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COMPUTER SIMULATION OF TANK CAR HEAD PUNCTURE MECHANISMS

Classification Yard Accidents

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16. Abstract

A number of railroad accidents have been aggravated by couplers puncturing the shell head of hazardous material tank cars. Development of means for identifying possible puncture mechanisms and quantifying the coupler forces involved is the subject of this report. A mathematical model, capable of simulating train action in the vertical plane, has been developed and used for simulation of three classification yard accidents. A detailed description of the model and the results of simulation are presented. The conclusions of this report must be considered tentative until the results of verification studies become available.

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EXECUTIVE SUMMARY

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 to study the effectiveness of coupling systems, particularly those involving shelf couplers, in reducing the probability of head puncture in railroad accidents. Mr. Don Levine is the program manager. The first task under this contract was to acquire an understanding of the mechanisms of coupler override in accident situations. The dynamics of impact in three major classification yard accidents (East St. Louis [1972], Decatur [1974] and Houston [1974]) were simulated on the basis of assumed initial conditions and estimated parameter values by means of a mathematical model developed for the study of longitudinal - vertical train action.

- Results achieved

It was found that the probable coupler override mechanisms were similar in the three accidents. In each case a free-standing or freely moving uncoupled light car was impacted by hazardous material tank cars. Coupling probably did not occur on impact and the light car accelerated away from the impacting cars. The mathematical model indicates detrucking in such

a way that the coupler on the impact end comes to an elevated position. It was concluded that the tank car head was punctured when the tank cars impacted the light car for the second time.

- Utilization of results

Understanding the override mechanisms in the three classification yard accidents analyzed in this report makes it possible to develop performance criteria for devices designed for the protection of hazardous material tank cars in similar situations. Evaluation of the effectiveness of protective devices through impact experiments also requires an understanding of the override mechanisms involved.

- Conclusions

The simulation studies indicated that coupler override would not have occurred in either of the three cases investigated if the light car, whose coupler penetrated the tank car head, had been coupled to back-up cars at the time of impact. Thus the probability of coupler override in overspeed impact situations can be reduced by the operational restriction of not permitting light cars, especially uncoupled light cars, to come in contact with hazardous material tank cars. Once the effectiveness of protective devices has been established, such operational precautions may no longer be necessary.

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1. INTRODUCTION

A research project began at Washington University on March 1, 1974 with the principal objective to develop a rational basis for designing devices suitable for the protection of hazardous material tank cars from head puncture in emergency and derailment situations. The first goal of this project was to acquire an understanding of head puncture mechanisms and, through this understanding, to estimate the direction and magnitude of coupler forces (as time functions) in critical situations. Studies of accident reports lead to the conclusion that puncture mechanisms involve excessive pitching motion of cars, excited by longitudinal coupler forces. A mathematical model, capable of simulating such pitching motion, has been developed. Following a series of test runs, three actual accidents, known as the East St. Louis, Decatur and Houston accidents, were simulated. On the basis of this simulation it was concluded that the sequence of events in the East St. Louis accident was different from the previously assumed ones. This underlines the importance of quantative studies of accident occurrences.

The quantative approach requires that the dynamic parameters of train consists be known with reasonable accuracy. Some of the parameters can be measured readily, others, such as the response of draft gears and ladings to forces applied in rapid succession, require additional experimental and theoretical work. Simulation of the East St. Louis, Decatur and Houston accidents was based on best available estimate of the parameters. In order to verify the model, a series of impact tests, to be conducted under controlled conditions, were designed. The conclusions of this report must be considered tentative until the results of these verification studies become available.

2. BACKGROUND INFORMATION

In derailments and high speed impact situations couplers may become disengaged and one coupler may override the other. Tank cars, carrying hazardous materials, have been punctured by the coupler of an adjacent car or, less frequently, by some other equipment. Many times in such instances the resulting loss of lading contributed to the severity of accident occurrence. Following an exceptionally large number of such accidents in 1969, a cooperative project was initiated by the Association of American Railroads (AAR) and five major tank car builders through the Railway Progress Institute (RPI) with the objective to improve the safety record of hazardous material tank cars. The project is known as the Railroad Tank Car Safety Research and Test Project.

The RPI-AAR project group solicited proposals from interested parties for the mechanical and thermal protection of hazardous material tank cars and received a number of proposals for evaluation. Two proposals for the mechanical protection of tank cars were subsequently selected as worthy of detailed study:

(i) addition of upper and lower shelves to the standard E coupler (fig. 1);

(ii) installation of a shield directly in front of the tank car head to protect that portion of the head which is most frequently punctured by couplers (fig. 2).

Cost-effectiveness studies were conducted in an effort to establish the economic value of these alternatives for existing and new hazardous material tank cars(1)(2).

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* Numbers in brackets refer to references listed in Appendix - References.

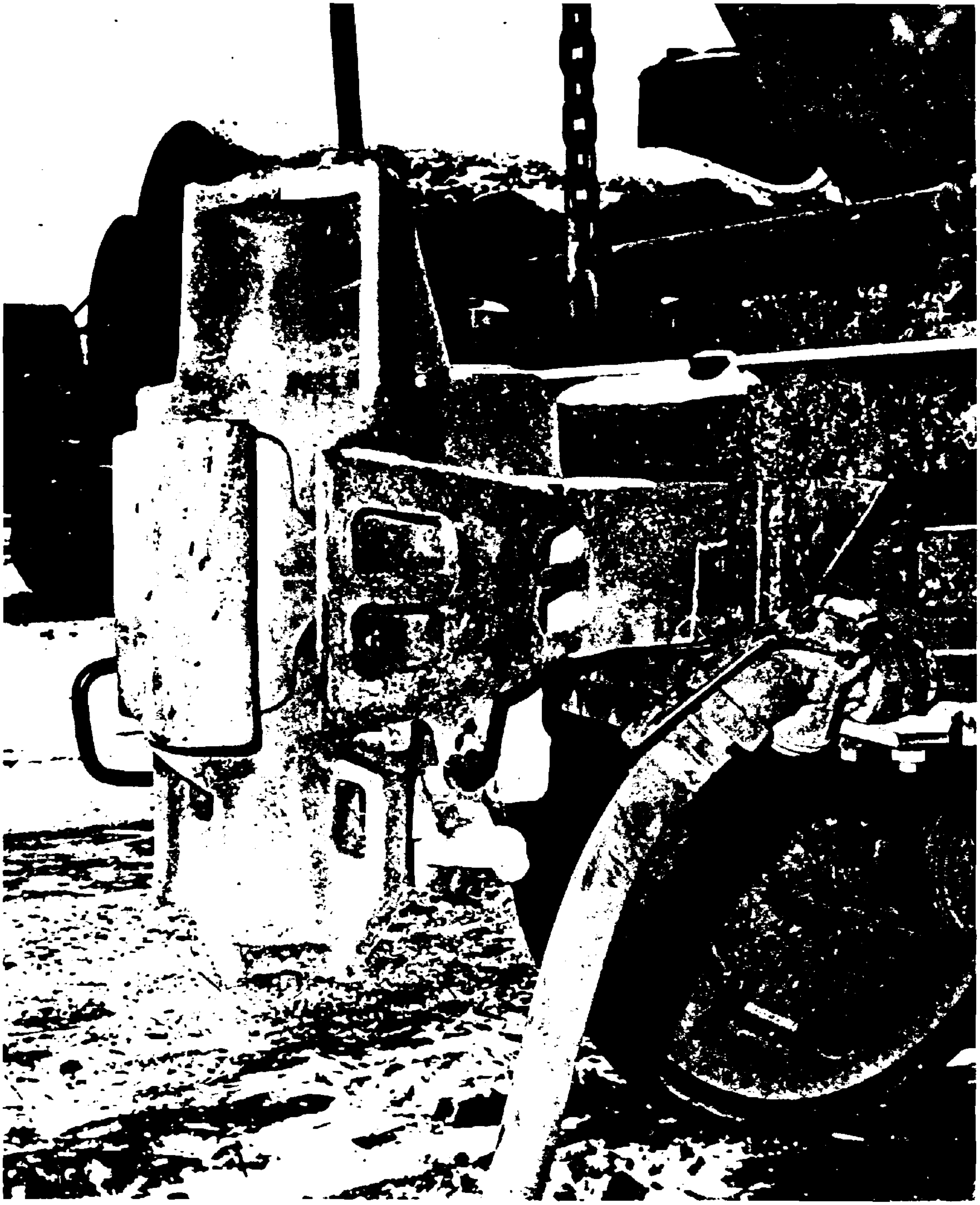


Figure 1
The shelf E Coupler
(Courtesy: Director, Railroad Tank Car
Safety Research and Test Project)

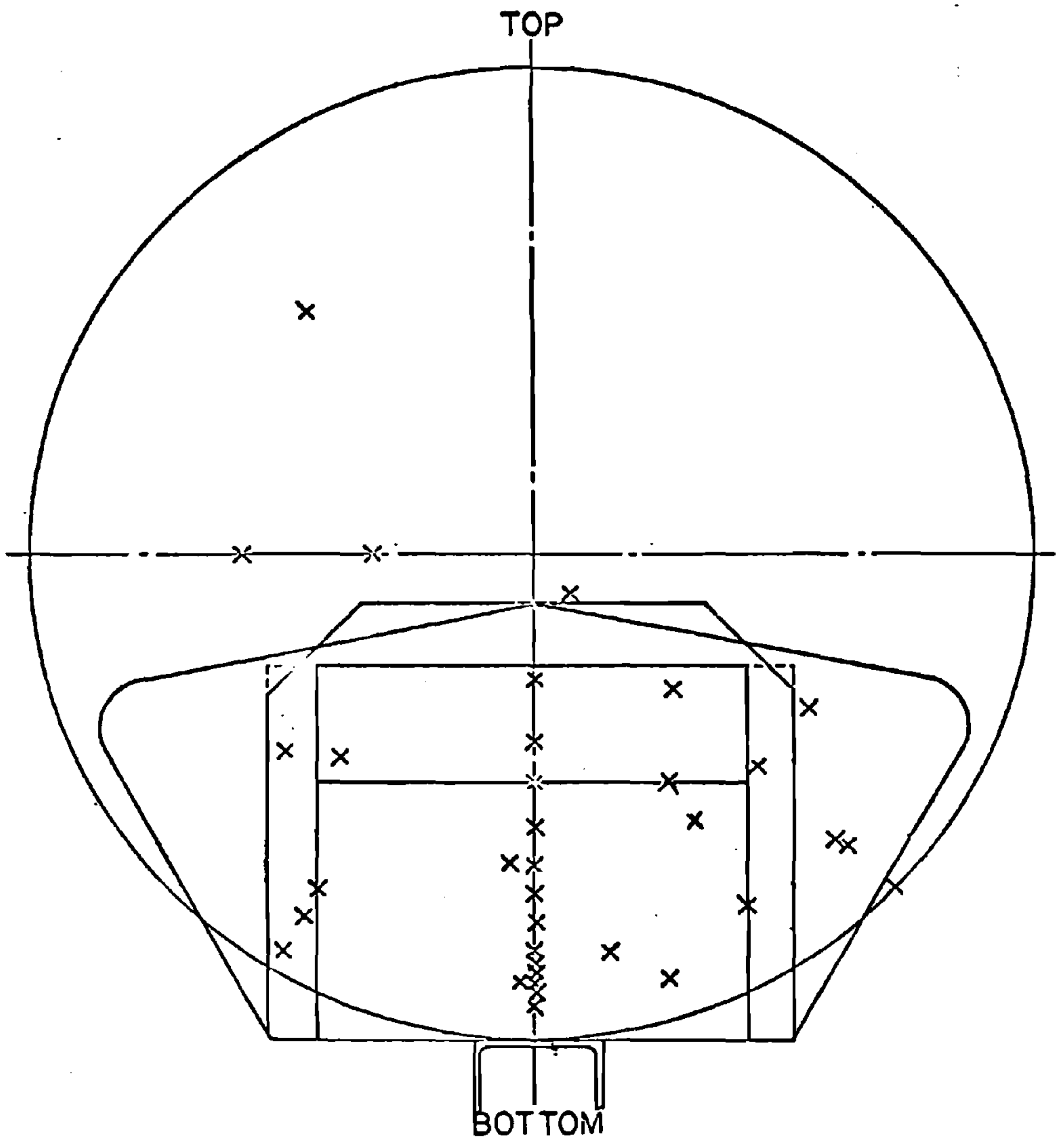


Figure 2

Proposed head shield schemes.
The location of recorded head punctures is marked by X.

Source: Final report, Contract DOT-FR-00035

The RPI-AAR project included a preliminary study of computer simulation of vertical motion during impact[3]. The objective of this study was "to investigate the existence of relative vertical motions between cars and to determine the conditions creating potential for coupler disengagement". The computer model was checked against a test case where a loaded hopper car impacted an empty hopper car, backed up by several loaded hopper cars, at 10 miles per hour. The measured horizontal and vertical impact forces agreed reasonably well with the computer generated forces. However, when applied to the East St. Louis accident simulation, the analytical model did not predict sufficiently large relative vertical coupler motion between cars to explain coupler disengagement and the simulation study was abandoned.

The Federal Railroad Administration awarded a contract to Washington University early in 1974 to study the effectiveness of coupling systems, particular those involving shelf couplers, in reducing the probability of head puncture in railroad accidents. The first task under this contract was to develop a mathematical model, suitable for the simulation of longitudinal-vertical train action in elastic impact situations. Except for planned future refinements, this work has now been completed. The basic assumptions for this model are the same as those in reference (3). For example, the motion is limited to the vertical plane, car bodies are assumed to be rigid with springs representing underframe elasticity; trucks are also rigid bodies, connected to the car body with vertical springs. The time histories are obtained with a Runge-Kutta algorithm based on a state variable form of the equations of motion. This allows response computation up to one second or more after the impact without loss of accuracy due to truncation errors. Reference (3) apparently uses a different algorithm and responses are shown only up to .2 to .3 seconds following the impact. As will be shown later, this time period is, at least in some cases, inadequate to gain an understanding of the train action leading to tank car head punctures. The model described herein was compared against the verification case given in reference (3) and approximately reproduced the measured horizontal and vertical coupler impact forces.

As in reference (3), vertical rail deformation and truck dynamics following the separation of the car body from one of the supporting trucks have been omitted. Given that the impact end of the standing car is moving down following an impact, it is important to determine the maximum vertical spring force at the end of the downward motion, since it is this force that may produce separation of the car body from the truck. The elastic rail deformation has an effect on this maximum vertical spring force and will, therefore, be included in a refinement of the mathematical model. The relative position of the truck after separation of the car body is also important. If, for example, the truck moves relative to the car body while the car body is separated from it, the car body may settle in such a way that the coupler remains in an elevated position which may lead to penetration of the head of the impacting tank car in a subsequent impact. Therefore the mathematical model will be refined by including the relative motion of the truck after separation from the car body.

3. ANALYTICAL MODELS OF TRAIN DYNAMICS

A complete analytical model of train dynamics should treat each car as a six degree-of-freedom system performing longitudinal, lateral and vertical linear motions and angular pitching, rolling and yawing motions. If the lading can move relative to the car, more degrees of freedom must be added. A complete train dynamics model has not been developed as yet. Instead, the problem has been broken up into several parts. The simplest model includes only longitudinal forces and motions. This model incorporates the assumption that the effects of lateral and vertical motions and of angular motions about the longitudinal, vertical and lateral axes have little effect on longitudinal train dynamics. The assumption is approximately satisfied for the dynamics of braking on a straight track. A rather sophisticated treatment of longitudinal train dynamics is given in reference (4).

A more elaborate analytical model is required to gain some understanding of the train action involving impacts which may lead to coupler penetration of a tank car head. Such a model must account for longitudinal, vertical and angular car pitching motions about a lateral axis. Longitudinal accelerations caused by the longitudinal coupler forces are centered at the car body center of gravity. Thus longitudinal inertia forces through the center of gravity excite the car pitching motion about a lateral axis, see fig. 3. The pitching motion about the car center of gravity in turn provides a longitudinal motion of the couplers, see fig. 4. The problem of interacting longitudinal, vertical and angular pitching motion is studied in reference (3) and in this report. The assumptions of the mathematical model are approximately satisfied in situations where vertical coupler override is of interest and (lateral) jackknifing does not occur.

A third partial model where longitudinal, lateral and angular yawing motions are coupled and where vertical, angular pitching and angular rolling motions are neglected has been studied in reference (5) and was successful in correlations with derailment situations with jackknifing. The latter analysis was made possible only by assuming an initial condition of a derailed car with a substantial misalignment of its longitudinal axis with respect to the rails. The derailment process itself leading to this condition cannot be handled with the mathematical model of reference (5). It must be noted that even if a global mathematical model including all six degrees of freedom per car body were available, such a model could not predict in a deterministic way derailment processes, since small changes in initial conditions, such as minor coupler misalignments, will produce large effects in the derailment process. A stochastic treatment of derailment dynamics may be a logical alternative.

The problem here is to model the effects of coupler design - mainly shelf couplers vs. conventional couplers - on the probability of tank car head punctures. Reference (5) assumes that the couplers remain engaged during the events following an initial derailment of one car. The final location and angular position of the cars behind the derailed car were reasonably well predicted in comparison with actual derailment observations. Since the purpose of shelf couplers is to lower the possibility of coupler disengagement, and since the coupler disengagement was not considered in reference (5), one must conclude that shelf couplers will not materially affect the lateral derailment sequences modeled in reference (5). Also, shelf couplers will not materially affect single car impact situations with impact speeds over 12 miles per hour where coupler engagement is unlikely. Experience with numerous impacts has shown that coupler engagement is the less likely to occur the higher the impact speed. The probability of coupler engagement vs. impact speed has not as yet been statistically determined. The value of 12 miles per hour as the limiting speed, beyond which coupler engagement probably will not occur, is an estimate by railroad engineers familiar with many impact situations. It is probable that in the case of the East St. Louis yard accident the first impact did not result in coupler engagement. Rather, longitudinal forces from a first impact excited large pitching motions in an unloaded hopper car (see fig. 3) which apparently resulted in lift-off and loss of truck and subsequently in a head puncture from the second impact of the tank car by the coupler of the unloaded hopper car. It is conceivable that similar events could also occur by braking induced longitudinal impact forces in the initial phases following a derailment of a car, before the lateral dispersion of the trailing cars studied in reference (5) takes place. From all these considerations it seems justified to tackle the problem of tank car head punctures with an analytical model which admits longitudinal, vertical and pitching motions of each car, but which does not consider lateral, rolling and yawing motions. Our goal then is to develop a model of the type used in reference (3) but with sufficient flexibility to accept future test results on coupler and underframe impact characteristics and on lading dynamic characteristics.

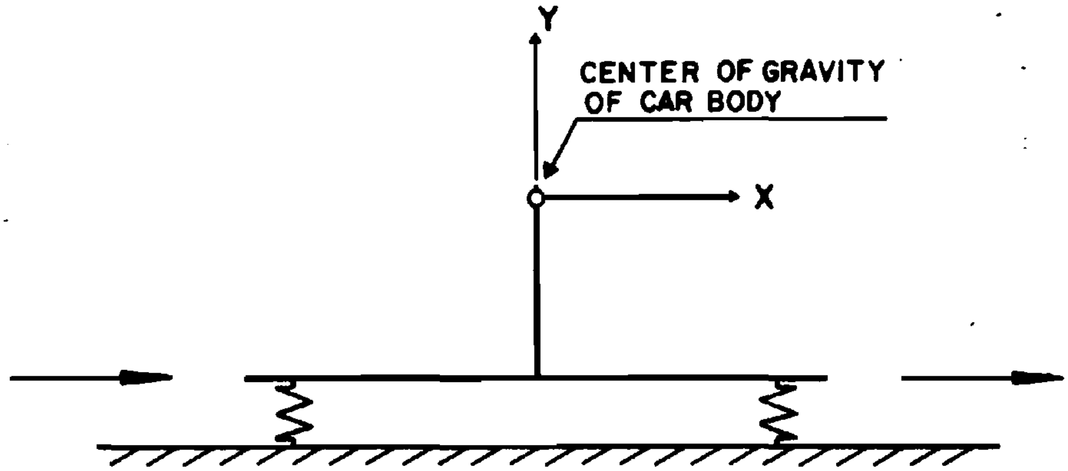


Figure 3

Excitation of pitching motion about the center of gravity by longitudinal coupler forces

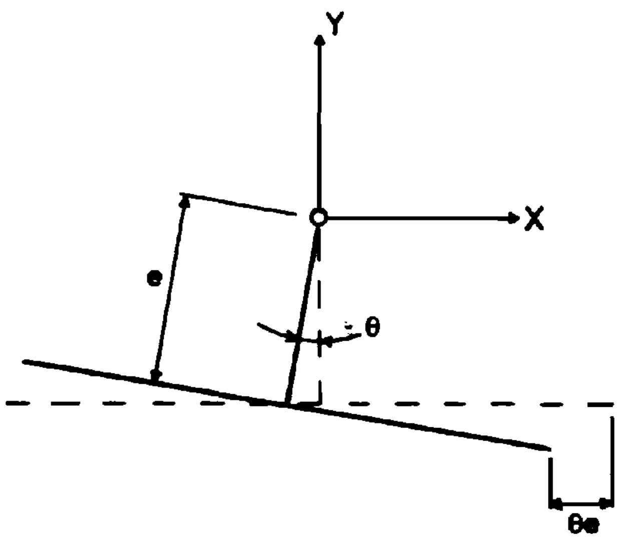


Figure 4

Excitation of longitudinal motion by pitching about the center of gravity

4. THE LONGITUDINAL-VERTICAL TRAIN ACTION MODEL

The model provides a state variable representation of the problem. It admits arbitrary force-displacement hysteresis loops and arbitrary loops for the horizontal coupler forces. The car body and lading are idealized as rigid bodies. The notation is shown on figures 5 and 6.

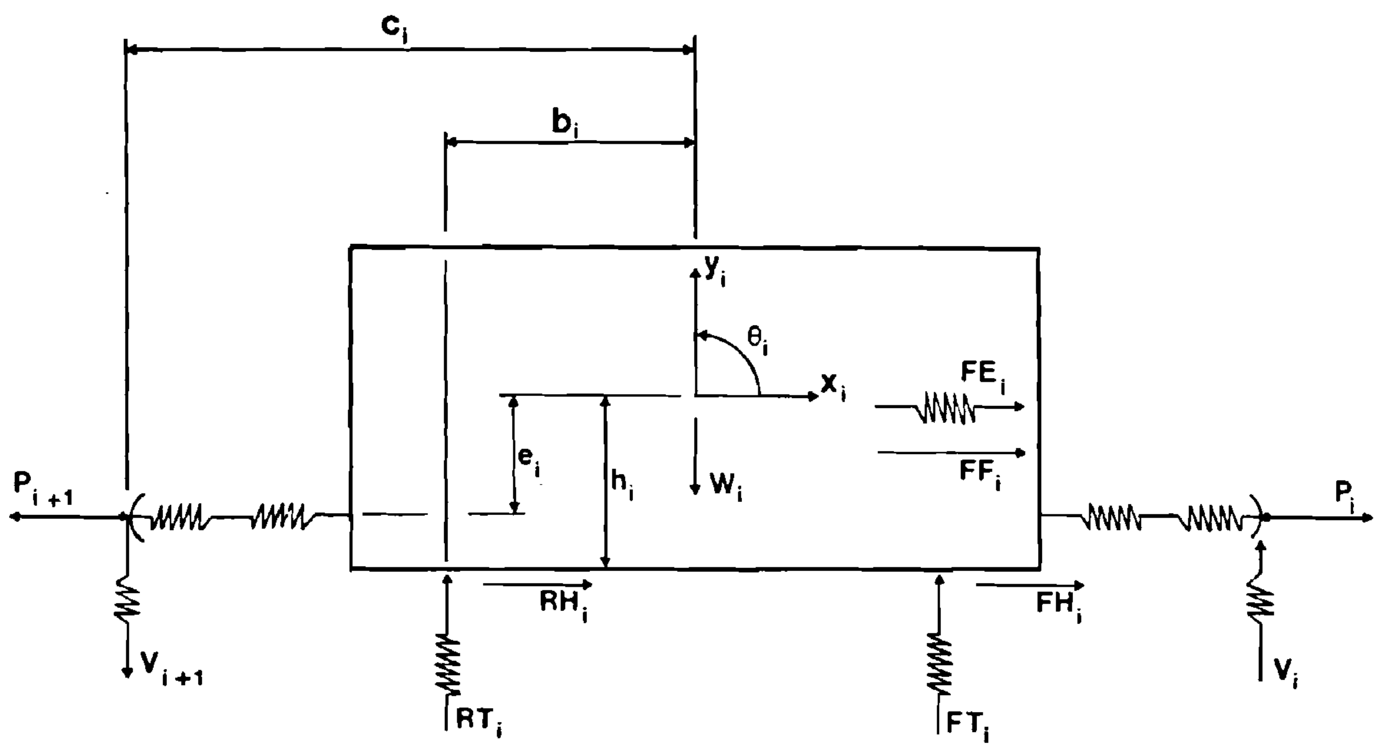


Figure 5

Idealized car body

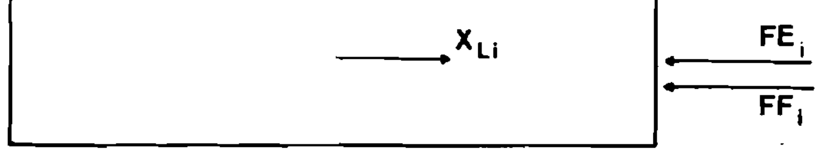


Figure 6

Idealized lading

Definitions:

CHF_i, CHR_i = 1.0 if the car body is in contact with the truck, zero otherwise
FH = Horizontal force from mass of front truck
RH = Horizontal force from mass of rear truck
M_b = Mass of car body
M_t = Mass of truck
M_L = Mass of lading
I_b = Mass moments of inertia of car body and lading
FT = Vertical force of front truck on car body
RT = Vertical force of rear truck on car body
FE = Force of lading spring on car body
FF = Force of lading friction on car body

Equations of motion:

$$FH_i = - CHF_i \, M_{t_i} \, \ddot{X}_i \tag{1}$$

$$RH_i = - CHR_i \, M_{t_i} \, \ddot{X}_i \tag{2}$$

$$[M_{b_i} + (CHF_i + CHR_i) \, M_{t_i}] \, \ddot{X}_i = P_i - P_{i+1} + FE_i + FF_i \tag{3}$$

$$(M_{b_i} + M_{L_i}) \, \ddot{Y}_i = FT_i + RT_i + V_i - V_{i+1} - (M_{b_i} + M_{L_i}) \, g \tag{4}$$

$$\begin{aligned} I_{b_i} \, \ddot{\theta}_i = {} & P_i \, (e_i - c_i \, \theta_i) - P_{i+1} \, (e_i + c_i \, \theta_i) \\ & + V_i \, (c_i + e_i \, \theta_i) + V_{i+1} \, (c_i - e_i \, \theta_i) \\ & + FT_i \, (b_i + h_i \, \theta_i) - RT_i \, (b_i - h_i \, \theta_i) \\ & - CHF_i \, M_{t_i} \, \ddot{X}_i \, (h_i - b_i \, \theta_i) \\ & - CHR_i \, M_{t_i} \, \ddot{X}_i \, (h_i + b_i \, \theta_i) \end{aligned} \tag{5}$$

$$M_L \, \ddot{X}_{L_i} = - FE_i - FF_i \tag{6}$$

The computer program flow charts are given in the Appendix.

Rigid body idealization presumes that the time of propagation of elastic waves through the car body is small as compared with propagation through the draft gear. The propagation velocity through steel is 17,000 feet per second. For a car body length of 60 feet, the time of a compression wave to travel this distance is .0035 seconds. The times involved in train dynamics with draft gear participation are about a tenth of a second. Therefore the car bodies behave essentially as if they were rigid. This is true as long as only elastic deformations occur. For high speed impacts, when plastic deformation becomes important, the car body can no longer be considered rigid.

It is known[6] that at least for some types of freight cars, the bolster ring and pin can suffer sizeable plastic deformation if the horizontal force transmitted from the truck to the car body exceeds 150 Kips. Though the results shown in this report neglect such plastic deformation, a few cases were recomputed assuming a horizontal yield force between truck and car body of 150 Kips. With this assumption the vertical car body motions are substantially increased as compared to those shown in this report. However, the conclusions regarding the coupler override mechanism in the three accidents analyzed in this report are not affected.

The behavior of the draft gear after bottoming in a high speed impact is not well understood. Here it is assumed that horizontal bottoming of the draft gear does not affect vertical slack and vertical coupler spring constant. Some cases were recomputed without vertical coupler slack and with increased vertical coupler spring constant after horizontal draft gear bottoming. Sizeable differences in the car motions after impact were found. Verification and possible corrections of the assumptions made in this report are particularly needed for impact speeds above 10 miles per hour.

5. VERIFICATION OF THE MODEL

Results obtained with the mathematical model described in the preceding section, using the inputs for the verification case of reference (3), have been compared with the results presented in reference (3). For the first oscillation following the impact the agreement is excellent. For the subsequent oscillations some deviations occur which will be discussed later. The test results of the verification case also agree very well with the computed results for the first oscillation, but show substantial deviations for the subsequent oscillations.

5.1 Effect of coupling between longitudinal and pitching motion.

The computer program was exercised with and without coupling between longitudinal and pitching motions. It was found that the omission of the coupling terms had only a small effect on the longitudinal responses, confirming the earlier statement that longitudinal train dynamics can be approximately determined by omitting all other motions. One can interpret this finding in the form of an energy statement. In a train consist, where an unloaded car is located between loaded cars, the energy of the pitching motion of the unloaded car is small as compared to the energy available in the longitudinal oscillations following an impact, a braking action or a derailment of a car. Therefore, a considerable energy reservoir exist in the longitudinal motion from which the pitching oscillation of the unloaded car can be fed. In extreme cases lift-off and loss of truck takes place due to excessive pitching oscillation, and if the adjacent car is a tank car, head puncture by the coupler of the unloaded car may result. It should be noted that the approximate independence of the longitudinal motion from the pitching motion feedback is established for rather large pitching motions of the unloaded car with bottoming of the truck springs on one end and lift-off from the truck springs on the other end, though without loss of truck.

5.2 Effect of lading mobility.

The computer program was exercised with and without lading mobility. It was found that lading mobility in the loaded cars adjacent to the unloaded car has a very large effect on the longitudinal force amplitudes. Without lading mobility these amplitudes following an impact are substantially greater. At present lading mobility, as in reference (3), is modeled by a rigid lading with a linear spring and friction damper between lading and car. In view of the large effect of lading mobility this simple modeling may be inadequate and may be one of the reasons for the discrepancy between test results and computed values shown in reference (3).

5.3 Effect of draft gear characteristics.

The computer program was exercised with a variety of assumptions as to the details of the coupler force-displacement hysteresis loop. It was found that these details have a significant effect on the longitudinal oscillations following an impact. Little is known about these details. All coupler impact tests have been conducted with a single impact resulting in a single loop. For the problem studied where a consist of moving cars impacts a consist of standing cars there is not a single impact but rather a sequence of impacts whereby reversal of the relative coupler motion occurs before complete relaxation of the compressive coupler longitudinal force has taken place. One would assume that at the instant of relative motion reversal from a longitudinal distance increasing to a distance decreasing motion of adjacent cars, a jump in compressive coupler force would take place. In other words, if the draft gear is first compressed, then partially but not completely relaxed as indicated in fig. 7, and then once more compressed, one would expect a jump in force as shown in fig. 7. No such jump was assumed in reference (3). It was found that the subsequent coupler forces and displacements are substantially different when this jump in coupler force is taken into account.

In reference (4) a four parameter dynamic analytical model of a coupler and draft gear is used whereby the four parameters are derived from coupler impact tests by using an identification method based on the equation of motion error method and on a Paynter filter as state variable generator. A Paynter filter is a mathematical device to obtain rates of deflection and accelerations when only deflections are measured. The writers conducted some computer experiments with the Paynter filter and found that it is not useful in the presence of measurement noise. There are, however, other methods to obtain similar results that are free from the limitations of the Paynter filter. Should the four parameter coupler model be proven to represent a satisfactory description of dynamic impact force-displacement loops, this model can be used in future computations. The model predicts a jump in coupler force at the instant of coupler motion reversal before complete relaxation.

5.4 Effect of frequency changes

The computer program was exercised with a variety of assumptions with respect to the ratio of pitching frequency over vertical frequency. Changing this ratio by only 10% resulted in entirely different impact responses, extending from a large climb up on the impact end of a light car to a large climb up on the opposite end of this car. The pitching over vertical frequency ratio appears to be an important parameter which should be studied in verification tests.

The pitching frequency is sensitive to the lading, to the truck spring rate, to the bolster stiffness, to the longitudinal impact force and to the draft gear characteristics. Actually the impact dynamics are not so much affected by the pitching frequency as by the difference between the pitching frequency and the longitudinal frequency. The smaller this difference is, the more energy is pumped from the longitudinal motion into the pitching motion, and the more likely is a separation of the car body from the truck. A recently performed analysis showed for example that longitudinal draft gear slack can have a large effect on the pitching motion if it brings the longitudinal frequency closer to the pitching frequency and thereby allows a large energy transfer to the pitching motion.

5.5 Conclusions of model verification

A computer program has been developed for the combined longitudinal, vertical and pitching motions of a train consist following an impact. With the inputs of the verification case of reference (3) the outputs are in general agreement with those presented in reference (3). Differences are due to differences in the assumed coupler force-displacement loops. Sensitivity studies performed with the computer program have indicated three areas where additional information is needed. First, the important coupler-underframe force-displacement dynamic characteristics following an impact must be better defined, in particular for a sequence of impacts without intermediate complete relaxation. Second, the lading mobility must be better defined, since it was found to have a substantial effect on the final result. Third, more information is required on the limit of the horizontal force between truck and car body.

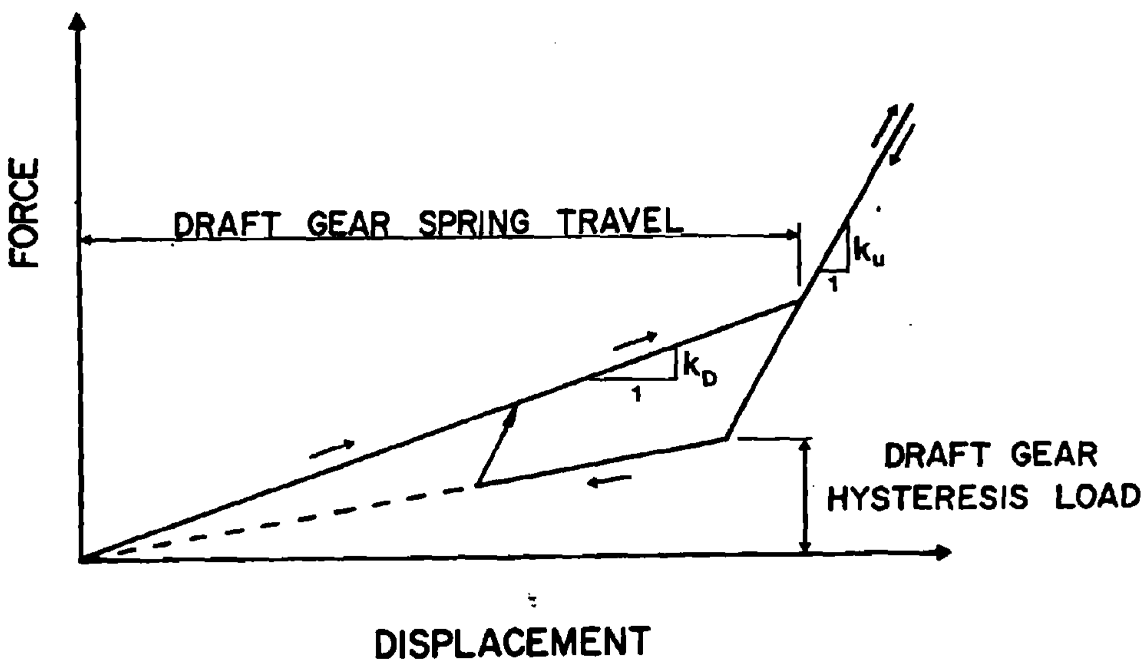


Figure 7

Loading, unloading and immediate reloading of draft gear.

k_D: draft gear spring constant

k_U: car underframe spring constant

6. THE EAST ST. LOUIS ACCIDENT

An accident, involving tank car head puncture, occurred in East St. Louis, Illinois in January, 1972. This accident was selected for detailed study because good documentation is available and, in this case, derailment did not occur. Consequently, the configuration of cars and other evidences of impact remained unobscured.

The accident occurred during humping operations. Three loaded tank cars impacted a standing empty hopper car, then this group impacted a string of box cars. It is not known whether the empty hopper car coupled with the impacting tank cars. Events leading to head puncture were reconstructed differently by two investigating teams, those of the National Transportation Safety Board and the RPI/AAR Railroad Tank Car Safety Research and Test Committee. The sequence of events presented by these teams is detailed in the following sections.

6.1 National Transportation Safety Board Report

The National Transportation Safety Board's Report (NTSB-RAR-73-1)[8] presents the sequence of events to be the following: Three coupled loaded tank cars left the hump with a velocity of 16.5 miles per hour. They travelled approximately 350 feet and struck a single stationary empty hopper car. The coupler of the hopper car overrode the coupler of the first tank car and caused a small head puncture. The four cars proceeded down the track and were hit by a fourth loaded tank car which had been released from the hump at a velocity of 17.5 miles per hour. The five cars proceeded for approximately 1000 feet beyond the first impact where they struck the string of stationary cars being made up into a train, at which time the puncture was made larger. A vapor cloud formed, moved downwind, and an explosion occurred shortly thereafter, ignited by a source other than the accident impact.

6.2 RPI/AAR Tank Car Safety Research and Test Committee Inspection

The RPI/AAR Tank Car Safety Research and Test Committee did not issue an official report on this accident. The Committee's files were made available to the writers, however, and a member of the investigating team was interviewed. From these sources, the following sequence of events was assembled:

Three coupled loaded tank cars left the hump with a velocity of 16 miles per hour. They travelled 350 feet, coupled with the stationary empty hopper car at 6:14 a.m., proceeded as a unit for 1000 feet and at 6:16 a.m. struck the cut of cars being made into a train. At this time the empty hopper car pitched up and caused a head puncture. A fourth loaded tank car had subsequently been released from the hump with a velocity of 17.5 miles per hour and caused a final impact at 6:18 a.m. An explosion of leaking contents occurred at this time.

6.3 Simulation of the accident

Based on considerations of possible head puncture mechanisms, five cases were selected for simulation studies:

(a) Three loaded tank cars and coupled empty hopper car, moving at 16 miles per hour, impact three standing box cars. (Case a, page 21)

(b) Same as case (a) except five standing box cars are impacted (Case b, page 22)

(c) Three loaded tank cars moving at 16 miles per hour, impact standing empty hopper car. Standard E couplers on all cars. Coupling is effected between the leading tank car and hopper car on impact. (Case c, page 23)

(d) Same as case (c) except the tank cars are equipped with shelf E couplers. (Case d, page 28)

(e) Same as case (d) except coupling is not effected between the leading tank car and hopper car on impact. (Case e, page 33)

The parameters used in the simulation studies are given in Table 1. (Page 24)

<u>Case (a)</u>

Three loaded tank cars and coupled empty hopper car, moving at 16 miles per hour, impact three standing box cars.

This case was studied by Raidt[3] who found that the pitch of the empty hopper car was not sufficient to clear the tank car coupler. Raidt used the value of 55,000 pounds for the weight of the empty hopper car. The National Transportation Safety Board report[8] gives the weight as 40,900 pounds. When corresponding values for the mass and moment of inertia of the hopper car body were used, the pitch up of the hopper was sufficient to clear the coupler of the tank car. It is very important to note that the excessive pitch up occurred on the second oscillation rather than the first. Raidt's simulation extended only to the first oscillation.

Raidt had used a coefficient of friction of 0.01 between the coupler faces and the same value was selected at first for the present simulation. The low coefficient of friction was used by Raidt in order to reduce the vertical forces thereby to compensate for what he considered to be unrealistically high longitudinal coupler forces. The longitudinal coupler forces, over three million pounds, were not considered excessive by the writers because of the very short time of application of approximately 0.05 seconds. The coefficient of friction was changed to a more realistic value of 0.2 which is representative of dry friction between steel and steel. With this value the pitch of the hopper car was not sufficient to clear the coupler of the tank car. The greatest pitch again occurred on the second oscillation. It must then be concluded that the excessive pitch found before was due to the unrealistic choice of coefficient of friction between the coupler faces.

Case (b)

Three loaded tank cars and coupled empty hopper car, moving at 16 miles per hour, impact five standing box cars.

Because in the previous simulation the pitching motion had built up over two cycles, it seemed logical to think that more back up cars would cause larger coupler forces and possibly a larger pitching motion. Two simulations with five stationary cars were conducted.

The first simulation was with box cars, the first and third loaded, the others empty. Sufficient pitch up of the hopper was not achieved to lead to coupler penetration of the tank car.

The second simulation was with the following order of stationary cars: loaded box, empty box, loaded box, two loaded large tank cars. Lift off of the hopper car at the tank car end was not sufficient to clear the coupler of the tank car, but lift off at the opposite end of the hopper was sufficient to clear the coupler of the adjacent box car.

Case (c)

Three loaded tank cars, moving at 16 miles per hour, impact standing empty hopper car. Standard E couplers on all cars. Coupling is effected between the leading tank car and hopper car on impact.

With the earlier observation that the pitching motion builds up over two cycles came the realization that an excessive pitch up could have occurred when the three tank cars hit the standing hopper car. If the hopper car lifted off of its trucks it could be in position for its coupler to penetrate the head of the first tank car during one of the subsequent impacts.

This situation was simulated for standard E couplers, using the same numbers as used by Raidt except for the previously mentioned reduction of the mass of the hopper car and increase of the coupler coefficient of friction. The coupler rise of the hopper car was 22 inches at the trailing (impact) end, on the second oscillation, which is twice that necessary for separation. Also, the hopper car center plate at the impact end lifted 17 inches on the second oscillation, well beyond the ten inches needed to clear the center pin. These conditions could easily lead to positioning for head puncture during a later impact. The leading (free) end at the center plate lifted eight inches on first and second oscillations, enough to clear a short center pin, and more than enough to lift off the center plate. The maximum vertical coupler force was 75,000 pounds. The maximum longitudinal coupler force was 2,400,000 pounds.

The computed force-time and displacement-time functions are shown in figures 8a to 8e. The parameters are listed in Table 1.

Table 1

Simulation Parameters East St. Louis Accident

Car Number	Impacting Tank Cars* 1	2	3	Hopper Car 4
Car weight (body only), kips	85.5	86.5	85.5	25.9
Lading weight, kips	132.9	135.7	132.9	0
Weight of one truck, kips	10.0	10.0	10.0	7.5
Half length between coupler faces, inches	440.0	383.0	440.0	231.0
Half length between truck centers, inches	354.0	311.5	354.0	147.0
C. G. height above centerplate, inches	50.0	50.0	50.0	50.0
C. G. height above coupler, inches	40.0	40.0	40.0	40.0
Mass moment of inertia, kip-in-sec^2	31,000	28,800	31,000	990
Truck spring const., kips per inch	63.8	63.8	63.8	63.8
Bolster spring const., kips per inch	3,000	3,000	3,000	3,000
Car underframe spring const., kips per inch	2,500	2,500	2,500	2,500
Draft gear spring const., kips per inch	86.4	86.4	86.4	86.4
Draft gear spring travel, inches	2.5	2.5	2.5	2.5
Truck spring travel, inches	2.5	2.5	2.5	2.5
Lading spring const., kips per inch	246.0	280.0	246.0	-
Lading friction coef.	.01	.01	.01	-
Vertical coupler spring const., kips per inch	50 & 400	50 & 400	50 & 400	50 & 400
Vertical coupler slack, inches	1.2	1.2	1.2	1.2
Draft gear hysteresis load, kips	40.0	40.0	40.0	40.0
Friction coef. between couplers	.20	.20	.20	.20
Initial car velocity, MPH	16.0	16.0	16.0	0

* Car No. 3 is the leading tank car

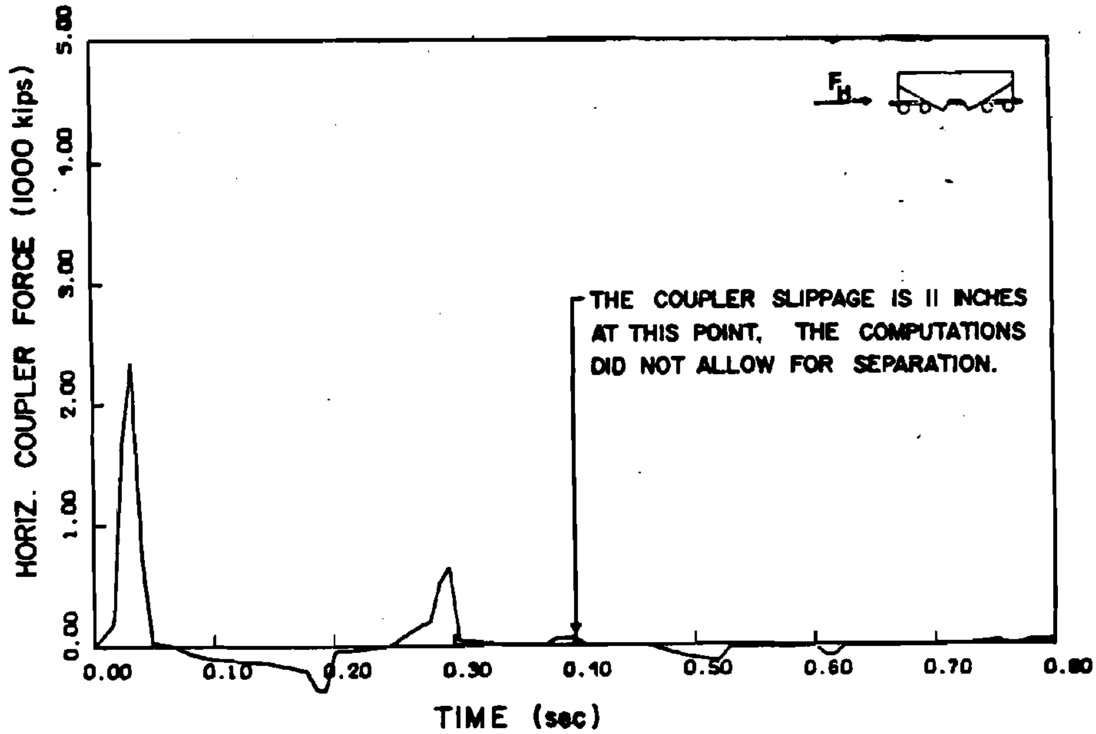


Figure 8a

Horizontal coupler force at
trailing (impact) end of hopper car

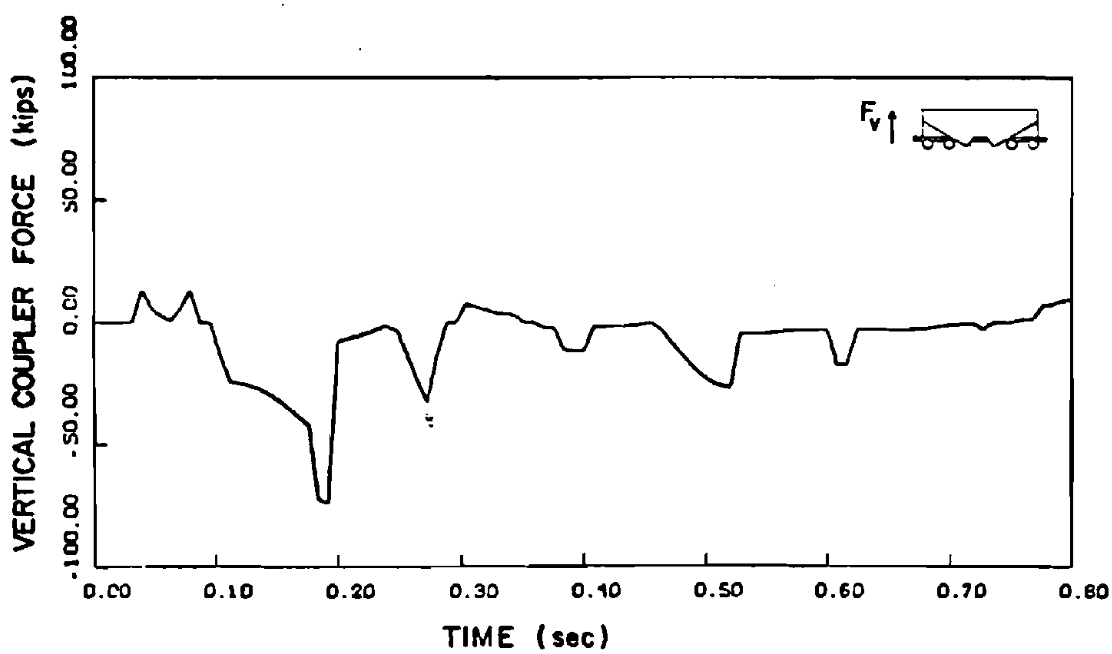


Figure 8b

Vertical coupler force at
trailing (impact) end of hopper car

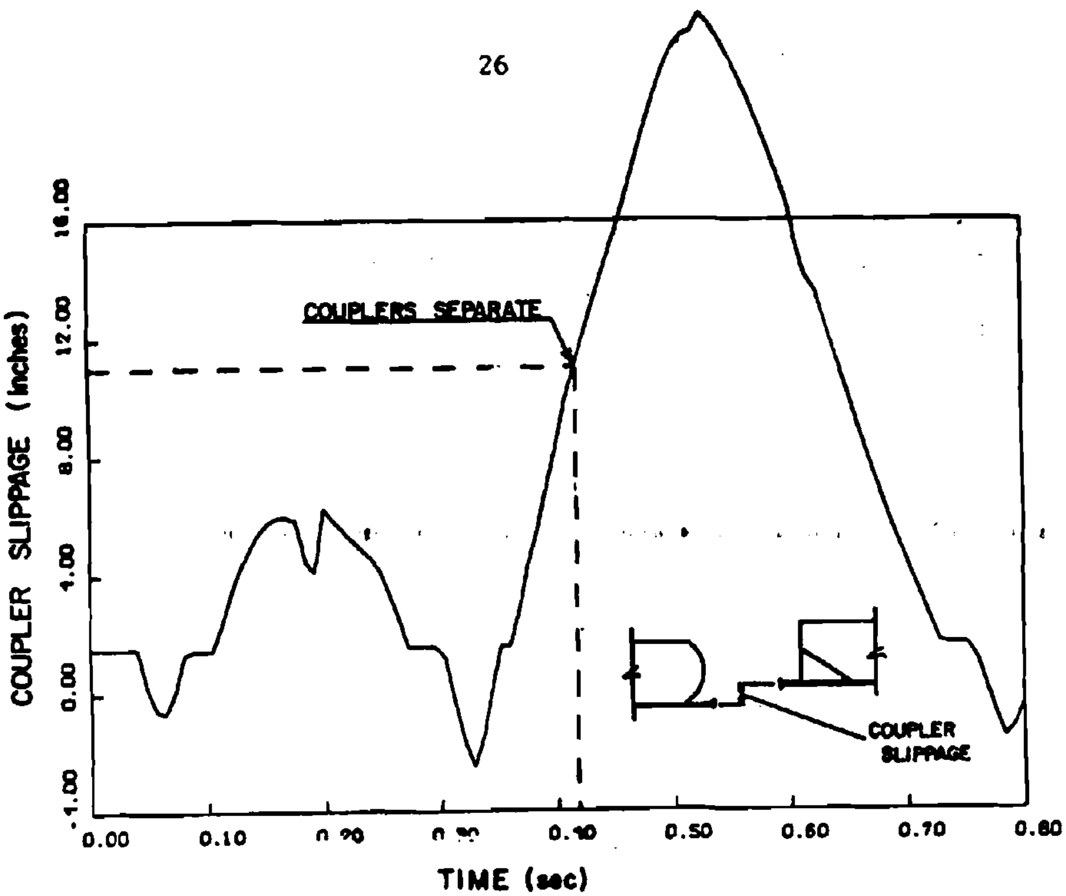


Figure 8c

Coupler slippage at
trailing (impact) end of hopper car

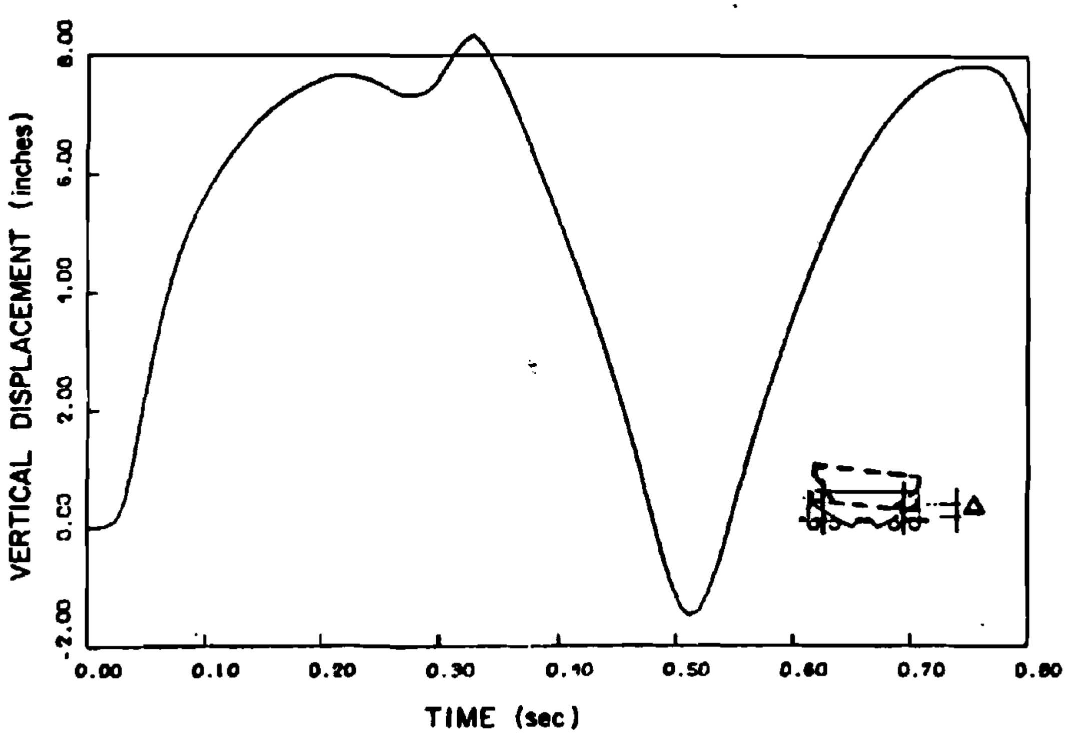


Figure 8d
Car body center plate displacement
at leading (free) end of hopper car

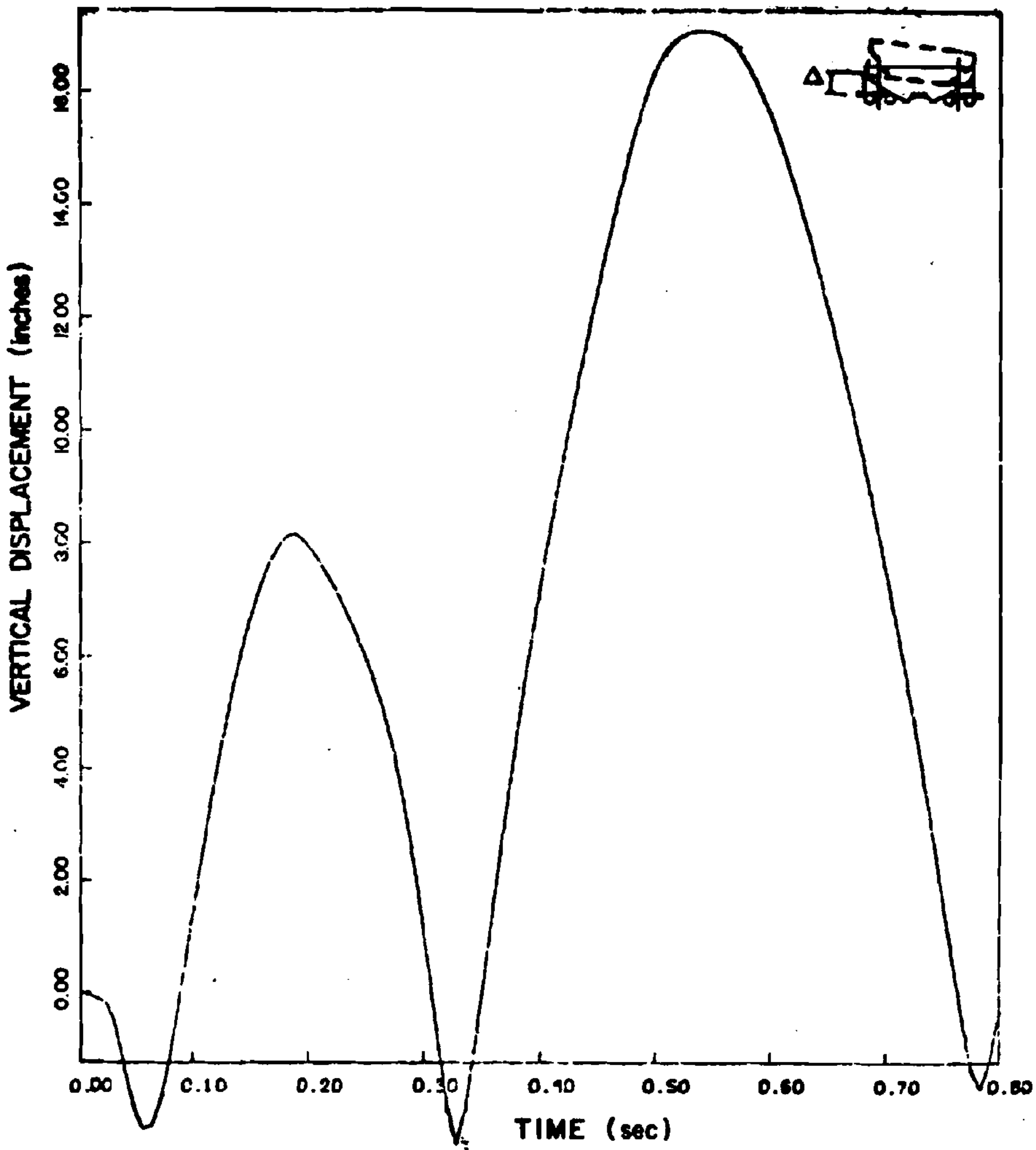


Figure 8e

Car body center plate displacement
at trailing (impact) end of hopper car

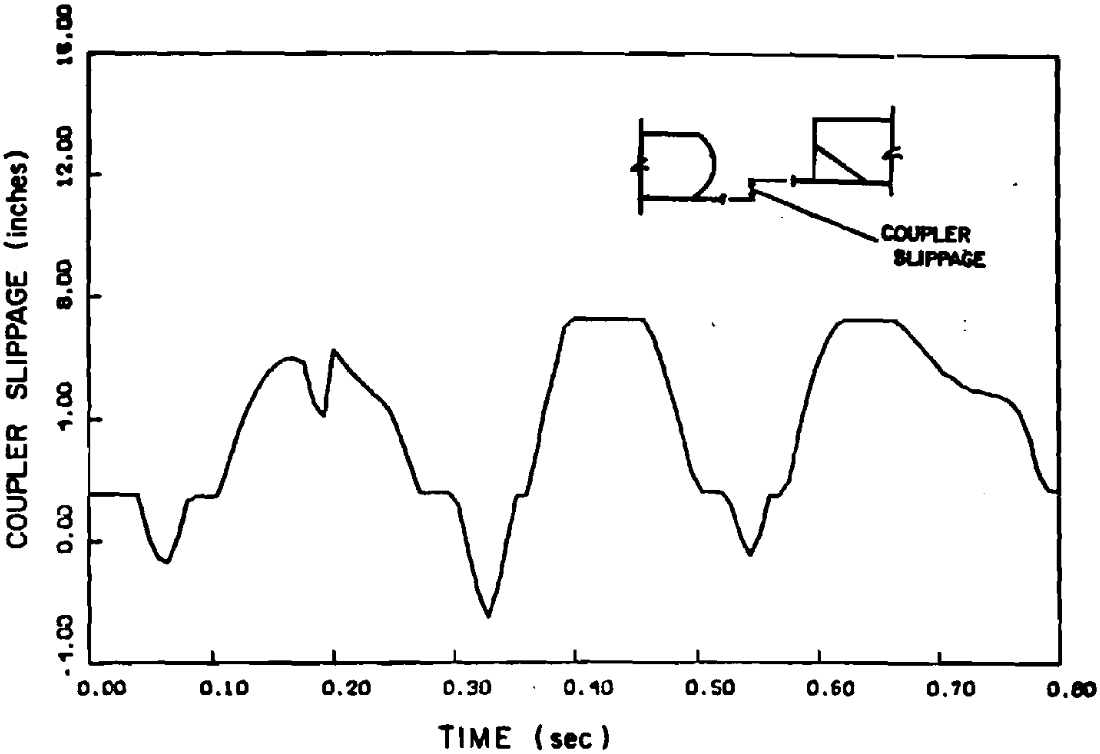
Case (d)

Three loaded tank cars, moving at 16 miles per hour, impact standing empty hopper car. Shelf E couplers on tank cars. Coupling is effected between the leading tank car and hopper car on impact.

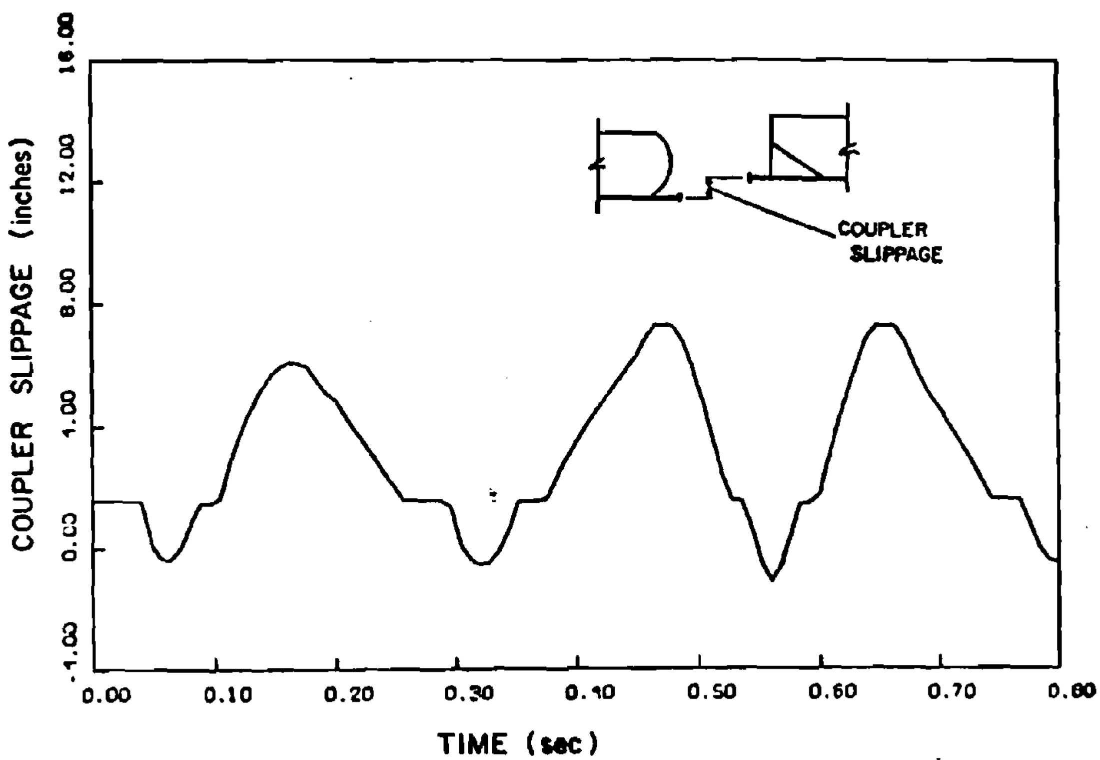
The shelf E coupler limits the motion of the mating E coupler to 7.25 inches above and below the aligned position. The resulting vertical coupler force is sensitive to the vertical coupler spring rate. The shelf E coupler with a soft mounting on the underframe (vertical coupler spring rate of 50 kips per inch) will prevent coupler separation at the impact end but will not prevent lift off from the truck at the free end. The maximum vertical coupler force in this case is 85,000 pounds, or only 13 per-cent greater than in case (c).

The shelf E coupler with a stiff mounting on the underframe (vertical coupler spring rate of 400 kips per inch) will prevent coupler separation and will prevent lift off from the trucks. The maximum vertical coupler force in this case is 150,000 pounds, or twice as large as in case (c). The longitudinal coupler forces were virtually unchanged from case (c).

The effect of vertical coupler spring rate on coupler slippage, vertical coupler force and car body center plate displacements is illustrated on figures 9a to 9d.

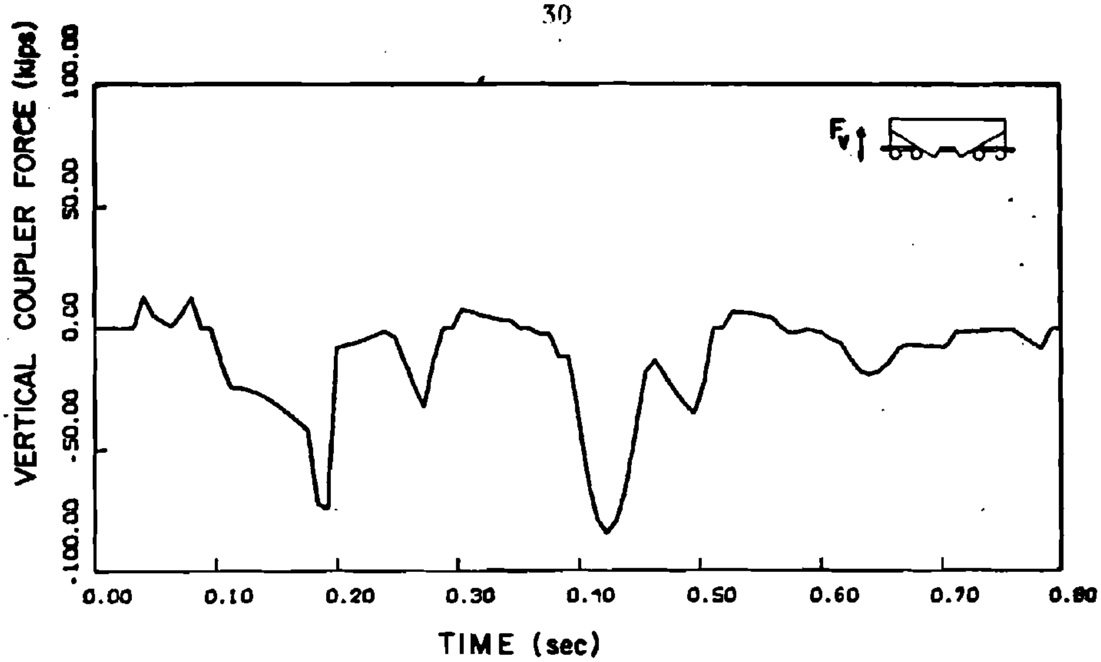


Vertical coupler spring rate: 50 kips/inch

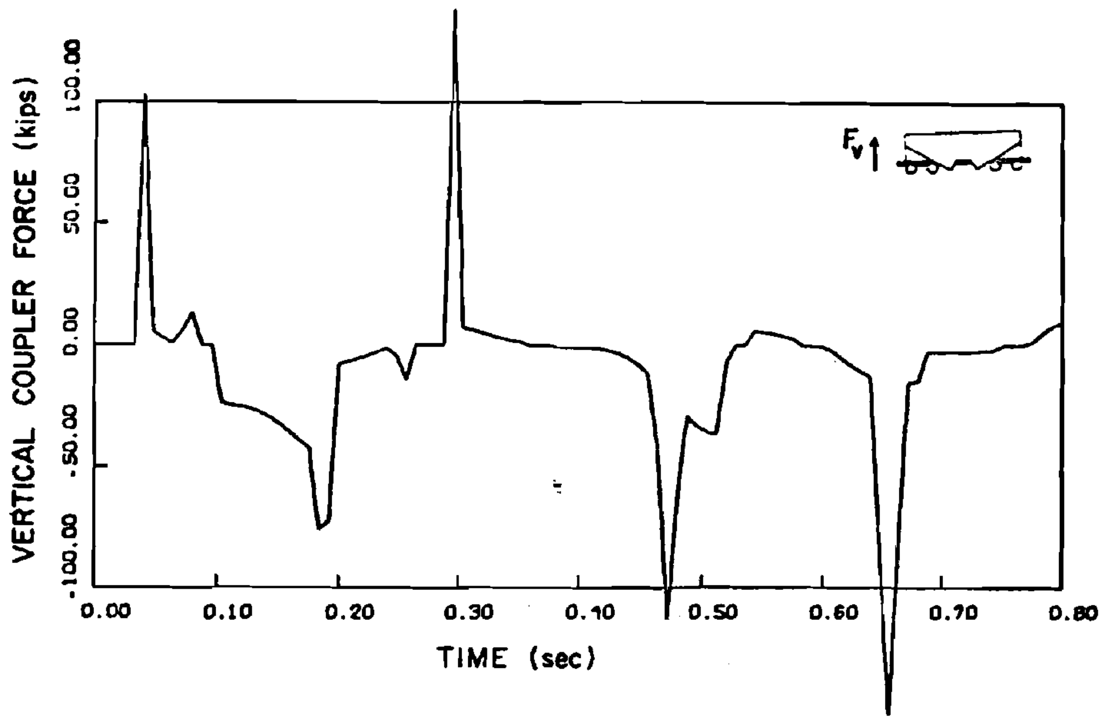


Vertical coupler spring rate: 400 kips per inch

Figure 9a
Effect of vertical coupler spring rate on coupler slippage
at trailing (impact) end of hopper

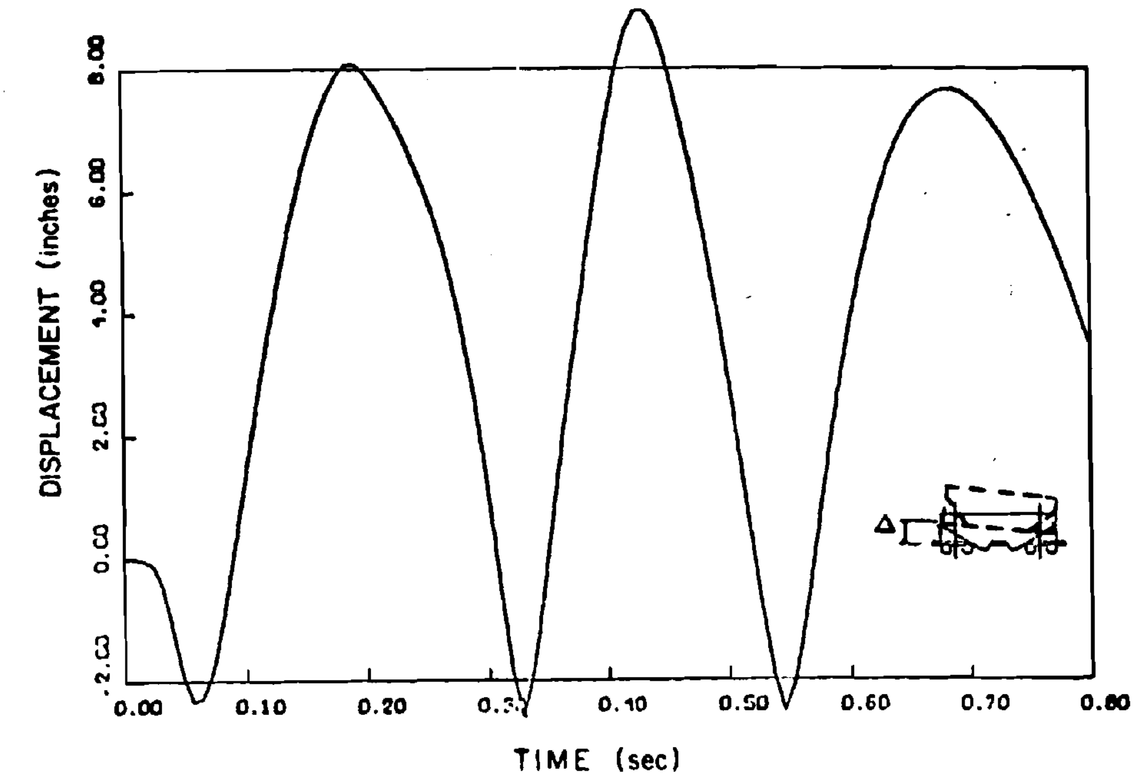


Vertical coupler spring rate: 50 kips/inch

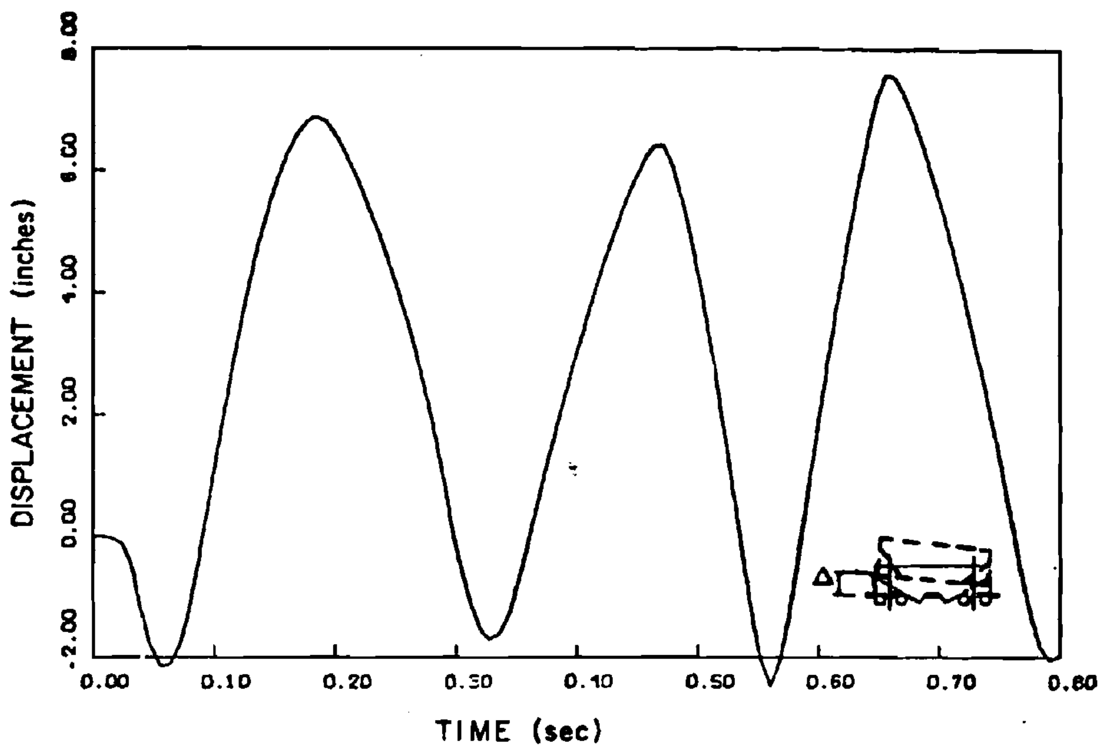


Vertical coupler spring rate: 400 kips/inch

Figure 9b
Effect of vertical coupler spring rate on
vertical coupler force at the trailing (impact) end of hopper car

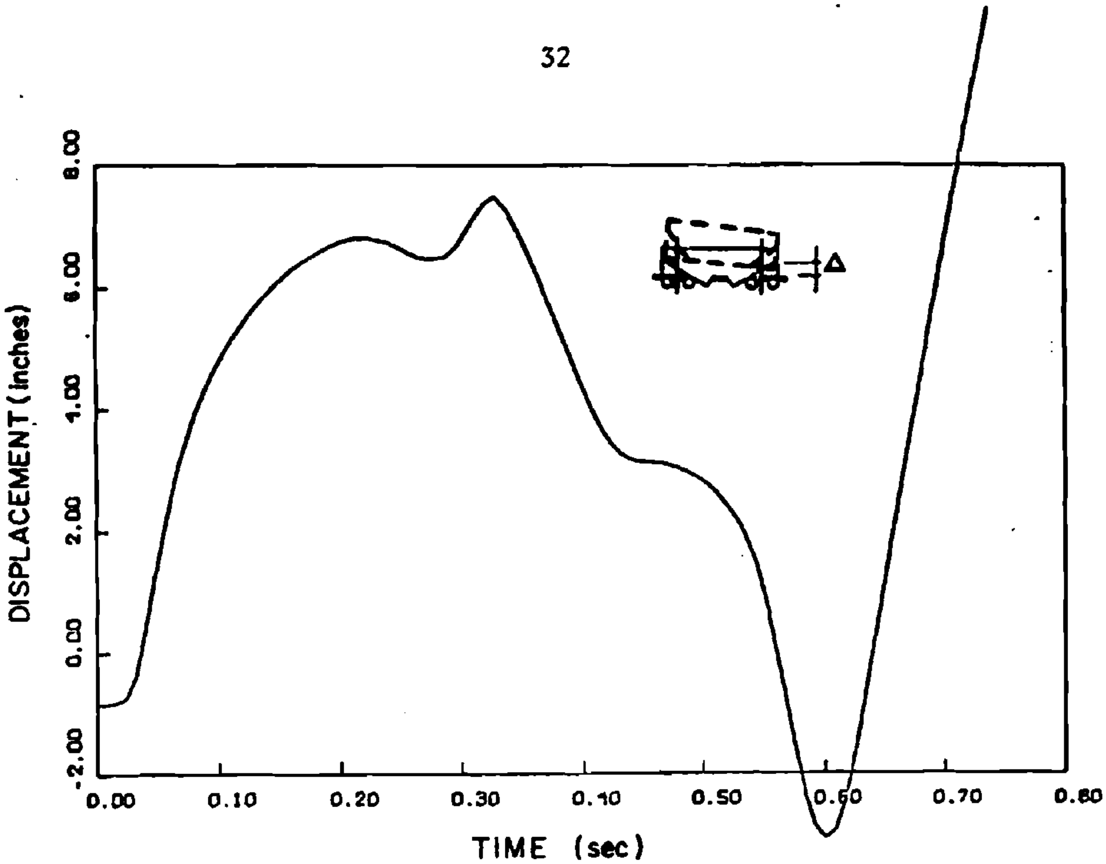


Vertical coupler spring rate: 50 kips/inch

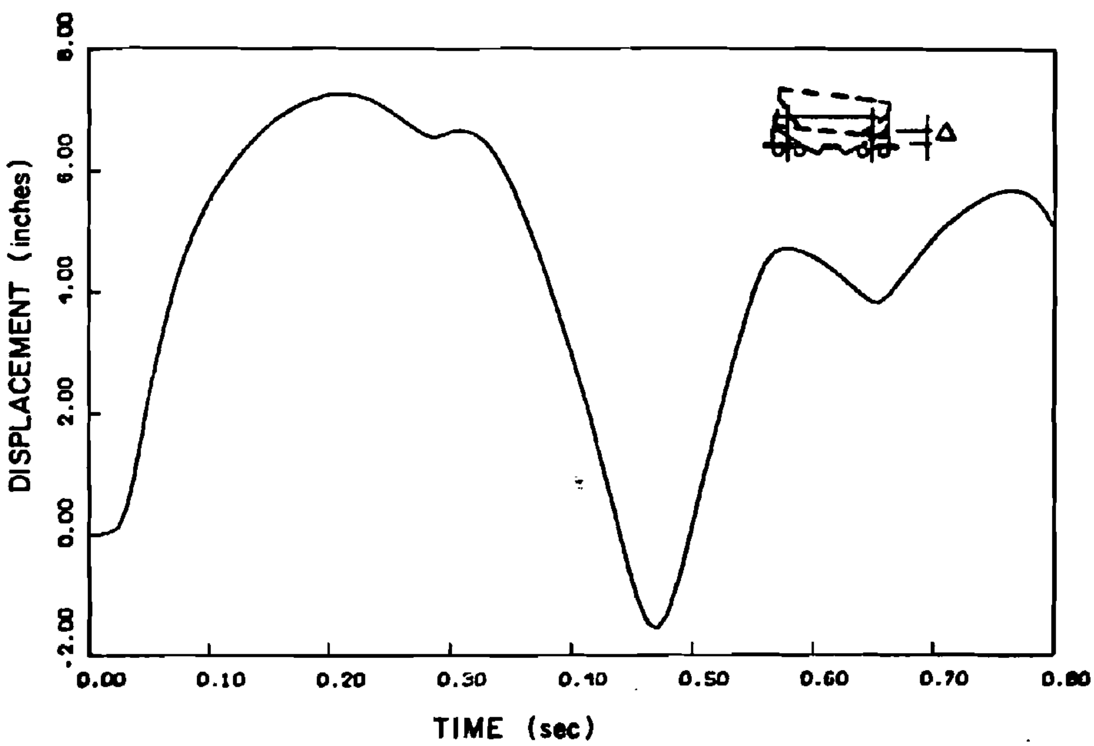


Vertical coupler spring rate: 400 kips/inch

Figure 9c
Effect of vertical coupler spring rate on
car body center plate displacement
at the trailing (impact) end of hopper car



Vertical coupler spring rate: 50 kips/inch



Vertical coupler spring rate: 400 kips per inch

Figure 9d
Effect of vertical coupler spring rate
on car body center plate displacement
at the leading (free) end of hopper car

Case (e)

Three loaded tank cars, moving at 16 miles per hour, impact standing empty hopper car. Shelf E couplers on tank cars. Coupling is not effected between the leading tank car and hopper car.

As can be expected, this simulation indicated the largest values for the displacement of the hopper car.

The horizontal coupler force reached the same maximum value as in case (c), however it dropped to zero in 0.07 seconds as the impact accelerated the hopper car and the hopper car moved away from the impacting cars (fig. 10a).

The relative displacement between the impacting and impacted couplers reached 23 inches, however the shelves of the E coupler did not engage because the coupler faces separated before the coupler slippage could reach 7.25 inches (fig. 10c).

The center plate displacements were sufficiently large for detrucking to occur on both ends of the hopper car (figures 10d, e).

The relative velocity between the hopper car and leading tank car is shown on fig. 10f. The absolute velocities after impact were: 21 miles per hour for the hopper car and an average of 14 miles per hour for the tank cars.

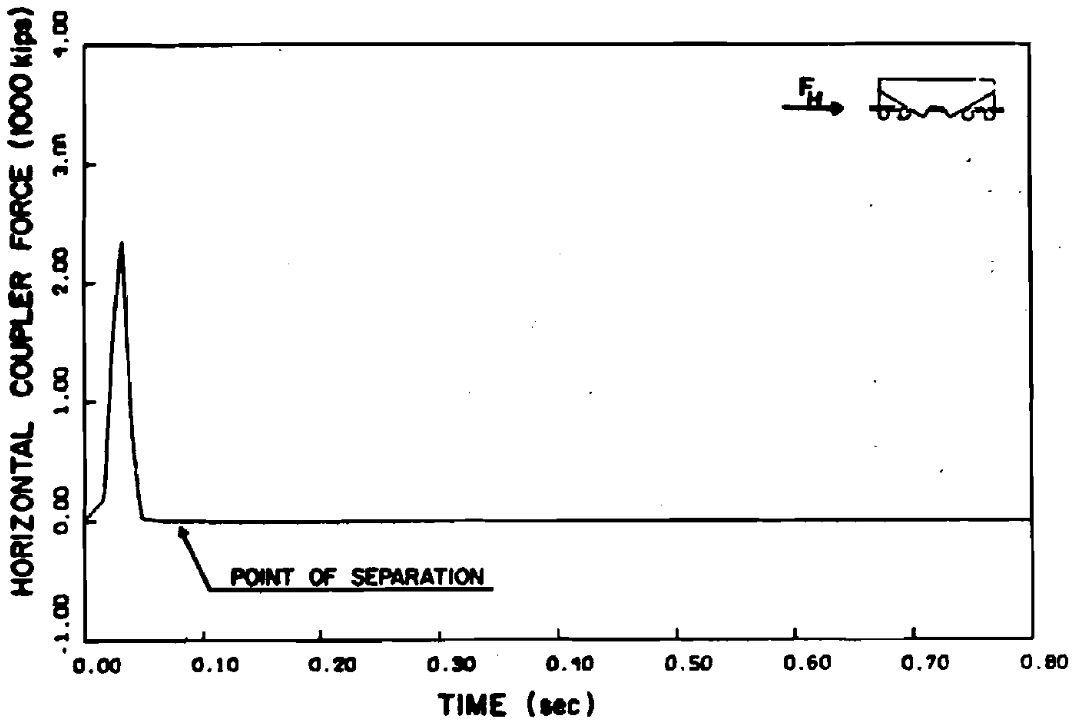


Figure 10a
Horizontal coupler force at trailing (impact) end of hopper car

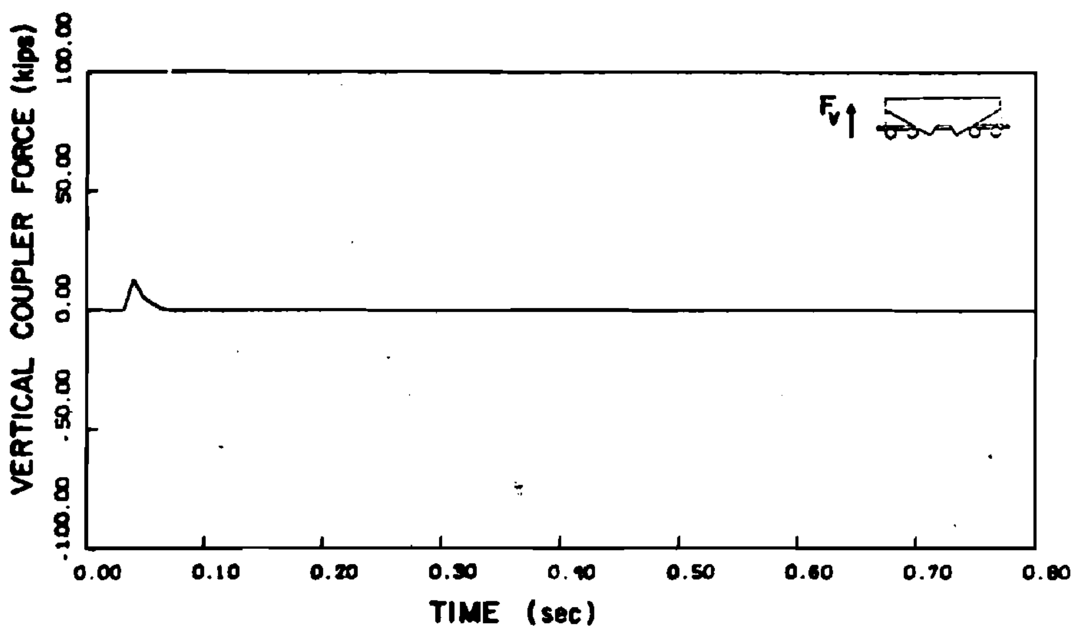


Figure 10b
Vertical coupler force at trailing (impact) end of hopper car

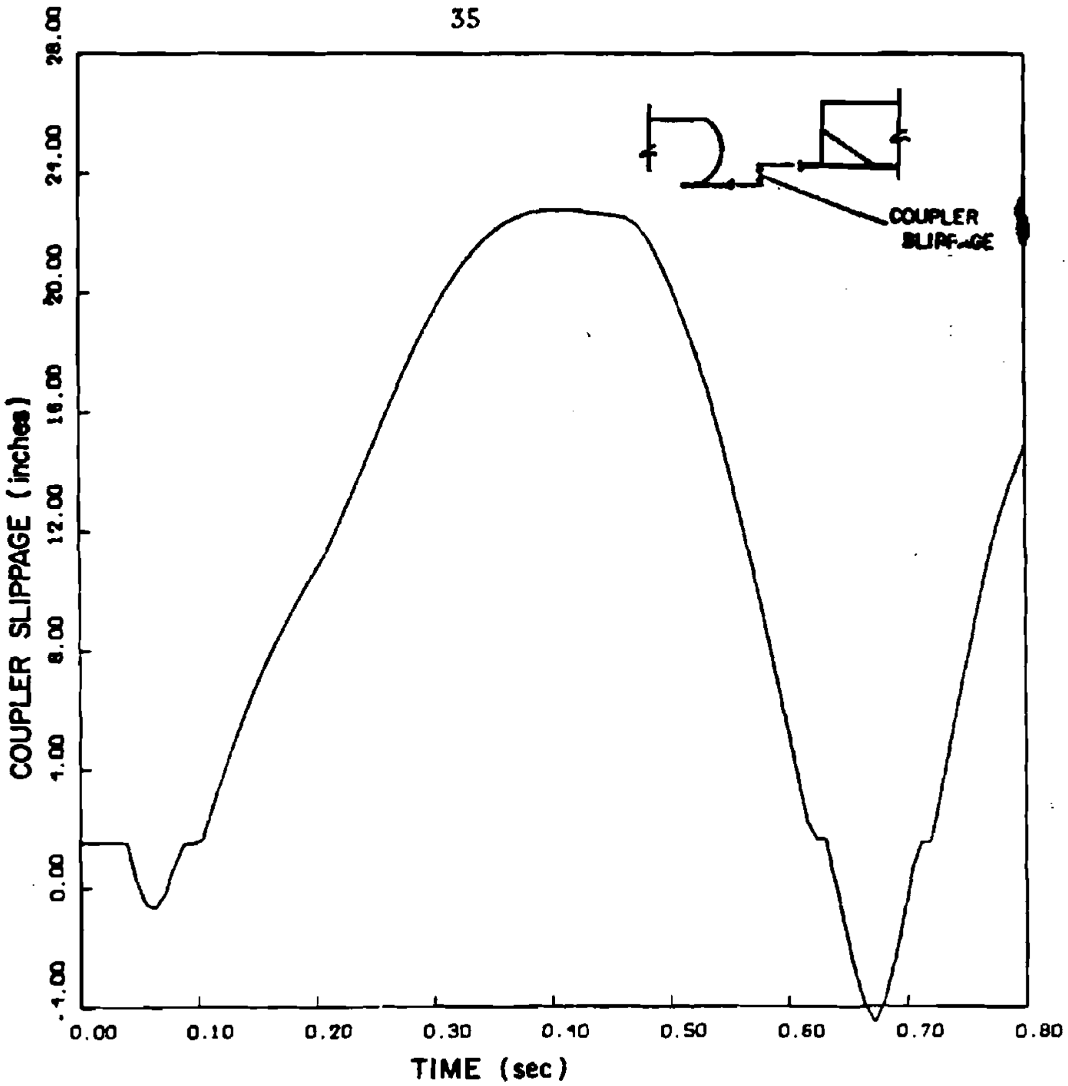


Figure 10c
Coupler slippage at trailing (impact) end of hopper car

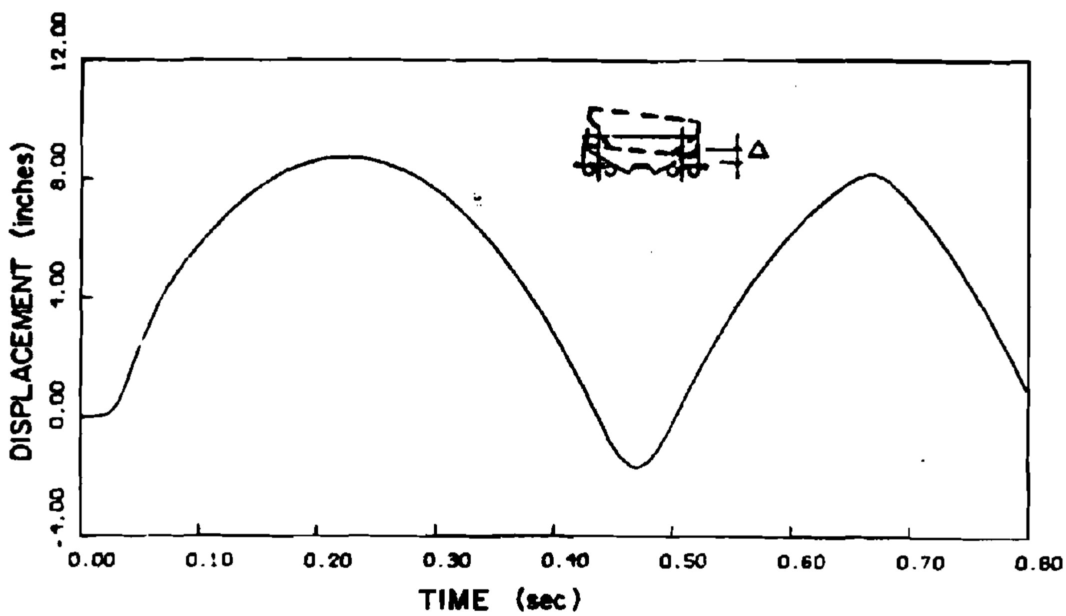


Figure 10d
Car body displacement at leading (free)
end of hopper car

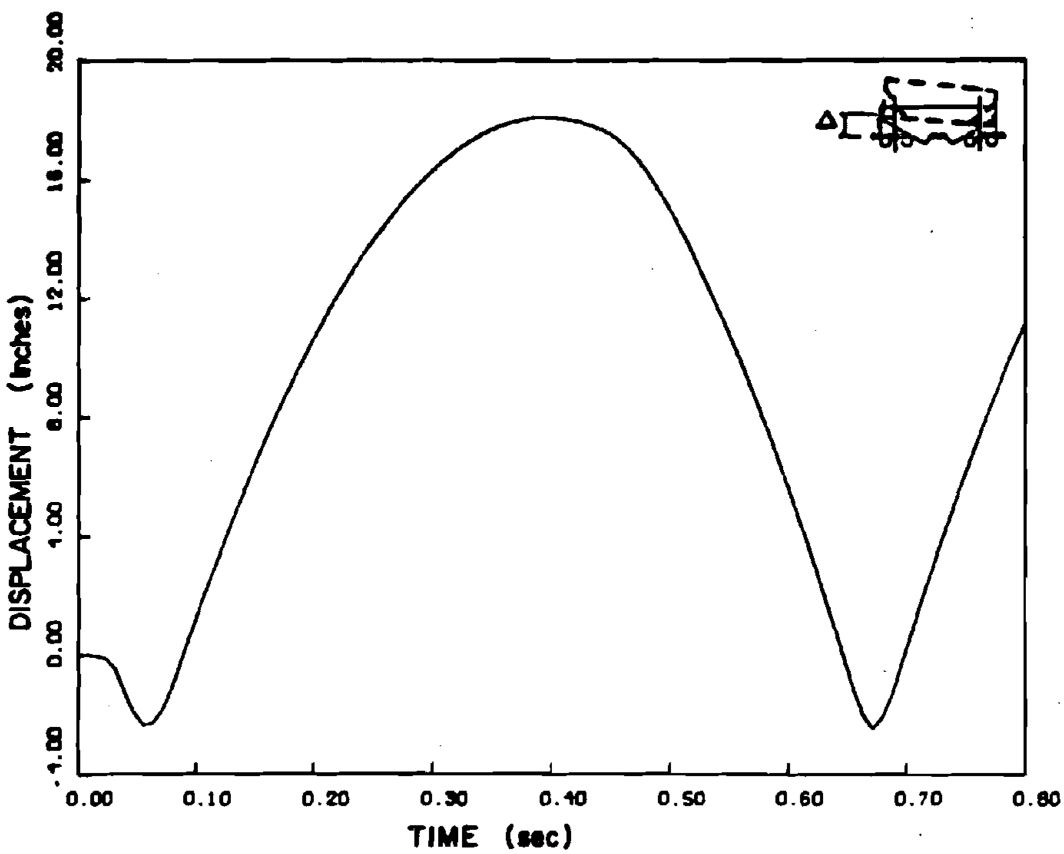


Figure 10e

Car body center plate displacement at trailing (impact) end of hopper car

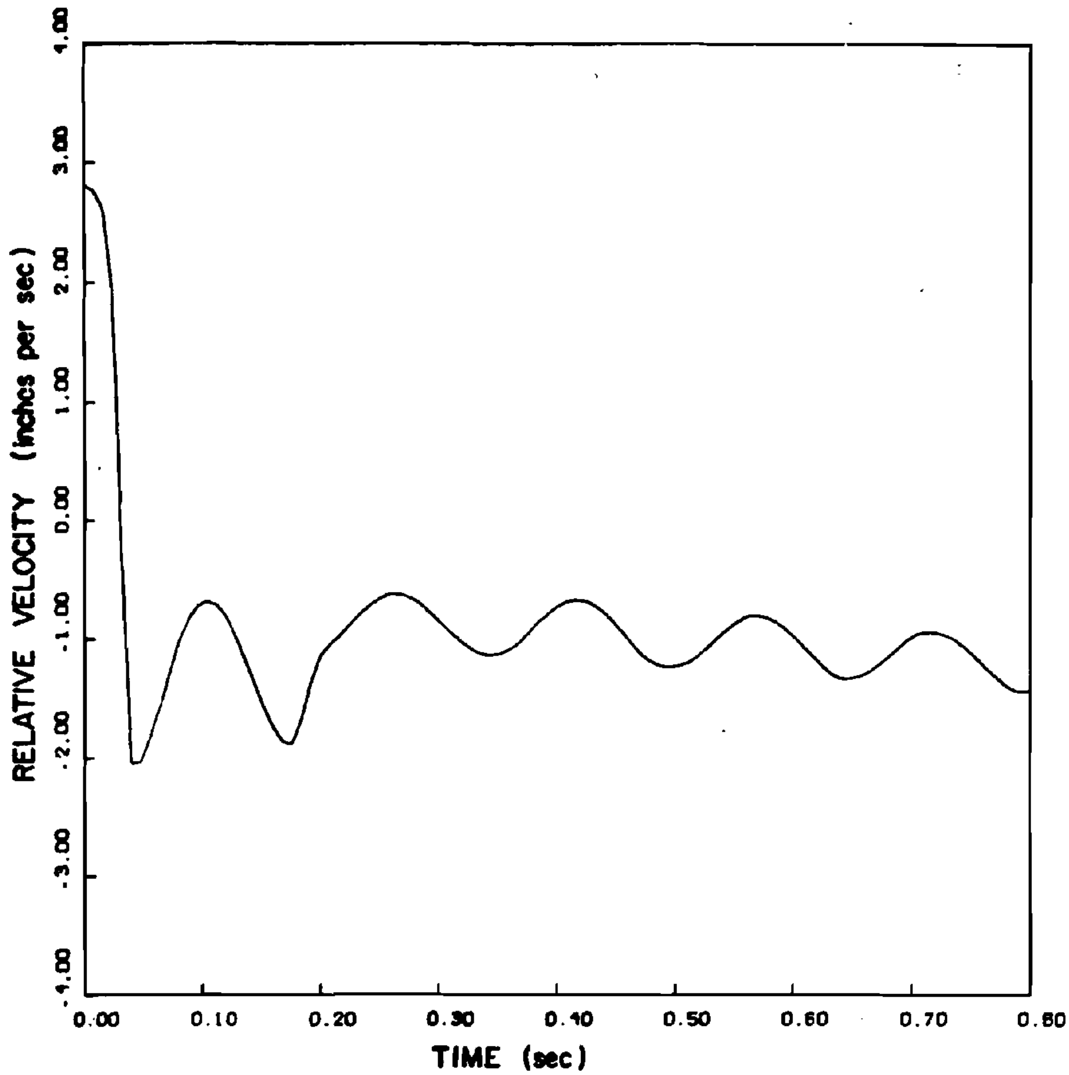


Figure 10f

Relative velocity between hopper car and leading tank car
(Oscillation is due to lading motion)

6.4 Probable sequence of events

The available information on the East St. Louis accident and the results of simulation studies indicate that the probable sequence of events was as follows:

Three coupled loaded tank cars left the hump at a velocity of approximately 16 miles per hour. They travelled 350 feet and impacted a stationary, empty hopper car weighing approximately 40,900 pounds. It is unlikely that coupling could have occurred between the leading tank car and the hopper car at such high impact velocity. However, we shall consider the seouence of events with and without coupling on impact:

Assuming that the cars did couple, the first longitudinal impact in compression was followed by a smaller impact in tension and further by a second impact in compression (fig. 8a). The impact end of the hopper car moved first down, then up - without clearing the center pin - then down again. At this time the second longitudinal impact in compression occurred, reinforcing the downward deflection of the impact end of the box car. Upon rebound, this end reached a much higher level than at the first time and moved beyond the constraint of the center pin (fig. 8d). The impact end of the hopper car de-trucked at this point and the coupler remained elevated either because the car body was supported on the center pin or because the coupler came to rest on the end sill of the tank car. The tank cars and hopper car continued to move down the track at a velocity of approximately 14 miles per hour and struck a cut of standing cars. This impact drove the coupler into the tank car head.

Assuming that the cars did not couple, the impact end of the hopper car moved first down then up, high enough for the car body center plate to separate from the truck center plate dish by 18 inches, clearing the center pin (fig. 10e). The hopper car was accelerated by the impact to 21 miles per hour, the cut of tank cars decelerated by the impact to 14 miles per hour. The hopper car detrucked on the impact end and, moving down the track, impacted a cut of stationary cars. It probably rebounded, with the coupler in an elevated position, and was then hit by the loaded tank cars. At this impact the coupler punctured the head of the leading tank car.

Two significant observations can be derived from the simulations:

(i) Regardless of whether coupling occurred on first impact or not, the vertical motion of the hopper car was sufficient for detrucking to occur. As a result, the coupler of the hopper car on the impact end was elevated such that a second impact could drive it into the tank car head.

(ii) The tank car would not have been punctured without the occurrence of a second impact. A cut of cars standing on the track provided sufficiently large reaction for the coupler of the hopper car to puncture the tank car head.

7. THE DECATUR, ILLINOIS ACCIDENT

An accident, involving tank car head puncture, occurred on July 19, 1974 in Decatur, Illinois. Official accident investigation documents were not available at the time when the computer simulations presented herein were conducted. However, based on information one of the writers (Diboll) collected at the hearings of the National Transportation Safety Board in August 1974, the sequence of events can be reconstructed as follows:

Five loaded tank cars were being flat switched (released from a locomotive at a sufficient speed to allow them to hit and couple with cars further down the yard track). The tank cars were released at approximately 4 miles per hour. A light empty box car previously flat switched had not reached the other cars on the track, and it was standing alone when the five cars impacted it. Apparently the couplers were misaligned. This is suggested by the evidence of the impact marks on the couplers and that the track at the impact area had been subjected to a large lateral force, indicated by broken rail joint bolts near the point of impact. There is general agreement among the investigators that coupling did not occur. Following impact, all six cars proceeded down the track. The box car derailed such that it was found at an angle of about 45° away from the track with its trailing (impact) end having dragged for some distance along the left (with respect to the direction of motion) of the track. The trailing end truck had derailed soon after impact, but the lead end probably derailed only when the box car was about to stop.

Rollability tests were conducted simulating the accident conditions(8). It was found that cars released at approximately 4 miles per hour reached a speed of approximately 8 miles per hour at the position of the box car because the yard is on a down grade. There was some concern by observers that the simulation was not precise and that the final speed could have been higher.

Computer simulations for this accident were conducted for 8, 12 and 16 miles per hour. The simulation for 8 miles per hour did not produce conditions which could cause a puncture, but the 12 miles per hour simulation did. The probable puncture mechanism was as follows:

The five tank cars impacted the box car. The box car coupler was misaligned to the left with reference to the direction of motion. The cars separated within 0.10 seconds after impact. The impact end of the box car went down, to bottoming of the truck springs, bounced up sufficiently high to clear the truck center pin and, due to the eccentricity of impact, was moved to the left. The car body came down on the wheels, braking the car. The impact end remained in an elevated position. The tank cars impacted the box car a second time, thrusting the coupler through the head of the first tank car. The six cars moved as a unit further down the track. Vibration and liquid pressure caused the cars to separate and the box car dropped to the ground to the left. At this time the leading truck of the box car derailed.

The results of the simulation are shown in figures 11a to 11d. The simulation parameters are given in Table 2. The most important result is that the vertical displacement of the car body at the impact end was high, probably sufficiently high to clear the center pin at 0.3 seconds after impact (fig. 11d).

Table 2

Simulation Parameters
Decatur, Illinois Accident

Car Number	Impacting tank cars (each of 5)	Box car
Car weight (body only), kips	85.5	29.0
Lading weight, kips	132.9	0
Weight of one truck, kips	10.0	7.5
Half length between coupler faces, inches	440.0	255.0
Half length between truck centers, inches	354.0	171.0
C. G. height above centerplate, inches	50.0	50.0
C. G. height above coupler, inches	40.0	40.0
Mass moment of inertia, kip-in-sec^2	31,000	1,370
Truck spring const., kips per inch	63.8	63.8
Bolster spring const., kips per inch	3,000	3,000
Car underframe spring const., kips per inch	2,500	2,500
Draft gear spring const., kips per inch	86.4	86.4
Draft gear spring travel, inches	2.5	2.5
Truck spring travel, inches	2.5	2.5
Lading spring const., kips per inch	246.0	-
Lading friction coefficient	.01	-
Vertical coupler spring const., kips per inch	50.0	50.0
Vertical coupler slack, inches	1.2	1.2
Draft gear hysteresis load, kips	40.0	40.0
Friction coef. between couplers	.20	.20
Initial car velocity, MPH	12.0	0

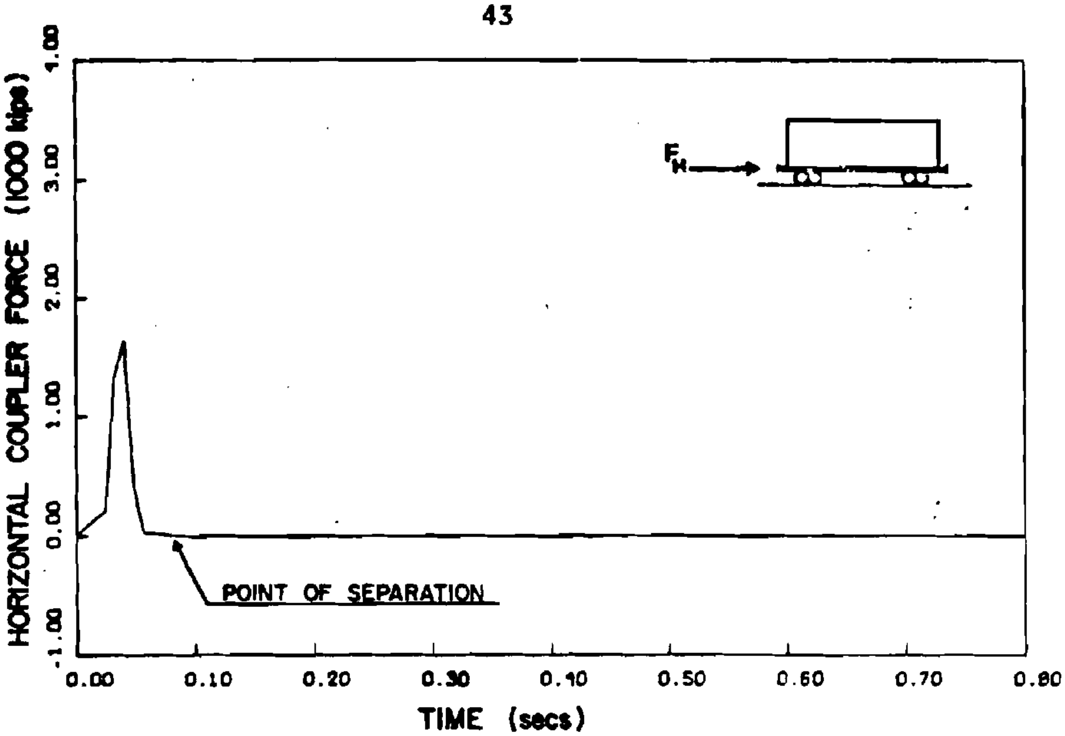


Figure 11a

Horizontal coupler force at the trailing (impact) end of the box car

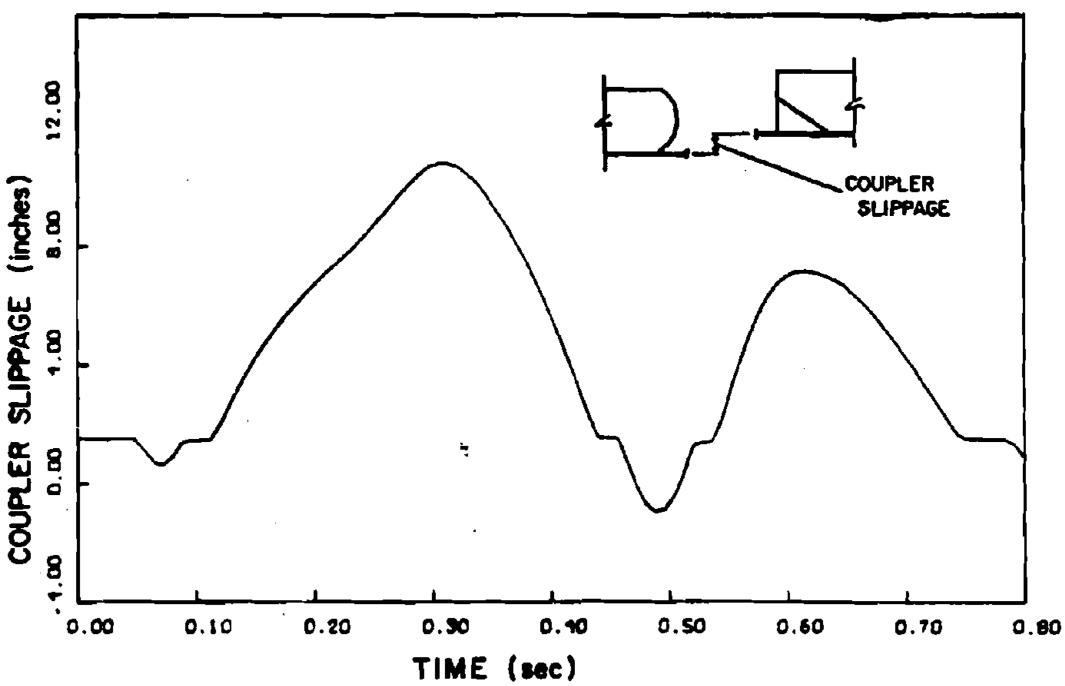


Figure 11b

Coupler slippage at the trailing (impact) end of the box car

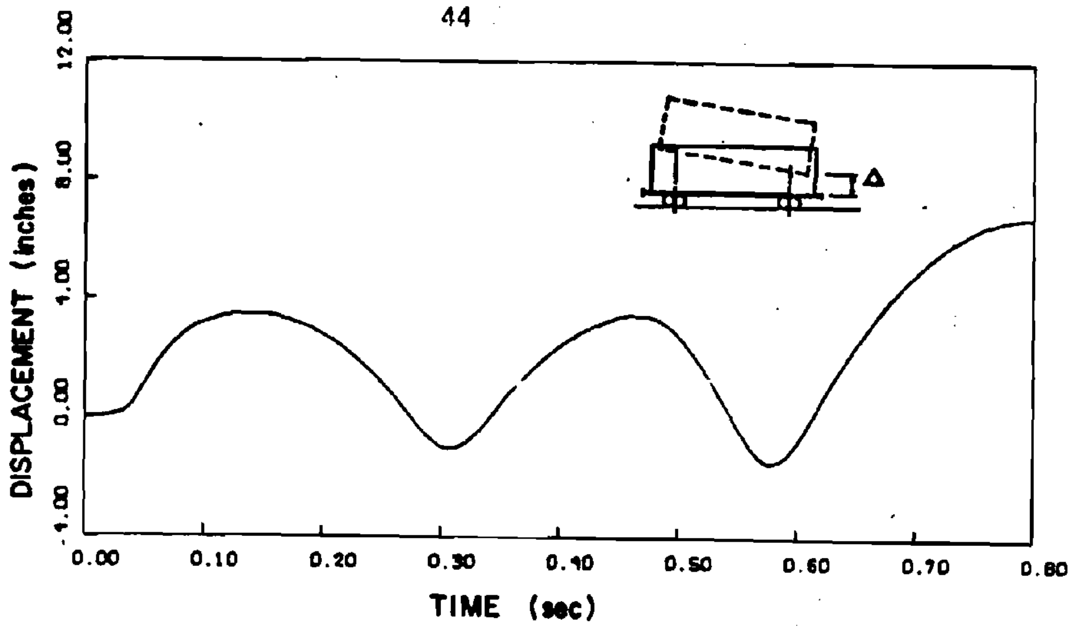


Figure 11c

Car body center plate displacement
at the leading (free) end of the box car

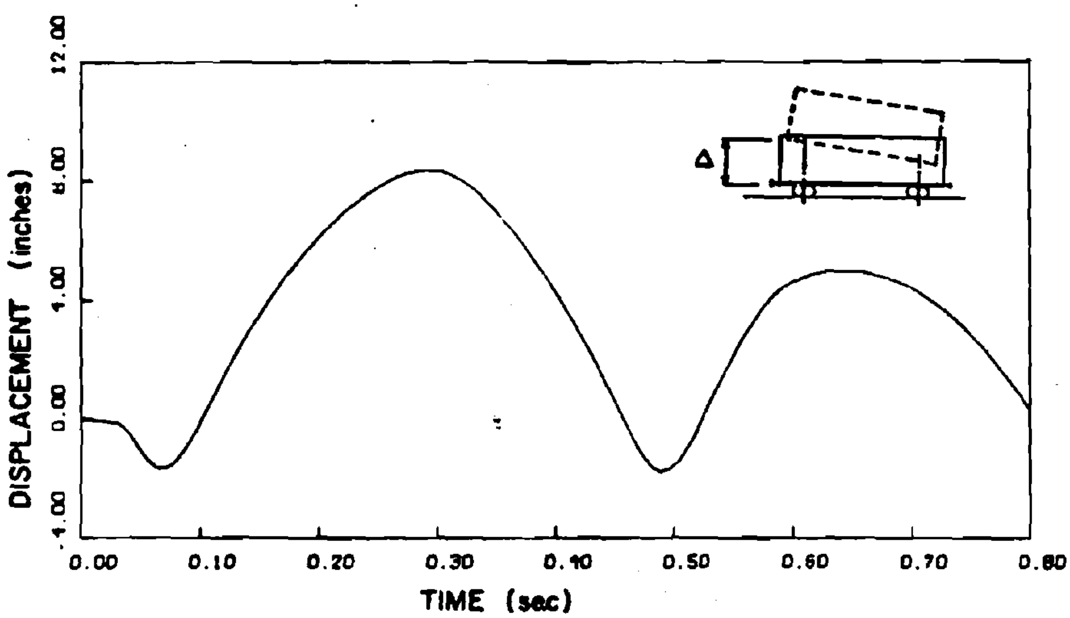


Figure 11d

Car body center plate displacement
at the trailing (impact) end of box car

8. THE HOUSTON, TEXAS ACCIDENT

On Saturday, September 21, 1974 the head of a tank car loaded with butadiene was torn as a result of impact in the Englewood classification yard, Southern Pacific Railroad, Houston, Texas, causing an explosion and fire which resulted in the death of one person and great property damage.

One of the writers (Diboll) went to the accident scene on September 24 and, with the assistance of officials of the Federal Railroad Administration and the National Transportation Safety Board, reconstructed the sequence of events as follows:

A malfunction of the retarding system resulted in a number of cars overspeeding and impacting standing cars at high speed. There were actually three permanently damaging high speed impacts prior to the fire, two on bowl 50 (track 50) involving very heavily loaded cars, which were not the cause of nor damaged by the fire. These impacts were not analyzed because head puncture was not involved, but they are mentioned because the initial cause was probably the same for both impacts. The other high speed impact was on bowl 1, which resulted in the tank car head tear. The impacts were reconstructed from information obtained shortly after the fire, from conversation with the investigating officer of the National Transportation Safety Board (NTSB), and from the depositions of the accident taken November 12 and 13, 1974. NTSB had not released its formal report at the time of this analysis.

The first reports indicated that during normal humping operations a car deposited a foreign substance on the retarders, making them lose their braking power. A large loaded covered hopper car, CELX 773, impacted a large loaded tank car, GATX 98415 on bowl 50 at high speed, estimated to be approximately 20 miles per hour by the hump towerman. He stopped the hump operation, but other cars had left the hump by this time. A large loaded box car, SP 675076, impacted CELX 773 hard enough to damage the contents of the SP car in spite of the fact that it had a Hydracushion Type 20-14A draft gear. Finally, two large loaded tank cars, SCMX 3641 and RTMX 3055, released together into bowl 1, impacted a small empty tank car, UTLX 88717. Also on bowl 1 were 18 other cars. The two cars, next to UTLX 88717, were a light empty tank car, UTLX 91693, similar to UTLX 88717, and a large box car heavily loaded with wood products, SP 224469.

There was an explosion near the two tank cars and after the fire was out it was discovered that the head of the first large tank car, SCMX 3641, had been torn open by an impact with the first small empty tank car, UTLX 88717. During the impact (or impacts) the coupler of UTLX 88717 split vertically near the bottom and the head broke off. It was found balanced between the tops of the truck side frames of the two cars where they were close together. The coupler head broke off before the head of SCMX 3641 was impacted. This is evidenced by the marks on the head of SCMX 3641 which bore imprints of the coupler pocket and of the stub of the broken coupler shank. The end sill of UTLX 88717 had risen 34 inches above the stub sill of SCMX 3641, at which position high longitudinal forces crushed the head of SCMX 3641, deforming it until the head tore at the weld near the stub sill. UTLX 88717 slid down 17 $\frac{1}{2}$ inches during and after the impact and was found with its center sill resting on the stub sill of SCMX 3641 (figures 12, 13, 14).

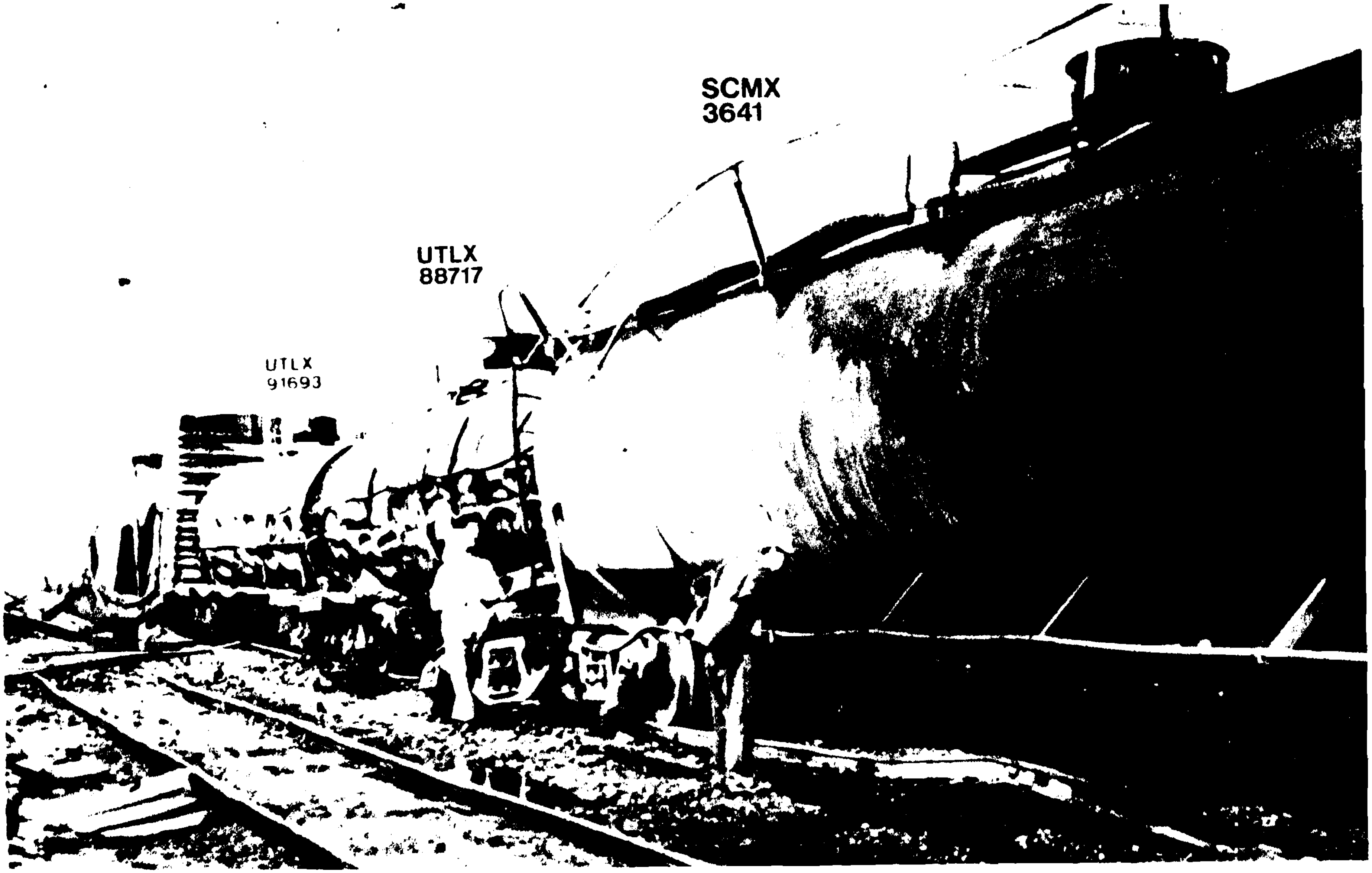


Figure 12
The scene of accident in Houston
September 24, 1974

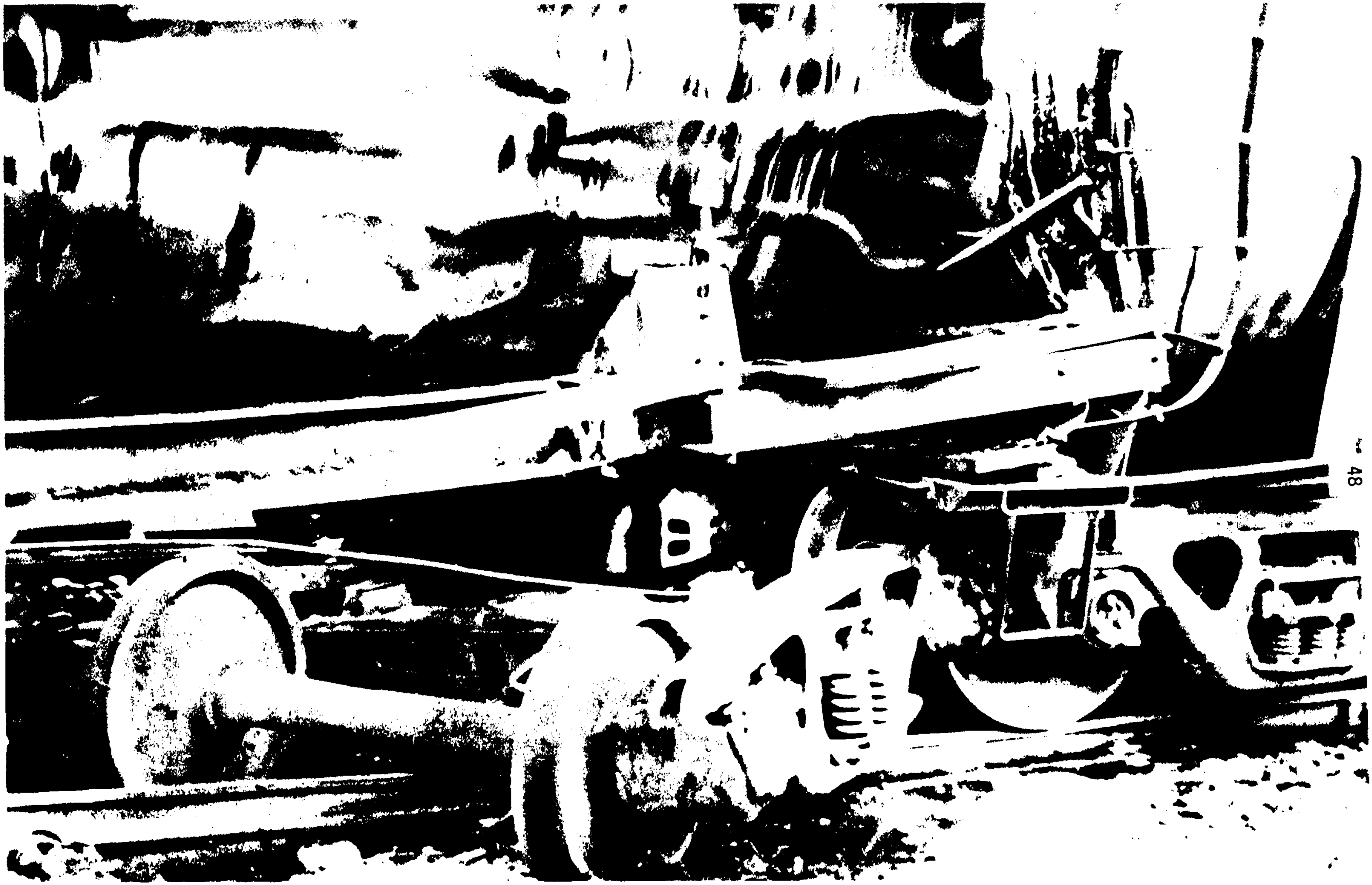


Figure 13
Cars in the Houston head tear accident.
SCMX 3641 to right, UTLX 88717 to left.

Figure 14
Tear in head of SCMX 3641, just above end sill.

The coupler of SCMX 3641 had marks on its knuckle at a point approximately eight inches above the lower edge, which indicated that the coupler of UTLX 88717 was eight to ten inches higher than that of SCMX 3641 at the moment of this impact. There were also deformations on the top of the coupler of SCMX 3641.

There were no marks on the bottom of UTLX 88717 which would indicate that there had been contact before the head puncture occurred. There was one small mark which appeared to be in the correct position to be a flange imprint made when the car dropped on a derailed wheel.

Many details of the cars and accident conditions necessary for mathematical modelling of the accident were obtained, but the most important question was whether the impacted small empty tank car, UTLX 88717, was separate from or coupled to the rest of the cars standing on the track when it was first impacted. At first the answer was, according to the towerman, that contact with the rest of the cars had been made. His angle of view and distance from the impact area (1300 feet) made it difficult for him however to be certain.

On October 29 Mr. Richard H. Peterson of the National Transportation Safety Board gave further information received from an eyewitness. His statement (later confirmed in the deposition, reference 9) was that the small empty tank car (UTLX 88717) was still moving at approximately four miles per hour when struck by the two large loaded tank cars (SCMX 3641 and RTMX 3055) which were moving at 16 to 20 miles per hour. Within one second all three impacted the standing cars and gas started leaking. The witness was much closer to the point of impact and had a better angle of view than the towerman.

Analysis of the motion of the cars was conducted by simulation of the cars and the conditions of operation in a mathematical model. The simulation parameters are listed in Table 3. First, the action of cars 1 and 2, moving at 16 miles per hour and impacting into a cut of standing cars, comprising cars 3 through 9 all coupled together, was simulated. The simulation did not produce a combination of longitudinal forces and vertical motions which would explain the pitch up of the first standing car, UTLX 88717, and the tear of the head of the impacting tank car, SCMX 3641. The force of the first impact peaked at 3,000,000 pounds. The duration of impact was only 0.02 seconds.

On the basis of the evidence indicating that car 3 (UTLX 88717) was not coupled to the cut of cars standing on bowl 1 when first impacted, additional simulation studies were conducted. The simulation studies assumed <u>three</u> consecutive impacts as shown in fig. 15: (i) cars 1 and 2 impact car 3, coupling does not occur, car 3 therefore moves faster than cars 1 and 2; (ii) car 3 impacts the standing cars and its free end pitches up; (iii) cars 1 and 2 impact car 3 for the second time. Analyses were conducted for two cases, with assumed velocities of 16 and 20 miles per hour for cars 1 and 2 prior to the first impact. Car 3 was assumed to move at four miles per hour for both cases. The second impact was simulated at two time intervals after the first impact. The third impact was not simulated. The results of simulation of the first two impacts and assumptions concerning the third impact are presented in the following sections.

TABLE 3

Simulation Parameters

(All values, except those marked by asterisk [*], were estimated)

Car Number in Simulation	1	2	3	4	5	6	7	8	9
Type of Car*	Loaded Tank	Loaded Tank	Empty Tank	Empty Tank	Loaded Box	Empty Box	Empty Box	Empty Cov. Hop.	Loaded Tank
Number*	RTMX 3055	SCMX 3641	UTLX 88717	UTLX 91693	SP 224469	KTY 20700	SP 175937	SLSF 79350	UTLX 98136
Total weight, from consist list*	506,700		70,000 (From UTLX)	69,900	203,700	69,600	107,300		187,000
Car weight (body only), kips	86.0	86.0	50.0	50.0	50.0	50.0	25.9	43.3	70.0
Lading weight, kips	147.0	147.0	0.0	0.0	134.0	0.0	0.0	0.0	97.0
Weight of each truck, kips	10.0	10.0	10.0	10.0	7.5	7.5	7.5	7.5	10.0
Half length between coupler faces, inches	440.0	440.0	240.0	240.0	300.0	300.0	231.0	360.0	300.0
Half length between truck centers, inches	354.0	354.0	156.0	156.0	216.0	216.0	147.0	276.0	216.0
C. G. height above centerplate, inches	50.0	50.0	50.0	50.0	50.0	45.0	45.0	55.0	50.0
C. G. height above coupler, inches	40.0	40.0	40.0	40.0	40.0	35.0	35.0	45.0	40.0
Mass moment of inertia, kip-in-sec^2	33,000	33,000	2,020	2,020	13,200	3,580	1,000	5,000	13,000
Truck spring const., kips per inch	62.5	62.5	62.5	62.5	62.5	62.5	62.5	62.5	62.5
Bolster spring const., kips per inch	3,000	3,000	3,000	3,000	3,000	3,000	3,000	3,000	3,000
Car underframe spring const., kips per inch	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500
Draft gear spring const., kips per inch	86.4	86.4	86.4	86.4	86.4	86.4	86.4	86.4	86.4
Draft gear spring travel, inches	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5
Initial truck spring displacement, inches	1.8	1.8	0.4	0.4	1.4	0.4	0.2	0.3	1.3
Truck spring travel, inches	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5
Lading spring const., kips per inch	250.0	250.0	0.0	0.0	700.0	0.0	0.0	0.0	250.0
Lading friction coefficient	0.01	0.01	0.0	0.0	1.4	0.0	0.0	0.0	0.01
Vertical coupler spring const., kips per inch	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0
Vertical coupler slack, inches	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2
Draft gear hysteresis load, kips	40.0	40.0	40.0	40.0	40.0	40.0	40.0	40.0	40.0
Friction coef. between couplers, vertical	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20

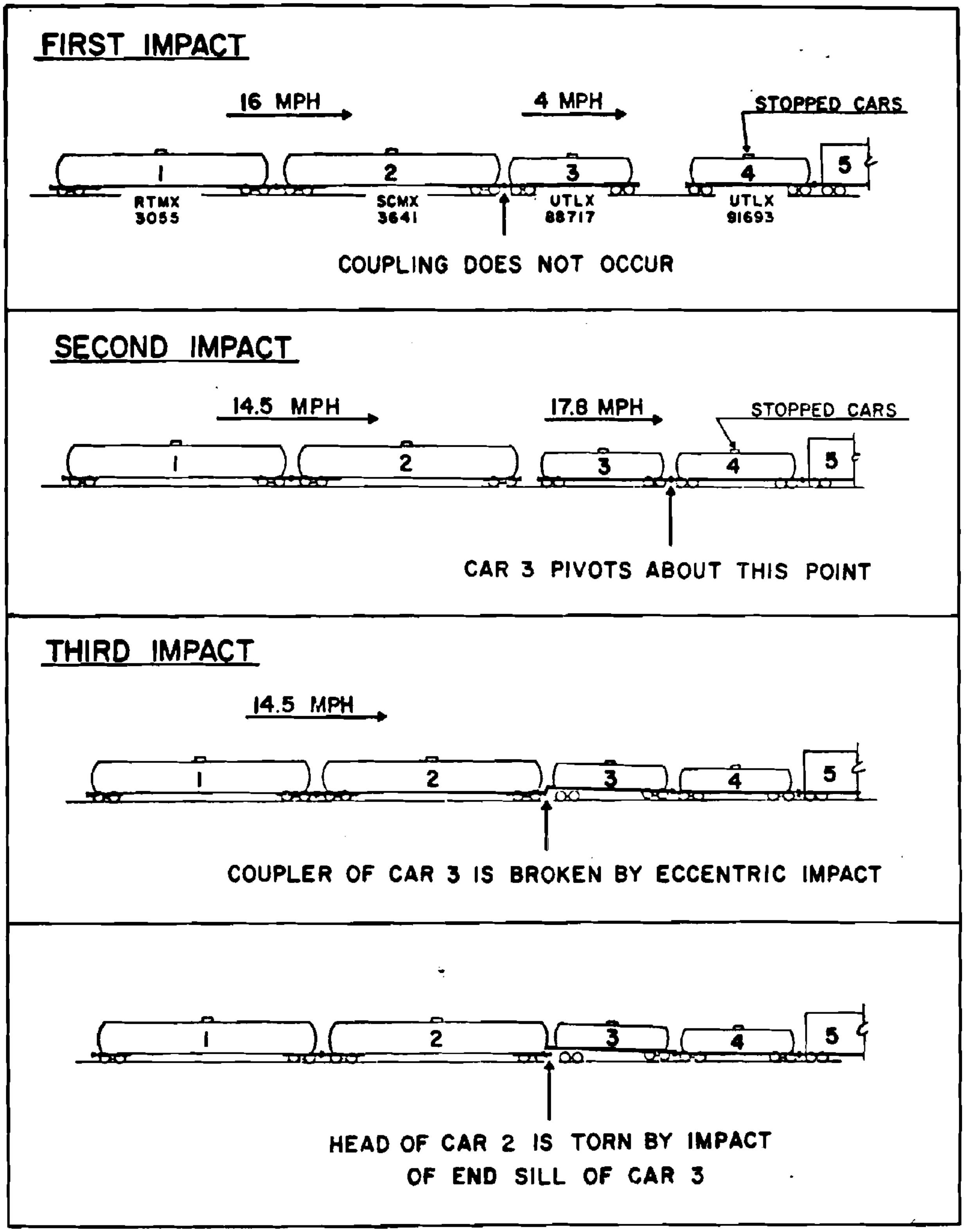


Figure 15

Sequence of impacts. Houston accident.

8.1 Simulation of the accident

Case (a): Initial velocity of cars 1 and 2: 16 MPH

(i) First impact:

Cars 1 and 2 impact car 3, moving at four miles per hour. Since the relative velocity is 12 miles per hour, it is reasonable to assume that the couplers do not engage. The velocity of car 3 increases to 17.8 miles per hour, cars 1 and 2 slow to 14.5 miles per hour following the impact. The duration of impact is 0.15 seconds. The peak longitudinal coupler force is 2,137,000 pounds, at 0.04 seconds. The impacted (trailing) end of car 3 pitches up and down three times in one second after impact, with amplitudes up to 7.5 inches. Reference is made to figures 16a - d.

(ii) Second impact:

Car 3, moving at 17.8 miles per hour, impacts the standing cars. This impact can be assumed to be at any short time after the first impact. Eyewitnesses' estimates of the distance between car 3 and car 4 at the time of first impact ranged between 10 feet and two car lengths (approximately 100 feet). The simulation shown is for a motion of 16 feet, at .64 seconds. (Figures 17a and b) During the second impact the trailing end of car 3 pitches up and at the end of the second impact the car has negligible velocity.

(iii) Third impact:

Cars 1 and 2 again impact car 3, this time while it is in an elevated position so the coupler of car 3 is impacted at its lower edge. Marks on the coupler of car 2 indicate the contact to be approximately eight inches from the bottom of the coupler knuckle, or three inches from the top, in agreement with model prediction.

It is very likely that the lower portion of the coupler of car 3 split

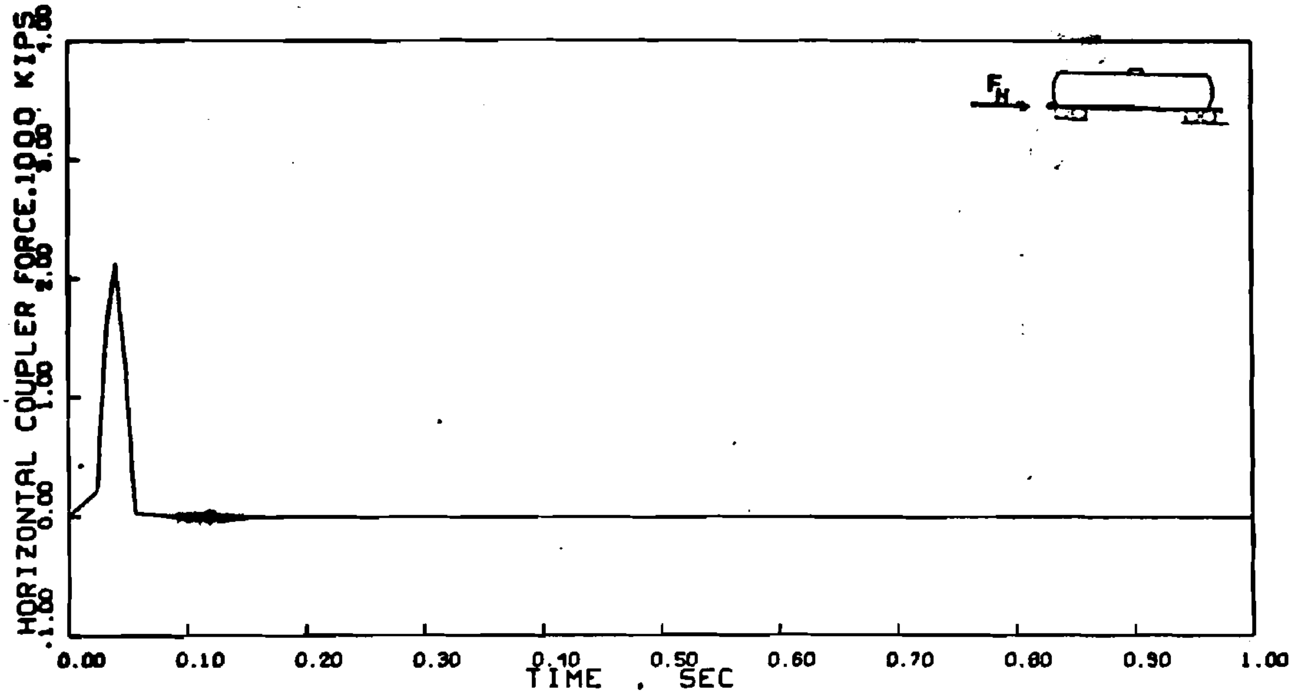


Figure 16a

Horizontal coupler force at trailing (impact) end of car No. 3 (first impact)

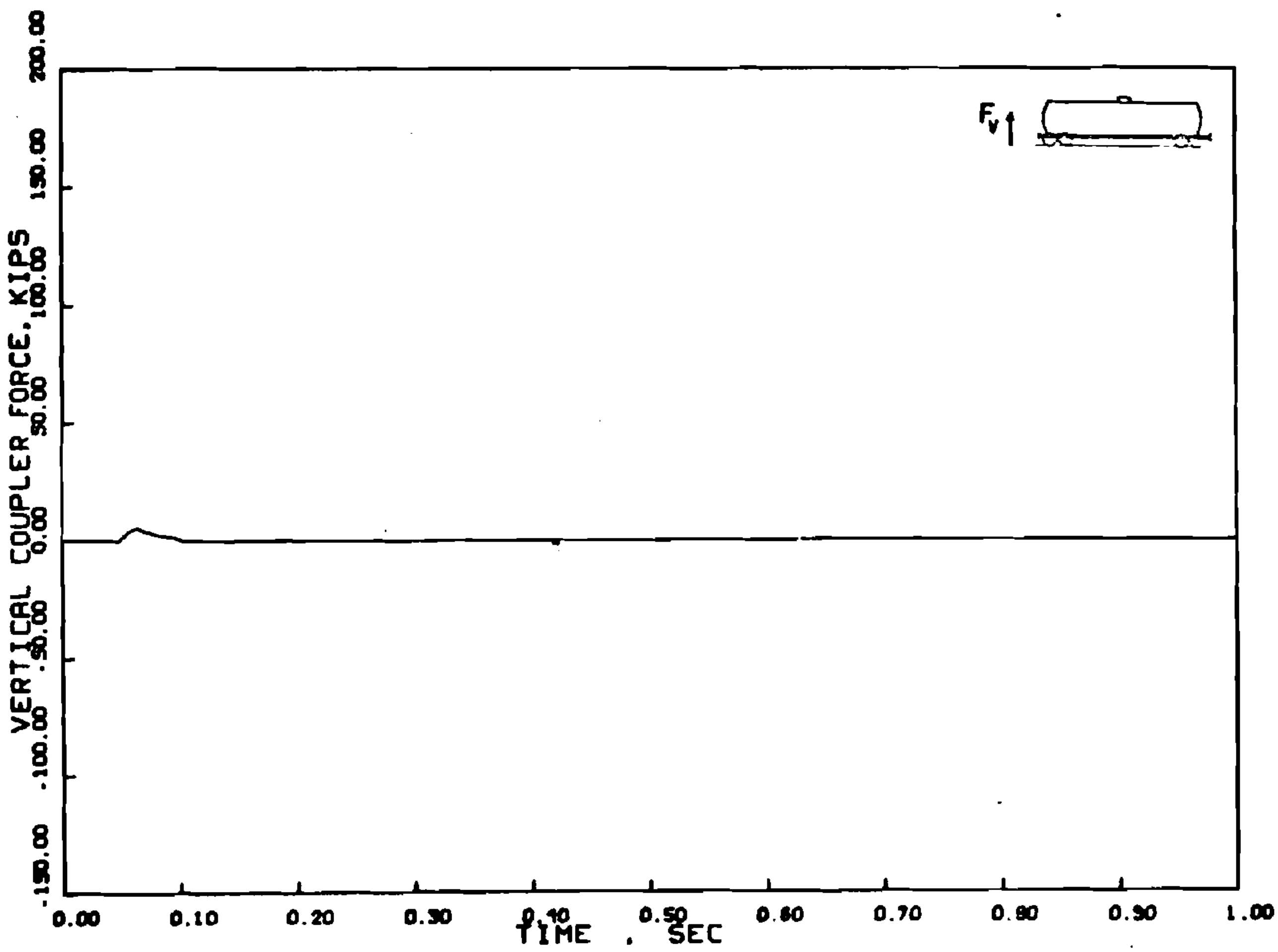


Figure 16b

Vertical coupler force at trailing (impact) end of car No. 3 (first impact)

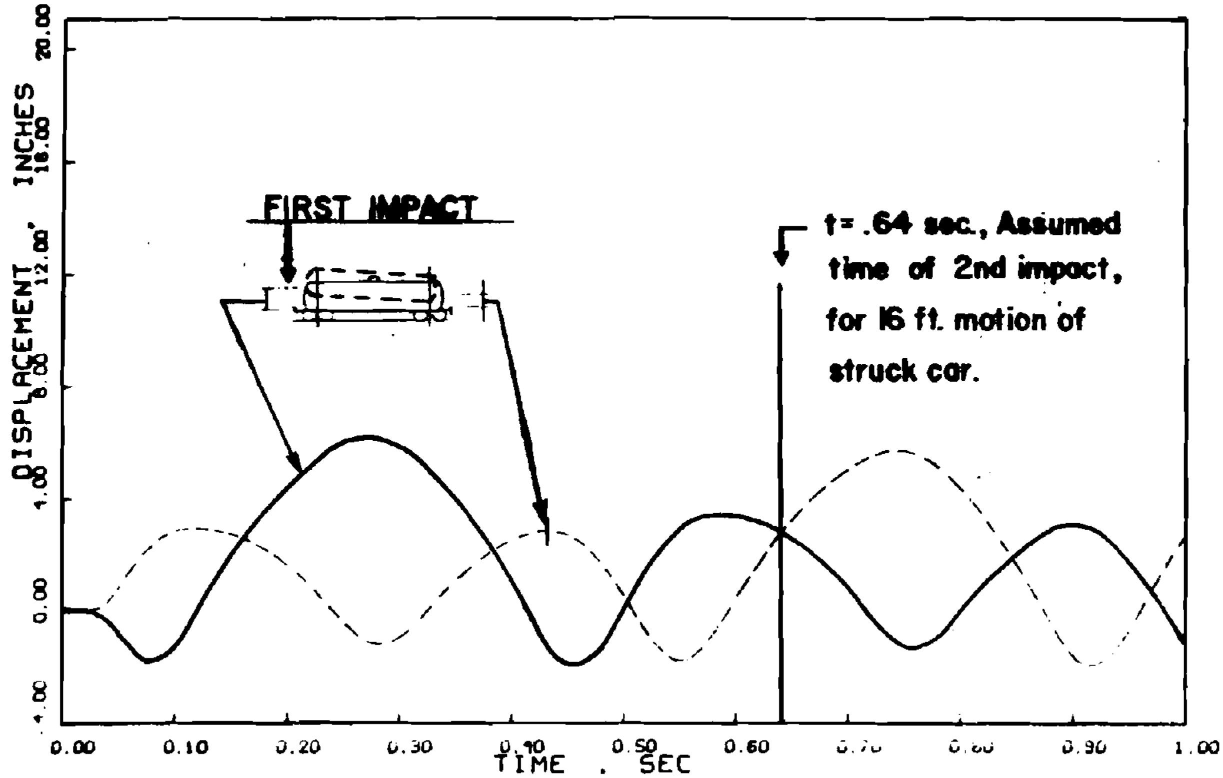


Figure 16c

Car body center plate displacements, car No. 3
(first impact)

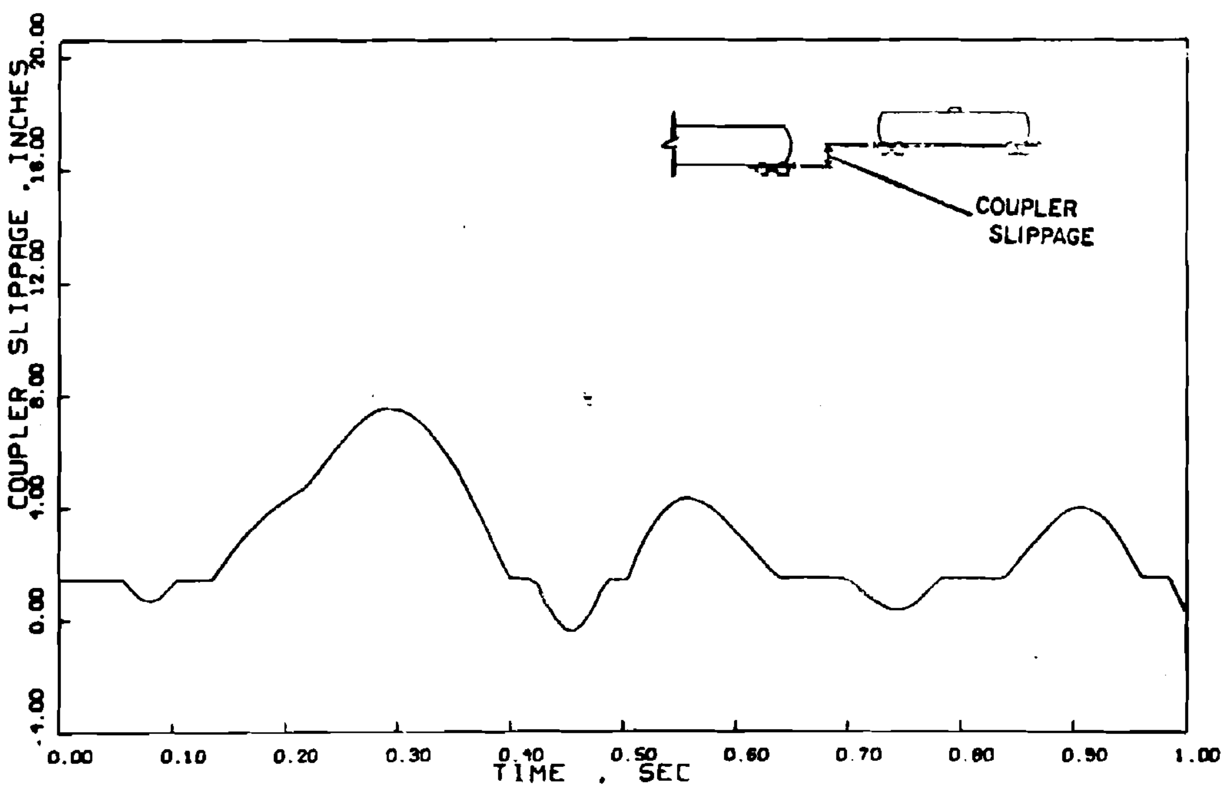


Figure 16d

Coupler slippage at trailing
(impact) end of car No. 3 (first impact)

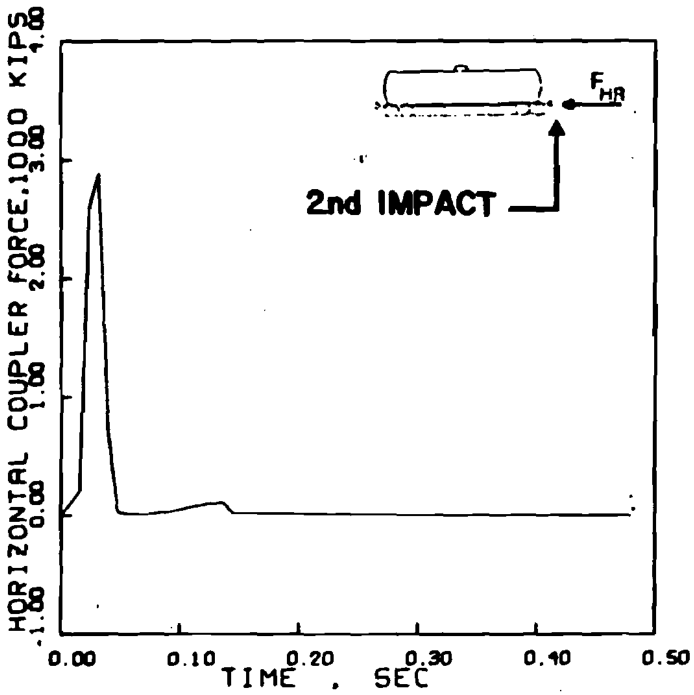


Figure 17a

Horizontal coupler force at the leading (impact) end of car 3. Second impact

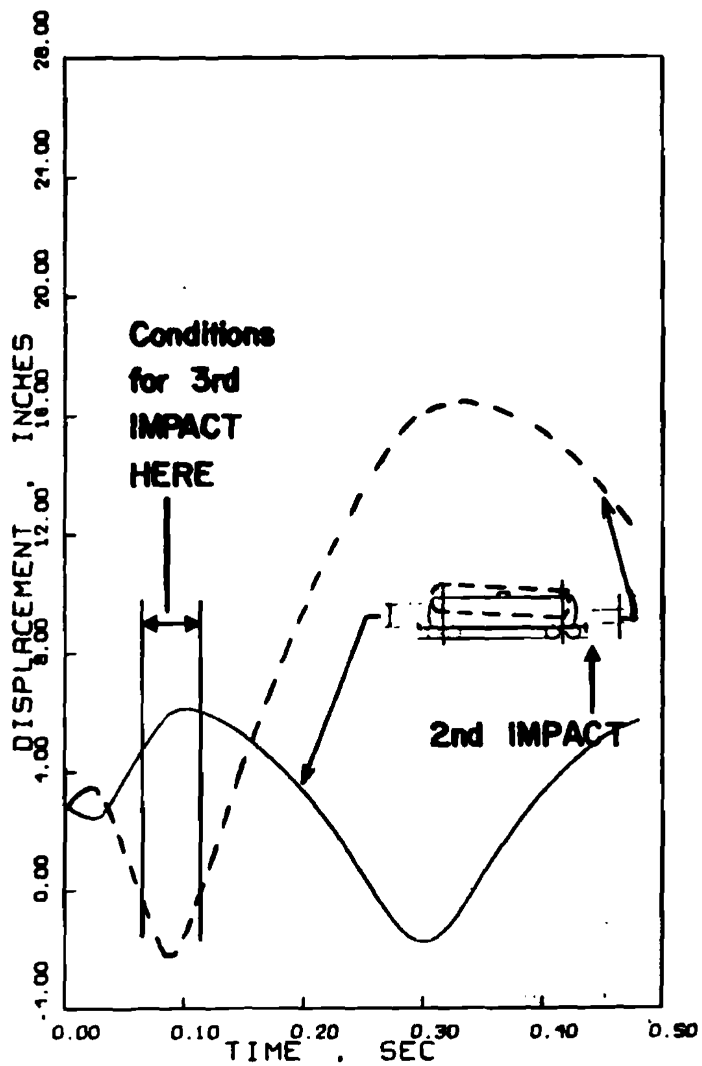


Figure 17b

Car body center plate displacements car 3, second impact and the head broke off at this time. The mathematical model is not presently able to predict motions once coupler override has occurred, therefore it cannot simulate the breaking of a coupler, but definite conlcusions can be made. With a force application at the bottom of the coupler, approximately eight inches below its center, the coupler can yield at a force of 500,000 pounds. In a non-eccentric impact the longitudinal coupler force would have reached 3,000,000 pounds, so it is reasonable to assume that the coupler did break at this point, and that car 3 pitched up further due to wedging action following coupler override, at which time the end sill of car 3 was in position to crush the head of car 2 and cause the tear near the stub sill.

<u>Case (b)</u>: Initial velocity of cars 1 and 2: 20 MPH

The assumptions were the same as in case (a). Only the results of simulation are given in the following.

(i) First impact:

The peak force was 3,020,000 pounds at 0.032 seconds. The total time of the impact to separation of the cars was 0.14 seconds. At the end of the impact the velocity of the small tank car was 22.6 miles per hour and of the two loaded cars was 18.2 miles per hour.

(ii) Second impact:

The peak force in the impact of the small tank car into the standing cars was 3,860,000 pounds at 0.024 seconds after the first contact. The coupler rose 5.9 inches above its normal position, at 0.18 seconds after first contact.

(iii) Third impact:

The third and final impact started at approximately one second after the first impact. The distance travelled between the first and last impact was 26 feet.

With the combination of the oscillations of car 3 following the first impact and the pitch up due to the second impact, the coupler height of car 3 would be eight inches above the coupler of car 2 more than once, so conditions agreeing with observations would occur.

As in case (a), the third impact was not simulated, but the longitudinal force would be greater than that of the previously simulated non-eccentric case, and would be more than sufficient to break the coupler.

8.2 Discussion of results

The reconstructed sequence of events agrees well with eyewitness accounts except for the fact that the towerman thought that the small tank car had coupled into the cut of standing cars. This discrepancy is not unreasonable, considering his distance from the cars and his angle of view from the tower.

The lack of success in achieving simulation of the head puncture mechanism on the basis of all of the cars in bowl 1 being coupled prior to the impact of the two loaded tank cars is significant. This confirms an earlier finding, obtained through simulation of the East St. Louis and Decatur accidents, which indicated that head puncture would not have occurred if the light car, whose coupler punctured the tank car head, had been coupled to back-up cars at the time of impact.

9. SUMMARY AND CONCLUSIONS

The analyses presented in this report are based on a number of assumptions regarding the parameters of the impacting cars (tables 1 to 3) and the mathematical representation of impact (section 3). Consequently, the results should not be viewed as precise quantitative descriptions of forces, displacements and velocities but rather as estimates of these values, which can be refined by refining the model and obtaining more accurate values for the input parameters. The results are considered to be sufficiently accurate however, to justify some conclusions regarding the mechanism of coupler override and to provide a basis for judgement regarding the effectiveness of certain protective devices, had the impacting tank cars been equipped with such devices in the three accident cases investigated.

- The mechanism of coupler override

When a free-standing railroad car is impacted, it accelerates in the direction of impact, the car body undergoes pitching oscillations about its center of gravity and it moves up and down on its springs. A typical impact, involving a heavy, moving tank car and a light, stationary hopper car, is illustrated in figure 18. It was assumed, in this case, that coupling occurred on impact. In the interest of clarity, the vertical displacements were exaggerated however the final configuration was drawn to proper scale.

It is seen that the impacted end moves first down, then up and down again. The simulations indicated that, for a rather wide range of parameters, these motions correspond to impact in compression followed by tension and by compression again. Consequently, energy is imparted to the light car over one and a half cycles of the longitudinal impact force. This results in an increase in the amplitude of oscillations over two cycles. The diagrams also

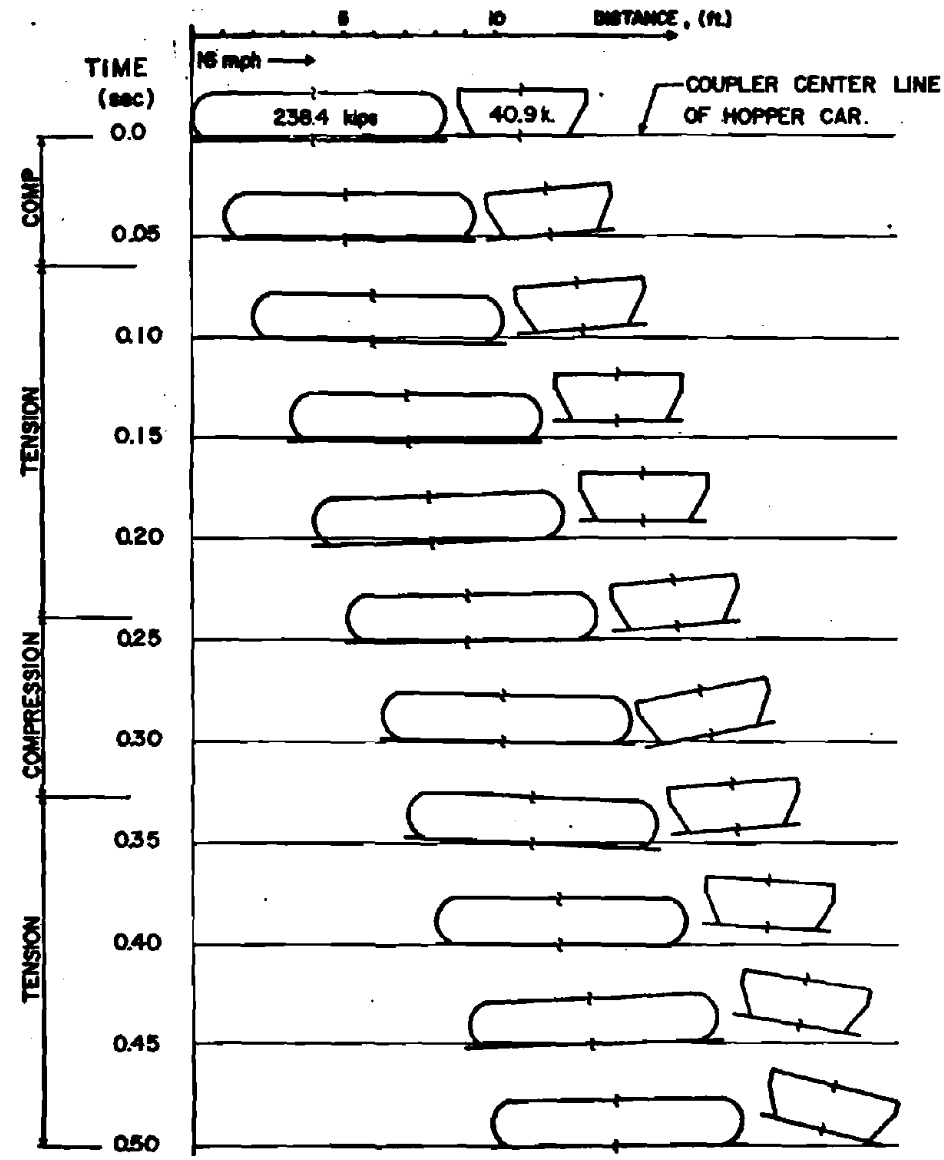


Figure 18a
Impact of heavy tank car into free-standing hopper car.
Cars not to scale. Other length scale as shown.
Note alternating compression and tension
(buff and draft) between the cars.

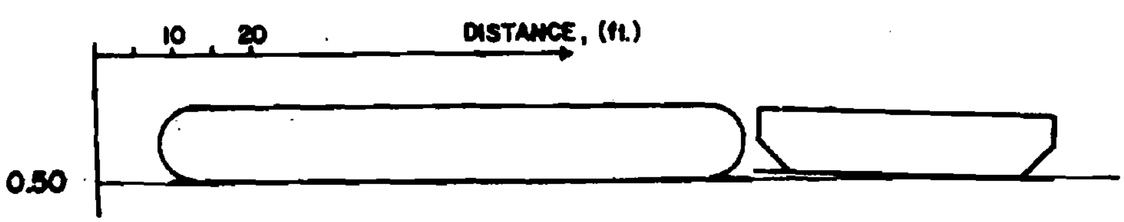


Figure 18b
Impact of heavy tank car into free-standing hopper car.
Configuration at 0.500 seconds after impact. Scale as shown.

indicate that the car body moves in such a way that, if the amplitude of motion is sufficiently large, it will loose contact with both of its trucks simultaneously. When coupling does not occur on impact, the maximum pitch of the impacted light car occurs on the first oscillation.

In each of the three classification yard accidents discussed in this report, an uncoupled, free-standing or moving light car was impacted by tank cars. The mechanism of head puncture involved consecutive impacts. The first impact accelerated the light car away from the impacting tank car and either detrucked it or caused it to impact cars standing further down the track. The impact (or impacts) resulted in bringing the coupler to an elevated position. The head of the leading hazardous material tank car was punctured or torn when it impacted the light car for the second time.

- Effectiveness of shelf couplers

Shelf couplers will be effective if coupling occurs on impact and the coupler is sufficiently strong to resist vertical loads of approximately 150,000 pounds, applied to the upper shelf, and peak horizontal loads of approximately 2.5 million pounds. The difficult requirement appears to be ensuring that coupling will occur on impact. In the East St. Louis, Decatur and Houston accidents the couplers were in compression for less than 0.10 seconds. In this period the impact caused the couplers to move suddenly downward, thus any gravity-controlled locking mechanism would have had little, if any, chance to engage. Consideration should be given to providing high capacity draft gears or other mechanical devices in conjunction with shelf couplers designed in such a way that the elasticity of impact is sufficiently reduced for coupling to occur. Such devices can be expected to reduce the likelihood of coupler override even when coupling does not occur by extending the period of contact between the impacting cars. Friction between the contacting surfaces will tend to reduce the pitching amplitude of the impacted car.

- Effectiveness of head shields

For head shields to be effective, it is necessary that they survive two consecutive impacts. In the East St. Louis accident the first impact occurred at 16 miles per hour. A head shield would not have been impacted directly at this time. The second impact, with the coupler in a raised position, occurred at approximately 14 miles per hour. To be effective, a head shield would have had to deflect the coupler or distribute the force of impact over the tank car head in such a way that puncture could not occur.

In one experiment, conducted under the RPI-AAR Railroad Tank Car Safety Research and Test Project, a loaded tank car equipped with $\frac{1}{2}$ inch head shield was impacted at 15.5 miles per hour by an elevated coupler mounted on the lead car of two rigidly coupled hammer cars[10]. The impact deformed the head shield and the $\frac{11}{16}$ inch tank car head but the head was neither torn nor punctured and no lading was lost. The kinetic energy of the impacting cars in this experiment was 2.9 million foot-pounds. By comparison, the kinetic energies in those impacts which caused lading loss at East St. Louis, Decatur and Houston can be conservatively estimated at 4.7, 5.7 and 3.6 million foot-pounds. Although the kinetic energy of the impacting cars is only one of the several factors influencing the amount of damage sustained by the tank car head, it is obvious that the experimentally established energy bound was exceeded by a large margin in each of the three accidents investigated. It is also true that had a single car been switched in each case at identical speeds, the kinetic energy would have remained below the bound of 2.9 million foot-pounds. Unfortunately, the experiment does not provide information on the amount of energy absorbed by the tank car head, which makes it impossible to draw definite conclusions regarding the effectiveness of head shields in other impacts.

- Operation of unprotected tank cars

In the absence of effective protective devices the probability of head puncture in overspeed impact situations can be significantly reduced by not permitting light cars, especially uncoupled light cars, to come in contact with hazardous material tank cars. In the opinion of the writers, tank car head punctures would not have occurred in East St. Louis, Decatur and Houston if the impacted light cars were securely coupled to back-up cars at the time of impact.

- Unsolved problems and future work.

Of course, the coupler override mechanism described in this report is not necessarily the only mechanism that may lead to head puncture. Continued study of head puncture accidents is expected to lead to an understanding of other mechanisms as well. An essential tool in this work is a mathematical model which is capable of representing the dynamics of impact adequately.

In its present form the mathematical model is capable of representing central impact but not capable of representing motions following coupler override, the attendant wedging action and inelastic structural response. Furthermore, multiple impact situations in which substantial lading shift and rapidly alternating loading and unloading of draft gears occurs, cannot be modeled accurately because the response of ladings and draft gears to rapidly alternating loads is not known. Additional work is required in these areas.

It is within the current capabilities of the model to study the effect of different types of draft gears in impacts similar to those discussed in this report. Studies to define draft gear characteristics which might permit safe humping of hazardous material tank cars are currently underway.

Other work planned under the tank car safety project includes non-destructive impact studies, improvement of the model by including rail and truck dynamics, refinement of input parameters by measurements, and extension of the model into the non-elastic range by means of analysis and destructive experiments.

10. ACKNOWLEDGMENTS

The writers wish to thank Mr. Earl A. Phillips, director of the Railroad Tank Car Safety Research and Test Project for providing detailed information on the relevant phases of the RPI-AAR tank car safety project and for granting access to the AAR accident files. The writers also wish to thank Mr. Charles E. Reedy, staff engineer, Shippers Car Line Division, ACF Industries Incorporated for providing valuable information on mechanisms of derailment and tank car head puncture.

Dr. W. P. Manos and Mr. J. B. Raidt of Pullman-Standard, a division of Pullman Incorporated, gave much valuable information on their vertical train action model and on the results of their impact studies. The mathematical model described in this report has been verified by successfully duplicating the results of the Pullman-Standard impact experiments.

Mr. Richard J. Harrison of the Federal Railroad Administration and Messrs. Richard Peterson and Elmer Garner of the National Transportation Safety Board provided very valuable assistance in compiling information on the Houston accident.

11. APPENDIX - REFERENCES

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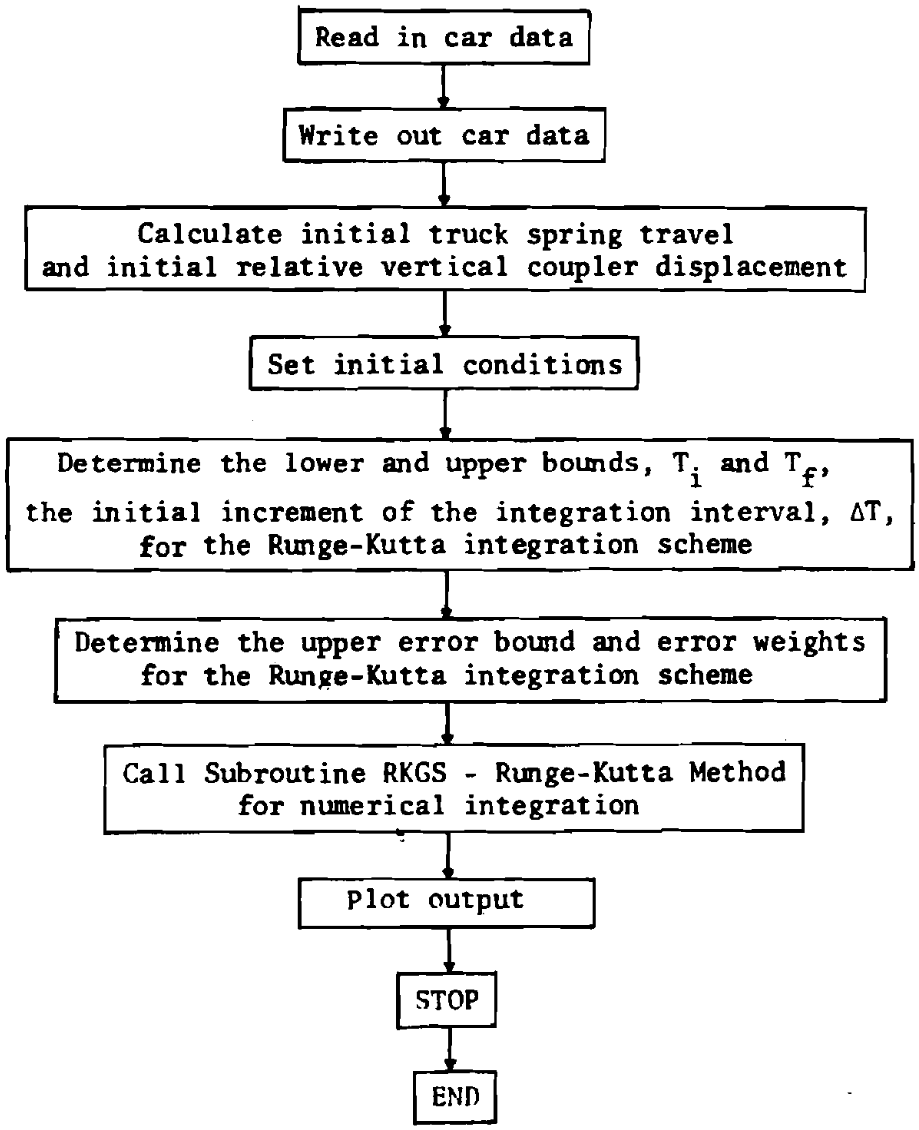
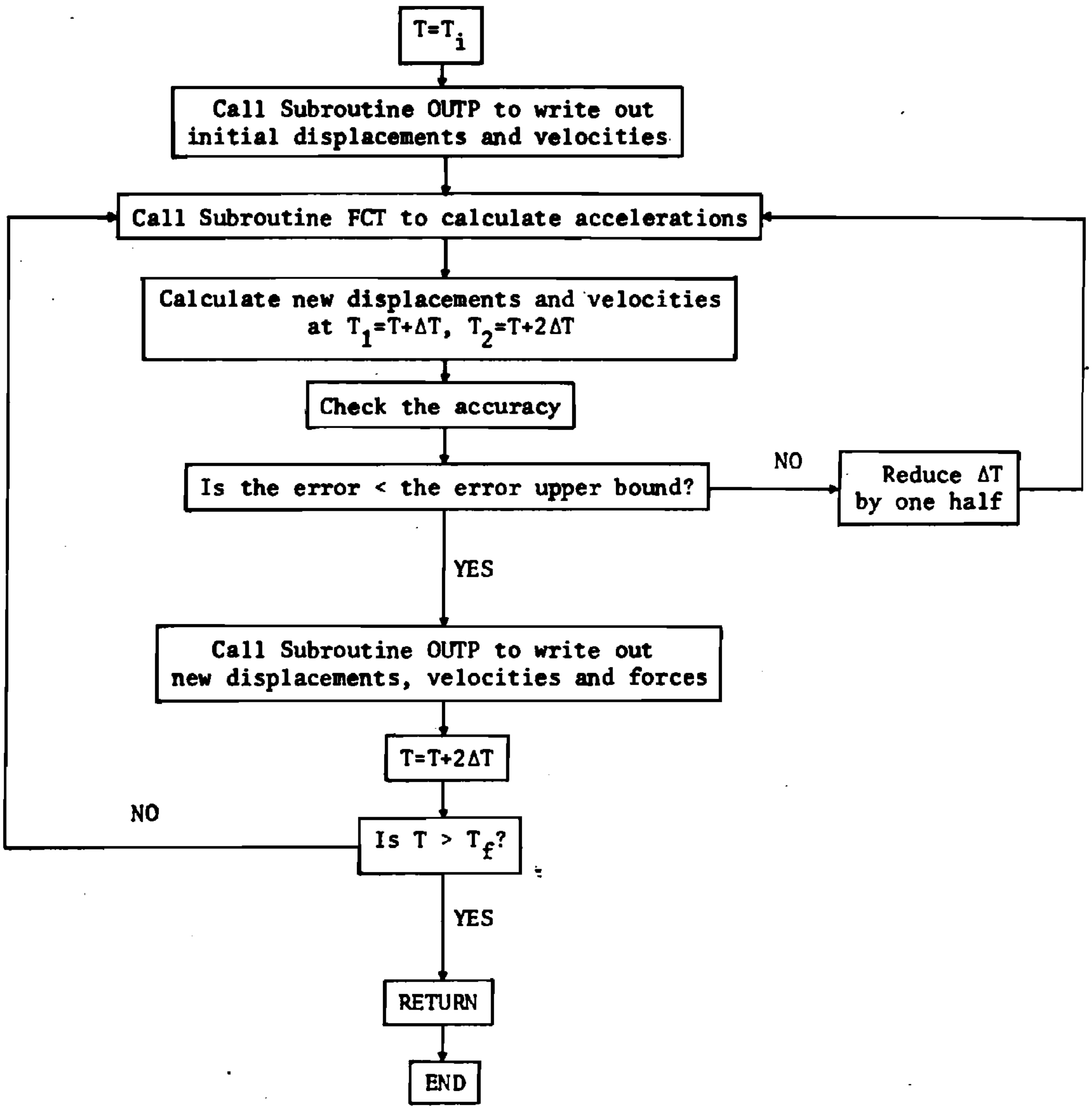
(10) Final Phase 05 Report on Tank Car Head Study, Railroad Tank Car Safety Research and Test Project, Report No. 2A-05-1-17, July 14, 1972

12. APPENDIX - THE COMPUTER PROGRAM FLOW CHART

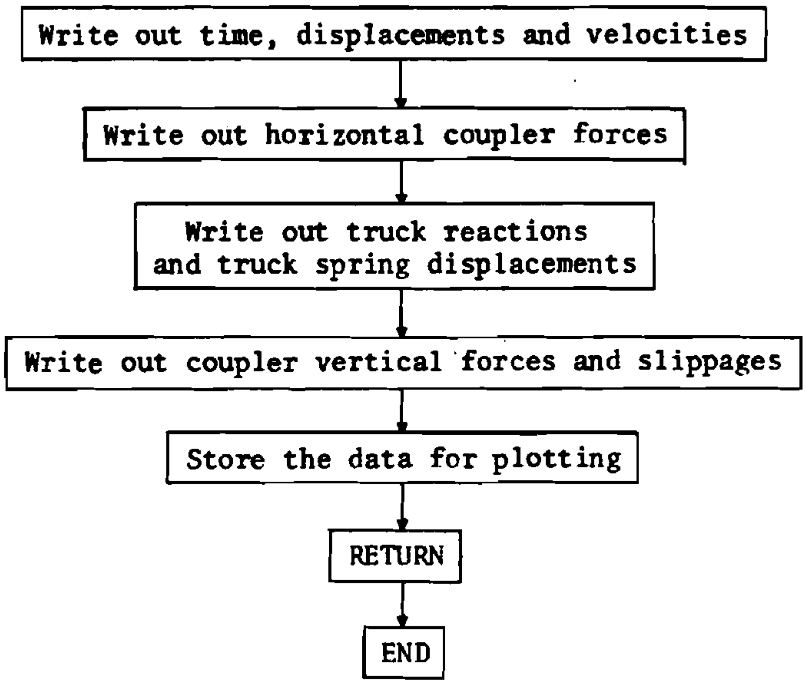
Main Program Flow Chart



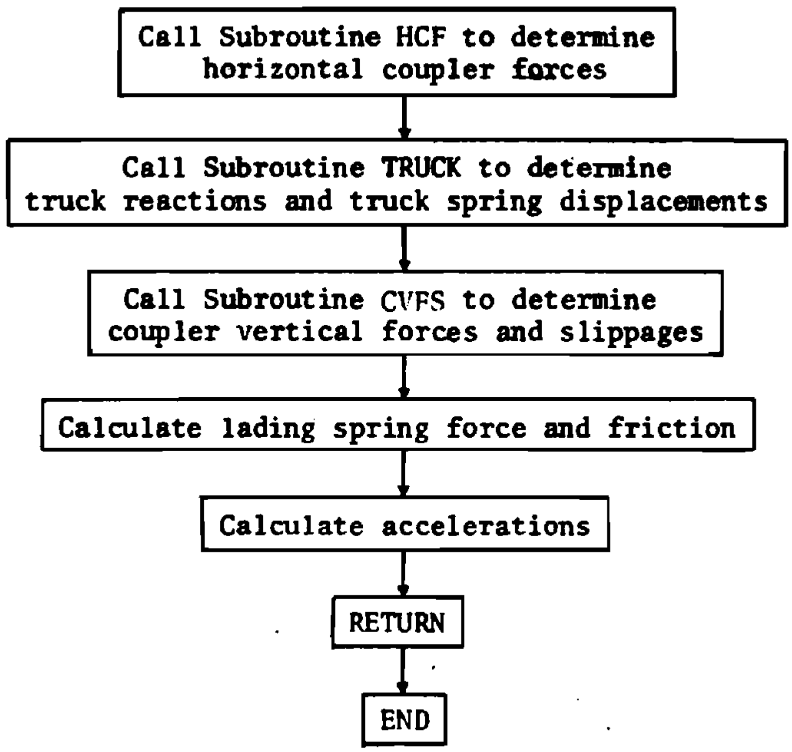
Subroutine RKGS Flow Chart



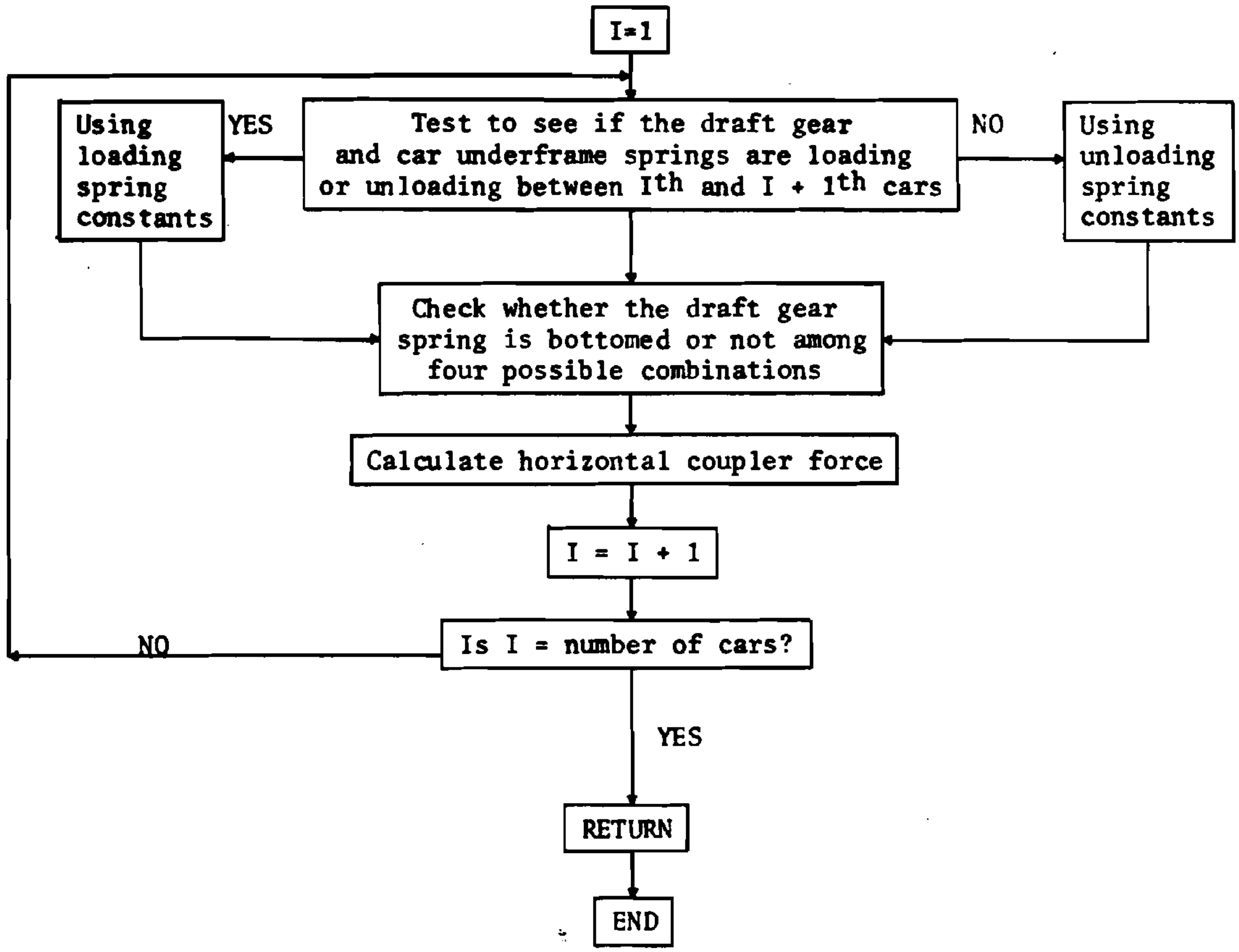
<u>Subroutine OUTP</u> Flow Chart (OUTP is called by RKGS)



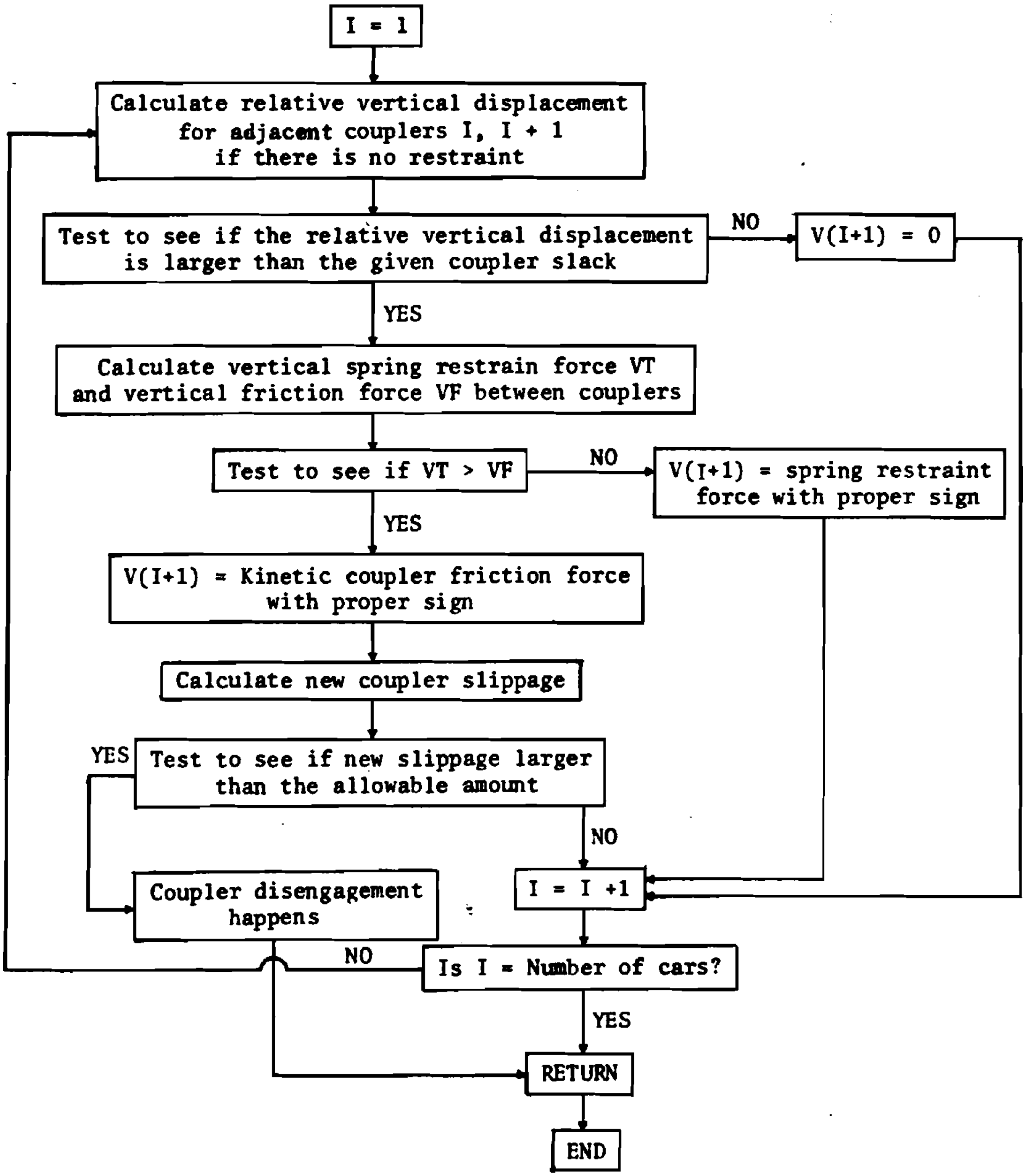
Subroutine FCT Flow Chart (FCT is called by RKGS)



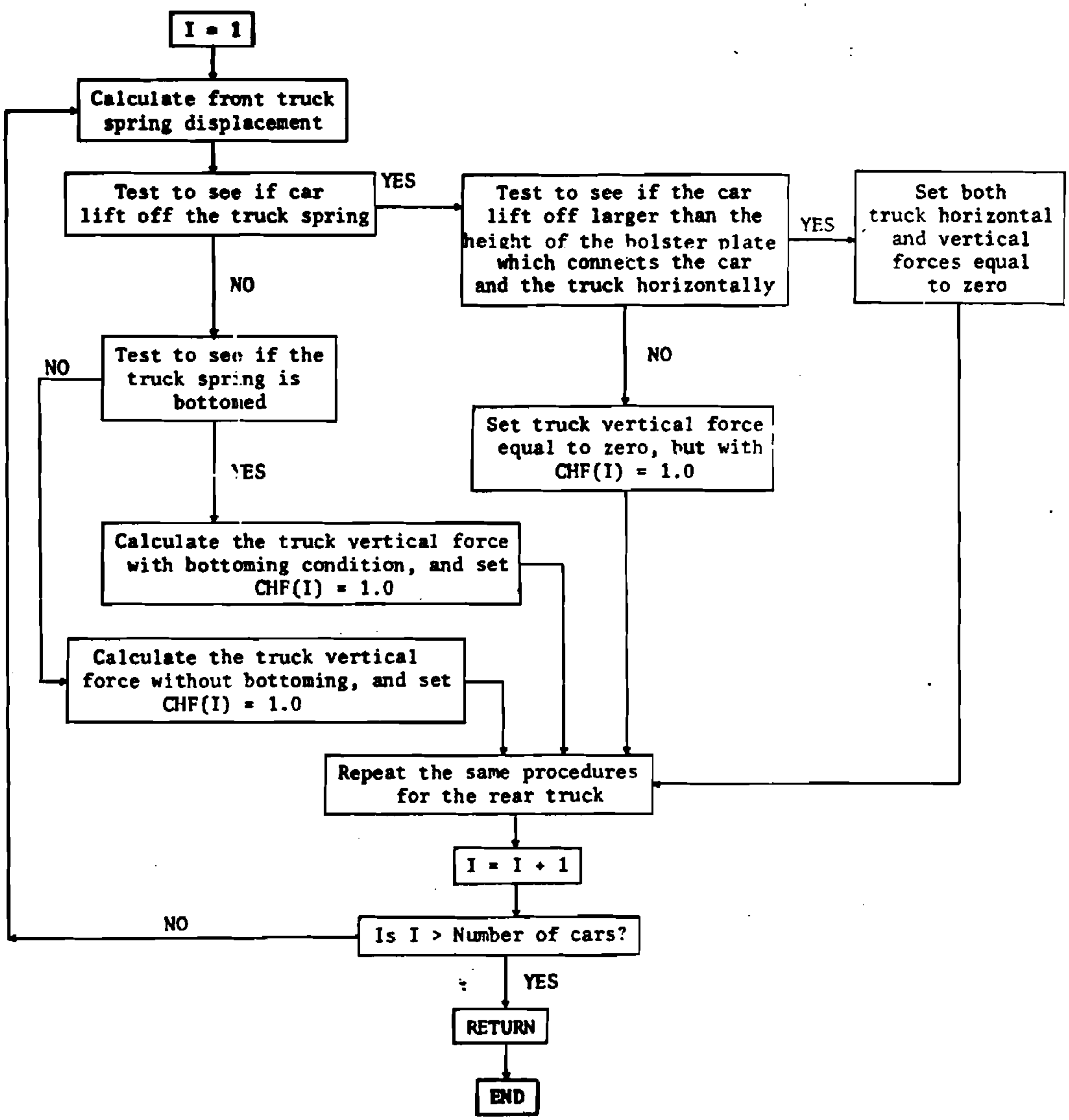
Subroutine HCF Flow Chart (HCF is called by FCT)



<u>Subroutine CVFS</u> Flow Chart (CVFS is called by FCT)



Subroutine TRUCK Flow Chart (TRUCK is called by PCT)



Note: This mathematical model does not include rail deflections and truck dynamics following a separation of the car body from a supporting truck. An expanded version of the model is currently in preparation.