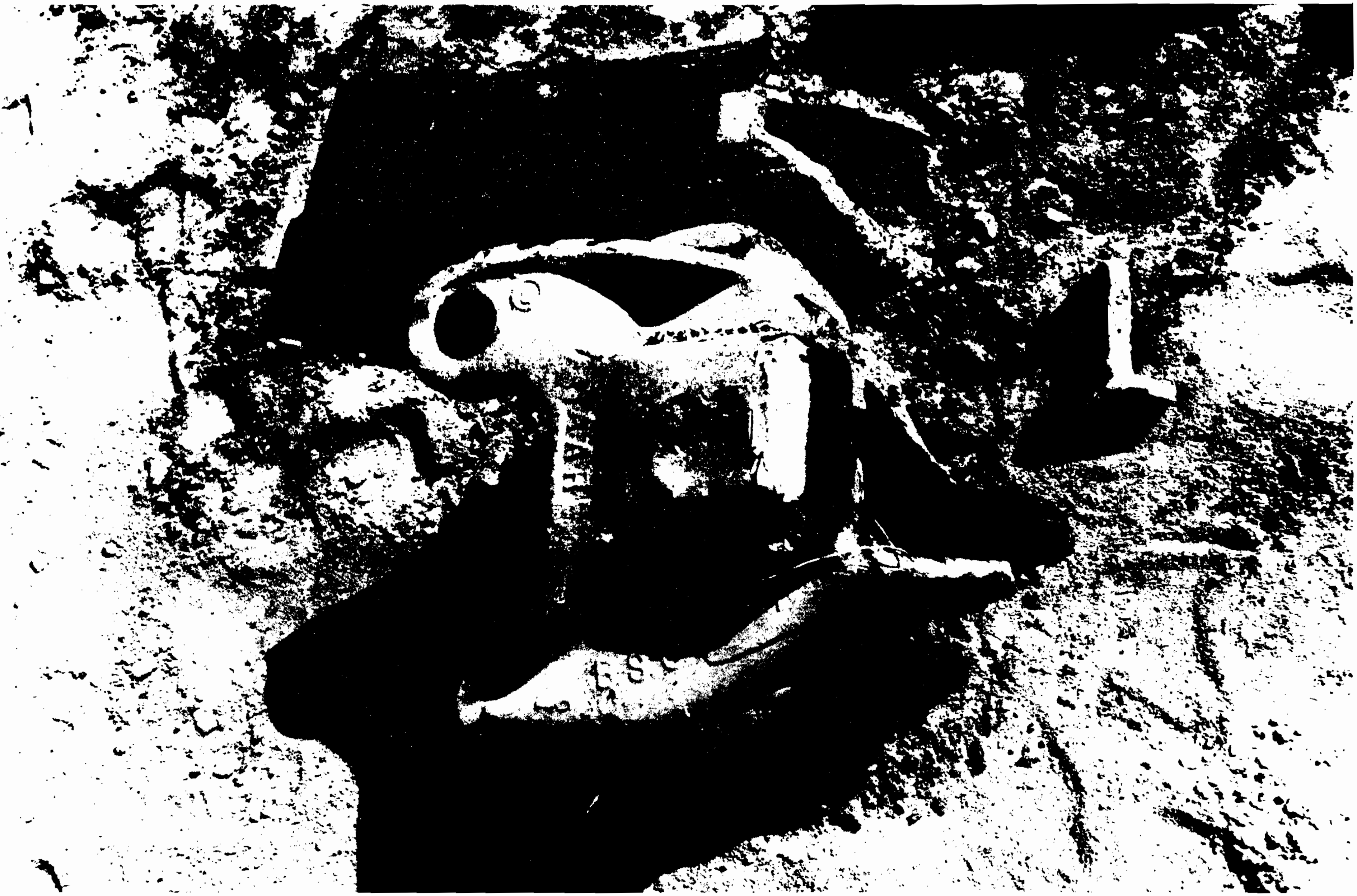

Figure 21 Broken coupler from impact on March 4, 1976

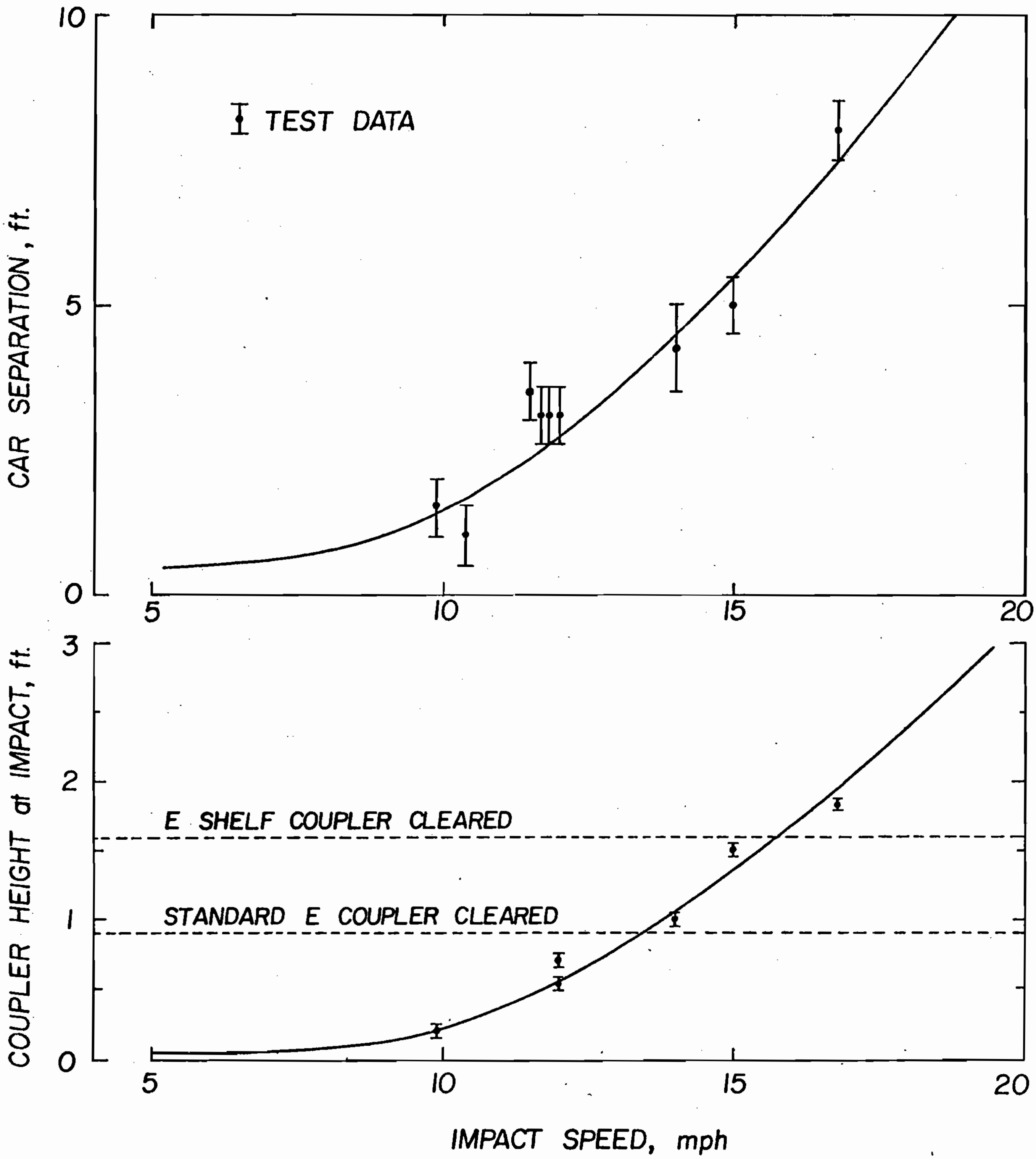

Figure 22 Relation between impact speed, car separation, and coupler height

This is not to imply, however, that lower speeds will not override. Even when couplers do not completely clear, override is possible because vertical slipping can occur. In the E shelf coupler, however, a coupler height of 7 in. or less (12 mph impact) may engage the shelf and probably prevent override. For speeds between 12 or 14 mph, however, the shelf acts as only a small barrier of the override mechanism. For the squeeze mechanism, however, in which initially the cars are coupled, the shelf may provide some protection against override.

The first impact of Series 3 was performed on April 15. The impact sequence is shown in Figures 23-26. In Figure 23, the hopper car and first back-up car are shown. The hopper was hit from the left and pitched up. The impact speed was 12.7 mph. The couplers narrowly cleared, with the faces sliding over only 2 inches vertically, as shown in Figure 23. The hopper car coupler then overrode the tank coupler, Figure 24, striking the tank slightly above the cradle pad, Figure 25. The car then fell down onto the end platform just as the tank car hit again, Figure 26.

A very similar sequence of events occurred in the second test of Series 3. The impact speed was 14.9 mph. In each case, the spatial and temporal separation of the second and third impacts decreased the force of the collision. The tank head suffered two small dents rather than one large one. The head block and cradle pad provided substantial protection to the head during the third impact, which occurred when the coupler already rested on the end platform of the back-up tank car. Thus, no puncture was obtained in these tests. The car spacing was reduced for the remainder of the Series 3 tests in order to control the impact heights and to make the second and third impacts closer in time.

In the test on May 6, the impact speed was 16.5 mph and the car spacing was 3.5 ft. This impact resulted in override at both ends of the hopper, as shown in Figures 27-28. In Figure 27, the tank car came in from the right causing the hopper to override the back-up tank car on the second impact. This second impact, however, caused the hopper to pitch up at the hammer car end. The pitching was sufficient for the hopper car to override the incoming tank car on the third impact, Figure 28. It so happened that on this test, the hammer car was equipped with a head shield as shown in Figure 28. The shield did absorb some energy.

Figure 23 Coupler face slipping, 12.7 mph

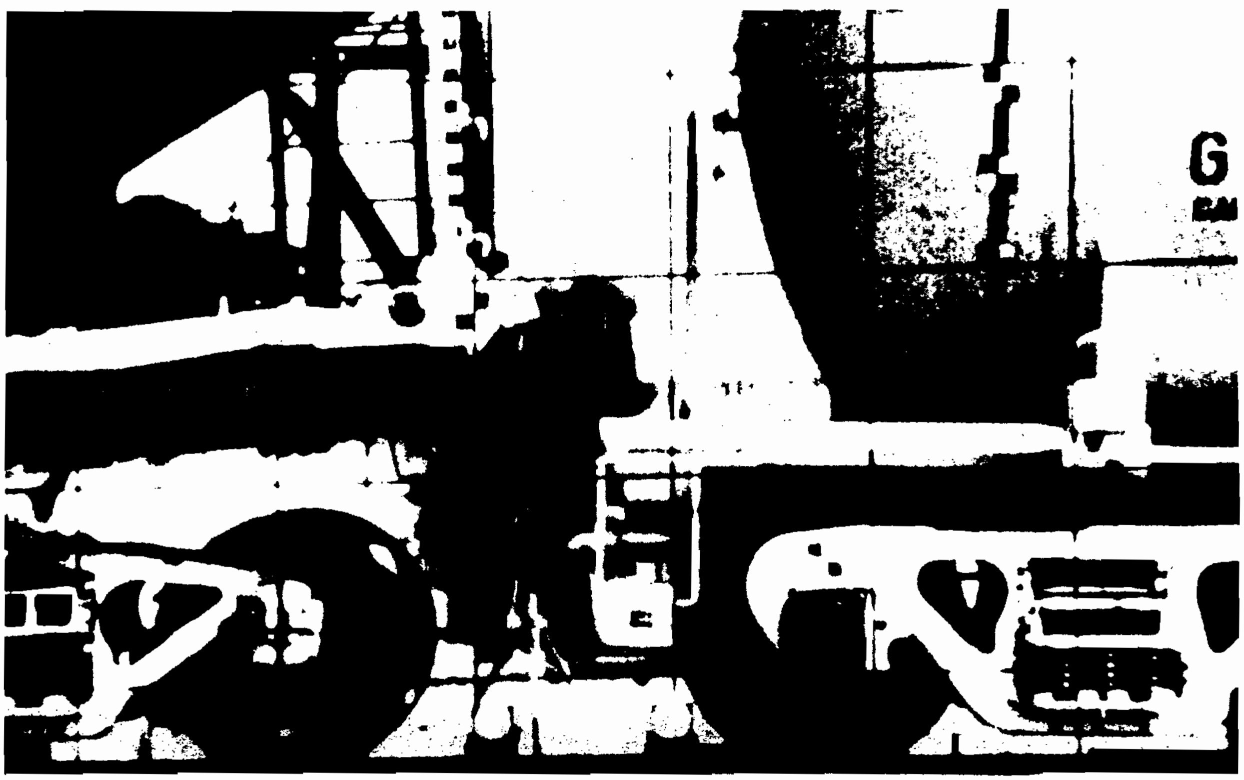

Figure 24 Coupler override, 12.7 mph

Figure 25 First hit of tank car head, 12.7 mph

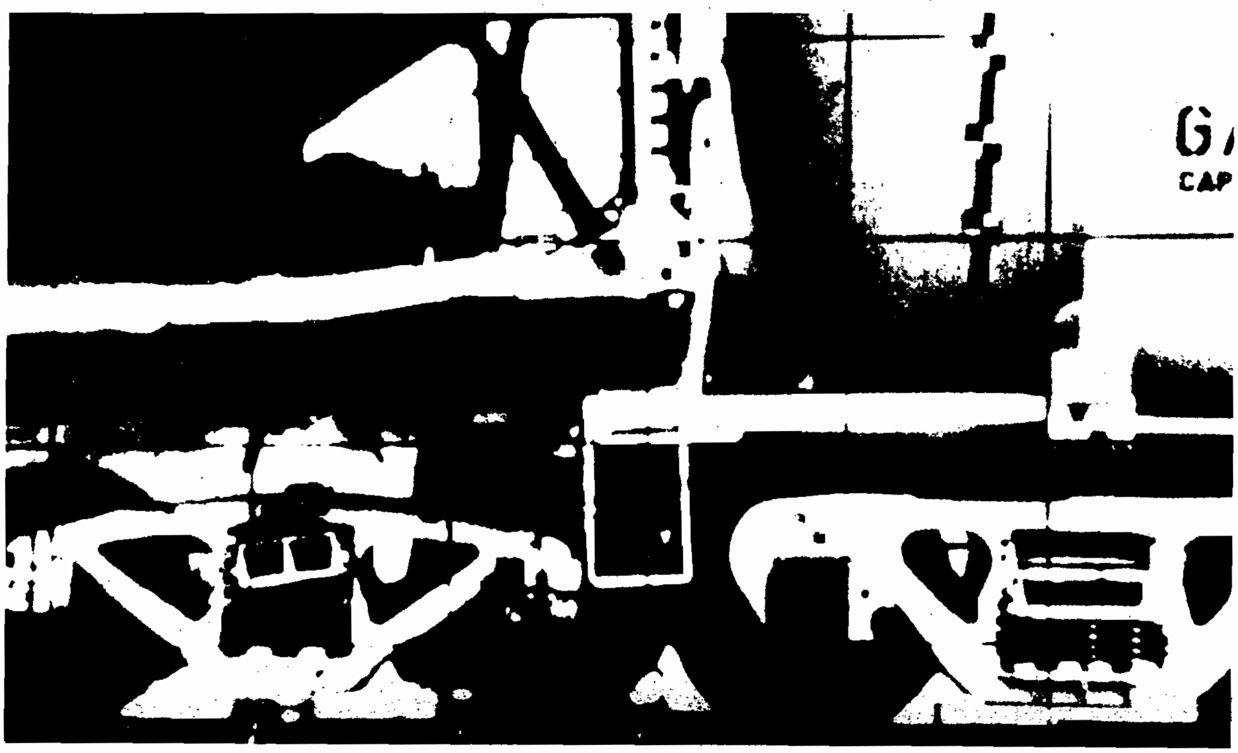

Figure 26 Second hit of tank car head, 12.7 mph

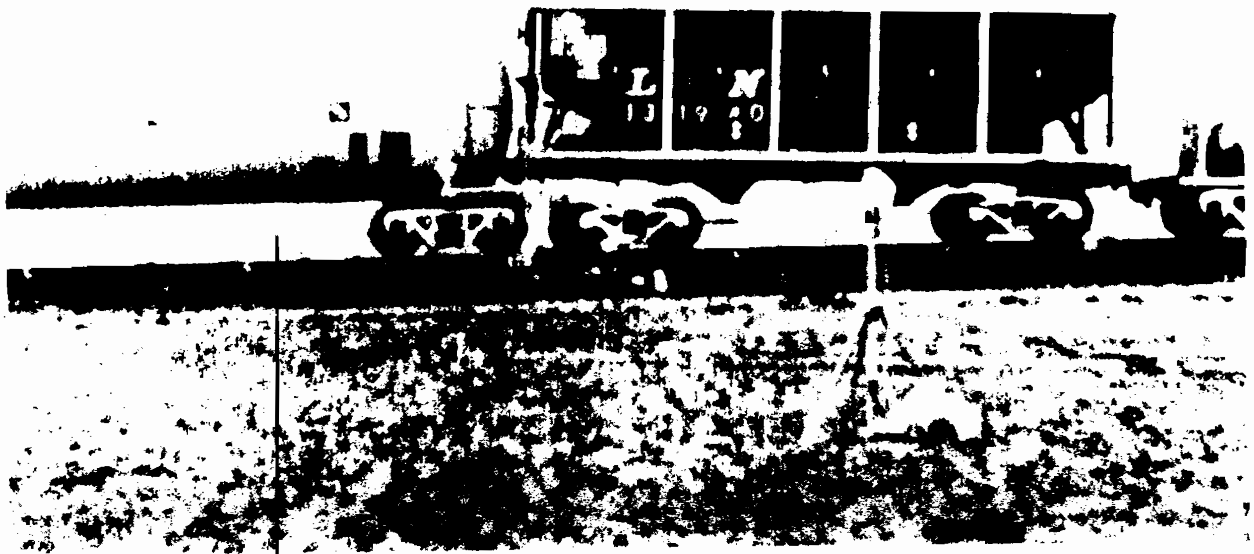

Figure 27 Back-up car override, 16.5 mph

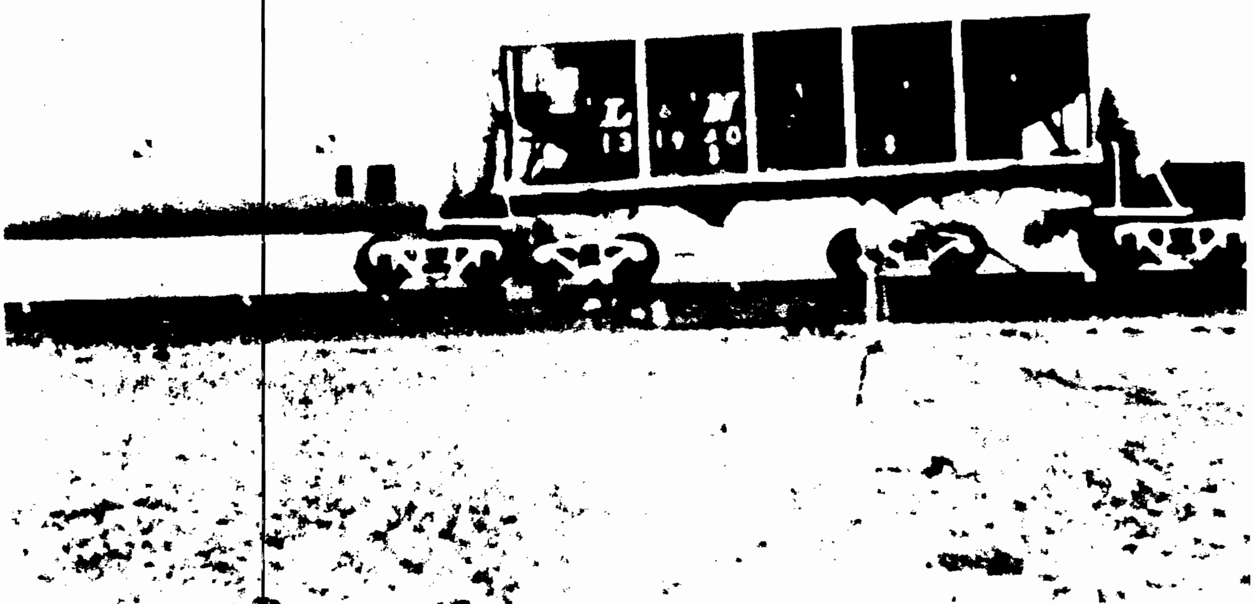

Figure 28 Hammer car override, 16.5 mph

The main benefit provided by the head shield however was that it minimized the chance of punching shear failures, thereby tending to increase the amount of energy needed to breach the head, which is expected to occur by tearing along the cradle pad or head block weld seam.

The fourth test of Series 3 was run at 15.4 mph, and with a spacing of only 2.5 feet. Although the speed was less than for run 3, the closer spacing brought the second and third impacts closer. Thus this impact produced the largest dent of the Series, although puncture did not occur. The hopper car overrode both the hammer car and the back-up car. The center sill of the hopper buckled in this test. The buckling served to lower coupler loads and was one of the probable reasons why puncture did not occur. When the angle of the buckled sill reached 15°, the loose car overrode the hammer car, Figures 29 and 30.

The final test of Series 3 occurred on May 27. The impact speed was 16.9 mph and the spacing was 2.5 ft. The hopper car coupler barely cleared the back-up coupler just as in all tests of Series 3. In this case, however, the coupler struck the tank car head at a low elevation due to the wedging of the hopper car bolster at the impact end as described earlier. Thus, both hammer car impacts (first and third impacts) occurred slightly above the head block. The sill of the hopper car did not buckle in this case. A tear occurred in the tank car head at the head block which resulted in loss of lading, Figure 31. Meanwhile, the second impact (hopper into back-up tank) caused the hopper car to pitch up at the impact end. Therefore, when the hammer tank car hit for the second time, the tank and hopper car couplers were misaligned by approximately 6 inches. This eccentricity caused the hopper car coupler head to break off. This coupler was made of grade B steel. As a result, the hopper was lifted very high and struck the tank car head near its center, Figures 32 and 33. No puncture occurred on the hammer tank car, however. Thus, this test provided the first head puncture obtained under realistic switchyard conditions.

4.4 <u>Future tests</u>

The remaining testing schedule of Phase 15 has yet to be formulated, but there are a number of important tests that might be included. First

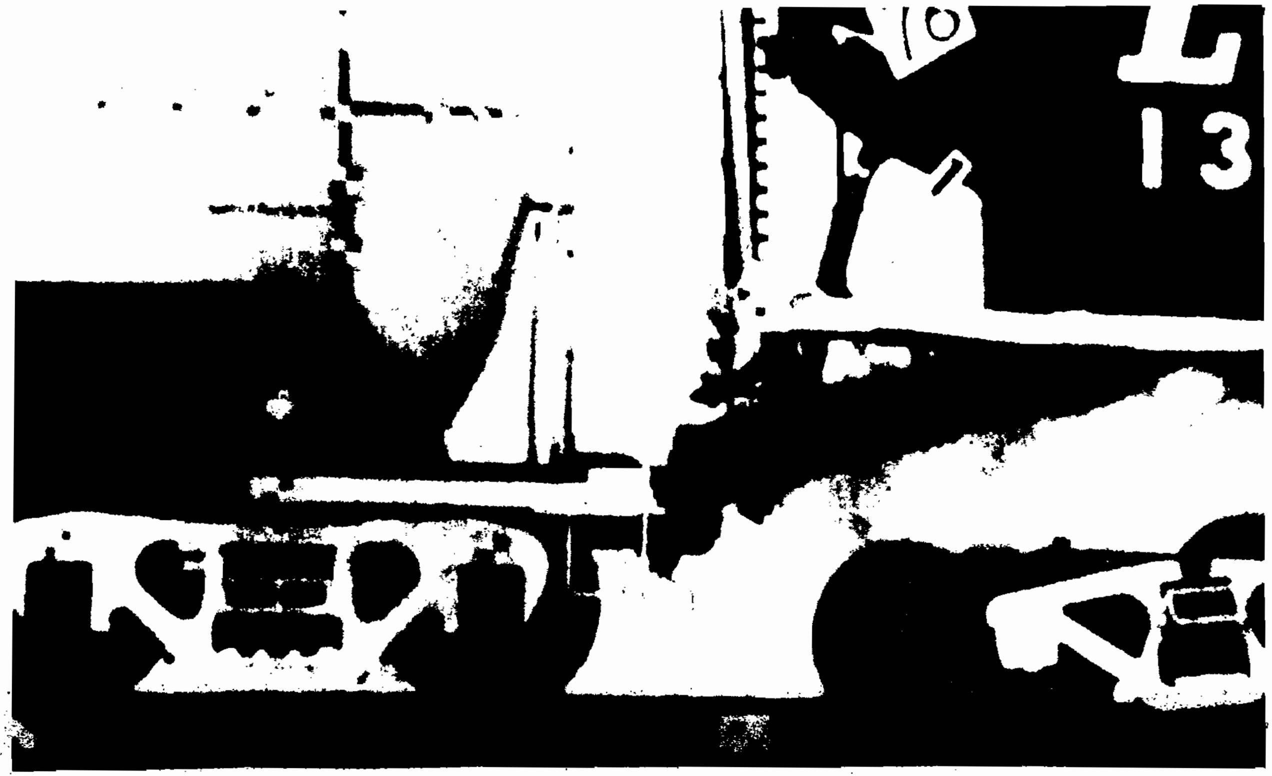

Figure 29 Sill Buckling, 15.4 mph

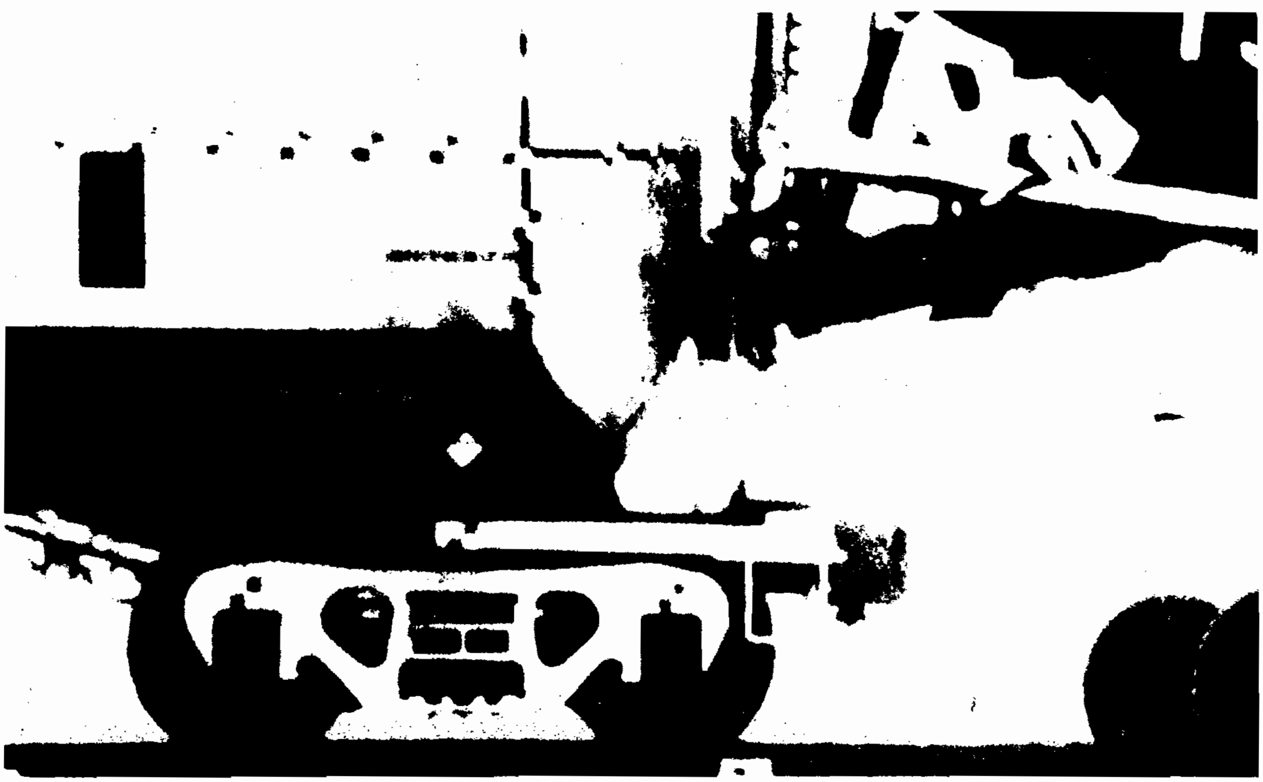

Figure 30 Hammer car override, 15.4 mph

Figure 31 Tank car head puncture, 16.9 mph

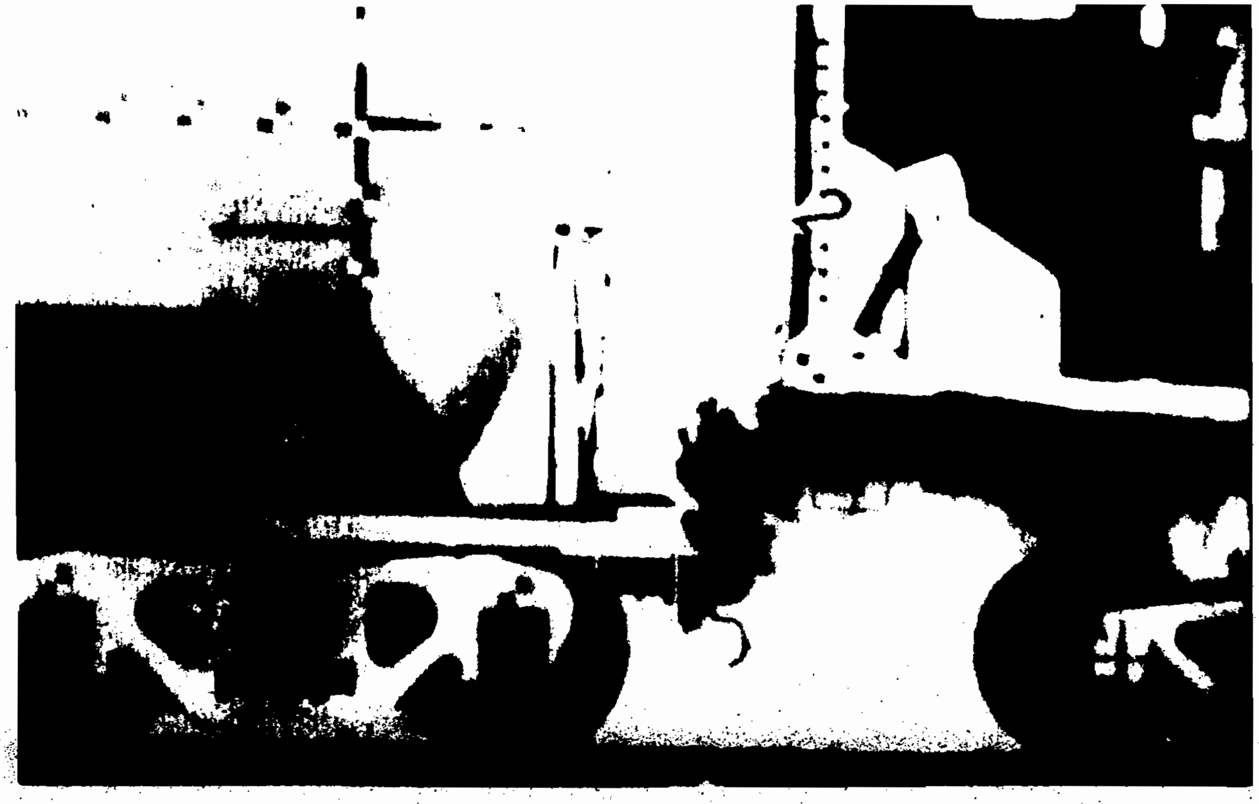

Figure 32 Broken coupler, 16.9 mph

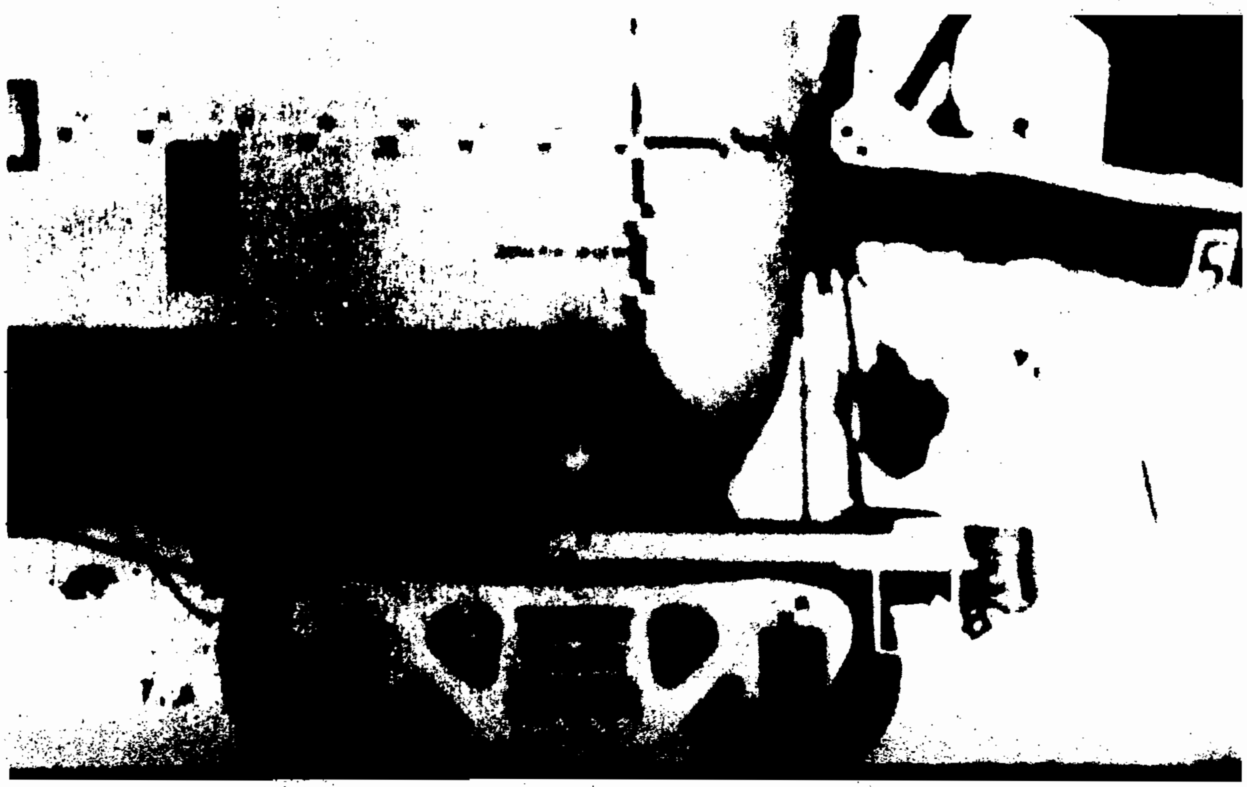

Figure 33 Hammer car override, 16.9 mph

of all, tests without head shield will probably be performed with multiple impacting cars. These tests would not only yield insight into the effects of multiple car humping, but would also provide a lower impact speed for puncture. The lower impact speed could conceivably be used as a baseline for tests to analyse protective devices. Thus, the Phase 15 tests will probably include tests of shelf couplers and head shields for both the high speed, double impact mechanism and the lower speed, multiple car squeeze mechanism.

5. DESIGN CRITERIA

The information obtained thus far in the Phase 15 testing program can provide a logical basis for the choice of countermeasures to prevent coupler override and tank car head puncture. Exact quantitative criteria must wait the completion of the Phase 15 program, but sufficient information now exists to formulate an approach to the development of criteria and to outline the method by which quantitative standards can be set. The basis for designing protective measures is the understanding of the nature of potentially destructive override mechanisms. The locations and magnitudes of expected forces must be known, and some idea of the probability of critical occurrences must be determined. Furthermore, the expected duration of forces must also be considered.

It is useful to distinguish between two types of coupler override events according to whether such events occur in classification yards or on the main line. To prevent classification yard accidents, certain operational guidelines can be introduced for the purpose of limiting the amount of energy available to puncture the head of a hazardous material tank car in case of retarder failure or other emergency. Consequently, countermeasures can be chosen on the basis of a performance criterion to be stated in terms of a minimum energy absorption capacity. On the other hand, it is not possible at this time to establish reasonable upper bounds on the available energy for main line accidents. Consequently, no protective device can be expected to be fully effective in preventing head puncture in main line accidents.

The Phase 15 experiments conducted to date involved simulated switchyard collisions. For this reason, the following discussion is mainly concerned with switchyard impact.

5.1 Number of high speed impacts

The first step in the analysis of protective devices is an estimate of the number of impacts that a given tank car can be expected to encounter. According to AAR statistics, Reference (9), the average freight car experiences .625 switchyard impacts per 100 miles of travel. An average tank car travels 10,000 miles per year, and some tank cars may travel up to 40,000 miles. Therefore, a tank car may experience 60 to 250 impacts per year. Over a 40 year life span, a tank car may be subjected to 2500 to 10,000 impacts. Similarly, a fleet of 22,000 tank cars will experience a combined total of over a million impacts in a single year. Therefore, impact speeds which have only a small probability of occurrence (e.g., 1 in 1,000) must be considered as everyday occurrences.

The next step in the analysis must be to determine the probability distribution of switchyard impact speeds. Two independent studies of classification yard impact speeds were made by Pullman Standard and by the New York Central Railroad.[10] The Pullman Standard data are based on 1568 measurements over a three month period in 1950, and the New York Central data are based on system wide averages over a three year period. These data are plotted in Figure 34 in terms of the probability that an impact will exceed a specified velocity, V. The two sets of data are practically identical and can be well approximated by the "largest value" probability distribution described in Reference (11). This particular distribution function is well suited to the descripton of the largest value to be expected in a one-sided (i.e., positive values only) random sample. Such a distribution has been fitted to the data and also plotted in Figure 34.

According to the published observations, the average velocity of impact is 4.7 mph and the standard deviation is 1.5 mph. The interpretation of the probability distribution is that 64% of all impacts are above 4 mph, 30% are above 5 mph, 13% are above 6 mph, 6% are above 7 mph, 3% are above 8 mph, and only 1% are above 9 mph. The low end of the curve, $V < 4$ mph is not expected to be accurate because the "largest value"

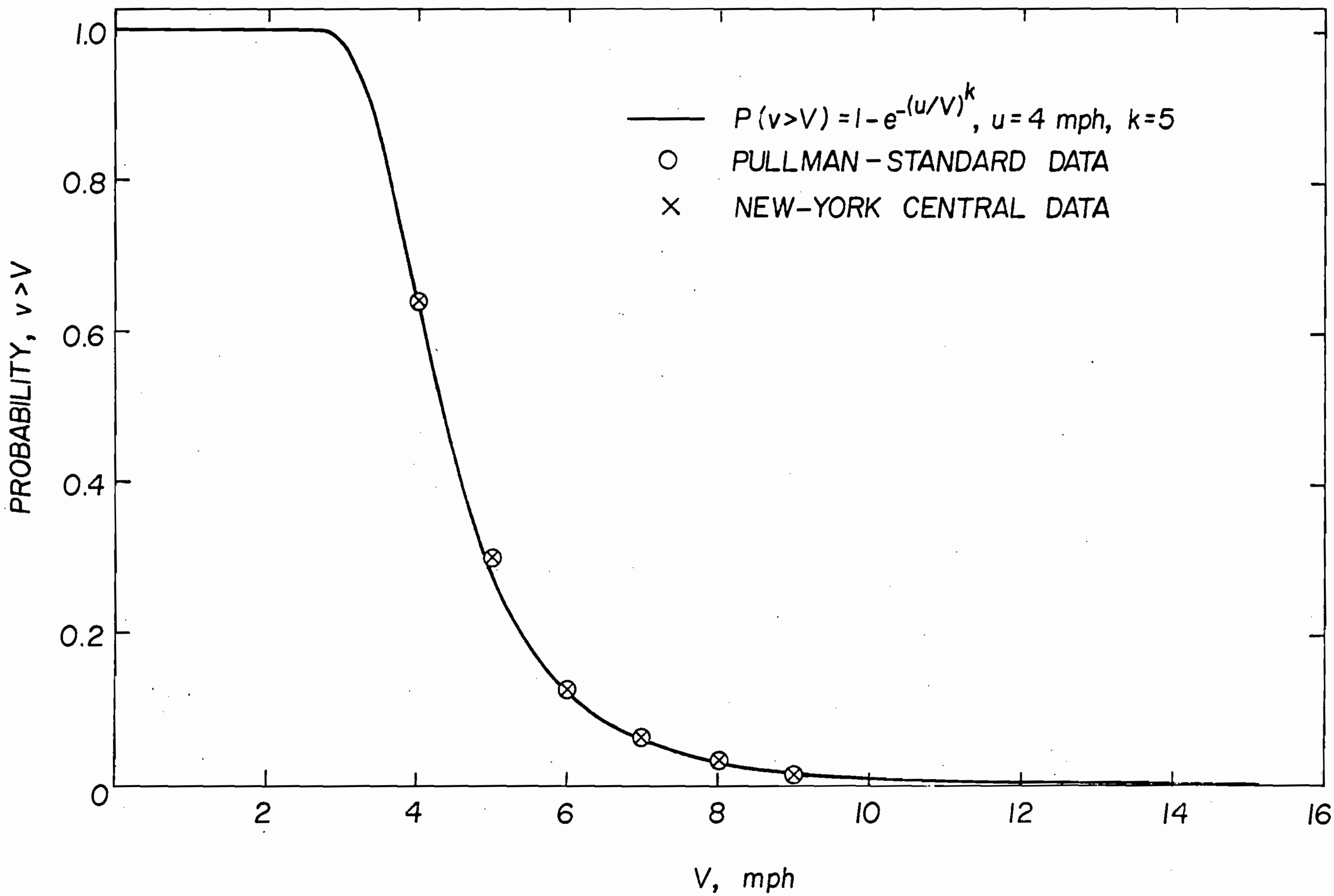

Figure 34 Probability of impact speeds

probability distribution is not suited for smallest value estimations. Although the data base goes only to 10 mph, the excellent fit of the assumed distribution indicates that some extrapolation to higher velocities is warranted. There is, of course, a practical limit to extrapolation. In hump yards, for example, the maximum height of the hump is approximately 14.5 ft. The law of conservation of energy implies that there is, therefore, a limit to the car speed even in the event of a retarder failure. For a rolling friction coefficient of .0022 and a rolling distance of 400 feet, it would be highly unlikely to exceed 20 mph for any reasonable initial velocity. Therefore, the distribution in Figure 34 should be truncated near 20 mph.

5.2 Expected impact energy

The largest expected impact energy in switchyard impacts can now be determined from the known facts on car weights and maximum velocity. The simplest case to analyze is that of single car humping. In single car humping, the maximum allowed car weight is 273,000 pounds, and the maximum speed is 20 mph. This implies that the maximum possible impact energy is 3,650,000 ft-lb. Tests have shown, however, that not all of this energy is available for puncture. Much energy is lost in the dynamic mechanisms leading to puncture as shown in Appendix C.

For example, only the first one or two back-up cars are generally involved in the high energy wave of the impact. Due to conservation of momentum, therefore, at most two-thirds of the impacting energy is available for immediate dissipation (2,430,000 ft-lb.). The rest of the energy remains as kinetic energy and is dissipated slowly as the remainder of the consist is involved in the collision. Furthermore, approximately 8% of the available energy is lost in the initial impact between the hammer car and light car body. Energy calculations are based on detailed analyses of the Phase 15 test data including high speed motion pictures. If multiple impacts are involved so that the light car trucks participate, another 5% of the energy is lost in the truck dynamics. It should be noted here that the Phase 15 tests often seem to imply that more than 8% of the energy is lost upon impact. Careful analysis, however, has shown that the "missing" energy is due to tank car sloshing (even at 94.6% volume). This energy again becomes available during the puncture impact.

Energy is also lost due to coupler action during the override process provided that coupler faces do not completely clear. In other words, the sliding of these faces and the energy required to lift the car cause a loss of energy in the order of 0.7 million ft-lb., based on the Phase 15 data. Furthermore, if the couplers meet with small eccentricity so that a dynamic squeeze ensues, a large amount of energy can be lost due to yielding and bending of sills, underframes, and couplers. The experimental evidence shows that 1.0 million ft-lb. is often dissipated in structural yielding in addition to the 0.7 million lost in coupler sliding and car body lift. If further car bouncing takes place, as in some multiple impact mechanisms, further energy is lost in these motions.

It follows that the actual energy remaining for possible head puncture is highly dependent upon the particular override mechanism. In turn, the override mechanism is dependent upon the spacing between the light car and the first back-up car. Therefore, the amount of energy required for puncture is a function of the spacing between the light car and the remainder of the hopper cars. The relation between car spacing and energy is illustrated in Figure 35 for 16, 18 and 20 mph impacts. For zero car spacing, the coupled configuration, the squeeze mechanism takes place and 1.7×10^6 ft-lb. of energy is dissipated in yielding. When the spacing is increased to represent an uncoupled car starting 2 ft. to 11 ft. from the other cars, the double impact mechanism can occur in which couplers completely clear. Therefore only 8% of the energy is lost. For spacings of the orders of 20 ft., the triple impact mechanism takes place which requires 8% + 13% = 21% of the available energy. For spacings between (or slightly greater than) the double and triple impact distances, substantial coupler sliding must take place and an extra 700,000 ft-lb. is required. For larger spacings, other multiple impact mechanisms become plausible. These absorb slightly more energy than the triple impact mechanism. In the event that the distance is large enough for a detruck or derail type override, little is known about the required energy, but it is certainly no less than for the triple impact mechanism.

Figure 35 can be interpreted in terms of a total energy balance. The dashed lines give the total available energy (including momentum

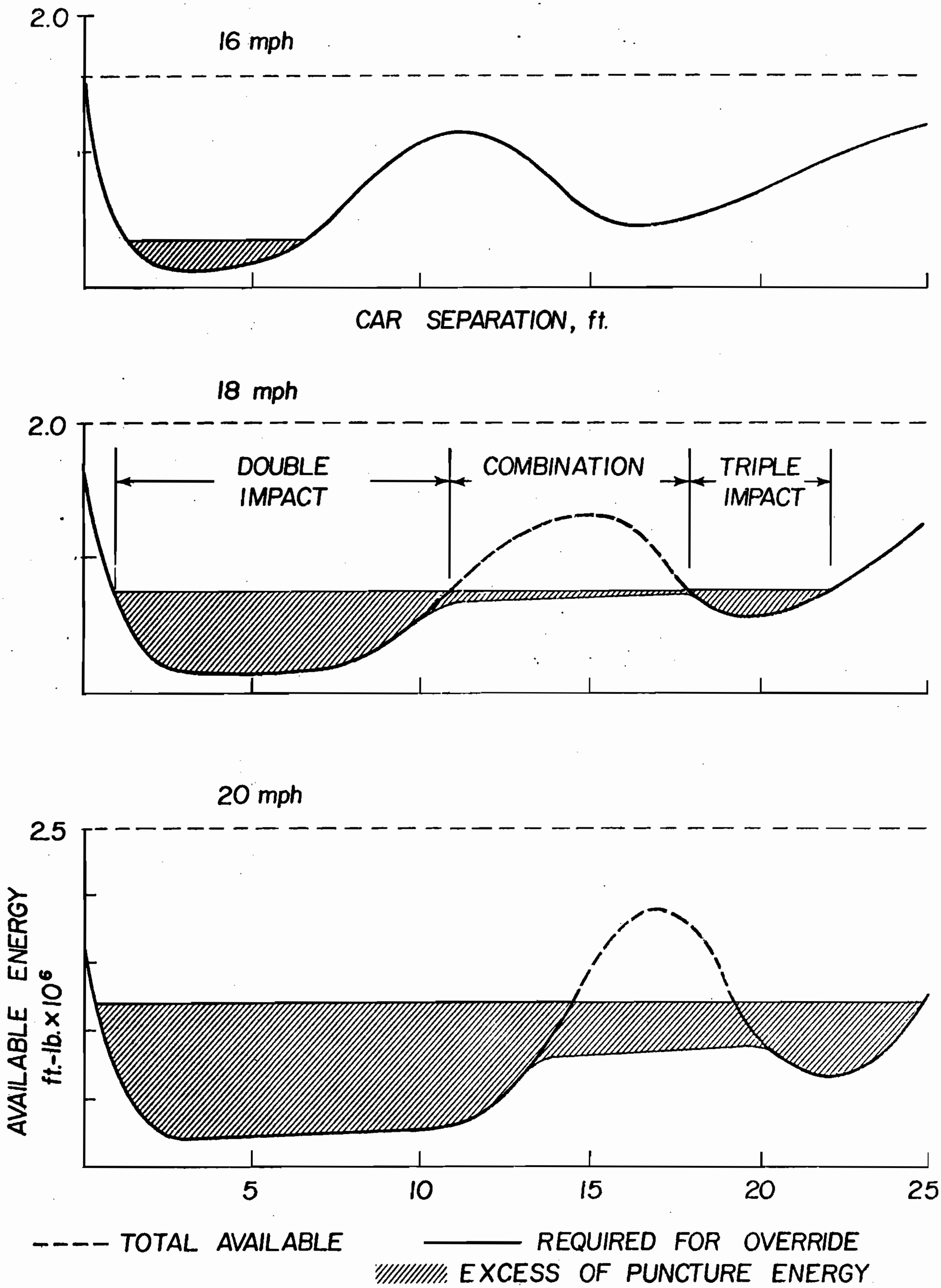

Figure 35 Relation between impact velocity, car spacing and absorbed energy

considerations) for each impact speed. The solid curves give the energy required to override. Therefore, the difference in the two curves is the energy available to puncture. Tests have shown that as little as 1.2×10^6 ft-lb of energy can cause a puncture. When this energy is subtracted from the available energy, the shaded area remains as the excess of puncture energy.

It should be pointed out that the double and triple impact mechanisms do not always occur in their pure form. Sometimes the high eccentricity inherent in the impacts can cause a coupler to break or bend. This action, in turn, can cause override at either end. This type of override cannot be classified as a pure dynamic squeeze, however, because the high eccentricity greatly diminishes the energy required to bend couplers or sills. The minimum required energy is probably in the order of 250,000 ft-lb (1,000,000 lb × 1 ft eccentricity × 15° rotation). Thus the shaded areas in Figure 35 include combination type overrides as special cases. The combination type override can also occur in the intermediate spacings (12 - 20 ft) for which neither pure double nor triple impact could occur.

Figure 35 can now be used to generate a plot of the velocity required to puncture as a function of back-up distance. This is given in Figure 36 for four mechanisms: double impact, triple impact, combination, and derail. The energy in the derail mechanism must be greater than the energy in the triple impact, and the spacing in the derail mechanism must be larger than in triple impact. Figure 36 is obtained by plotting the corners of the shaded areas of Figure 35 for each of the three impact speeds. Thus, for car separations from 0 to 11 ft the double impact mechanism is possible with 2 1/2 ft being the most critical distance. From 18 to 25 ft the triple impact mechanism is probable. Intermediate distances can result in combination overrides and larger distances in detrucking or derailing. It should be pointed out again that the combination mechanism can occur within the "double impact" or "triple impact" areas but requires more energy than the minimum energy represented by the plotted curves.

A detailed analysis of the first head puncture in Phase 15, shows that at temperatures above 60°F only 1.2×10^6 ft-lb. of energy is required to tear or puncture the head if the hit is just above the sill block. Other Phase 15 tests and the RPI/AAR tests, Reference (9), show that if the hit is higher up on the head as much as 2.0×10^6 ft-lb may be required for puncture. Thus, even for a 20 mph impact and single car humping, a coupled configuration (no loose cars) would probably not cause a puncture. In the event that a loose car is present, a spacing of from 2 ft. to 11 ft. and a speed of 18 mph could definitely cause puncture at any spot on the head. It is postulated that car spacings of ½ ft. to 2 ft. or 18 to 22 ft., however, could only result in puncture if the hit were very low on the head.

For multiple car humping, the available impact energy is much greater than it is for single car humping; but there is, nevertheless, an upper limit to the amount of available energy. Test results and computer simulations have shown that generally only the first three heavy impacting cars participate in the override mechanism. Thus, the maximum impacting energy is roughly $3 \times (3{,}650{,}000)$ ft-lbs or 11×10^6 ft-lbs. From conservation of momentum considerations, however, only about one-half of this energy is available for override and puncture. The rest of the energy is absorbed in moving the entire consist down the track. Thus, about 5.5×10^6 ft-lbs of energy could possibly be available for head puncture in a switchyard.

For mainline accidents, on the other hand, there is no practical limit to the available energy. That is, the velocities can be so high and the number of cars involved so large that no protective device could prevent all head punctures. Nevertheless, if shelf couplers can be proven to be effective in preventing vertical coupler disengagement in high energy dynamic squeeze events then shelf couplers may provide some protection for tank cars against head puncture in main line accidents.

5.3 Probability of puncture event

Now that a rough idea of the energy required for puncture has been obtained the crucial question is: what is the probability that all critical events necessary for puncture will occur simultaneously? Three

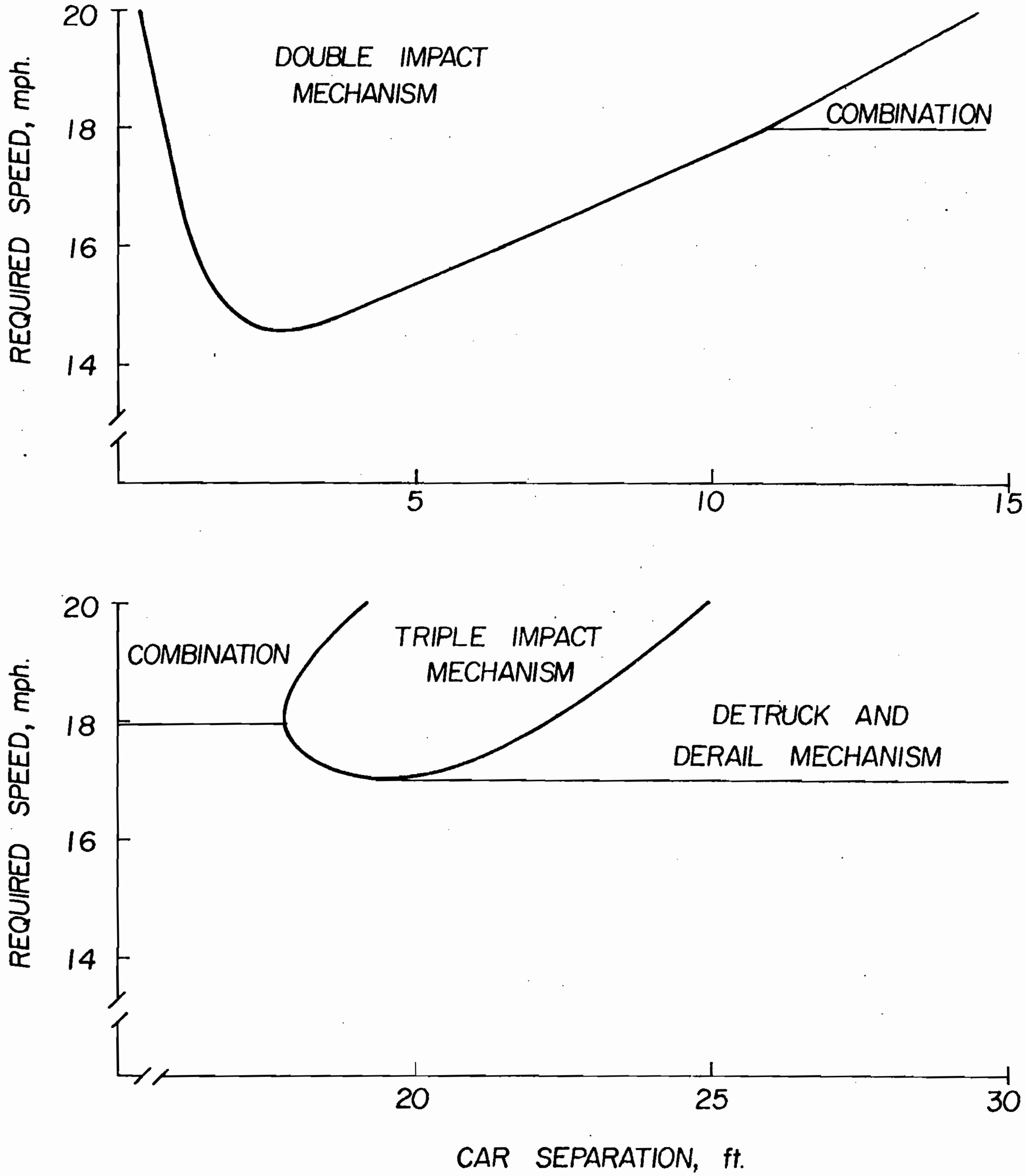

Figure 36 Velocity required to puncture
at ambient temperatures above 60°F

switching scenarios that may lead to tank car punctures are given below with listings of the necessary events required for puncture of a hazardous material tank car.

Scenario #1

a) Lightly loaded car is switched onto track and does not couple

b) Heavily loaded hazardous material tank car follows at excessive impact speed

c) The tank car does not couple with the loose car

d) Spacing is just right to cause triple impact or combination override

e) Impact energy is sufficient to rupture tank

Scenario #2

a) Lightly loaded car is switched onto track and does not couple

b) Heavily loaded hazardous material tank car follows at excessive impact speed

c) Light car detrucks and or derails in a position so as to cause coupler override

d) Spacing is large enough to keep back-up cars from interfering

e) Impact energy is sufficient to rupture tank

Scenario #3

a) Heavily loaded hazardous material tank car stands as last car on track

b) Lightly loaded car switched onto track and does not couple

c) Heavily loaded car follows at excessive speed

d) Spacing just right to cause double impact or combination override

e) Impact energy is sufficient to rupture tank

By adding the probabilities of scenarios 1, 2 or 3, one can obtain the overall probability that a hazardous material tank car will receive a puncture in a typical switchyard situation of two tank car impacts: as hammer car and as anvil car.

The next step, therefore, is to estimate the single probability of each of the events a-e in the three scenarios. Now, the probability of a light car being switched onto the track (either before or after the tank) can be obtained from AAR car data. There are 147,000 light box cars (40 ft. cars), 70,000 light covered hoppers (less than 3,000 cu. ft.), and roughly 70,000 light open hoppers (50 ton capacity). The only other cars that can be classified as light (in the sense that they have low moment of inertia and can bounce easily) are the small 100 ton tank cars of which there are probably less than 30,000. Considering, therefore, a total of 1,700,000 cars, and a probability of .47 of being empty, the probability of a light car switching is (.47)(317,000)/(1,700,000) = .08. This number is decreasing every year, however, as the older light cars go out of service. Similarly, the probability of a heavy car being switched (greater than 200,000 lb.) can be computed and is roughly .12.

The probability that a light car will detruck in a manner conducive to puncture is also a function of the type of cars involved. Hopper cars are very unlikely candidates, as was found in Phase 15, because of the interference of the hoppers themselves. Thus only the box cars and tank cars are possibilities. Furthermore not all will detruck in exactly the prescribed manner. Therefore we place a nominal value of .5 on the probability of override provided that the impact speed is 16 mph or above, that at least 20 feet of back-up space is available, and that no hopper car is involved. The total probability is .5 $(\frac{177}{217})$ = .40 when the spacing and speed are right. (Later calculation show that this assumption has little effect on the overall probability of puncture.)

The probability of any given impact speed occurring has already been presented in Figure 34 and is fairly reliable. Of course, we have not taken into account that when a loose car already stands on the track (as in scenarios 1 and 2), the crew may have a tendency to switch the next car a little faster. This, however, is probably not a factor in impact speeds above 16 mph which are due mainly to mechanical failures. The probability of a loose car being present is closely related to the impact velocity distributions, because it can occur due to an overly slow switching speed or an overly fast speed. If the velocity distribution is assumed to be symmetric about the median speed (4.3 mph) in terms of impact energy, then roughly 2% of all attempted impacts would appear as having negative energy (in other words, as being slow and stopping short of impact). For loose cars due to overly fast impacts, the early Phase 15 tests showed that 80% of all impacts between light and heavy cars at speeds of over 6 mph did not couple. Although later tests have not fully corroborated this finding, to be conservative we say that at least 20% of light car to heavy car impacts will couple. The odds for a speed over 6 mph are .13; thus, the conservative probability of no coupling with a light car is .02 + .80(.13) = .12.

The only probability remaining to be determined in the probability of car spacing. This must be used in conjunction with Figure 31 to determine the probability that enough energy will remain to puncture. Since most cars fail to couple in the over 6 mph category, we can assume that the energy remaining after impact is proportional to the car rebound distance. Thus, the velocity distribution in Figure 34 can be used to generate a rough distance distribution for cars that don't couple. If we assume that the distribution for distance (s) from 0 to 40 ft. is the same as v^2 for 6 mph to 20 mph, the probability function becomes

$$P(s>D) = 8.1\left[1-e^{-(u/v)^k}\right] \text{ where } v = \sqrt{36+9.1D}$$

$$u = 4, \; k = 5$$

which is illustrated graphically in Figure 37. The 40 ft. assumption at 20 mph is derived from an expected maximum rebound speed of 1.6 ft/sec., based on the Phase 15 tests.

We can now calculate the total probability by integrating each scenario through the various possible back-up distance and velocity combinations. For each distance, the potential override mechanism can be determined, thus yielding the required energy and impact speed. Since the impact probability is also known, it is only a matter of bookkeeping to calculate the entire probability. As a convenient notation, we define P_{vd} as the probability that spacing and velocity will fall within an area of the curves of Figure 36. Superscript TI denotes triple impact, superscript DR denotes derail, and superscript DI denotes double impact. (The combination override area is divided between TI and DI at 15 mph.) The analytical formula for P_{vd} is

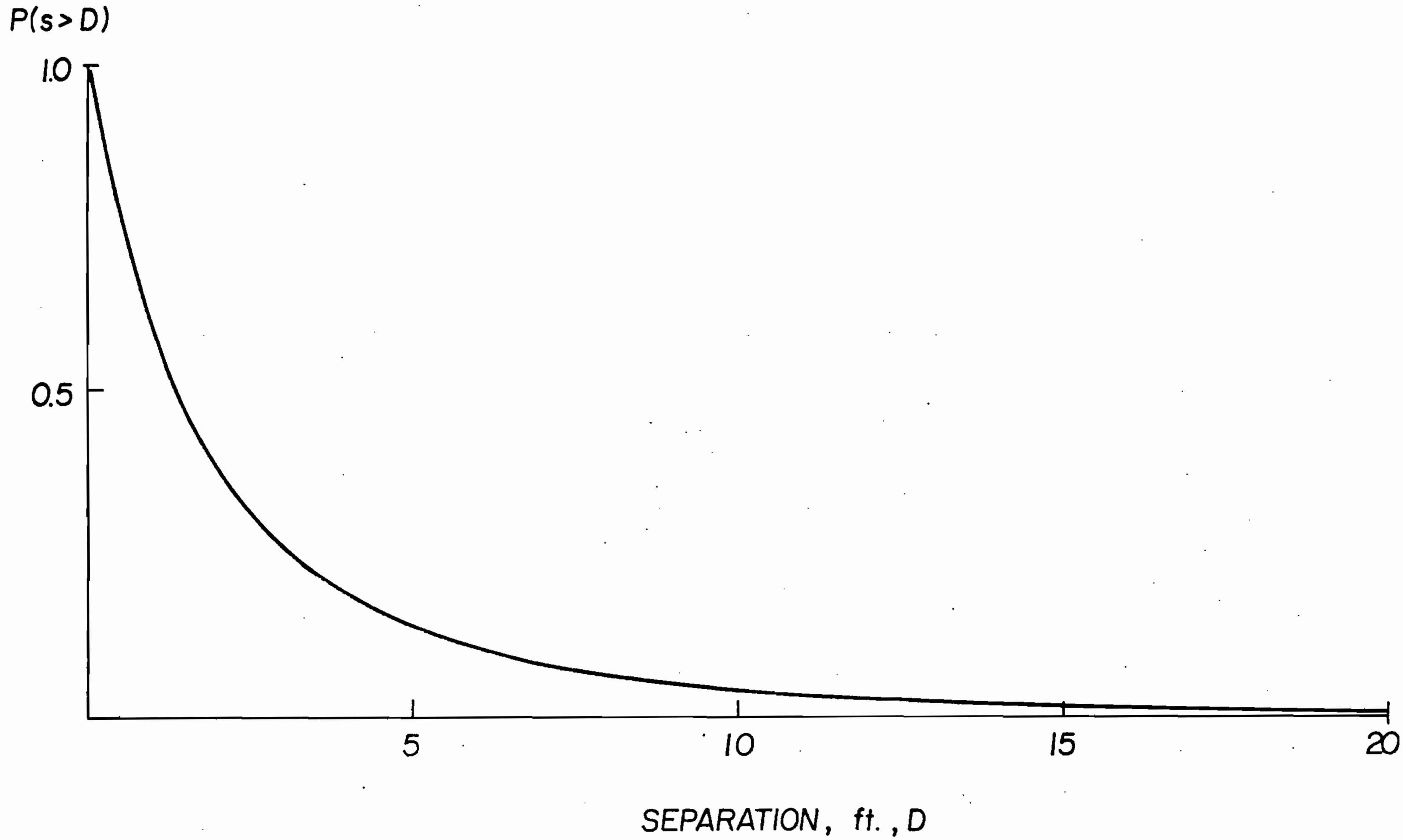

Figure 37 Probabilities of loose car distances

$$P_{vd} = \int_{S_1}^{S_2} P(s)P(v > v_{crit})ds = \int_{v_1}^{v_2} P(v)P(d < critical)dv$$

From section 5.3, therefore, the total probability is:

$$P = P(light) \cdot P(did\ not\ couple) \cdot P(did\ not\ couple) \cdot P_{vd}^{TI}$$
$$+ P(derail) \cdot P_{vd}^{DR} + P(heavy\ car) \cdot P_{vd}^{DI}$$
$$= (.08)(.12)(.8\ P_{vd}^{TI} + .4\ P_{vd}^{DR} + .12\ P_{vd}^{DI})$$

The probabilities of P_{vd} are computed in Tables 8-10.

Table 8. Triple impact probability

S range ft	Probability p_s	Velocity mph	Probability p_v
15-18	.0068	17.9	.000557
18-19	.0016	17.3	.000661
19-20	.0013	17.0	.000721
20-21	.0011	17.3	.000661
21-22	.0010	17.5	.000624
22-23	.00087	18.1	.000527
23-24	.00077	18.8	.000436
24-25	.00067	19.5	.000361

$p_s \times p_v$ TOTAL = 1.57×10^{-5}

Table 9. Derail probability

S range ft	Probability p_s	Velocity mph	Probability p_v
20-21	.0011	17 17.3	.000060
21-22	.0010	17 17.5	.000010
22-23	.00087	17 18.1	.00020
23-24	.00077	17 18.8	.000285
24-25	.00067	17 19.5	.000360
25-40	.00736	17 20	.000721

$p_s \times p_v$ TOTAL = 6.02×10^{-6}

Table 10. Double impact probability

S range ft	Probability p_s	Velocity	Probability p_v
0.5-1.0	.170	18.5	.000472
1-2	.210	15.5	.00114
2-3	.119	14.6	.000154
3-4	.0722	14.8	.00114
4-5	.0467	15.1	.00130
5-6	.0317	15.5	.00114
6-7	.0223	16.0	.000976
7-8	.0162	16.5	.000837
8-9	.0121	17.0	.000721
9-10	.00926	17.3	.000661
10-11	.00721	17.8	.000573
11-12	.00571	18.3	.000499
12-15	.0114	17.9	.000557

$p_s \times p_v$ TOTAL = 7.80×10^{-4}

Therefore, the major probability is for a double impact override occurrence. The overall probability of a puncture for a given tank car switching is

$$P = (.0096)[.8(1.57\times10^{-5}) + .4(6.02\times10^{-6}) + .12(7.80\times10^{-4})]$$
$$= 1.04\times10^{-6} \text{ or } 1/1{,}000{,}000.$$

Although the probability is small, the number of tank car impacts that occur in practice is very large so that the odds are not at all beyond reasonable probability. For example, we have already shown that there are 1,400,000 hazardous material tank car impacts per year. This implies that there are roughly 350,000 switching operations involving cars that are loaded. (One-half of the cars are full, and every two impact comprises a switching event.) Thus, it would be reasonable to assume that a hazardous material puncture could occur, even with single car humping, within any given three year period.

The probability of head puncture involving multiple car humping is much more difficult to estimate because of the increased number of combinations that might lead to a squeeze type override. We can only say that, since all major switchyard accidents of hazardous material cars have involved multiple car humping, the probability of such an occurrence must be considerably higher than for single car humping. It may even be possible that relatively heavy cars could be forced to override and puncture under such conditions.

5.4 Alternative measures

The probable velocity distribution given in Figure 34 and the estimated impact energies provide a basis for development of countermeasures. Further testing will be necessary before more firm quantitative performance specifications can be established. The performance specifications will also depend on operational restrictions. We shall consider three alternative operational procedures.

The first alternative is to limit impact speeds in switchyards to 8 mph or less. This figure is based on flat switching operations where several cars may be involved in a single switching move. It is probably quite conservative for humping operations. In this case special protection might not be required for tank cars. However, fail-safe retarding systems would have to be installed in hump yards, and hazardous material tank cars and cars following it would have to be flat switched under locomotive control in flat yards.

The second alternative is to permit single car humping only. The restriction must apply to the tank car and to the first heavy non-sliding sill car after the tank car. (A "heavy" car is one with rail weight of 200,000 lbs. or more.) If this restriction were in use, then a logical performance criterion could be that the tank head be able to withstand 2,500,000 ft-lb coupler impact at 20 mph under all ambient temperatures which the head may experience. The point of impact for design considerations would be just above the head block, where the maximum chance for tearing is present. The car designer could conceivably use shields, reenforcing plates, or extra thick head sections to meet this criterion.

The third alternative is not to restrict the number of cars to be humped or flat switched. In this case tank car heads would have to be designed to withstand about 5,500,000 ft-lb impact energy, a very stringent requirement. Figure 38 provides a graphical representation of the percentage of punctures that probably would have been prevented by the various energy absorption requirements. Without the single car switching

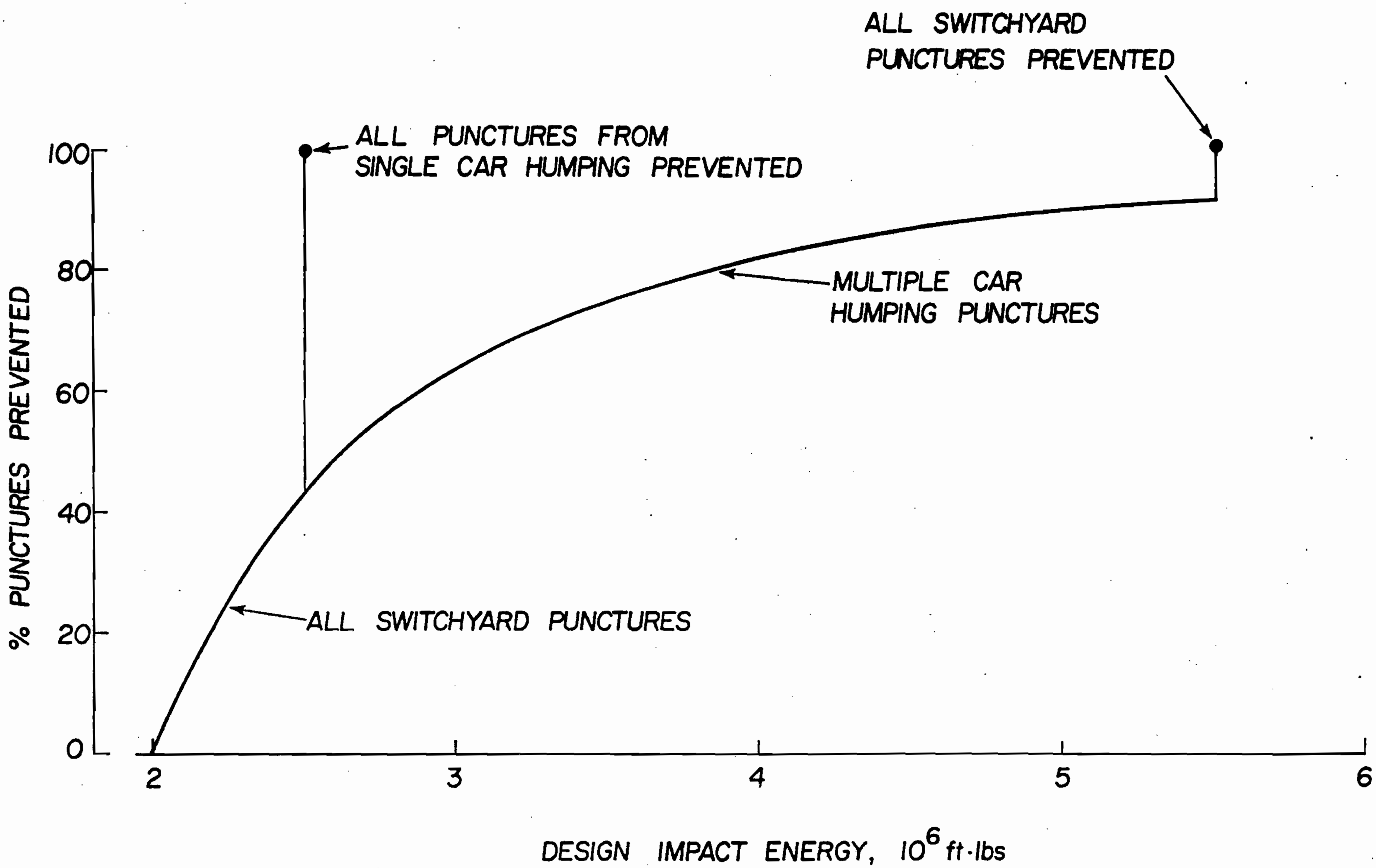

Figure 38 Design impact energy

restriction, the 2.5×10^6 ft-lb energy absorption capacity would prevent only about 60% of the head punctures involving multiple impacting cars. Energy absorption criteria would necessitate uniform testing procedures. These must include a realistic coupler and sill as the penetrating device, with the impact directed slightly above the sill block.

6. CONCLUSIONS

The following are the major conclusions of our tank car safety research project.

6.1 Modeling of override dynamics

The computer train action model of Raidt(1) has been extended in several vital areas to be applicable to coupler override dynamics. Comparisons with experimental data show that the model can provide good predictions of car body motions in the elastic range. Furthermore, when coupled with data on where plastic hinges are likely to form, the model can also be used in the inelastic range. Nevertheless, further refinements of the model in the areas of draft gear, car body elastic modes, and lateral degrees of freedom would provide for much better correlation.

6.2 Override mechanisms

The basic override mechanisms that have been identified are: the double impact (light car pitches up and into back-up car), multiple impact (light car bouncing causes override of back-up car or hammer car), derail (derailing or detrucking creates elevated coupler position), and dynamic squeeze (plastic yielding or breaking of sills and couplers cause squeeze car to override). In practice, a coupler override can involve various combinations of these mechanisms.

6.3 The Phase 15 program

The Phase 15 tests, Table 7, have provided valuable and heretofore unknown information on the mechanisms of override and puncture. Most known override mechanisms were simulated and an actual pressurized tank car head puncture was obtained. The tests are continuing beyond the termination date of this contract with multiple impacting cars and with head shields and shelf couplers.

6.4 Design criteria

Head puncture occurrences in classification yards are usually caused by overspeed impact. The maximum safe speed, below which coupler override (and therefore head puncture) is unlikely to occur, is approximately 8 mph. If impact speeds could be reliably controlled to remain below 8 mph then other operational restrictions or special structural requirements might not be necessary.

Given that impact speeds are difficult to control with great reliability, head puncture occurrences in classification yards could be significantly reduced by imposing an operational restriction and a complementary structural performance requirement in lieu of a speed restriction. The proposed operational restriction is that not more than one hazardous material tank car, or cars following it, should be humped or flat switched onto any one track. The structural requirement is based on the observation that, with the single car switching restriction, although coupler override can occur above approximately 12 mph, the energy available to puncture the tank car head is limited by the weight and maximum velocity of free cars, and by energy losses in override mechanisms. Consequently, it is possible to state a performance criterion to govern the design or retrofitting of tank car heads in terms of a minimum required energy absorption capacity.

The value of the minimum required energy absorption capacity depends on the maximum velocity that runaway cars achieve in classification yards. Authoritative estimates of runaway car velocities are not available at the time of writing this report, however. The subject should be carefully investigated before the recommendations of this report are adopted. If protective devices are to be designed on the basis of a minimum energy absorption capability, as recommended in this report, then it will be necessary to develop standard acceptance test procedures. At the minimum, the following variables will have to be standardized: Number of back-up cars (behind tank car) and slack; pressure in tank car, geometry of indenter, elevation of indenter, impact energy. The present report contains information on tests performed through May, 1976 only. It will be necessary to evaluate the entire Phase 15 testing programs prior to establishing standard test procedures in order to assure that those procedures are based on the best and most complete information available.

Head puncture occurrences in main line accidents can be caused by buff forces sufficiently high to induce plastic buckling in the underframe. The energy levels in such accidents can be so high that no protective device of any kind is fully effective. However, if shelf couplers can be demonstrated to control the buckling mechanism to a significant degree then shelf couplers are expected to provide some protection in such accidents.

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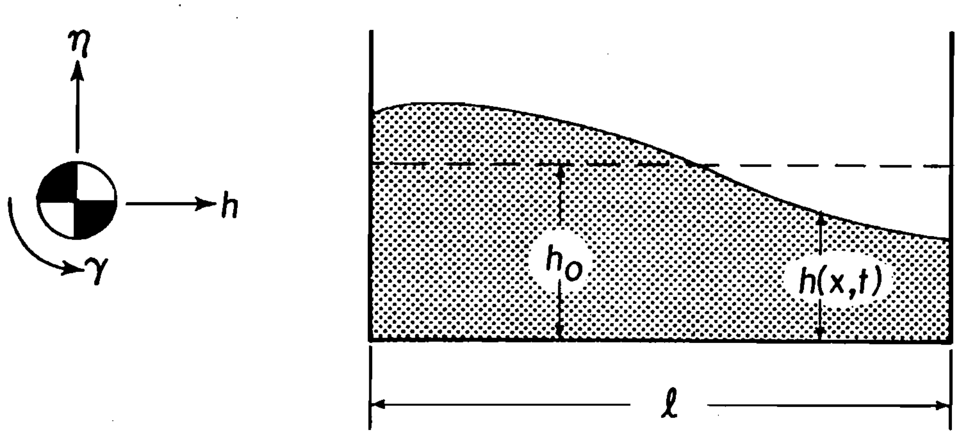
8. APPENDIXES

Appendix A - Sloshing of Liquid Lading

Although no closed form solutions exist for the sloshing of liquids in horizontal cylindrical tanks, solutions do exist for small oscillations of liquids in rectangular tanks. These can be used as an approximation to the partially full cylindrical tank.

We consider a rectangular tank of length ℓ, width w, and filled to a height h_0 with a fluid having density ρ.



Generalized coordinates for the tank motion are defined as vertical translation η, longitudinal translation ζ and rotation γ. Generalized coordinates for the fluid motion with respect to the tank are defined as α_n:

$$h(x,t) = \sum_{n=1}^{\infty} \alpha_n(t) \cos(n\pi x/\ell)$$

The corresponding equations of motion can be obtained from potential flow theory and are given by:

$$(M_0+\rho\ell h_0 w)\,\frac{d^2\eta}{dt^2} = F_\eta$$

$$(M_0+\rho\ell h_0 w)\,\frac{d^2\zeta}{dt^2} - \sum_{n=1}^{\infty} \frac{2\rho w\ell^2}{n^2\pi^2}\,\frac{d^2\alpha_n}{dt^2} = F_\zeta$$

$$\left[I_0 + \frac{\rho h_0 w\ell}{12}\,(h_0^2 - 3\ell^2) + \sum_{n=2}^{\infty} \frac{64\ell^4}{n^5\pi^5}\,\rho \tanh\left(\frac{n\pi h_0}{2\ell}\right)\right] \frac{d^2\gamma}{dt^2}$$

$$+\left[\sum_{n=1}^{\infty} \frac{\rho\ell^2 wh_0}{n^2\pi^2} - \frac{4\rho w\ell^3}{n^3\pi^3} \tanh\left(\frac{n\pi h_0}{2\ell}\right)\right]\frac{d^2\alpha_n}{dt^2} = M_\gamma$$

$$-\frac{2\rho w\ell^2}{n^2\pi^2}\frac{d^2\zeta}{dt^2} + \left[\frac{\rho\ell^2 h_0 w}{n^2\pi^2} - \frac{4\rho\ell^3 w}{n^3\pi^3}\tanh\left(\frac{n h_0}{2\ell}\right)\frac{d^2\gamma}{dt^2} + \frac{\rho\ell^2 w}{2n\pi}\coth\left(\frac{n\pi h_0}{\ell}\right)\right]\frac{d^2\alpha_n}{dt^2} = 0$$

$(n = 1,3,5,\ldots\infty)$

$(\alpha_n = 0$ for n even)

Where F_η, F_ζ, and M_γ are the forces and moment on the tank and M_0 and I_0 are the mass and pitching inertia of the tank alone.

To simulate a cylindrical tank, h_0 and w are chosen such that $\rho\ell h_0 w$ is the correct mass of water and so that the first sloshing frequency is matched.

As a simple illustrative example, consider longitudinal motions and the first mode of sloshing only. The equations then become:

$$M\ddot{\zeta} - \bar{m}\ddot{\alpha} = F$$

$$m\ddot{\alpha} - \bar{m}\ddot{\xi} + k = 0$$

where M = mass of tank and water = $M_0 + \rho\ell wh_0$

$\bar{m}$ = generalized coupling = $2\rho w\ell^2/\pi^2$

m = generalized mass = $\frac{\rho\ell^2 w}{2\pi}\coth(\pi h_0/\ell)$

k = generalized stiffness = $\rho wg\ell/2$

$\alpha = \alpha_1$, $F = F_\zeta$

For impulsive loadings it is more convenient to express the above equations as uncoupled inertially.

$$(M - \bar{m}^2/m)\ddot{\zeta} + \frac{\bar{m}}{m}k\alpha = F$$

$$(\bar{m} - \frac{\bar{m}^3}{Mm})\ddot{\alpha} + \frac{\bar{m}}{m}k\alpha = \frac{\bar{m}^2}{mM}F$$

Thus, in an impulsive collision, the effective mass of the tank is reduced by $\bar{m}^2/m$.

To simulate the rectangular tank in Phase 15, we used the following data.

ℓ = 60 feet

$\rho \ell w h_0$ = 177 kips

Period of motion = 8 seconds

From this data, the equivalent width and depth are

ρw = .42 kips/ft^2

h_0 = 7 ft.

It follows that $\bar{m}^2/m$ = 130 kips. Therefore of the 177 kips of water, only 47 kips (27%) are effective in an impulsive impact.

Appendix B - The Des Moines, Iowa, Accident

On 1 September 1975, on the northern outskirts of Des Moines, Iowa, 14 cars (the 24th through the 37th of a 61 car train) were derailed. The first 11 of the derailed cars were DOT 112-114 tank cars loaded with butane and propane. All but the first of the 11 tank cars burned and four of them exploded.

Investigators from the Federal Railroad Administration, the National Transportation Safety Board (NTSB) and the Chicago, Rock Island and Pacific Railroad Company did not find any mechanical damage that could have caused the derailment. It was determined that the operating procedure was normal. The track was on a descending 1% grade with a 1 degree curve to the left. The train was heading south at approximately 25 miles per hour. The derailment occurred very close to the turnout for a side track, and it was suspected at first that a wheel had "picked" the points of the turnout; but in this derailment the points were not damaged. The track was found to be damaged between the points and frog. The frog, guard rail and trackwork was torn up. The first section of outside (right or west side) rail beyond the point was rolled over, top to the right. The details of the derailment may be found in the document prepared for the National Transportation Board by the Chicago, Rock Island and Pacific Railroad (1).*

W. B. Diboll of Washington University joined the NTSB investigation of October 14 and 15, 1975 and returned to Des Moines on November 5, 1975 to further inspect the cars. The Washington University inspection centered on damage to the cars, especially the couplers and sills. An analysis of the sequence of events of the derailment is presented below.

Analysis of Derailment

The results of a detailed inspection of the accident site and equipment by investigators of the NTSB, the railroad, and other governmental bodies are presented in Reference (1). This investigation did not uncover damage to rolling stock which existed prior to the derailment. The

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*Reference numbers refer to Appendix B only.

train was moving at approximately 25 miles per hour, the locomotives were in 8th notch (full power) and shortly before the derailment the engineer applied a minimum brake reduction. The train of 61 cars was light on the front and heavy on the rear. Of the first 16 cars only four were loaded, and of the remaining 45 cars only 7 were empty, including the caboose. These conditions can lead to excessive lateral to vertical force (L/V) ratios, with resultant rail overturn or derailment. In this case none of the conditions were of sufficient magnitude to lead to excessive L/V for normal track, but the track on which the accident occurred was classified as 30 mph track by the Federal Railroad Administration. No information is available of the values of L/V which will result in rail turnover on this class of track, but it is known from the Track-Train Dynamics Program of the Association of American Railroads (AAR) that values of L/V of 0.6 to 0.8 can result in wheels overclimbing rails. There is the possibility that cross ties in poor condition will allow rail rollover at values of L/V below 0.6.

The inspection on 14 and 15 October of the cars, trucks, switch frog and guard rails did not lead to determination of the cause of derailment. Inspection of the truck bolsters showed a pattern of center pin damage caused by trucks separating from the car body on the derailed cars, except the first. There was great damage to many of the trucks.

The couplers and stub sills of the tank cars were closely examined. The second trip to Des Moines on 5 November by Diboll was mainly for this purpose. Many of the stub sills were broken off and had to be matched to cars. Evidence of jackknifing is clear from the trailing end of car 5 to the last cars which derailed. The last three derailed cars were found in alternate jackknifed positions.

The conclusions from these inspections and the subsequent analysis are shown in Figure B1, which shows the assumed position of jackknifing, starting from the positions of the last three cars after the fire. This alternate jackknife pattern agrees well with the positions of cars after the fire and explosions. The damage to coupler pockets and couplers during the intense buff action and jackknifing can occur in two ways as

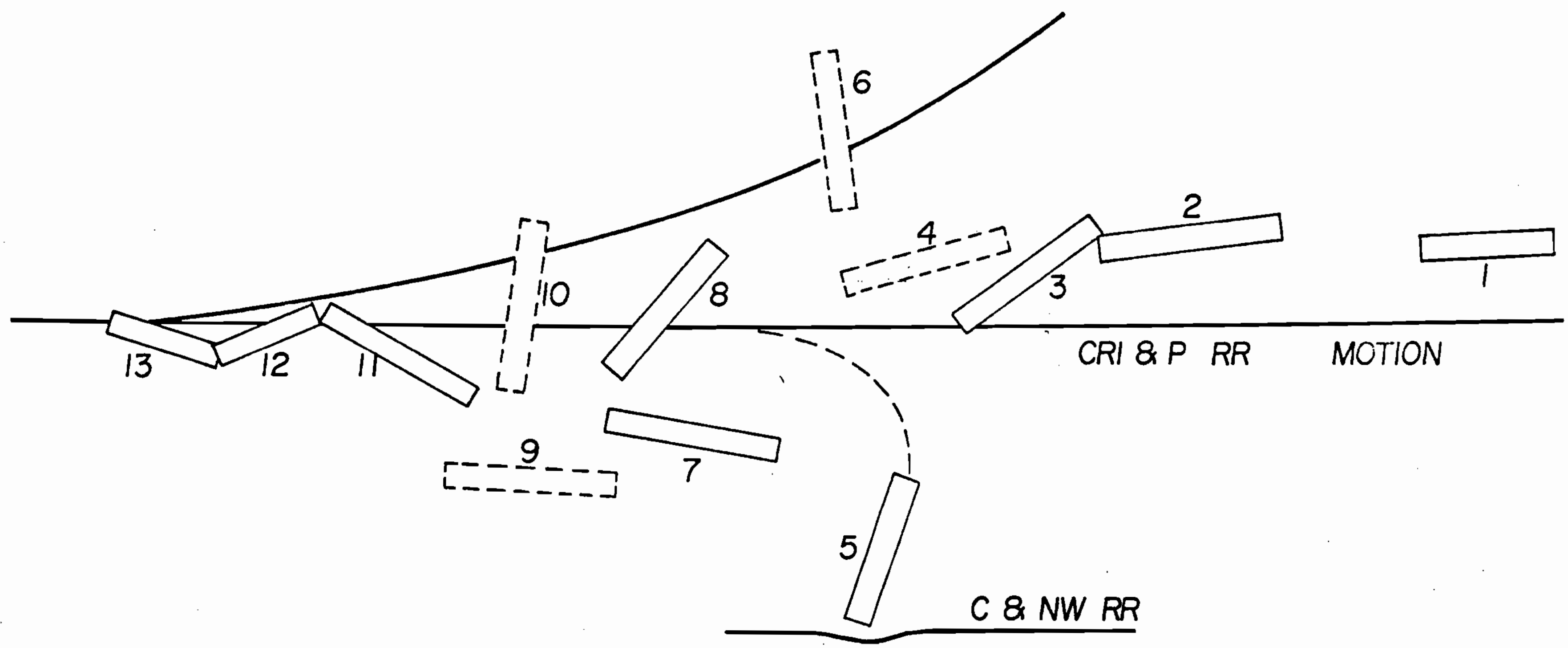


Figure B1 Assumed positions after jackknifing shown in Figure B2. In intense buff action cars can move to opposite sides of the track and leave coupler horn imprints on pockets on opposite sides. In the assumed jackknife configuration the horn - pocket damage did not precisely match, but did indicate that jackknifing did occur. The breaking of stub sill or coupler should occur on only one of mated cars, and this fact, along with the final position of cars, was used to establish the jackknife positions.

In this analysis the numbers of cars 7 and 8 have been interchanged compared to Reference 1. This change is based on photographs and preliminary car locations made shortly after the derailment.

Sequence of Events

In this analysis the cars are numbered from the first derailed car, which was the 24th car in the train of 61 cars.

The sequence of events is assumed to be:

1. Car 3, 4, or 5 derailed first. It has not been possible to determine which one was first.

2. The leading truck of car 5 derailed and separated from the body. The leading end (A end) dug into the track structure, the stub sill "rolled under" and separated. The coupler knuckle has a dent in it, as if made by striking a non-metallic object such as a tie. It is not as sharply indented as if it had hit another metal object.

The tank head of the trailing end of car 5 has clear marks of a coupler impact approximately half way up the tank, which is approximately five feet above the coupler. The marks made by the coupler are those of the lower edge of the knuckle and guard arm, which progress upward for approximately 18 inches and terminate in a ductile penetration of the tank head. This initiated a brittle fracture which resulted in separation of approximately 20% of the upper left portion of the tank head, see Figure B3. As a result of these observations it was concluded that car 6 overrode car 5. The stub sill of the trailing end of car 5 is bent up to approximately 20 degrees. This allowed car 6 to move upward.

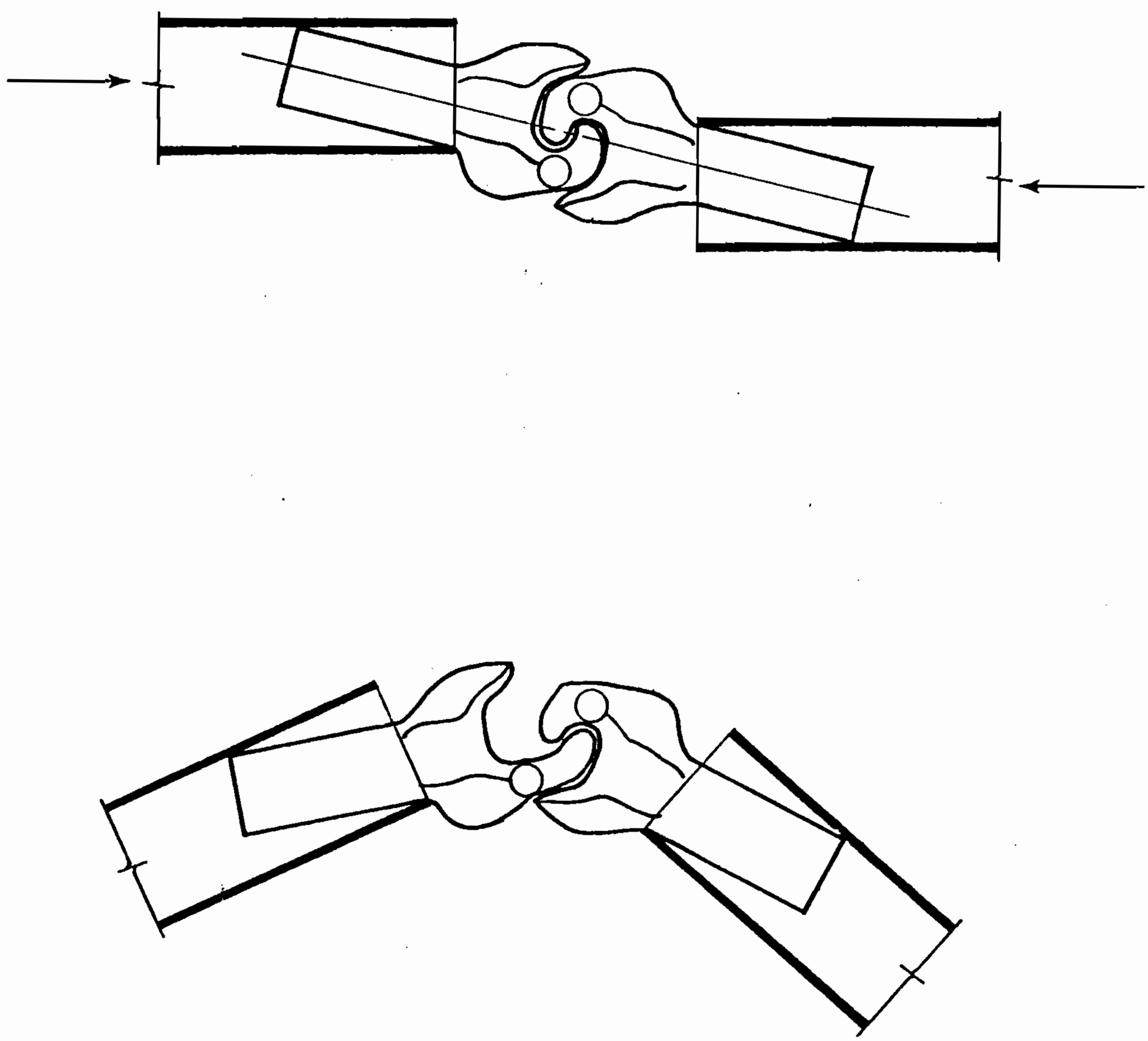
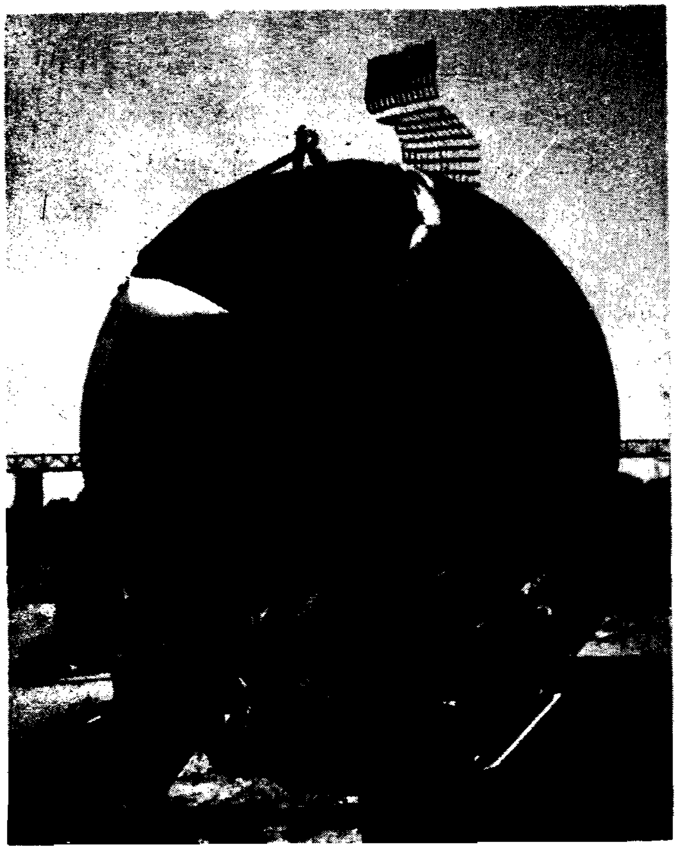


Figure B2 Damage to pockets and couplers



(a)

Figure B3

Damage to trailing end of car 5

(b)

Very likely, this puncture in the trailing end of car 5 was one of the first ruptures which resulted in the release of gas. The gas probably did not ignite immediately because witnesses reported seeing white vapors before an explosion took place. Car 5 rocketed along the ground, leaving marks in a long arc to the West, finally resting with the leading end in the track of an adjacent railroad, Figure B1.

It was surprising to find that a fully loaded tank car had risen more than five feet and punctured the car ahead of it. Similar types of override have been observed and photographed in impact tests at the Transportation Test Center at Pueblo, Colorado, however there the bodies of the overriding cars weighed only 25,000 pounds.

3. After rising and penetrating car 5, the leading end of car 6 dropped down and dug into the trackwork. The stub sill was torn off, the coupler torn out, and the end of the stub sill torn down, Figure B4. The trailing end of car 6 went up the side track, where the car assumed its final position.

4. The indentations of the coupler horns into striker plates indicate that cars 7 through 13 jackknifed. In Figure B5 cars 11 through 13 are shown in the positions in which they were found after the fire.

5. While this was occurring, the forward part of the train continued on, towing the first 3 of the derailed cars, and possibly the fourth. Car 3 derailed, rolled to the left and sequentially rolled cars 2 and 1 to the left. The basis for this conclusion is the observed twisting of the couplers involved plus the lack of horn to striker plate damage in these couplers.

6. Car 4 also derailed and struck car 3 after car 3 had turned over, penetrating the top of 3 near the manway toward the trailing edge.

7. During the derailment some of the cars were punctured, releasing their contents. The gas was ignited, which "cooked" the unpunctured cars until they exploded. Some of the tanks were heated to a temperature at which the material yielded under the stress from the internal pressure resulting in "thinning" of the material and rupture. Some of the flattened plates from the ruptured cars did have thinning, to approximately 1/8 inch.

Figure B4 Stub sill of leading end [illegible]
showing downward def[illegible]

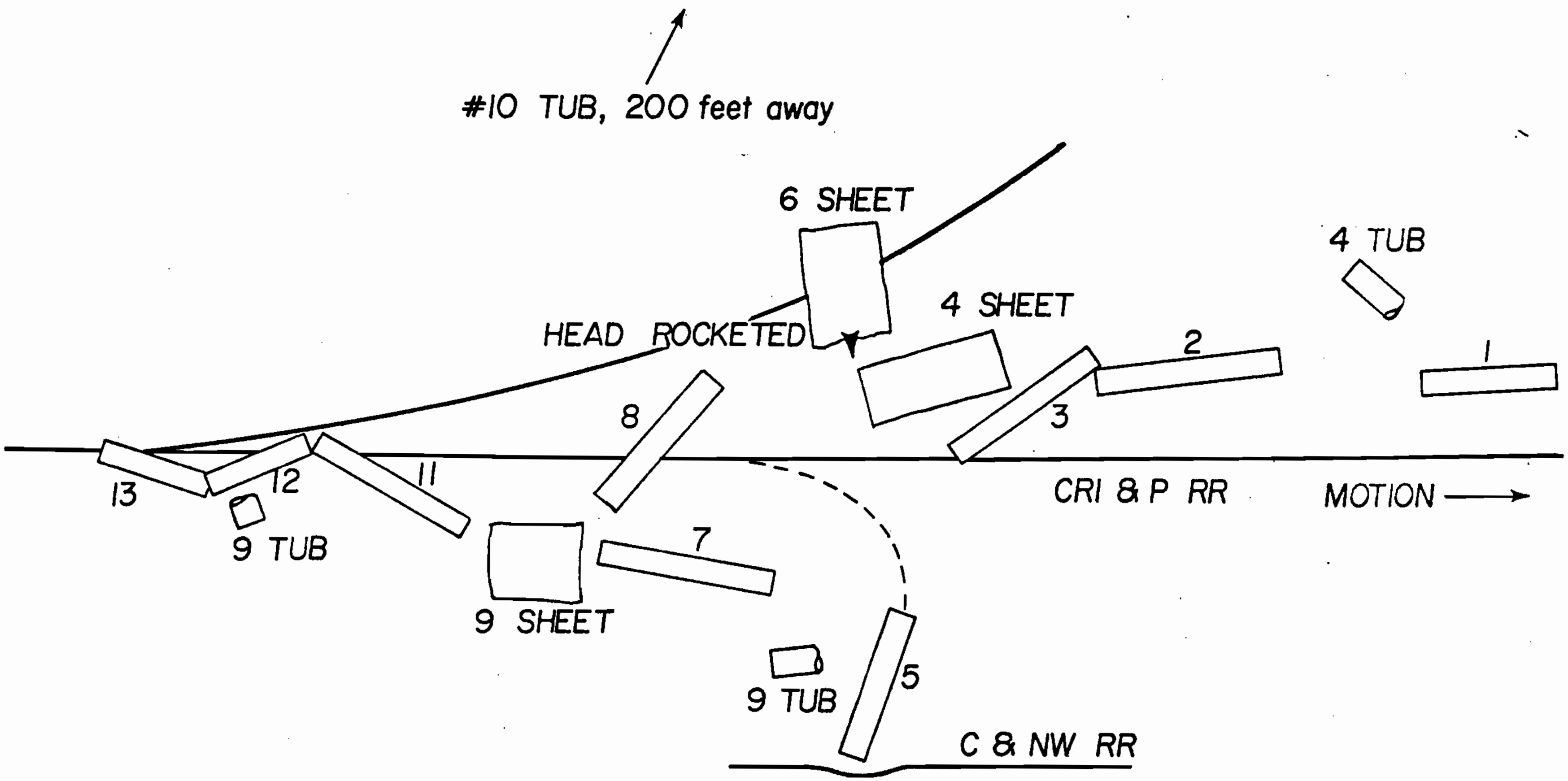


Figure B5 Positions after fire

The tubs of cars 4, 9, 10 had ductile type failures. No brittle failures were evident in the tubs of these three cars.

Cars 4, 6, 9, and 10 exploded, due to external heating by torch action or pool fire.

8. It is known that car 6 was last to explode because video tapes were available which showed the final explosion in which parts were blown into the air. These were located and identified as parts of car 6. Apparently car 10 exploded first (Reference 2) and rocketed approximately 200 feet east where it struck a large sign. Cars 4 and 9 exploded between the times of the explosions of cars 10 and 6, but the order of occurrence cannot be established.

Mechanism of Override of Car 6 into 5

The positions of cars 5 and 6 in Figure B6 show the override mechanism which has been called "pole vaulting". The large amount of energy in the train following car 6 can lift car 6 provided that car 5 has been slowed sufficiently. Apparently car 5 lost its front trucks allowing the forward coupler and sill to dig into the track structure. Evidence indicates that the leading stub sill broke at the rear draft lugs and bent under the car. With the front of Car 5 down, car 6 bent the trailing stub sill of car 5 up. When a sufficient angle was reached the vertical friction force between couplers was exceeded and car 6 "popped up". There was very little damage to the trailing coupler and stub sill of car 5 except for the bending of the stub sill near the bolster.

There are coupler marks at approximately the center of the tank head of the trailing end of car 5. The knuckle impacted first, then the guard arm, and continued to move upward approximately 18 inches where the coupler penetrated the tank head. There was evidence of only a small amount of penetration, plus "pull-out" marks caused by withdrawal.

There was evidence (2) of car 5 having "rocketed" along the ground in a large arc to the right until it struck the adjacent Chicago and Northwestern railroad track approximately at a right angle and displaced it 3 feet out of line.

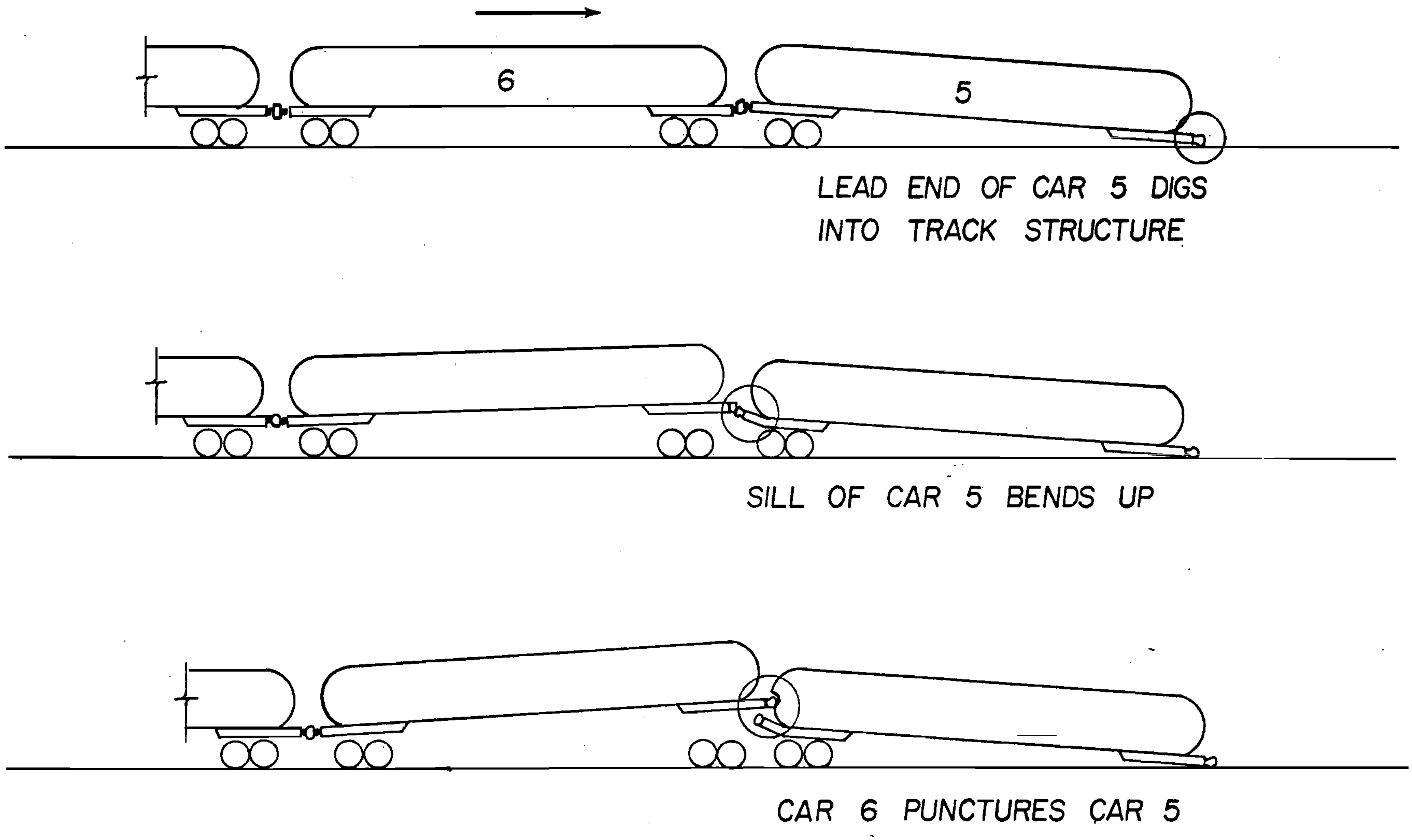


Figure B6 Pole vaulting override mechanism

Effectiveness of E Shelf Couplers

Estimating the effectiveness of the E top-and-bottom shelf coupler in a Des Moines type mainline derailment rests upon the determination of how this type of coupler would have affected the motion of the cars following the first derailment. The vertical restraint of the top and bottom shelves can be effective only in those dynamic actions which involve large (exceeding 7 inches) vertical relative motions of mating couplers.

Immediately following the first derailment, cars 1, 2, 3, and 4 moved together, but car 4 became uncoupled and struck car 3. Shelf couplers probably would have kept car 4 coupled to car 3 preventing car 4 from puncturing car 3. With car 3 not burning, cars 4 and 6 probably would not have been heated enough to cause them to explode. Judging from its position, car 2 would not have supplied enough fuel to cause nearby cars to explode.

In the preceeding analysis of the sequence of events it was concluded that cars 4 and 5 became uncoupled, car 5 lost its leading trucks, dug into the trackwork, and was overriden by car 6. Shelf couplers probably would have kept cars 4 and 5 coupled for a longer period of time and would have allowed braking to be effective for a longer time. This reduces the kinetic energy available in the train thereby reducing further damage. The shelves and shanks of the couplers are not strong enough to support the end of a loaded car after a truck has separated from the car, but a delay while the couplers are yielding is better than having them separate immediately.

When cars remain coupled, the emergency brakes are not applied involuntarily, i.e. braking would depend upon train crew action. In this accident the derailment was sensed by the locomotive crew even before the first fire was observed, so brake action probably would have been initiated quickly.

Once a car has dug into the track, such as is assumed to have happened to cars 5 and 6, there is little that the shelves of the couplers will do to affect jackknifing. Jackknifing is mainly a lateral motion which would not be influenced by the shelves. Extreme lateral bending will fracture stub sills and/or couplers irrespective of the type of coupler.

The high buff forces involved in the start of jackknifing can cause coupler override, which was evidented by car 6 overriding car 5. In this override an E shelf coupler would probably have changed the action only slightly.

The only override other than that between cars 5 and 6 was between cars 10 and 11, resulting in a slight tear in car 11 just above the stub sill. Apparently this occurred at low speed during the last stages of stopping, and was a low energy action which probably could have been prevented by an E shelf coupler. There was little damage to the coupler of car 11, indicating that the override could have been caused by the teetering of car 10 rather than by high buff forces between the cars.

The conclusions concerning the effect of the E shelf couplers had they been installed on all tank cars are:

1. The override of car 11 by car 10 probably would have been prevented, and probably car 1I, 10, and 9 would not have burned or exploded since they were sufficiently far away from the other burning cars that they would not have been heated to the point of rupture.

2. The whole train would have remained coupled longer. With braking initiated by the train crew the train velocity, therefore kinetic energy, would have been reduced, causing less damage than actually occurred.

3. Car 4 would not have punctured car 3, and probably cars 3, 4, 6 would not have burned or exploded.

Effectiveness of Head Shields

The head shields required by the Federal Railroad Administration were designed to prevent head punctures in the lower part of the head where most head punctures had occurred in the past.

In the Des Moines derailment the only place where a half height head shield might have been effective was on car 11. The tear in car 11 apparently was caused by a coupler pushing the head just above the stub sill. A brittle fracture also resulted from the impact, Figure B7.

The only other head penetration was in the override of car 5 by car 6. This tear was above the center of the tank which would have been above the top of the head shield. There was damage to the tank heads of other cars which had exploded, but this damage apparently occurred after the explosion. All other tank penetrations were in locations other than the heads. Conclusions concerning head shields:

1. A head shield would probably have prevented the tear in the head of car 11.

2. No other tank penetrations occurred where a head shield could have been located.

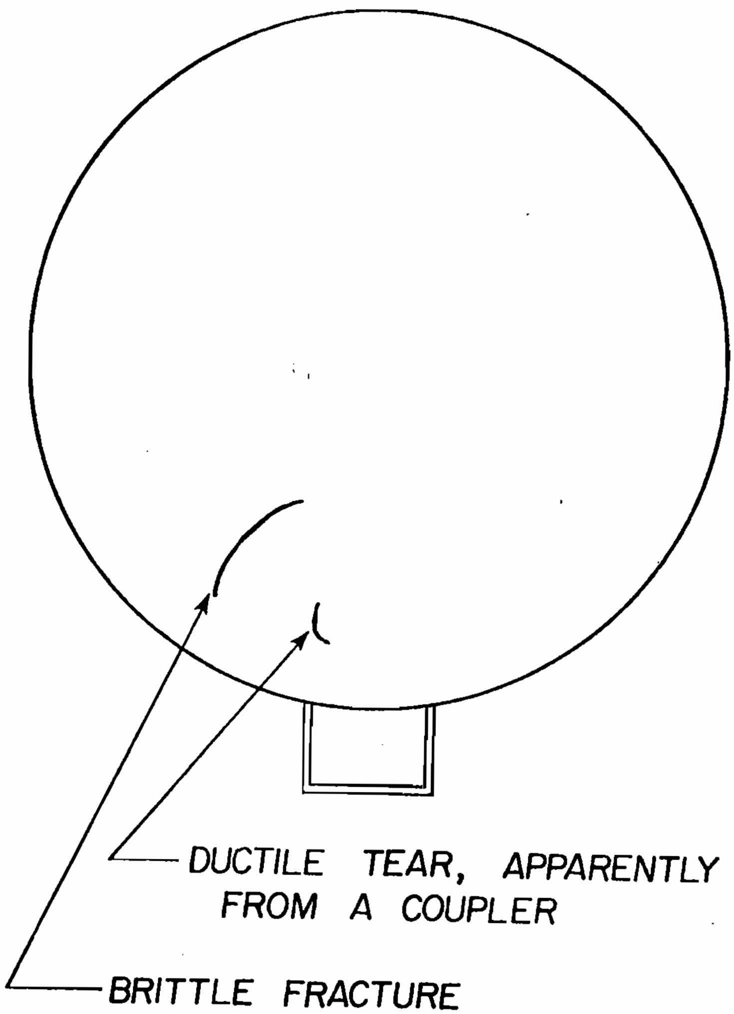


Figure B7 Ductile and brittle fractures in head of car 11

References, Appendix B

1. Chicago, Rock Island and Pacific Railroad, Accident Report, Train 81A31, September 1, 1975, milepost 77.9, Northeast of Des Moines, Iowa.

2. Anonymous, "Railroad Accident Report: Derailment of Tank Cars with Subsequent Fire and Explosion on Chicago, Rock Island and Pacific Railroad Company, Near Des Moines, Iowa, September 1, 1975," NTSB-RAR-76-8, June 1976.

Appendix C - Energy Calculations

1. Available energy

The available energy is calculated in a straightforward manner. The incoming energy is simply $1/2\ m_1 v_1^2$ where m_1 is the mass of the humped cars and v_1 the velocity with appropriate units. Due to conservation of momentum, however, not all of this is available for override and puncture. The generally low value of rolling friction implies that some of the energy must remain as kinetic, and there is an upper limit to the energy that can be dissipated in friction (a perfectly plastic collision). The maximum percentage of energy available is equal to the ratio m_2/m_1+m_2 where m_2 is the mass of the standing cars that are involved in the collision. The test data agree with physical intuition that only the first one or two cars are involved in override and puncture (except where sliding sills are involved). Therefore, for a single incoming car, the worst case would be two equally heavy back-up cars, and the maximum percentage available is $2m_1/m_1+2m_1 = 2/3$. In multiple car humping, the first three moving cars and the first three standing cars might be involved so that the percentage is decreased to $3m_1/3m_1+3m_1 = 1/2$ of the energy of the first three cars.

Another aspect of available energy is the calculation of maximum humping speed of 20 mph. This number is calculated including the effects of rolling friction, initial hump velocity, and aerodynamic forces. The most important parameter, however, proves to be the height of the hump, which is limited. The actual velocity formula is

$$v_f^2 = 2g(h - s\mu) \ / \ (1 - 1/6\ \rho As/m)$$

where v_f = final velocity, g = acceleration of gravity, s = hump distance, μ = coefficient of friction, ρ = density of air, A = flat plate drag area of car, m = mass of car. This calculation is described in section 5.1.

2. Dissipated energy

The energy dissipated in the various phases of impact was calculated from a careful analysis of film data along with the laws of physics as applied to such collisions. The car velocities were calculated in a three-phase procedure. First, films of the impacts were projected onto graph paper and stepped a few frames at a time. At each frame number, reference points on cars, trucks, and couplers were marked on the graph paper and labelled with the appropriate frame number. Targets on cars and tracks were used for scaling, and any oblique angles were taken into consideration. Second, the data from these graphs (both vertical, horizontal, and pitching motions) were plotted versus time for the entire impact sequence. The slopes give velocities of car bodies and trucks, and slope discontinuities imply the points of impact. Coupler face angles were also plotted when override was a consideration. The third phase of calculation involved converting the velocity data into energy data by utilizing the mass properties of the cars involved.

This procedure was performed for most of the Phase 15 impacts. The results are consistent throughout all the data including the train-to-train tests. Basically four numbers were obtained from those calculations. The first is that 8% of available energy is lost in a loose car impact. This is the energy involved in draft gears and car body shaking. If the car leaves the trucks, an additional 5% is lost in bolster friction. This is documented in Reference 3 which shows that the car can move independently of the trucks; but then the trucks catch up, causing a large amount of dissipation even without draft gears. In a triple impact, a second loose car collision occurs so that another 8% is lost. Sloshing is also an important factor. The original calculations showed much more than 8% dissipation. In fact, the dissipation appeared to be higher than allowed from conservation of momentum. When sloshing dynamics were included, however, the consistent value of 8% was obtained.

The third number calculated was $.7 \times 10^6$ ft-lb of energy required for coupler forces to slide and the car body to lift in a dynamic squeeze. At first, this measured value seems rather large; but an analysis of its various components shows that it is very reasonable. To bring sills up to yield requires approximately 240,000 ft-lb of energy. Although this energy is theoretically not "dissipated" it is not available for puncture.

The four draft gears take up about 80,000 ft-lb, and coupler sliding = 1,250,000 lb $\times$ μ $\times$ 1 ft = 300,000 ft-lb. The additional energy required to lift the car body and the energy lost to truck kinetic energy (none of which can cause puncture) add up to the required 700,000 ft-lb. This figure holds true for all squeeze overrides including the train-to-train impact. The fourth number used is 1.0×10^6 ft-lb dissipated in structural yielding in a dynamic squeeze. The movies provided this number also. The physical interpretation is that the 1.25×10^6 lb maximum sill force and the moments due to eccentricity can cause large buckling deformations. The coupler bending energy, calculated to be 250,000 ft-lb must be added to this if a coupler bends.

3. Puncture energy

At the time of writing this report, only two punctures are documented: one from Phase 15 and one from AAR tests. There are also many tests which did not puncture, however, and these are also valuable because they provide a measure of how much energy the head can absorb without puncturing. Hits low on the head (in the vicinity of the head shoe) can absorb up to 1×10^6 ft-lb without puncturing. The Phase 15 puncture absorbed 1.2×10^6 ft-lb. Hits higher on the head can absorb considerably more energy, and the AAR tests showed that 2.0×10^6 ft-lb was required for puncture. This gives the range 1.2×10^6 to 2.0×10^6 ft-lb that is used in the calculations.

It should be pointed out, however, that the calculations are meant to demonstrate a methodology. Any further tests should certainly be used to refine the numbers and obtain better estimates.