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INCREASED RAIL TRANSIT VEHICLE CRASHWORTHINESS IN HEAD-ON COLLISIONS

Volume I - Initial Impact

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FINAL REPORT

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16. Abstract

A two-dimensional analytic simulation model of the leading cars of two impacting transit car consists is formulated. This model is capable of simulating the mechanics of head-on initial impact of two transit cars on straight level track. Specifically the model is capable of establishing the critical parameters which govern whether the cars crush, override or crush with subsequent override. This simulation model is used to assess impact control devices currently in service such as anticlimbers, couplers and draft gear. The assessment involves (1) isolating the critical parameters which govern whether the cars crush, override, or crush with subsequent override and (2) developing recommendations concerning future directions of effort in design of effective impact control devices.

A detailed experimental test plan for evaluating the strength and effectiveness of future designs of impact control devices is developed.

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Cambridge, Massachusetts 02142

16. Abstract

As systems manager for the Urban Mass Transportation Administration (UMTA) Rail System Supporting Technology Program, the Transportation Systems Center (TSC) is conducting research and development efforts directed toward the introduction of improved technology in urban rail system applications. As part of this program, TSC is conducting analytical and experimental studies toward improved safety in urban rail systems. A specific goal in this area of safety is to reduce the number of injuries that may result from the collision of two trains.

In this report, Volume I, a two-dimensional analytic simulation model of the leading cars of two impacting transit car consists is formulated. This model is capable of simulating the mechanics of head-on initial impact of two transit cars on straight level track. Specifically, the model is capable of establishing the critical parameters which govern whether the cars crush, override, or crush with subsequent override. This simulation model is used to assess impact control devices currently in service, such as anticlimbers, couplers, and draft gear. The assessment involves isolating the critical parameters which govern whether the cars crush, override, or crush with subsequent override and developing recommendations concerning future directions in effort in design of effective impact control devices. This report also presents a detailed experimental test plan for evaluating the strength and effectiveness of future designs of impact control devices which has been developed. This report is comprised of three other volumes: Volume II: Primary Collision (UMTA-MA-06-0025-80-2); Volume III: Guidelines for Evaluation and Development of New Railcar Designs (UMTA-MA-06-0025-80-3); and Volume IV: IITRAIN Users' Manual (UMTA-06-0025-80-4).

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PREFACE

As systems manager for the Urban Mass Transportation Administration (UMTA) Rail System Supporting Technology Program, the Transportation Systems Center (TSC) is conducting research and development efforts directed toward the introduction of improved technology in urban rail system applications. As part of this program, TSC is conducting analytical and experimental studies toward improved safety in urban rail systems. A specific goal in this area of safety is to reduce the number of injuries that may result from the collision of two trains.

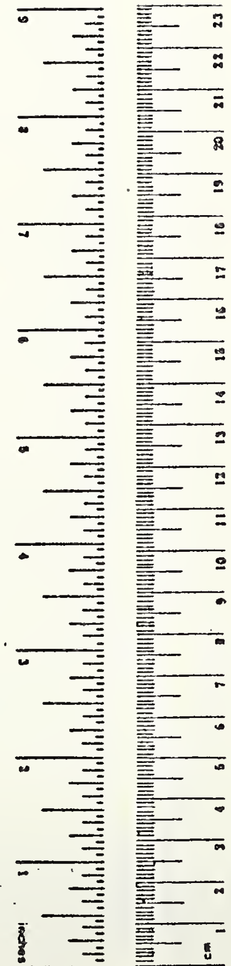
On 30 June 1975, TSC contracted with IIT Research Institute (IITRI) to perform this study to develop engineering methods and data pertaining to improved technology in urban rail systems which will lead to increased rail transit vehicle crashworthiness and passenger injury minimization. This final report is submitted in four volumes. Part 1 describes the results of Task 1 which is concerned with the initial impact of two transit cars. The results of Task 2 which is concerned with the primary collision of two impacting transit car consists are described in Part 2. Part 3 describes the results of Tasks 3 and 4 of this study which are concerned with prediction of passenger injury and guidelines for evaluation of railcar designs. The final volume is a manual containing a description of the organization and use of the IITRAIN computer code which was developed as a tool to help meet the goals of this contract.

Major IITRI contributors to the work covered in this report include Edward E. Hahn, Arne H. Wiedermann, Anatole Longinow, Robert W. Bruce and Steven C. Walgrave. The author takes this opportunity to acknowledge the contributions to this report made by Dr. A. Robert Raab, Mr. Samuel Polcari, Dr. Ming Chen, Mr. George Neat and Mr. Ronald Madigan of the U.S. Department of Transportation, TSC, Cambridge, Massachusetts.

METRIC CONVERSION FACTORS

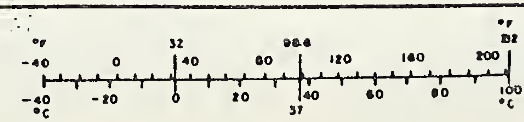
Approximate Conversions to Metric Measures

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



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1. INTRODUCTION

The collision of two consists of transit cars can be broken into three separate, but interdependent, phenomena: initial impact, primary collision, and secondary collision. Initial impact is concerned with the mechanics of the initial impact of the leading cars of two consists. The interaction of all of the cars and car components of two impacting consists comprise the primary collision. Secondary collisions include the interaction of passengers with the car components, passengers with passengers and passengers with other loose objects. This final report, submitted in four volumes, describes the results of the IIT Research Institute (IITRI) program which is concerned with the collision of transit car consists on straight level track. Part 1 of the final report is concerned with the initial impact of the leading cars of two consists. The results of the study of the primary collision of two impacting consists are given in Part 2, and Part 3 is concerned with secondary collisions including the prediction of passenger injury and guidelines for evaluation of new railcar designs. The final volume is a manual containing a description of the organization and use of the IITRAIN computer code which was developed as a tool to help meet the goals of this contract.

1.1 Program Objectives

The program objectives, as taken from the contract, are restated here.

Item 1a: Formulate an analytical model in two dimensions, longitudinal and vertical, of the leading cars of two impacting consists in sufficient detail to examine the mechanics of head-on initial impact on straight track. This model will include the distribution of mass in the cars as well as the nonlinear force-deformation relationships existing among major structural subassemblages. Consideration will be given to the shapes and configurations of the impacting surfaces and to the forces generated by the impact. The model shall be capable of establishing the critical parameters which govern whether the cars crush, displace vertically and override, or crush with subsequent override.

Item 1b: Utilize the above analytical model of initial impact to assess impact controlling devices currently in service, such as anticlimbers, couplers and draft gears of various designs. This assessment shall uncover the critical parameters of such devices which govern whether the cars crush, displace vertically and override or crush with subsequent override. The contractor shall develop recommendations concerning future directions of effort in design of impact controlling devices which would be particularly pertinent to crashworthiness goals.

Item 1c: Develop an experimental test plan for the evaluation of the strength and effectiveness of future designs for impact controlling devices. These tests are to assure that the forces generated during impact do not produce structural failure of the impact controlling device or vertical misalignment and override of the car body. The test plan is to be sufficiently detailed so that all equipment, fixtures, instrumentation and procedures are completely described.

Item 2a: Develop an analytical model in two dimensions, longitudinal and vertical, of the primary collision of two impacting consists of urban railcars of similar and different configurations. This model will include the formulation of the leading cars developed in Part 1 of this program, as well as the distributions of mass and nonlinear force-deformation relationships existing among major structural subassemblages. This model shall be capable of determining the extent of crushing and/or override suffered by the individual cars in the consists, as well as the time histories of displacement, velocity, and acceleration in both the longitudinal and vertical directions.

Item 2b: Develop methods for generating the dynamic force-deformation relationships for structural subassemblages comprising the critical modules of railcars. These methods shall include finite-element analysis, scale modeling and full-scale testing procedures including specifications for required testing equipment and instrumentation. Utilize the finite-element analytical method to generate the nonlinear force-deformation relationships among major components of a typical urban railcar.

Item 3: Develop the analytical methodology of passenger injury due to secondary collision to include modes of injury due to longitudinal, vertical, and pitching motions of the vehicles after impact. This methodology shall be capable of considering the location of the passenger prior to impact, his orientation (seated, standing, facing forward, facing sideways, facing rearward), the configuration of interior features of the cars, passengers density, and passenger restraint. This methodology shall also be capable of determining the severity of the injury sustained by the passenger.

Item 4: Utilize the results of Items 1 through 3 to develop guidelines for the evaluation of proposed railcar designs, and guidelines for the development of new railcars. These guidelines are to be developed in parametric form, so that individual parameters may be considered and the effects of specific values assigned or computed for these parameters may be assessed. These parameters are to include:

a - the number of cars in the consist
b - operational velocity ranges
c - dimensions and weights of each car
d - placement and dimensions of windows and doors
e - placement and weights of mechanical/electrical equipment
f - interior configurations of passenger compartment
g - carbody force-deformation relationships among major structural subassemblages
h - locations of carbody centers of gravity (c.g.)

1.2 Report Organization

This portion of the final report describes the work conducted under Items 1a, 1b and 1c. In order to meet the objectives of these tasks it was necessary to develop a computer code to carry out the required calculations. Section 2 of this final report describes this computer code. The development of the transit car model to be input to this code for the purpose of assessing impact control devices is described in Section 3. Section 4 gives the parametric analysis used to assess impact control devices. Recommendations for the design of impact control devices are given in Section 5. A test plan for the evaluation of the strength and effectiveness of future impact control device designs is given in Section 6.

2. DEVELOPMENT OF COMPUTER METHODOLOGY

2.1 General Considerations

The task of developing an accurate computer simulation of a head-on railcar crash poses many difficult problems due to the complexity of the railcar interactions and the lack of information on the mechanisms causing crush, override or crush with subsequent override. The effects of coupler motions, draft gear behavior, sill flexibility, truck dynamics, braking action, rail flexibility, c.g. locations and initial conditions for the positions of all components must be accounted for in any realistic simulation. Many of these factors which affect crash dynamics are highly nonlinear and may also be very sensitive to small changes in the initial conditions just prior to impact.

Some significant research in the area of railcar crash dynamics has been conducted during the past few years. Boeing-Vertol (Ref. 1) has conducted studies funded by DOT/TSC which attempted to identify significant parameters affecting crashworthiness of rail vehicles. Locomotives, freight cars, long distance passenger cars, and urban transit cars were all considered and it was concluded that, among these, crashworthiness of the urban transit car was the area which offered the greatest probability of reasonably immediate success.

Calspan (Ref. 2) also pursued research funded by DOT/TSC in the urban railcar area. This research covered three broad categories: crashworthiness of urban railcars; state of the art crash energy management devices for urban railcars; and parametric structural studies of urban railcars.

The RPI/AAR Railroad Tank Car and Safety Research and Test Project included a preliminary study of computer simulation of vertical motion during impact (Ref. 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 mph. The measured horizontal and vertical impact forces agreed reasonably well with the computer generated forces.

Washington University has also developed a computer crash model (Ref. 4). The basic assumptions for Szabo's model are the same as for Raidt's model described above. 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 entire analysis is linear except for hysteresis losses in the draft gear, friction between lading and the car bottom and lifting of the car body from the draft gear.

None of the described models have been successful in simulating head-on railcar crashes at any significant speed, particularly with respect to predictions of override. Some probable causes of the lack of accuracy of present models in this respect include assumptions of linearity, neglect of track elasticity, lack of control of initial conditions (i.e., draft gear positions), insufficient detail in the local interaction of couplers and other contacting appurtenances and accurate representation of the input parameters for the model.

To solve or circumvent the many difficulties which arise in the simulation of railcar crashes, IITRI chose a developmental approach to the computer model formulation and implementation. For the first stage of this development, simplified computer modules were written to simulate each of the subcomponents associated with the overall model. An executive program was also written to control all calculations. This modular form of computer analysis allows ease of modification of the analytical model. The use of simplified, but realistic, subcomponent computer modules enabled the completion of a running computer program at an early stage in the project.

This computer program was exercised to study railcar crash dynamics and the computer results were analyzed critically to determine which modules needed to be modified to successfully simulate a head-on crash. Modifications were carried out and the resulting simulation further evaluated until a satisfactory simulation was obtained.

2.2 IITRAIN Computer Code

2.2.1 Program Organization-A lumped mass approach to the model formulation was selected but the procedure will allow finite elements to be used if required, resulting in a "hybrid" formulation. The main program modules and their corresponding functions are shown in Table 1. Figure 1 is a control diagram showing the manner in which the various modules interact.

TABLE 1.-PROGRAM MODULES

Name	Function
EXEC	Controls overall program
INPT	Controls input and echo print of input
INIT	Initializes program variables
INTG	Controls integration scheme
EULR	Euler integration subroutine
FINT	Controls internal force calculation
FEXT	Controls external force calculation
ACCL	Computes accelerations
OUTP	Controls Output
FNSH	Terminates the calculations and saves information required for further processing of output

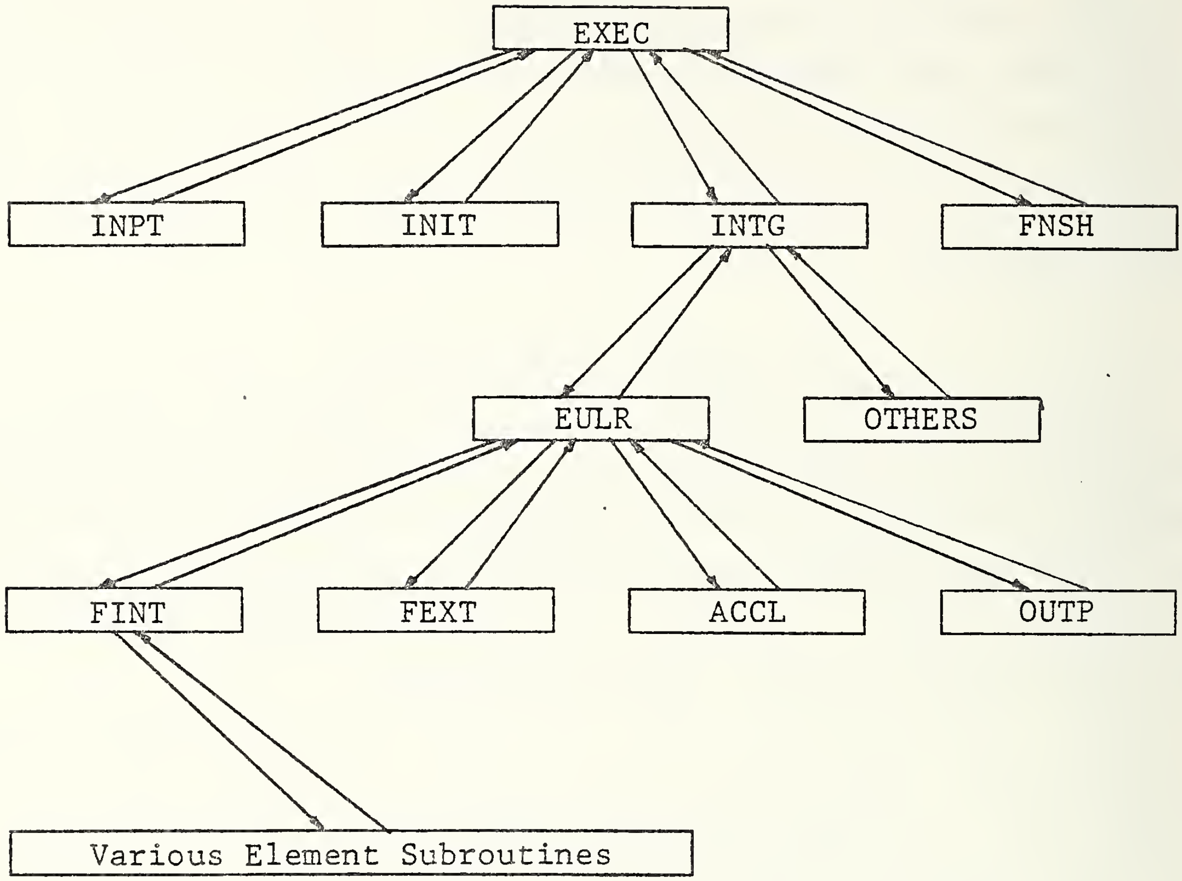


FIGURE 1. IITRAIN CONTROL DIAGRAM

2.2.2 Capabilities and Limitations-The IITRAIN computer code is designed to simulate a system of m masses, connected by n elements, subjected to applied external forces (and moments) specified as functions of time. Each of the individual mass degrees of freedom can be constrained or given specified initial conditions. At present, motions of the masses as functions of time cannot be specified. Limitation of the number of masses, m, and the number of elements, n, are only dependent on FORTRAN dimension statements and the storage capability of the computer being used.

The program is designed so that the user has his choice of integration procedures. However, at present, the only option available is simple Euler integration. Another option provided for in the program organization is the capability of integrating the equations of motion for different masses over different time intervals.

Both printed motion and force output are available. Displacements, velocities and/or accelerations (linear and rotational) of any number of points on any number of masses (limited by FORTRAN dimension statements and computer storage limitations only) can be called out. Also the internal forces and moments in any number of connecting elements can be specified as output. No graphical output is, as yet, available but the capability of adding either printer-plotter or Calcomp graphs is built into the code.

Interaction among masses is provided by connecting elements. Connecting points on the masses are specified in a local coordinate system with origin at the mass c.g.. A single connecting element can connect up to three masses at up to three connecting points per mass. Figure 2 is a schematic showing a possible system of three masses connected by a single element.

There are two general classifications of element types which must be considered; deformable elements such as linear springs or elastic-plastic beams and constraint elements such as pinned joints. The internal forces in deformable elements can be determined from the element properties and the state variables of the masses to which it is attached while the constraint element internal forces are determined from the kinematic relationships expressing the constraint imposed between masses.

There are 20 connecting elements available in the IITRAIN computer code. These elements are listed in Table 2. Element types 6 through 10 in Table 2 are constraint elements while the remaining are deformable elements.

2.2.3 Train Model Validation-Since no data were readily available for a passenger car collision, it was decided to attempt to model freight car crashes for computer program validation purposes. The East St. Louis accident modeled in Ref. 4 and the instrumented experimental test impact of freight cars conducted by Pullman Standard (Ref. 3) were chosen.

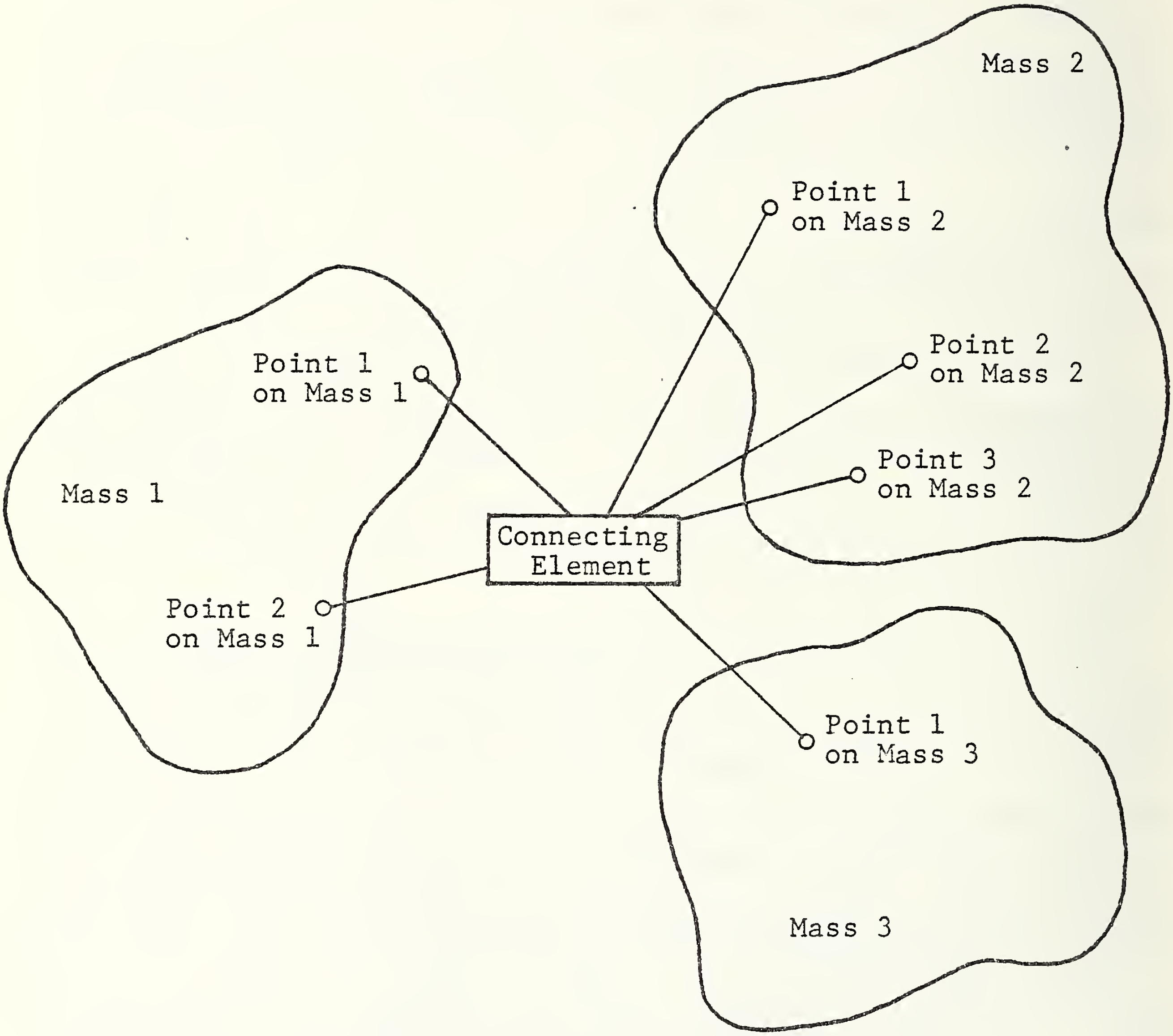
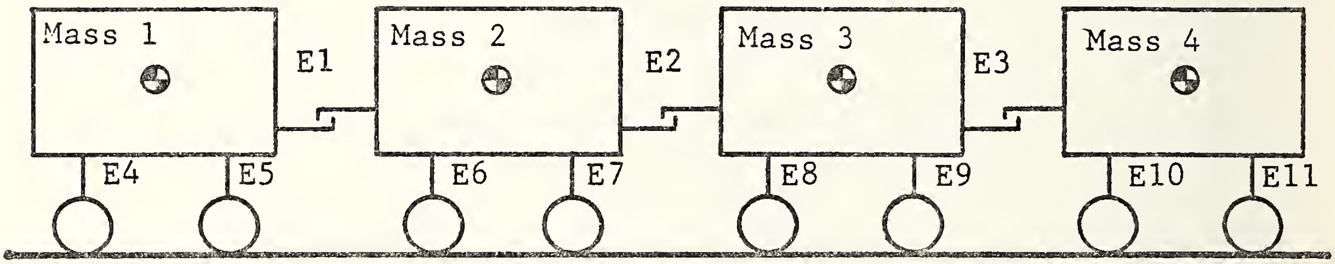


FIGURE 2. SAMPLE CONNECTION AMONG MASSES

TABLE 2.-IITRAIN ELEMENTS

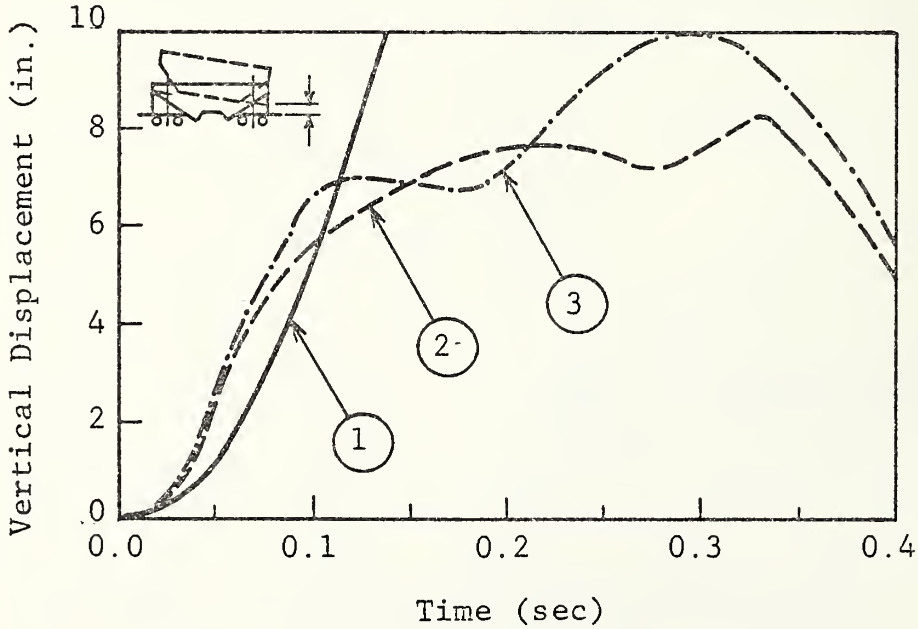
Type	Description
1	Linear spring
2	Linear dashpot
3	Torsional spring
4	Torsional dashpot
5	Elastic-plastic beam
6	Pin joint
7	Slider joint
8	Sliding pin joint
9	Double slider joint
10	Rigid joint
11	Type 1 coupling
12	Type 2 coupling
13	Type 3 draft gear
14	Type 3 coupler end element
15	Type 1 anticlimber
18	Wheel-rail interaction
19	Nonlinear spring
20	Nonlinear dashpot
21	Special linear spring
22	Tapered beam element

East St. Louis Accident: The IITRAIN model for the East St. Louis accident is shown in Figure 3. A simple linear spring coupler was used for the first attempt at modeling this accident. The results of this simulation, given in Figures 4 and 5, show the same trends as those of Ref. 4 results during the early part of the simulation but deviate radically after the first 100 msec. A second simulation was then attempted with a more realistic coupler element. This element includes the nonlinear horizontal spring characteristics of the draft gear-coupler combination, horizontal and vertical slack, vertical flexibility, coupler misalignment, and coupler sliding friction. The results of this simulation are also shown in Figures 4 and 5. Good correlation with Ref. 4 results are obtained especially during the early part of the simulation.



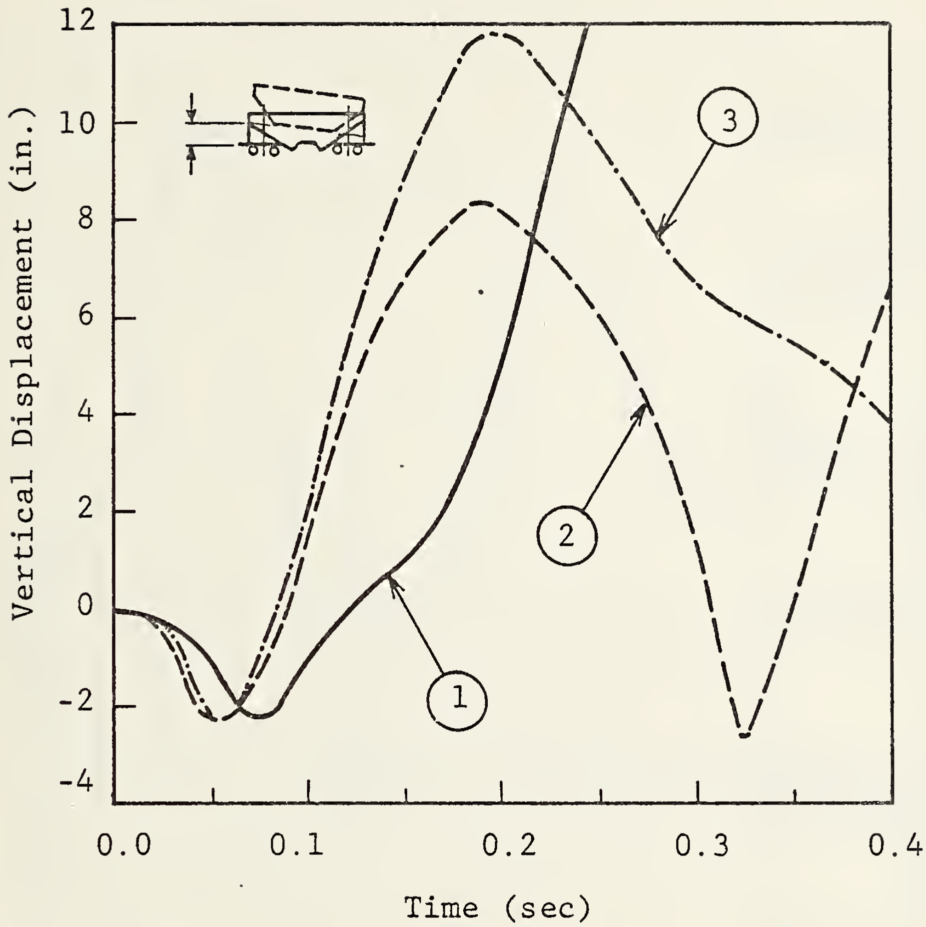
Elements E1-E3 Type 1 Coupler or Linear Spring Coupler
Elements E4-E11 Wheel Rail Interaction

FIGURE 3. IITRAIN VALIDATION, EAST ST. LOUIS ACCIDENT



Results: 1. IITRAIN Linear Coupler
2. Ref. 4
3. IITRAIN Type 1 Coupler

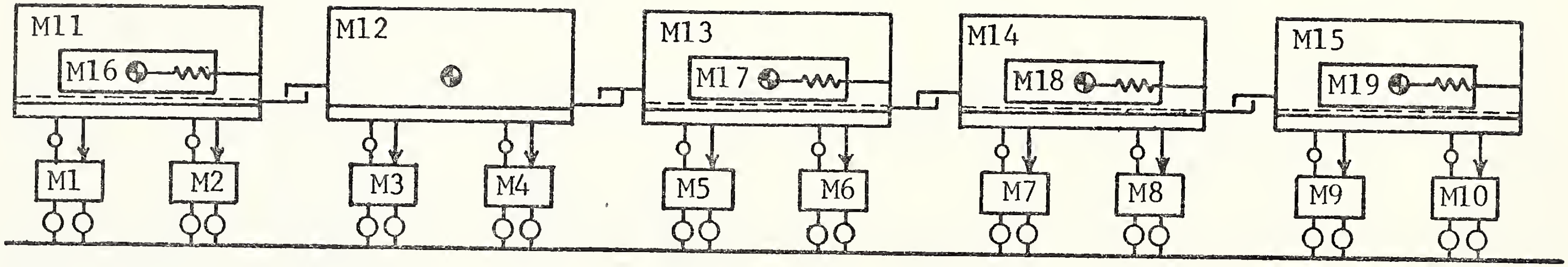
FIGURE 4. CAR BODY CENTER PLATE DISPLACEMENT AT LEADING (FREE) END OF HOPPER CAR



Results: 1. IITRAIN Linear Coupler
2. Ref. 4
3. IITRAIN Type 1 Coupler

FIGURE 5. CAR BODY CENTER PLATE DISPLACEMENT AT TRAILING (IMPACT) END OF HOPPER CAR

<u>Pullman Standard Test</u>: A second validation run was made to compare IITRAIN results with the experimental results for an instrumented impact of freight cars conducted by Pullman Standard (Ref. 3). The model for this validation is shown in Figure 6. Figures 7 and 8 show the horizontal forces in the couplers of the struck car. Extremely good correlation was obtained between the computer results and the test results for the force at the struck end of the car. The horizontal force in the backup coupler also agreed well but the first peak was slightly high. The vertical coupler force at the struck end of the car is shown in Figure 9.



<u>19 Masses (M)</u>:

M1 - M10: unsprung truck masses
M11 - M15: car body masses
M16 - M19: car lading masses

<u>52 Connecting Elements (E)</u>:

E1 - E20: wheel rail interaction (Element Type 6)
E21 - E30: nonlinear spring (Element Type 9)
E31 - E40: sliding pinned joint (Element Type 7)
E41 - E44: sliding joint (Element Type 8)
E45 - E48: linear spring (Element Type 1)
E49 - E52: Type 1 coupling (Element Type 11)

FIGURE 6. PULLMAN STANDARD MODEL DESCRIPTION

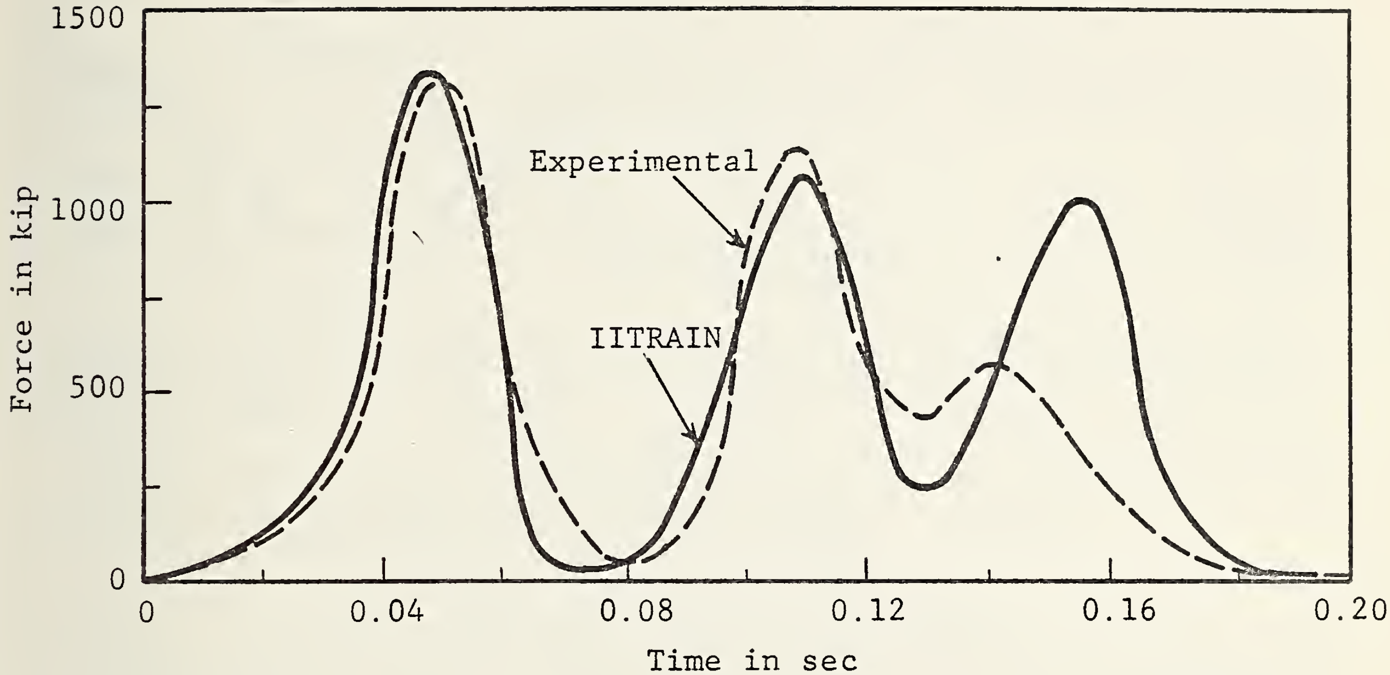


FIGURE 7. PULLMAN STANDARD TEST RESULTS, HORIZONTAL COUPLER STRIKING FORCE

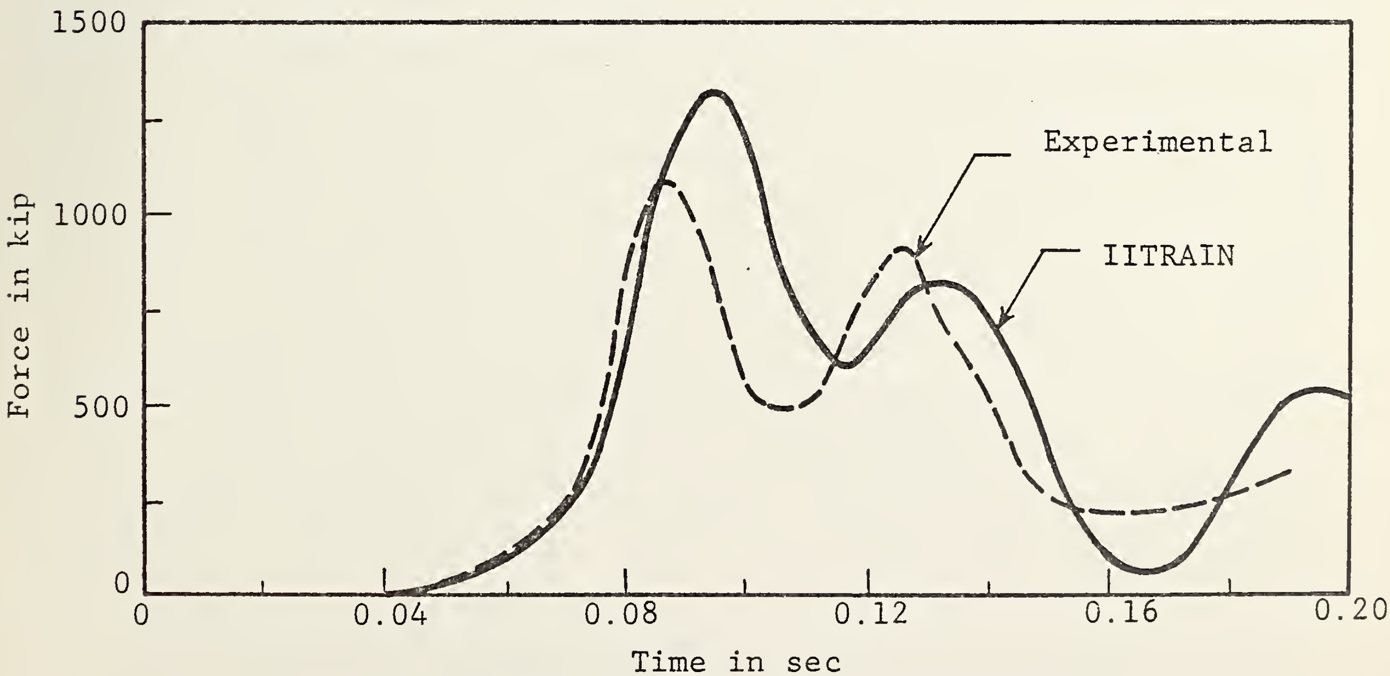


FIGURE 8. PULLMAN STANDARD TEST RESULTS, HORIZONTAL COUPLER BACKUP FORCE

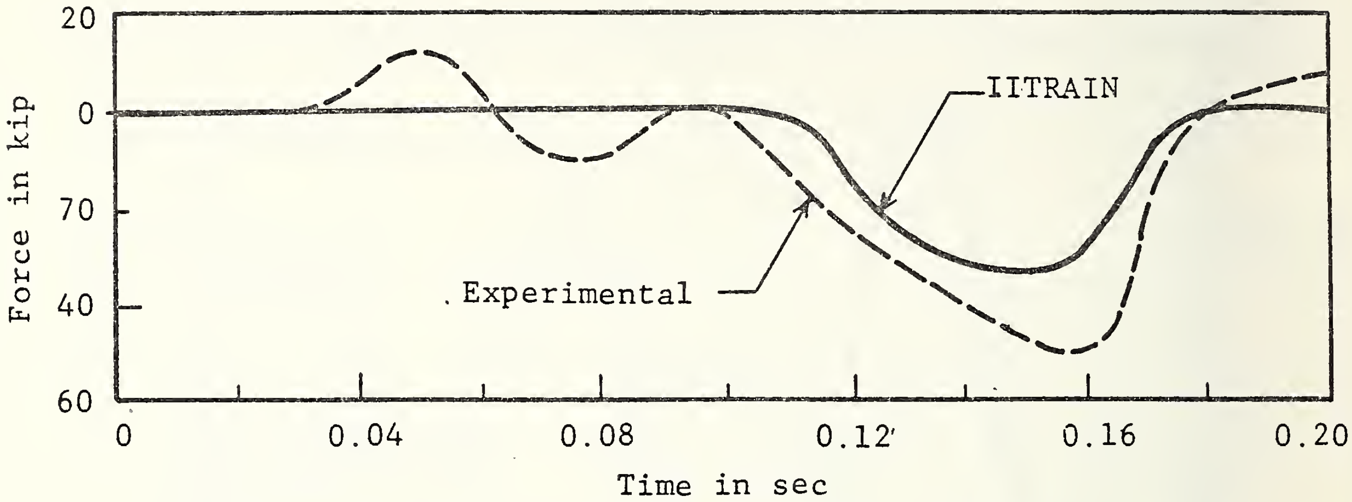


FIGURE 9. PULLMAN STANDARD TEST RESULTS
VERTICAL COUPLER STRIKING FORCE

The computer results show somewhat less vertical force than obtained from the experimental measurements. Part of the difference may be due to experimental error. In particular, the experimental results prior to 0.09 sec should be zero, since the coupler is in a zone of vertical free play during this period.

3. DEVELOPMENT OF SIMULATION MODELS

Since the dynamics of head-on railcar collisions are not fully understood with regard to crush and override, a very detailed model of the two impacting transit cars was developed. This model was exercised to determine the refinement required for an adequate crash simulation. A second model was then constructed maintaining the detail required for an accurate prediction of crush and override but simplified where this detail was found unnecessary. This simplification reduced the total system size and complexity enabling economical computer simulation while maintaining simulation accuracy. In the following sections the two models developed are described.

3.1 Preliminary Transit Car Model

The initial detailed transit car model is shown in Figure 10. The transit car body is modeled as six masses having both linear and rotary inertia to account for the distribution of the mass throughout the car body. These masses are rigidly connected to a series of five elastic-plastic beams representing the superstructure and five elastic-plastic beams representing the floor and understructure. The rigid connection between the masses and the beams allows shear to be transmitted from the substructure to the superstructure to simulate the action of the side panels of the car. The axial crushing and vertical bending of the superstructure and floor and understructure observed at the January 9, 1976 Chicago Transit Authority (CTA) accident can be reproduced with this model since the elastic-plastic beam elements allow large deflections, axial crushing and large plastic deformation in bending. In particular large plastic deformation in the understructure near the truck centerline is possible with this model and different directions of bending of the beams are possible on each side of the truck attachment.

An anticlimber element allows interaction between the anticlimbers on two impacting cars. This element represents the action of the anticlimbers and of part of the end understructure of the

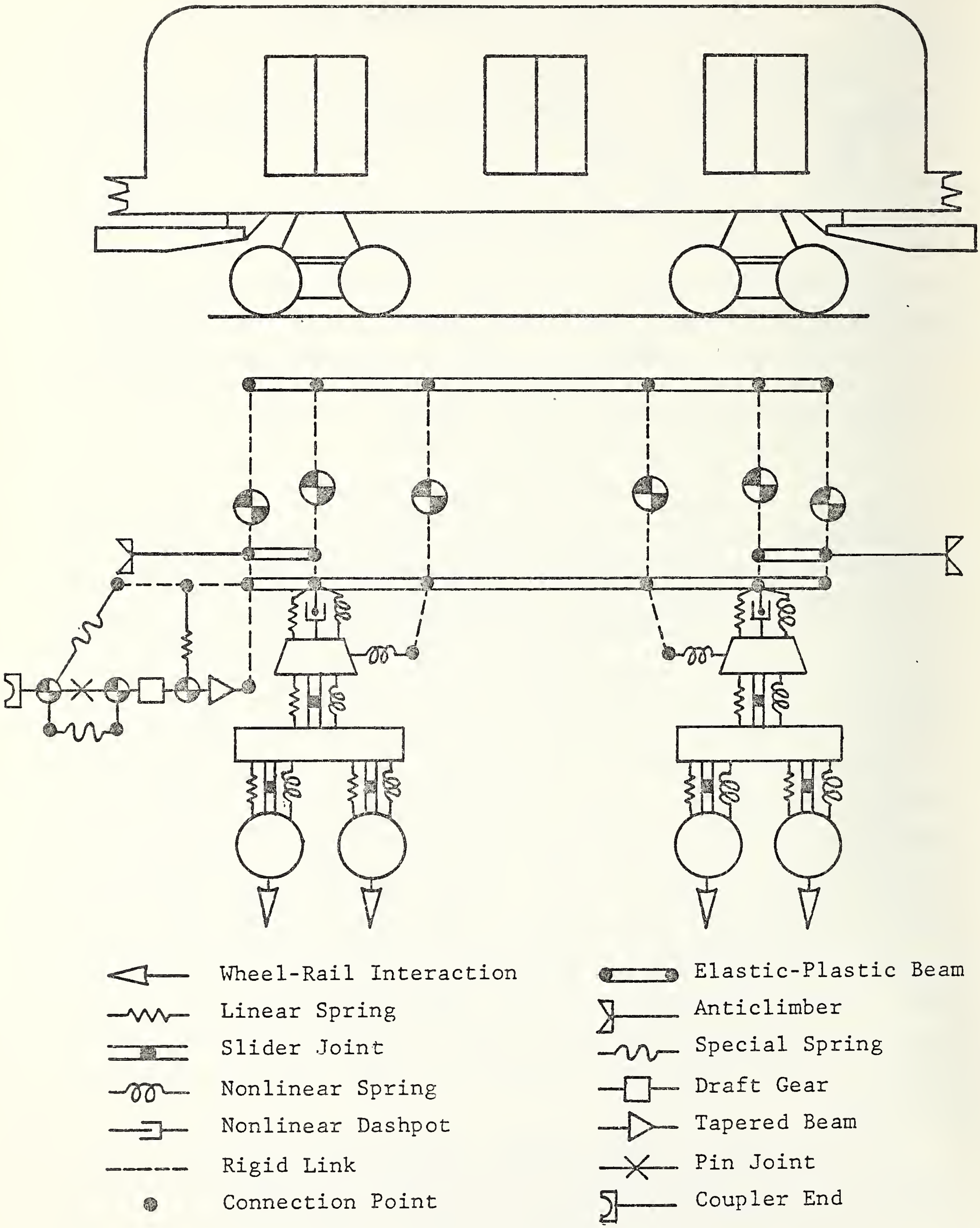


FIGURE 10. TRANSIT CAR MODEL

car. This element has independent nonlinear force-deformation relationships for vertical, horizontal and rotational deformations. The computer program internal logic checks at each time step to determine if the anticlimbers are in contact, if override occurs, and if the anticlimbers fail in either shear, crushing or bending. Vertical, horizontal and torsional loads are all accounted for at the anticlimber face in this analysis.

Identical models are used for each of the two trucks. The wheel-axle combinations are considered as rigid bodies having both mass and rotary inertia. The wheels are connected to the rails with wheel-rail interaction elements which account for the stiffness and damping of the wheel-rail combination and any braking between the wheel and rail. The wheel-axles are connected to the truck frame with a slider element allowing only vertical motion relative to the frame, a linear spring representing the journal box coil spring, and a nonlinear spring representing the limit stops on the journal box springs. The truck frame is attached to the truck bolster with a slider allowing only vertical motion, a linear spring representing the spring loaded locking center pin, and a nonlinear spring representing the limit stops for its travel. Finally the truck bolster is connected to the car underframe with a longitudinal anchor element representing the longitudinal anchors, a nonlinear dashpot representing the shock absorbers, and a linear and nonlinear spring combination representing the coil springs, the air springs, the spring limit stops and the safety trap.

The coupler draft gear assembly is being modeled as a series of separate elements and masses to allow the possibility of those actions observed in the CTA accident. Figure 11 shows the coupler model. Each coupler will consist of three masses and six elements. The draw bar draft pocket assembly is modeled as a mass which is connected to the car body by a tapered beam representing the draw bar, and is connected at the rail slides by a spring element. The yoke assembly, the second mass, is connected to the draw bar by a draft gear/shear pin element. The coupler proper, the third mass, is connected to the yoke by a pin joint and a special spring element which represent the leveling spring contact in the yoke assembly.

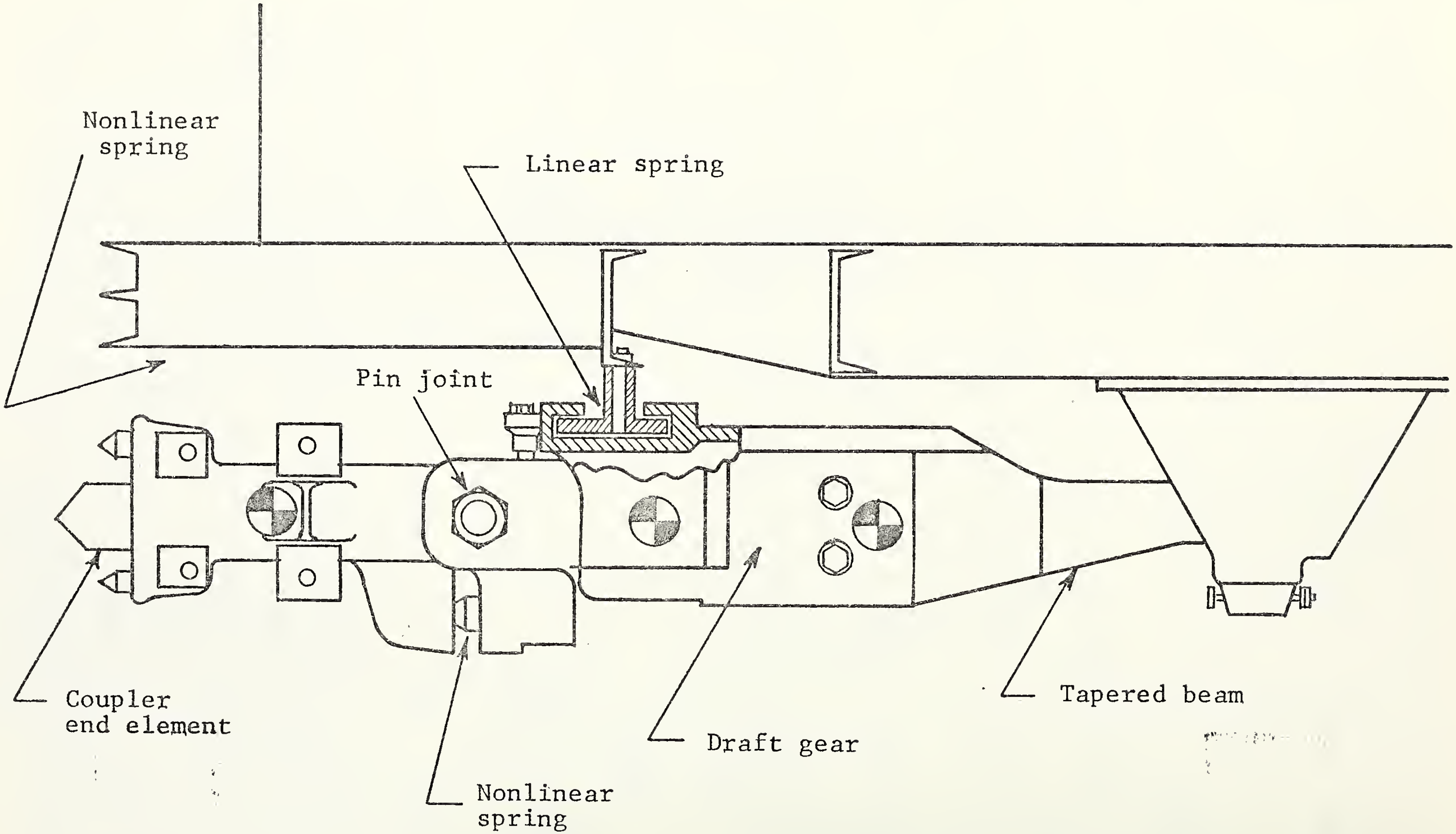


FIGURE 11. COUPLER ASSEMBLY MODEL

A second special spring represents contact of the coupler with the underside of the end sill. The interaction between coupler faces is then handled with a coupler end element which accounts for free play, elastic deformation, shear and friction between the two faces.

The geometric and physical properties necessary to describe this model were obtained for a representative transit car. Although other car designs may vary from these parameters, it is felt that the results of the analysis will be typical of transit cars in general. Table 3 gives the weight and inertia properties of the various masses composing the model. Table 4 lists the locations of the attachment points relative to the mass c.g. for the elements interconnecting these masses. The physical properties of the interconnecting elements are given in Table 5.

The coupler mass weights were approximated through a composite volume times density technique. The volumes of major sections of the coupler components were obtained from the assembly drawing of the coupler assembly. The distribution of the car body mass was accomplished by reviewing the body and frame drawings and then assigning weight factors for the general body and for all major subassemblages. This technique was also used for the trucks. The moments of inertia were then approximated using the individual masses. The procedure of sectioning and weight factoring also yields the positions of the individual mass c.g.

The physical properties of the suspension springs and dampers were obtained directly from component data. The spring constant of the wheel-rail element was obtained from a general force-deflection analysis of the wheel-rail interaction. The force-deflection characteristics of the anticlimber were generated through a finite element analysis of the assembly. The properties of the beams were set to yield areas and area moments of inertias equal to the beam being represented. The beams interconnecting the body masses are representative of the major longitudinal structural members of the car body. The box representation of the beams, as used here, assumes purely elastic loading. This case was found to be true after investigating the maximum stresses in the multisection, nonsymmetric beam models. The nonlinear spring representing the truck anchors includes a moderate initial stiffness representing the compressions of the rubber mounting system after which high stiffness compression of the metal strut occurs. The impacting coupler ends were modeled to yield high forces at low deflections. They also included a low coefficient of repercussion to allow for a low force recovery. The other elements used in the model are representative of particular components and the properties are based on physical data for that component.

TABLE 3.-PRELIMINARY TRANSIT CAR MODEL MASS DATA

Description	Mass	Weight (lb)	Inertia $lb\text{-}sec^2 inch$	Global* X Position inch	Global* Y Position inch
Coupler end mass	1	75.	60.	8.35	31.6
Draft gear yoke mass	2	90.	70.	25.40	31.6
Draft gear housing mass	3	150.	100.	40.00	31.6
Front car end mass	4	5595.	3000.	42.50	58.4
Front mass over body bolster	5	2230.	2000.	109.88	80.0
Front midbody mass	6	10175.	10000.	268.88	66.8
Rear midbody mass	7	10175.	10000.	562.88	66.8
Rear mass over body bolster	8	2230.	2000.	721.88	80.0
Rear car end mass	9	5595.	3000.	789.25	58.4
Front truck bolster mass	10	2000.	800.	113.88	28.5
Front truck mass	11	2700.	3000.	109.88	22.5
Front truck-front wheel	12	4000.	2000.	68.88	14.0
Front truck-rear wheel	13	4000.	2000.	150.88	14.0
Rear truck bolster mass	14	2000.	800.	717.88	28.5
Rear truck mass	15	2700.	3000.	721.88	22.5
Rear truck-front wheel	16	4000.	2000.	680.88	14.0
Rear truck-rear wheel	17	4000.	2000.	762.88	14.0

*Global positions are measured from rail level and from the initial position of the coupler face.

TABLE 4.-CONNECTION POINT DATA

Connection Description	Element	Local* X-Position inch	Local* Y-Position inch
• Mass 1 - Coupler End Mass			
Coupling between coupler faces	Coupler end	-7.35	0.00
Pin between coupler end and draft gear yoke	Pin joint	8.70	0.00
Coupler leveler spring	Special spring (Type 1)	8.15	-6.30
Interference between coupler end and underside of end sill	Special spring (Type 2)	-8.00	5.80
• Mass 2 - Draft Gear Yoke Mass			
Pin between coupler end and draft gear yoke	Pin joint	-8.35	0.00
Coupler leveler spring	Special spring (Type 1)	-7.65	-6.30
Draft gear connection	Draft gear, slider joint (Type 1)	0.00	0.00
• Mass 3 - Draft Gear Housing Mass			
Draft gear connection	Draft gear, slider joint (Type 1)	0.00	0.00
Rail slider connection to end sill	Nonlinear spring (Type 1)	-18.00	5.30
Draw bar and draft pocket assembly connection to car body	Tapered beam	0.00	0.00
• Mass 4 - Front Car End Mass			
Draw bar and draft pocket assembly connection to car body	Tapered beam	17.5	-26.80
Rail slider connection to end sill	Nonlinear spring (Type 1)	-20.50	-18.50
Interference between coupler end and underside of end sill	Special spring (Type 2)	-40.60	-10.40
End sill/anticlimber	Anticlimber	17.50	-11.90
Roof sill beam	Beam (Type 1)	0.00	86.60
Side sill beam	Beam (Type 2)	0.00	-14.80
Draft sill beam	Beam (Type 3)	17.50	-14.80

*Local positions are measured from the mass center of gravity.

TABLE 4.-CONNECTION POINT DATA (Contd)

Connection Description	Element	Local* X-Position inch	Local* Y-Position inch
• Mass 5 - Front Mass over Body Bolster			
Roof sill beam	Beam (Type 1)	0.00	65.00
Side sill beam	Beam (Type 2)	0.00	-36.40
Draft sill beam	Beam (Type 3)	0.00	-36.40
Suspension attachment at bolster	Linear spring (Type 1) Nonlinear spring (Type 2) Nonlinear dashpot	0.00	-34.00
• Mass 6 - Front Midbody Mass			
Roof sill beam	Beam (Type 1)	0.00	78.20
Side sill beam	Beam (Type 2)	0.00	-11.80
Truck anchor connection	Nonlinear spring (Type 3)	-105.00	-44.80
• Mass 7 - Rear Midbody Mass			
Roof sill beam	Beam (Type 1)	0.00	78.20
Side sill beam	Beam (Type 2)	0.00	78.20
Truck anchor connection	Nonlinear spring (Type 3)	105.00	-44.80
• Mass 8 - Rear Mass over Body Bolster			
Roof sill beam	Beam (Type 1)	0.00	65.00
Side sill beam	Beam (Type 2)	0.00	-36.40
Draft sill beam	Beam (Type 3)	0.00	-36.40
Suspension attachment at bolster	Linear spring (Type 1) Nonlinear spring (Type 2) Nonlinear dashpot	0.00	-34.00
• Mass 9 - Rear Car End Mass			
Roof sill beam	Beam (Type 1)	0.00	86.60
Side sill beam	Beam (Type 2)	0.00	-14.80
Draft sill beam	Beam (Type 3)	-17.50	-14.80

TABLE 4.-CONNECTION POINT DATA (Concl)

Connection Description	Element	Local* X-Position inch	Local* Y-Position inch
• Mass 10 - Front Truck Bolster Mass			
Suspension attachment at bolster	Linear spring (Type 1) Nonlinear spring (Type 2) Nonlinear dashpot	-4.00	2.25
Bolster locking pin assembly	Nonlinear spring (Type 4) Slider joint (Type 2)	-4.00	-15.00
Truck anchor connection	Nonlinear spring (Type 3)	21.00	-10.50
• Mass 11 - Front Truck Mass			
Bolster locking pin assembly	Nonlinear spring (Type 4) Slider joint (Type 2)	0.00	-9.00
Front axle attachment	Linear spring (Type2) Nonlinear spring (Type 5) Slider joint (Type 3)	-41.00	5.50
Rear axle attachment	Linear spring (Type 2) Nonlinear spring (Type 5) Slider joint (Type 3)	41.00	5.50
• Masses 12 and 13 - Front Truck Axle/Wheels			
Axle attachment	Linear spring (Type 2) Nonlinear spring (Type 5) Slider joint (Type 3)	0.00	5.13
Wheel-rail interaction	Wheel-rail	0.00	0.00
• Mass 14 - Rear Truck Bolster Mass			
Suspension attachment at bolster	Linear spring (Type 1) Nonlinear spring (Type 2) Nonlinear dashpot	4.00	2.25
Bolster locking pin assembly	Nonlinear spring (Type 4) Slider joint (Type 2)	4.00	-15.00
Truck anchor connection	Nonlinear spring (Type 3)	-21.00	-10.50
• Mass 15 - Rear Truck Mass			
Bolster locking pin assembly	Nonlinear spring (Type 4) Slider joint (Type 2)	0.00	-9.00
Front axle attachment	Linear spring (Type 2) Nonlinear spring (Type 5) Slider joint (Type 3)	-41.00	5.50
Rear axle attachment	Linear spring (Type 2) Nonlinear spring (Type 5) Slider joint (Type 3)	-41.00	5.50
• Masses 16 and 17 - Rear Truck Axle/Wheels			
Axle Attachment	Linear spring (Type 2) Nonlinear spring (Type 5) Slider joint (Type 3)	0.00	5.13
Wheel-rail interaction	Wheel-rail	0.00	0.00

TABLE 5.-PHYSICAL PROPERTIES OF ELEMENTS

Coupler End			
Horizontal stiffness, end K	= 360,000 lb/inch	Horizontal stiffness, end ℓ	= 360,000 lb/inch
Free length, end K	= 1 inch	Free length, end ℓ	= 1 inch
Coupler height, end K	= 12 inches	Coupler height, end ℓ	= 12 inches
Total horizontal slack	= 0.0 inch		
Pin Joint			
Friction parameter (μR)	= 0.3		
Special Spring (Type 1)			
Compressive stiffness, compression < δ_c	= 5000 lb/inch	Preload	= 1250 lb
		Fracture load	= 400,000 lb
Compressive stiffness, compression > δ_c	= 3,000,000 lb/inch	Free length	= 1.25 inch
δ_c	= 1.25 inch		
Special Spring (Type 2)			
Compressive stiffness, compression < δ_c	= 0.0 lb/inch	Preload	= 0.0 lb
		Fracture load	= 700,000 lb
Compressive stiffness, compression > δ_c	= 175,000 lb/inch	Free length	= 4.0 inches
δ_c	= 4.0 inches		
Draft Gear			
Initial stiffness	= 24,000 lb/inch	Hysteresis load	= 10,000 lb
Travel	= 1.25 inch	Pin shear load	= 150,000 lb
Stiffness after bottoming	= 320,000 lb/inch	Postshear travel	= 1.375 inch
		Fracture load	= 250,000 lb
Tapered Beam			
Elastic modulus	= 3×10^7 psi	Height, end K	= 3.273 inches
Plastic modulus	= 180,000 psi	Width, end K	= 2.830 inches
Yield stress	= 100,000 psi	Height, end ℓ	= 5.475 inches
Ultimate stress	= 200,000 psi	Width, end ℓ	= 1.827 inch

TABLE 5.-PHYSICAL PROPERTIES OF ELEMENTS (Contd)

Anticlimber			
Vertical elastic stiffness, end K	= 175,000 lb/inch	Vertical elastic stiffness, end ℓ	= 175,000 lb/inch
Vertical plastic stiffness end K	= 1633 lb/inch	Vertical plastic stiffness, end ℓ	= 1633 lb/inch
Vertical yield deflection, end K	= 0.20 inch	Vertical yield deflection, end ℓ	= 0.20 inch
Vertical rupture deflection, end K	= 5 inches	Vertical rupture deflection, end ℓ	= 5 inches
Horizontal elastic stiffness, end K	= 4,450,000 lb/inch	Horizontal elastic stiffness, end ℓ	= 4,450,000 lb/inch
Horizontal plastic stiffness, end K	= 20,620 lb/inch	Horizontal plastic stiffness, end ℓ	= 20,620 lb/inch
Horizontal yield deflection, end K	= 0.053 inch	Horizontal yield deflection, end ℓ	= 0.053 inch
Horizontal rupture deflection, end K	= 56 inches	Horizontal rupture deflection, end ℓ	= 56 inches
Torsional elastic stiffness, end K	= 110,800 inch-lb/rad	Torsional elastic stiffness, end ℓ	= 110,800 inch-lb/rad
Torsional plastic stiffness, end K	= 25,500 inch-lb/rad	Torsional plastic stiffness, end ℓ	= 25,500 inch-lb/rad
Torsional yield deflection, end K	= 0.001 rad	Torsional yield deflection, end ℓ	= 0.001 rad
Torsional rupture deflection, end K	= 1 rad	Torsional rupture deflection, end ℓ	= 1 rad
Face height, end K	= 6 inches	Face height, end ℓ	= 6 inches
Length, end K	= 58.125 inches	Length, end ℓ	= 58.125 inches
Beam (Type 1)			
Elastic modulus	= 1×10^7 psi	Ultimate stress	= 100,000 psi
Plastic modulus	= 20,000 psi	Height	= 19.55 inches
Yield stress	= 60,000 psi	Width	= 0.676 inch

TABLE 5.-PHYSICAL PROPERTIES OF ELEMENTS (Contd)

Beam (Type 2)			
Elastic modulus	$= 1 \times 10^7$ psi	Ultimate stress	= 100,000 psi
Plastic modulus	= 20,000 psi	Height	= 9.790 inches
Yield stress	= 60,000 psi	Width	= 2.082 inches
Beam (Type 3)			
Elastic modulus	$= 3 \times 10^7$ psi	Ultimate stress	= 150,000 psi
Plastic modulus	= 180,000 psi	Height	= 10.360 inches
Yield stress	= 100,000 psi	Width	= 0.776 inch
Linear Spring (Type 1)			
Spring constant	= 3400 lb/inch	Free length	= 18.55 inches
Linear Spring (Type 2)			
Spring constant	= 15,000 lb/inch	Free length	= 9.725 inches
Nonlinear Spring (Type 1)			
Compressive constant, compression $< \delta_c$	$= 3 \times 10^6$ lb/inch	Extension constant, extension $< \delta_t$	= 75,000 lb/inch
Compressive constant, compression $> \delta_c$	$= 3 \times 10^6$ lb/inch	Extension constant, extension $> \delta_t$	= 75,000 lb/inch
δ_c	= 10 inches	δ_t	= 10 inches
Free length	= 3 inches		
Nonlinear Spring (Type 2)			
Compressive constant, compression $< \delta_c$	= 70 lb/inch	Extension constant, extension $< \delta_t$	= 70 lb/inch
Compressive constant, compression $> \delta_c$	$= 3 \times 10^6$ lb/inch	Extension constant, extension $> \delta_t$	$= 3 \times 10^6$ lb/inch
δ_c	= 2.5 inches	δ_t	= 2 inches
Free length	= 13.25 inches		

TABLE 5.-PHYSICAL PROPERTIES OF ELEMENTS (Contd)

Nonlinear Spring (Type 3)			
Compressive constant, compression < δ_c	= 500,000 lb/inch	Extension constant, extension < δ_t	= 500,000 lb/inch
Compressive constant, compression > δ_c	= 4,500,000 lb/inch	Extension constant, extension > δ_t	= 4,500,000 lb/inch
δ_c	= 0.625 inch	δ_t	= 0.625 inch
Free length	= 33 inches		
Nonlinear Spring (Type 4)			
Compressive constant, compression < δ_c	= 600 lb/inch	Extension constant, extension < δ_t	= 3,600,000 lb/inch
Compressive constant, compression > δ_c	= 3,600,000 lb/inch	Extension constant, extension > δ_t	= 0.0 lb/inch
δ_c	= 2.25 inches	δ_t	= 10 inches
Free length	= 4.87 inches		
Nonlinear Spring (Type 5)			
Compressive constant, compression < δ_c	= 0.0 lb/inch	Extension constant, extension < δ_t	= 0.0 lb/inch
Compressive constant, compression > δ_c	= 3×10^6 lb/inch	Extension constant, extension > δ_t	= 3×10^6 lb/inch
δ_c	= 1.25 inch	δ_t	= 0.0 inch
Free length	= 8.875 inches		
Slider Joint (Type 1)			
Slider length	= 10 inches	Coefficient of friction	= 0.01
Slider width	= 1 inch		
Slider Joint (Type 2)			
Slider length	= 7 inches	Coefficient of friction	= 0.01
Slider width	= 8 inches		
Slider Joint (Type 3)			
Slider length	= 6 inches	Coefficient of friction	= 0.05
Slider width	= 2 inches		

TABLE 5.-PHYSICAL PROPERTIES OF ELEMENTS (Concl)

Wheel-Rail			
Spring constant deflection < δ_L	= 3,234,000 lb/inch	Wheel radius	= 14 inches
Spring constant deflection > δ_L	= 6,000,000 lb/inch	Damping constant	= 1000 lb-sec/inch
δ_L	= 0.5 inch		
Nonlinear Dashpot			
Damping constant compression, velocity < V_c	= 1180 lb-sec/inch	Damping constant extension, velocity < V_t	= 1180 lb-sec/inch
Damping constant compression velocity > V_c	= 173 lb-sec/inch	Damping constant extension, velocity > V_t	= 173 lb-sec/inch
V_c	= 4.5 inch/sec	V_t	= 4.5 inch/sec

3.2 Final Transit Car Model

The preliminary transit car model was exercised using the IITRAIN computer code to determine its adequacy to represent vehicle motions during crash situations. The initial simulation was for an unloaded car impacting a solid barrier. The initial velocity of the car was 20 mph. For this situation, it was found that the stresses in the beams representing the superstructure and substructure were well within the elastic ranges of the beam materials. It was also noted that the deflections between the body masses were sufficiently small to realistically allow a less complicated model. Therefore, the car body was reduced to a four-mass system, as shown in Figure 12. The four masses respectively represent a mass over the impacting end sill, a mass over the impacting end bolster, a mass representing the center section of the body between the bolsters, and a mass representing the rear end of the car body.

The motions of the truck masses were also monitored during the simulation, and it was found that displacements of the axles relative to the truck frame were effectively negligible. Thus, it was concluded that the truck assembly was sufficiently stiff to be considered as a single rigid mass. This simplification, also shown in Figure 12, was thus implemented in the model in order to reduce the total system size and complexity.

The three-mass model for each coupler assembly was retained for the impacting ends of the cars in order to ensure an accurate representation of the motions and failure modes involved.

Two cars were modeled for the transit car initial impact simulation. One of these cars was assumed to be unloaded and standing motionless on the track. The second car was assumed to be loaded and moving at 35 mph when it impacted the standing car. No braking was applied to either car. Except for the passenger loading and a slight misalignment of the anticlimbers, the cars were assumed to be identical. Although the couplers of the two cars were in a position to contact, it was assumed that the couplers

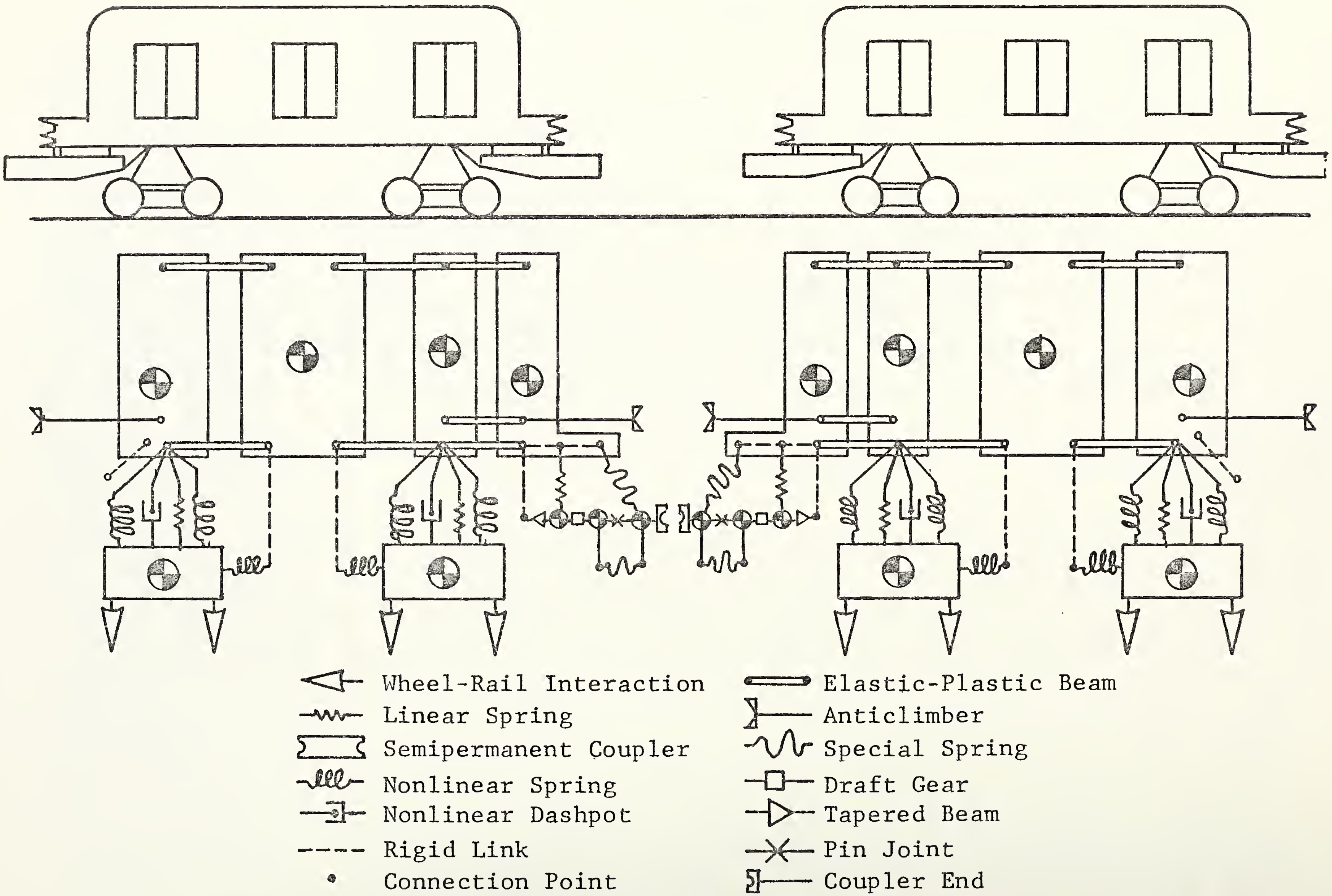


FIGURE 12. FINAL TRANSIT CAR CRASH MODEL

would not engage since engagement does not occur in actual high speed accidents. Table 6 gives the weight and inertia properties of the various masses composing the two cars. Table 7 lists the locations of the attachment points relative to the mass c.g. for the elements interconnecting the masses of the unloaded car. The attachment points for the loaded car are identical to those for the unloaded car with an adjustment for the position of the c.g. of the loaded car masses. The physical properties of the interconnecting elements are given in Table 8. The mass and physical property data were obtained by combining the data for the preliminary car model.

TABLE 6.-FINAL TRANSIT CAR MODEL MASS DATA

Description	Mass	Weight (lb)	Inertia ($lb\text{-}sec^2\text{-}inch$)	Global* X-Position (inch)	Global* Y-Position (inch)
Unloaded Car					
Coupler end mass	1	75	60	8.35	31.60
Draft gear yoke mass	2	90	70	25.40	31.60
Draft gear housing mass	3	150	100	40.00	31.60
Front car end mass	4	5,595	3,000	42.50	58.40
Front mass over body bolster	5	2,230	2,000	109.88	80.00
Front truck assembly mass	6	12,700	44,200	110.51	18.00
Center body mass	7	20,350	1,158,100	415.88	66.80
Rear body mass	8	7,825	29,750	775.06	64.60
Rear truck assembly mass	9	12,700	44,200	721.25	18.00
Loaded Car					
Coupler end mass	10	75	60	-8.35	31.60
Draft gear yoke mass	11	90	70	-25.40	31.60
Draft gear housing mass	12	150	100	-40.00	31.60
Front car end mass	13	6,180	4,000	-44.25	59.67
Front mass over body bolster	14	3,230	3,000	-109.88	77.48
Front truck assembly mass	15	12,700	44,200	-110.51	18.00
Center body mass	16	47,923	253,200	-415.88	69.71
Rear body mass	17	9,408	35,000	-773.50	65.79
Rear truck assembly mass	18	12,700	44,200	-721.25	18.00

*Global positions are measured from rail level and from the initial position of the coupler face.

TABLE 7.-CONNECTION POINT DATA

Connection Description	Element	Local* X-Position inch	Local* Y-Position inch
• Mass 1 - Coupler End Mass			
Coupling between coupler faces	Coupler end	-7.35	0.00
Pin between coupler end and draft gear yoke	Pin joint	8.70	0.00
Coupler leveler spring	Special spring (Type 1)	8.15	-6.30
Interference between coupler end and underside of end sill	Special spring (Type 2)	-8.00	5.80
• Mass 2 - Draft Gear Yoke Mass			
Pin between coupler end and draft gear yoke	Pin joint	-8.35	0.00
Coupler leveler spring	Special spring (Type 1)	-7.65	-6.30
Draft gear connection	Draft gear, slider joint	0.00	0.00
• Mass 3 - Draft Gear Housing Mass			
Draft gear connection	Draft gear, slider joint	0.00	0.00
Rail slider connection to end sill	Nonlinear spring (Type 1)	-18.00	5.30
Draw bar and draft pocket assembly connection to car body	Tapered beam	0.00	0.00
• Mass 4 - Front Car End Mass			
Draw bar and draft pocket assembly connection to car body	Tapered beam	17.5	-26.80
Rail slider connection to end sill	Nonlinear spring (Type 1)	-20.50	-18.50
Interference between coupler end and underside of end sill	Special spring (Type 2)	-40.60	-10.40
End sill/anticlimber	Anticlimber	17.50	-11.90
Roof sill beam	Beam (Type 1)	0.00	86.60
Side sill beam	Beam (Type 2)	0.00	-14.80
Draft sill beam	Beam (Type 3)	17.50	-14.80

*Local positions are measured from the mass center of gravity.

TABLE 7.-CONNECTION POINT DATA (Concl)

Connection Description	Element	Local* X-Position inch	Local* Y-Position inch
• Mass 5 - Front Mass over Body Bolster			
Roof sill beam	Beam (Type 1)	0.00	65.00
Side sill beam	Beam (Type 2)	0.00	-36.40
Draft sill beam	Beam (Type 3)	0.00	-36.40
Suspension attachment at bolster	Linear spring	0.00	-34.00
	Nonlinear spring (Type 2)		
	Nonlinear spring (Type 3)		
	Nonlinear dashpot		
• Mass 6 - Front Truck Assembly Mass			
Suspension attachment at bolster	Linear spring	-0.63	12.75
	Nonlinear spring (Type 2)		
	Nonlinear spring (Type 3)		
	Nonlinear dashpot		
Truck anchor connection	Nonlinear spring (Type 4)	20.37	0.00
Front wheel-rail interaction	Wheel-rail	-41.63	-4.00
Rear wheel-rail interaction	wheel-rail	40.37	-4.00
• Mass 7 - Center Body Mass			
Roof sill beam	Beam (Type 1)	0.00	78.20
Side sill beam	Beam (Type 2)	0.00	-23.20
Front truck anchor connection	Nonlinear spring (Type 4)	-25.20	-48.80
Rear truck anchor connection	Nonlinear spring (Type 4)	252.00	-48.80
• Mass 8 - Rear Body Mass			
Roof sill beam	Beam (Type 1)	0.00	80.44
Side sill beam	Beam (Type 2)	0.00	-20.96
Suspension attachment at bolster	Linear spring		
	Nonlinear spring (Type 2)	-53.18	-18.56
	Nonlinear spring (Type 3)		
	Nonlinear dashpot		
• Mass 9 - Rear Truck Assembly Mass			
Suspension attachment at bolster	Linear spring	0.63	12.75
	Nonlinear spring (Type 2)		
	Nonlinear spring (Type 3)		
	Nonlinear dashpot		
Truck anchor connection	Nonlinear spring (Type 4)	-20.37	0.00
Front wheel-rail interaction	Wheel-rail	-40.37	-4.00
Rear wheel-rail interaction	Wheel-rail	41.63	-4.00

TABLE 8.-PHYSICAL PROPERTIES OF ELEMENTS

Coupler End			
Horizontal stiffness, end K	= 360,000 lb/inch	Horizontal stiffness, end ℓ	= 360,000 lb/inch
Free length, end K	= 1 inch	Free length, end ℓ	= 1 inch
Coupler height, end K	= 12 inches	Coupler height, end ℓ	= 12 inches
Total horizontal slack	= 0.0 inch		
Pin Joint			
Friction Parameter (μR)	= 0.3		
Special Spring (Type 1)			
Compressive stiffness,		Preload	= 1250 lb
compression < δ_c	= 5000 lb/inch	Fracture load	= 400,000 lb
Compressive stiffness,		Free length	= 1.25 inch
compression > δ_c	= 3,000,000 lb/inch		
δ_c	= 1.25 inch		
Special Spring (Type 2)			
Compressive stiffness,		Preload	= 0.0 lb
compression < δ_c	= 0.0 lb/inch	Fracture load	= 700,000 lb
Compressive stiffness,		Free length	= 4.0 inches
compression > δ_c	= 175,000 lb/inch		
δ_c	= 4.0 inches		
Draft Gear			
Initial stiffness	= 24,000 lb/inch	Hysteresis load	= 10,000 lb
Travel	= 1.25 inch	Pin shear load	= 150,000 lb
Stiffness after bottoming	= 320,000 lb/inch	Postshear travel	= 1.375 inch
		Fracture load	= 250,000 lb
Tapered Beam			
Elastic modulus	= 3×10^7 psi	Height, end K	= 3.273 inches
Plastic modulus	= 180,000 psi	Width, end K	= 2.830 inches
Yield stress	= 100,000 psi	Height, end ℓ	= 5.475 inches
Ultimate stress	= 200,000 psi	Width, end ℓ	= 1.827 inch

TABLE 8.-PHYSICAL PROPERTIES OF ELEMENTS (Contd)

Anticlimber			
Vertical elastic stiffness, end K	= 175,000 lb/inch	Vertical elastic stiffness, end ℓ	= 175,000 lb/inch
Vertical plastic stiffness, end K	= 1633 lb/inch	Vertical plastic stiffness, end ℓ	= 1633 lb/inch
Vertical yield deflection, end K	= 0.20 inch	Vertical yield deflection, end ℓ	= 0.20 inch
Vertical rupture deflection, end K	= 5 inches	Vertical rupture deflection, end ℓ	= 5 inches
Horizontal elastic stiffness, end K	= 4,450,000 lb/inch	Horizontal elastic stiffness, end ℓ	= 4,450,000 lb/inch
Horizontal plastic stiffness, end K	= 20,620 lb/inch	Horizontal plastic stiffness, end ℓ	= 20,620 lb/inch
Horizontal yield deflection, end K	= 0.053 inch	Horizontal yield deflection, end ℓ	= 0.053 inch
Horizontal rupture deflection, end K	= 56 inches	Horizontal rupture deflection, end ℓ	= 56 inches
Torsional elastic stiffness, end K	= 110,800 inch-lb/rad	Torsional elastic stiffness, end ℓ	= 110,800 inch-lb/rad
Torsional plastic stiffness, end K	= 25,500 inch-lb/rad	Torsional plastic stiffness, end ℓ	= 25,500 inch-lb/rad
Torsional yield deflection, end K	= 0.001 rad	Torsional yield deflection, end ℓ	= 0.001 rad
Torsional rupture deflection, end K	= 1 rad	Torsional rupture deflection, end ℓ	= 1 rad
Face height, end K	= 6 inches	Face height, end ℓ	= 6 inches
Length, end K	= 58.125 inches	Length, end ℓ	= 58.125 inches
Beam (Type 1)			
Elastic modulus	= 1×10^7 psi	Ultimate stress	= 100,000 psi
Plastic modulus	= 20,000 psi	Height	= 19.55 inches
Yield stress	= 60,000 psi	Width	= 0.676 inch

TABLE 8.-PHYSICAL PROPERTIES OF ELEMENTS (Contd)

Beam (Type 2)			
Elastic modulus	= 1×10^7 psi	Ultimate stress	= 100,000 psi
Plastic modulus	= 20,000 psi	Height	= 9.790 inches
Yield stress	= 60,000 psi	Width	= 2.082 inches
Beam (Type 3)			
Elastic modulus	= 3×10^7 psi	Ultimate stress	= 150,000 psi
Plastic modulus	= 180,000 psi	Height	= 10.360 inches
Yield stress	= 100,000 psi	Width	= 0.776 inch
Linear Spring * Adjusted for Load			
Spring constant	= 3110 lb/inch	Free length	= 20.04 inches
Nonlinear Spring (Type 1)			
Compressive constant, compression < δ_c	= 3×10^6 lb/inch	Extension constant, extension < δ_t	= 75,000 lb/inch
Compressive constant compression > δ_c	= 3×10^6 lb/inch	Extension constant, extension > δ_t	= 75,000 lb/inch
δ_c	= 10 inches	δ_t	= 10 inches
Free length	= 3 inches		
Nonlinear Spring (Type 2)			
Compressive constant, compression < δ_c	= 0.0 lb/inch	Extension constant, extension < δ_t	= 0.0 lb/inch
Compressive constant, compression > δ_c	= 3×10^6 lb/inch	Extension constant, extension > δ_t	= 3×10^6 lb/inch
δ_c	= 3.75 inches	δ_t	= 2 inches
Free length	= 14.25 inches		

TABLE 8.-PHYSICAL PROPERTIES OF ELEMENTS (Concl)

Nonlinear Spring (Type 3)			
Compressive constant, compression < δ_c	= 0.0 lb/inch	Extension constant, extension < δ_t	= 0.0 lb/inch
Compressive constant, compression > δ_c	= 26,890 lb/inch	Extension constant, extension > δ_t	= 0.0 lb/inch
δ_c	= 2.79 inches	δ_t	= 1 inch
Free length	= 14.25 inches		
Nonlinear Spring (Type 4)			
Compressive constant, compression < δ_c	= 500,000 lb/inch	Extension constant, extension < δ_t	= 500,000 lb/inch
Compressive constant, compression > δ_c	= 4,500,000 lb/inch	Extension constant, extension > δ_t	= 4,500,000 lb/inch
δ_c	= 0.625 inch	δ_t	= 0.625 inch
Free length	= 33 inches		
Nonlinear Dashpot			
Damping constant compression, velocity < V_c	= 1180 lb-sec/inch	Damping constant extension, velocity < V_t	= 1180 lb-sec/inch
Damping constant compression, velocity > V_c	= 173 lb-sec/inch	Damping constant extension, velocity > V_t	= 173 lb-sec/inch
V_c	= 4.5 inch/sec	V_t	= 4.5 inch/sec
Slider Joint			
Slider length	= 10 inches	Coefficient of friction	= 0.01
Slider width	= 1 inch		
Wheel-Rail			
Spring constant deflection < δ_L	= 3,234,000 lb/inch	Wheel radius	= 14 inches
Spring constant deflection > δ_L	= 6,000,000 lb/inch	Damping constant	= 1000 lb-sec/inch
δ_L	= 0.5 inch		

4. ASSESSMENT OF IMPACT CONTROL DEVICES

4.1 Response Parameters

To investigate crush, override, and crush with subsequent override of transit cars, it is convenient to have a quantitative measure of these responses. The response parameters used in this study are defined.

Crush:

R_1 = axial crush/reference deformation

Here the axial crush will be taken as the maximum longitudinal deformation of the transit car and the reference deformation will be related to the maximum "safe" crush (the distance from the anticlimber to the collision post).

Override:

R_2 = maximum vertical force at anticlimbers/anticlimber vertical yield force

if the maximum vertical force is less than the rupture force.

R_2 = 1 + relative horizontal velocity/reference horizontal velocity

if override occurs. The relative horizontal velocity between the transit cars will be taken at the time the end sill of the overriding car hits the collision post of the other car. The reference horizontal velocity will be the velocity required to fail the collision post for impact at the height predicted in the analysis.

Crush with Subsequent Override:

$R_3 = R_1$ for $1 > R_2 < R_1 < 1$

$R_3 = R_2$ for $1 > R_1 < R_2 < 1$

$R_3 = R_1 + R_2$ for $1 \leq R_1$ or $1 \leq R_2$

These response parameters were chosen so that values less than unity indicate a relatively "safe" collision and values greater than unity indicate increasingly severe conditions. The crush response parameter, R_1, becomes equal to unity when it is likely that the passenger compartment will be penetrated and increases with increasing penetration. The override response parameter, R_2, is defined two different ways depending on whether or not override occurs. Should override not occur, the tendency toward override is defined as the ratio of the maximum vertical force at the anticlimbers to the vertical force necessary to yield the anticlimbers for the typical anticlimber as given in Table 8. If override occurs, R_2 becomes greater than unity and indicates the severity of the override in terms of the relative velocity of the transit cars and the relative velocity required for penetration of the passenger compartment through failure of the collision posts. The crush with subsequent override response parameter, R_3, merely takes on the maximum value of the crush and override parameters when these responses are both less than 1. Should either response parameter be greater than 1, the crush with subsequent override severity is measured by the sum of R_1 and R_2.

4.2 Impact Control Device Physical Parameters

The impact control device physical parameters which are likely to affect crush, override, and crush with subsequent override are listed for anticlimbers in Table 9 and transit car couplers in Table 10. Note that two sets of these coupler parameters are required to describe the two impacting cars.

TABLE 9.-ANTICLIMBER PARAMETERS

Parameter	Description
P_1	Elastic vertical force-deformation slope
P_2	Plastic vertical force-deformation slope
P_3	Vertical yield force
P_4	Vertical rupture force
P_5	Elastic horizontal force-deformation slope
P_6	Plastic horizontal force-deformation slope
P_7	Horizontal yield force
P_8	Engagement surface depth
P_9	Initial misalignment

TABLE 10.-TRANSIT CAR COUPLER PARAMETERS

Parameter		Description
Car 1	Car 2	
P_{10}	P_{11}	Drawbar strength
P_{12}	P_{13}	Rail hanger spring constant
P_{14}	P_{15}	Rail hanger rupture load
P_{16}	P_{17}	Draft gear spring constant (initial)
P_{18}	P_{19}	Draft gear spring constant (bottomed)
P_{20}	P_{21}	Draft gear travel
P_{22}	P_{23}	Draft gear hysteresis load
P_{24}	P_{25}	Shear pin failure load
P_{26}	P_{27}	Post shear pin failure draft gear travel
P_{28}	P_{29}	Coupler leveler spring constant
P_{30}	P_{31}	Coupler leveler spring preload

4.3 Assessment Procedure

The response parameters, R_1, R_2, and R_3, can be expressed in terms of a power series expansion of the 31 physical parameters for the impact control devices. For instance, R_1 can be expressed as

$$
\begin{aligned}
R_1 = {} & A_o + A_1P_2 + A_2P_2 + --- + A_{31}P_{31} \\
& + B_{1,1}P_1^2 + B_{1,2}P_1P_2 + B_{1,3}P_1P_3 + --- + B_{1,31}P_1P_{31} \\
& + B_{2,2}P_2^2 + B_{2,3}P_2P_3 + --- + B_{2,31}P_2P_{31} \\
& + ----------------- + B_{31,31}P_{31}^2 \\
& + C_{1,1,1}P_1^3 + C_{1,1,2}P_1^2P_2 + --- + C_{1,1,31}P_1^2P_{31} \\
& + C_{1,2,2}P_1P_2^2 + C_{1,2,3}P_1P_2P_3 + --- + C_{1,2,31}P_1P_2P_{31} \\
& + ---------------------- \\
& + C_{31,31,31}P_{31}^3 \\
& + ----------------------
\end{aligned}
$$

where the A, B, C, etc. are constants to be determined. These constants could theoretically be obtained by determining the response, R_1, for N independent sets of the input parameters, P_i, where N is the number of constants to be determined. For all of the constants in the infinite series this would, of course, be impossible. An approximate expression for R_1 can be obtained by truncating the above series. If terms containing triple products of the input parameters (third order terms) and higher are discarded the series becomes

$$
\begin{aligned}
R_1 = {} & A_o + A_1P_1 + A_2P_2 + \text{---} + A_{31}P_{31} \\
& + B_{1,1}P_1^2 + B_{1,2}P_1P_2 + B_{1,3}P_1P_3 + \text{---} + B_{1,31}P_1P_{31} \\
& + B_{2,2}P_2^2 + B_{2,3}P_2P_3 + B_{2,4}P_2P_4 + \text{---} + B_{2,31}P_2P_{31} \\
& + \text{---} \\
& + \\
& + B_{31,31}P_{31}^2.
\end{aligned}
$$

This expression has 528 constants, A_o through $B_{31,31}$. It would be possible to use the IITRAIN computer code to compute 528 values of the response R_1 for 528 independent sets of the input parameters P_i and then solve for the various constants. However, this would not be economical in terms of time or finances. Therefore only the first order terms were considered. This results in reducing the required computer runs to 32 to obtain the 32 values of R_1 necessary to solve for A_o through A_{31}. The resulting constants A_1 through A_{31} are the sensitivity of the response to the parameters P_1 through P_{31}. A relatively low value for the constant A_i indicates that the parameter P_i is not critical for the response. At this point, only the parameters which were found relatively critical in the first order analysis were retained for a limited second order analysis.

To allow ease of comparison of the sensitivities to the various input parameters, nondimensional forms of these parameters were used. The nondimensional forms were obtained by dividing the actual physical parameter by the typical physical parameter as given in the physical parameter input table (Table 8). Thus a value of 1.0 for the physical parameter indicates the parameter is equal to its typical value.

4.4 Assessment Results

Table 11 gives the results of the first 32 IITRAIN computer runs. All parameters were set to their typical values for Run 0. The remaining runs (Runs 1 through 31) were executed with all the parameters except for the parameter with the same identification as the run number set to their typical values and the run number parameter increased by 20 percent. (P_i = 1.0 for i ≠ Run number and P_i = 1.2 for i = Run number.) This was done to make the solution of the equations for the sensitivities simple.

Runs 6, 7, 20, 21, 24 and 25 show the most change in the crush parameter, R_1. In addition the same runs show relatively large changes in the override parameter, R_2. In addition, runs 16 and 17 show large changes in the override parameter. Therefore eight additional runs were made to study the second order effects of the parameters corresponding to these runs. All of the physical parameters were set to their typical values for these runs except for the parameters corresponding to the runs listed above, P_6, P_7, P_{16}, P_{17}, P_{20}, P_{21}, P_{24} and P_{25}. The values for these parameters are given in Table 12 for the eight additional runs. Again the parameters were selected for ease of solution for the constants in the equations for the response parameters. The results of these runs are shown in Table 13.

TABLE 11.-ASSESSMENT RESULTS - FIRST ORDER ANALYSIS

Run	Response Parameters			Percent Change		
	R_1	R_2	R_3	R_1	R_2	R_3
0	1.255	0.610	1.865	-	-	-
1	1.255	0.628	1.883	0.00	2.95	0.97
2	1.255	0.610	1.865	0.00	0.00	0.00
3	1.255	0.610	1.865	0.00	0.00	0.00
4	1.255	0.610	1.865	0.00	0.00	0.00
5	1.254	0.609	1.863	-0.08	-0.16	-0.11
6	1.219	0.654	1.873	-2.87	7.21	0.43
7	1.210	0.652	1.862	-3.59	6.89	-0.16
8	1.255	0.610	1.865	0.00	0.00	0.00
9	1.254	0.601	1.855	-0.08	-1.48	-0.54
10	1.255	0.610	1.865	0.00	0.00	0.00
11	1.255	0.610	1.865	0.00	0.00	0.00
12	1.255	0.610	1.865	0.00	0.00	0.00
13	1.255	0.610	1.865	0.00	0.00	0.00
14	1.254	0.610	1.864	-0.08	0.00	-0.05
15	1.255	0.610	1.865	0.00	0.00	0.00
16	1.254	0.628	1.882	-0.08	2.95	0.91
17	1.254	0.570	1.824	-0.08	-6.56	-2.20
18	1.256	0.619	1.875	0.08	1.48	0.54
19	1.256	0.625	1.881	0.08	2.46	0.86
20	1.236	0.645	1.881	-1.51	5.74	0.86
21	1.236	0.626	1.862	-1.51	2.62	-0.16
22	1.255	0.610	1.865	0.00	0.00	0.00
23	1.255	0.610	1.865	0.00	0.00	0.00
24	1.244	0.617	1.861	-0.88	1.15	-0.21
25	1.243	0.561	1.804	-0.96	-8.03	-3.27
26	1.255	0.606	1.861	0.00	-0.66	-0.21
27	1.255	0.603	1.858	0.00	-1.15	-0.38
28	1.255	0.610	1.865	0.00	0.00	0.00
29	1.255	0.610	1.865	0.00	0.00	0.00
30	1.255	0.610	1.865	0.00	0.00	0.00
31	1.255	0.608	1.863	0.00	-0.33	-0.11

TABLE 12.-PHYSICAL PARAMETER VARIATIONS FOR SECOND ORDER SENSITIVITY STUDY

Run	Physical Parameter Value							
	P_6	P_7	P_{16}	P_{17}	P_{20}	P_{21}	P_{24}	P_{25}
32	0.8	1.0	1.0	1.0	1.0	1.0	1.0	1.0
33	1.0	0.8	1.0	1.0	1.0	1.0	1.0	1.0
34	1.2	1.2	1.0	1.0	1.0	1.0	1.0	1.0
35	1.0	1.0	1.2	1.2	1.0	1.0	1.0	1.0
36	1.0	1.0	1.0	1.0	0.8	1.0	1.0	1.0
37	1.0	1.0	1.0	1.0	1.0	0.8	1.0	1.0
38	1.0	1.0	1.0	1.0	1.2	1.2	1.0	1.0
39	1.0	1.0	1.0	1.0	1.0	1.0	1.2	1.2

TABLE 13.-ASSESSMENT RESULTS - SECOND ORDER ANALYSIS

Run	Response Parameter			Percent Change		
	R_1	R_2	R_3	R_1	R_2	R_3
32	1.302	0.547	1.849	3.75	-10.33	-0.86
33	1.304	0.553	1.857	3.90	-9.34	-0.43
34	1.181	0.642	1.823	-5.90	5.25	-2.25
35	1.254	0.609	1.863	-0.08	-0.16	-0.11
36	1.255	0.623	1.878	0.00	2.13	0.70
37	1.255	0.623	1.878	0.00	2.13	0.70
38	1.253	0.601	1.854	-0.16	-1.48	-0.59
39	1.253	0.600	1.853	-0.16	-1.64	-0.64

Using the computer results in Table 11 and the first order terms of the power series expansion from Section 4.3, the following expressions are obtained for the crush, override, and crush with subsequent override parameters.

$$\begin{aligned} R_1 = {} & 1.980 - 0.005\ P_5 - 0.180\ P_6 - 0.225\ P_7 - 0.005\ P_9 \\ & - 0.005\ P_{14} - 0.005\ P_{16} - 0.005\ P_{17} + 0.005\ P_{18} \\ & + 0.005\ P_{19} - 0.095\ P_{20} - 0.095\ P_{21} - 0.055\ P_{24} \\ & - 0.060\ P_{25} \end{aligned} \tag{1}$$

$$R_2 = 0.150 + 0.090\ P_1 - 0.005\ P_5 + 0.220\ P_6 + 0.210\ P_7 - 0.045\ P_9 + 0.090\ P_{16} - 0.200\ P_{17} + 0.045\ P_{18} + 0.075\ P_{19} + 0.175\ P_{20} + 0.080\ P_{21} + 0.035\ P_{24} - 0.245\ P_{25} - 0.020\ P_{26} - 0.035\ P_{27} - 0.010\ P_{31} \quad (2)$$

$$R_3 = 2.130 + 0.090\ P_1 - 0.010\ P_5 + 0.040\ P_6 - 0.015\ P_7 - 0.050\ P_9 - 0.005\ P_{14} + 0.085\ P_{16} - 0.205\ P_{17} + 0.050\ P_{18} + 0.080\ P_{19} + 0.080\ P_{20} - 0.015\ P_{21} - 0.020\ P_{24} - 0.305\ P_{25} - 0.020\ P_{26} - 0.035\ P_{27} - 0.010\ P_{31} \quad (3)$$

If one neglects the physical parameters which show little effect on the responses, and considers only those parameters varied in Table 13, the power series expansion of Section 4.3 can be reduced to

$$R_i = A_o + A_6P_6 + A_7P_7 + A_{16}P_{16} + A_{17}P_{17} + A_{20}P_{20} + A_{21}P_{21} + A_{24}P_{24} + A_{25}P_{25} + B_{6,6}P_6^2 + B_{6,7}P_6P_7 + B_{7,7}P_7^2 + B_{16,17}P_{16}P_{17} + B_{20,20}P_{20}^2 + B_{20,21}P_{20}P_{21} + B_{21,21}P_{21}^2 + B_{24,25}P_{24}P_{25}$$

where a limited number of second order terms have been retained. Using the results of Tables 11 and 13 to solve for the constants in this equation the following expressions are obtained for the response parameters.

$$R_1 = 3.255 - 0.658\ P_6 - 0.510\ P_7 - 0.030\ P_{16} - 0.030\ P_{17} - 0.473\ P_{20} - 0.473\ P_{21} - 0.580\ P_{24} - 0.585\ P_{25} + 0.138\ P_6^2 + 0.175\ P_6P_7 + 0.050\ P_7^2 + 0.025\ P_{16}P_{17} - 0.238\ P_{20}^2 + 0.900\ P_{20}P_{21} - 0.238\ P_{21}^2 + 0.525\ P_{24}P_{25}$$

$$R_2 = -0.635 + 2.093\ P_6 + 1.973\ P_7 - 0.435\ P_{16} - 0.725\ P_{17} + 0.355\ P_{20} + 0.783\ P_{21} - 0.765\ P_{24} - 1.045\ P_{25} - 0.238\ P_6^2 - 1.350\ P_6P_7 - 0.188\ P_7^2 + 0.525\ P_{16}P_{17} + 0.600\ P_{20}^2 - 1.500\ P_{20}P_{21} + 0.363\ P_{21}^2 + 0.800\ P_{24}P_{25}$$

$$R_3 = 2.620 + 1.435\ P_6 + 1.463\ P_7 - 0.465\ P_{16} - 0.755\ P_{17} - 0.118\ P_{20} + 0.310\ P_{21} - 1.345\ P_{24} - 1.630\ P_{25} - 0.100\ P_6^2 - 1.175\ P_6P_7 - 0.138\ P_7^2 + 0.550\ P_{16}P_{17} + 0.363\ P_{20}^2 - 0.600\ P_{20}P_{21} + 0.125\ P_{21}^2 + 1.325\ P_{24}P_{25}$$

These equations, by themselves, are hard to interpret. However, their derivatives show the effects of the various physical parameters on the responses. Table 14 contains the derivatives evaluated at the "design" point ($P_i = 1.0$). The first derivatives in this table are a more accurate estimate of the sensitivity of the responses to the various parameters and are approximately equal to the coefficients in Equations (1), (2) and (3). The second derivatives indicate the nonlinearity of the responses or how rapidly the sensitivities change as the physical parameters diverge from the design point.

TABLE 14.-SECOND ORDER ANALYSIS SENSITIVITIES

Derivative	R_1	R_2	R_3
$\partial(\ )/P_6$	-0.207	0.267	0.060
$\partial(\ )/P_7$	-0.235	0.247	0.012
$\partial(\ )/P_{16}$	-0.005	0.090	0.085
$\partial(\ )/P_{17}$	-0.005	-0.200	-0.205
$\partial(\ )/P_{20}$	-0.049	0.055	0.008
$\partial(\ )/P_{21}$	-0.049	0.009	-0.040
$\partial(\ )/P_{24}$	-0.055	0.035	-0.020
$\partial(\ )/P_{25}$	-0.060	0.245	-0.305
$\partial^2(\ )/P_6^2$	0.276	-0.476	-0.200
$\partial^2(\ )/P_7^2$	0.100	-0.376	-0.276
$\partial^2(\ )P_{20}^2$	-0.476	1.200	0.726
$\partial^2(\ )/P_{21}^2$	-0.476	0.726	0.250
$\partial^2(\ )/P_6P_7$	0.175	-1.350	-1.175
$\partial^2(\ )/P_{16}P_{17}$	0.025	0.525	0.550
$\partial^2(\ )/P_{20}P_{21}$	0.900	-1.500	-0.600
$\partial^2(\ )/P_{24}P_{25}$	0.525	0.800	1.325

5. IMPACT CONTROL DEVICE DESIGN RECOMMENDATIONS

The results given in Section 4.4 indicate those physical parameters of the anticlimbers and draft gears which are critical in determining whether transit cars will crush, override, or crush with subsequent override. Tables 15, 16, and 17 list these parameters for each of the response categories in order of their importance.

The physical parameters which have the most significant effect on crush of transit cars are the anticlimber horizontal yield force and plastic horizontal force-deformation slope. Increases in either of these parameters will reduce the total crush of the cars. The draft gear travel and shear pin failure load of either the struck or striking car also has an appreciable effect on the crush. Again an increase in these parameters will reduce the total crush. Increases in all other physical parameters will reduce the total crush. However these physical parameters have an essentially negligible effect.

TABLE 15.-CRUSH SENSITIVITY

Parameter	Physical Description	Sensitivity
P_7	Anticlimber horizontal yield force	-0.225
P_6	Anticlimber plastic horizontal force-deformation slope	-0.180
P_{20}	Striking car draft gear travel	-0.095
P_{21}	Struck car draft gear travel	-0.095
P_{25}	Struck car shear pin failure load	-0.060
P_{24}	Striking car shear pin failure load	-0.055
P_5	Anticlimber elastic horizontal force-deformation slope	-0.005
P_9	Initial anticlimber misalignment	-0.005
P_{14}	Striking car rail hanger rupture load	-0.005
P_{16}	Striking car draft gear spring constant (initial)	-0.005
P_{17}	Struck car draft gear spring constant (initial)	-0.005
P_{18}	Striking car draft gear spring constant (bottomed)	-0.005
P_{19}	Struck car draft gear spring constant (bottomed)	-0.005

TABLE 16.-OVERRIDE SENSITIVITY

Parameter	Physical Description	Sensitivity
P_{25}	Struck car shear pin failure load	-0.245
P_{6}	Anticlimber plastic horizontal force-deformation slope	0.220
P_{7}	Anticlimber horizontal yield force	0.210
P_{17}	Struck car draft gear spring constant (initial)	-0.200
P_{20}	Striking car draft gear travel	0.175
P_{1}	Anticlimber elastic vertical force-deformation slope	0.090
P_{16}	Striking car draft gear spring constant (initial)	0.090
P_{21}	Struck car draft gear travel	0.080
P_{19}	Struck car draft gear spring constant (bottomed)	0.075
P_{9}	Initial anticlimber misalignment	-0.045
P_{18}	Striking car draft gear spring constant (bottomed)	0.045
P_{24}	Striking car shear pin failure load	0.035
P_{27}	Struck car postshear pin failure draft gear travel	-0.035
P_{26}	Striking car postshear pin failure draft gear travel	-0.020
P_{31}	Struck car coupler leveler spring preload	-0.010
P_{5}	Anticlimber elastic horizontal force-deformation slope	-0.005

TABLE 17.-CRUSH WITH SUBSEQUENT OVERRIDE SENSITIVITY

Parameter	Physical Description	Sensitivity
P_{25}	Struck car shear pin failure load	-0.305
P_{17}	Struck car draft gear spring constant (initial)	-0.205
P_{1}	Anticlimber elastic vertical force-deformation slope	0.090
P_{16}	Striking car draft gear spring constant (initial)	0.085
P_{19}	Struck car draft gear spring constant (bottomed)	0.080
P_{20}	Striking car draft gear travel	0.080
P_{9}	Initial anticlimber misalignment	-0.050
P_{18}	Striking car draft gear spring constant (bottomed)	0.050
P_{6}	Anticlimber plastic horizontal force-deformation slope	0.040
P_{27}	Struck car postshear pin failure draft gear travel	0.035
P_{24}	Striking car shear pin failure load	-0.020
P_{26}	Striking car postshear pin failure draft gear travel	-0.020
P_{7}	Anticlimber horizontal yield force	-0.015
P_{21}	Struck car draft gear travel	-0.015
P_{5}	Anticlimber elastic horizontal force-deformation slope	-0.010
P_{31}	Struck car coupler leveler spring preload	-0.010
P_{14}	Striking car rail hanger rupture load	-0.005

Override is most sensitive to the struck car shear pin failure load. An increase in this load will significantly reduce the tendency to override. The striking car shear pin failure load is much less significant but has an opposite effect on override. The anticlimber horizontal plastic force-deformation slope and yield force also have a large effect on the tendency to override, an increase in these parameters causing an increase in the tendency to override. The initial draft gear spring constant for the struck car (and less significantly for the striking car) also is of great importance in override. An increase in the spring constant for the struck car decreases the tendency to override but the opposite effect occurs for the striking car. An increase in the draft gear travel for either the striking or struck car significantly increases the tendency to override. The anticlimber elastic vertical force-deformation slope has a moderate effect on tendency to override, an increase in this parameter increasing the likelihood of override. The draft gear spring constant after bottoming of the draft gear spring also has a moderate effect on override. An increase in this parameter for either the struck or striking car will increase the tendency to override. The remaining parameters have only a small effect on override.

Crush with subsequent override is most affected by the struck car shear pin failure load, an increase in this load greatly reducing this response. The striking car shear pin failure load is essentially negligible. Of second most importance for crush with subsequent override is the struck car draft gear spring constant. An increase in this spring constant decreases this response. An opposite but greatly reduced effect occurs for the striking car draft gear spring constant. An increase in the anticlimber elastic vertical force-deformation slope increases the crush with subsequent override response. The draft gear spring constant after bottoming has a moderate effect on crush with subsequent override. An increase in this parameter for either the struck or striking car increases this response. An increase in the striking car draft gear travel moderately increases crush with subsequent override while an increase in this same parameter for the struck car slightly reduced this response. The remaining parameters have little effect on crush with subsequent override.

Careful analysis of the results given above suggests several recommendations for improving the design of transit car impact control devices. However, since these results were obtained for the special case of impact on a straight level track any such recommendations should not be put into effect until their consequences in other situations are studied.

Analysis of the computer results indicates that an increase in the draft gear shear pin failure load is the only design recommendation which can be given that will have a beneficial effect on all three responses; crush, override, and crush with subsequent override. An increase in the shear pin failure load allows more of the impact energy to be dissipated in the draft gear thus reducing the total energy absorbed by the anticlimber and end sill structure. For this recommendation to be effective the transit car coupling mechanisms must be centered to ensure contact in case of an impact.

Decreasing the draft gear spring constant after spring bottoming will decrease the tendency to override and crush with subsequent override while having negligible effect on the crush. Before bottoming, this spring constant should be increased to have the same effect, if all transit cars likely to impact have similar draft gear. Should cars with dissimilar draft gear impact, this recommendation would cause a greater likelihood of override and crush with subsequent override in those cases where the struck car had the increased spring constant. The draft gear spring constant has little effect on crush in any case. The anticlimber elastic vertical force-deformation slope should be reduced to decrease override and crush with subsequent override. This allows the anticlimbers to maintain contact even under large relative vertical motions of the impacting cars. This change will not effect crush.

Any other changes of significant physical parameters will increase one response while decreasing another. Any such changes should be studied in detail before inclusion in a new design.

Increasing the anticlimber horizontal yield force and plastic horizontal force deformation slope will decrease the crush but increase the tendency to override. Similarly increasing the draft gear travel and shear pin failure load will decrease the crush but increase the tendency to override.

It should be noted that increasing the vertical yield force for the anticlimbers showed insignificant effect on either crush, override, or crush with subsequent override. By definition the override response parameters, for the case of no actual override, was set equal to the maximum vertical force divided by the vertical yield force for the typical set of physical parameters. Had the vertical yield force for the altered set of parameters been used, then a large sensitivity to P_3 (the vertical yield force) would have been noted for override and crush with subsequent override. This sensitivity would have been -0.510 for both of these response parameters. The sensitivity of crush (R_1) to P_3 would still have been zero.

6. EXPERIMENTAL TEST PLAN

The object of this plan is to describe a test program for determining the effectiveness of improved impact control devices for transit cars. For the purpose of developing this test plan three possible impact control device variations are considered. These variations were selected to correspond with the critical parameters determined in Section 5 as well as the discussion of the results given in that section. Variation 1 will be a draft gear with a longer stroke designed to impact with other such draft gears in case of a collision, and is intended to reduce car body acceleration for low speed impacts. This corresponds to the third most important parameter for crush, fifth for override and sixth for crush with subsequent override. Variation 2 will be an anticlimber with a greater vertical force capacity and is designed to reduce the possibility of override. This is in accordance with the discussion of the design recommendations given in Section 5. Variation 3 will be an end sill structure with controlled longitudinal crush characteristics and is intended to reduce car body accelerations for high speed impacts. This corresponds to the first and second most important parameter for crush, second for override and ninth for crush with subsequent override. A separate test plan is given for each of these variations.

6.1 Test Plan 1 - Effectiveness of Long Stroke Draft Gear

Purpose-Determine the car body longitudinal accelerations for 5 to 10 mph impacts for a draft gear with an extra long stroke.

Test Specimen-This test will be conducted using two representative transit cars. Existing cars will be procured on a loan (or rent) basis. The proposed long stroke draft gear will be designed to replace the existing draft gear on these cars. The special draft gears will be procured and installed at the impacting ends of the two transit cars. The cars will be loaded to approximately two-thirds capacity with sand bags.

Instrumentation-Accelerometers will be mounted rigidly to each car frame near its c.g. Three accelerometers will be used; one for longitudinal, one for lateral and one for vertical motions. The longitudinal accelerometer will have a full-scale capacity of 20 g and will be capable of measuring acceleration in the frequency range from DC to 200 Hz. The other two accelerometers will have full-scale capacities of 5 g and will be capable of measuring accelerations in the frequency range from DC to 200 Hz with less than 2 percent cross axis sensitivity as mounted.

Strain gages will be mounted on the coupler mechanisms in suitable locations and will be connected in a suitable bridge to measure axial force in the draft gear. The arrangement will be such that any bending, twist or shear will be cancelled giving only axial force. The full-scale force for the gage setup will be 250,000 lb.

Since only average velocities before and after impact are required, car velocities will be measured by using a counter to time rotations of the car wheels.

A tape unit with sufficient capacity to store all data will be mounted in each car. The tape units will be shock mounted to ensure no damage to the units during the impact.

Two high-speed movie cameras with sufficient floodlights will be used to record the test pictorially.

Test Procedure-One of the cars will be stationary on straight level track without the brakes applied. The second car will be brought to speed either with the use of a ramp or an auxiliary pusher car. Two tests will be run at each of three speeds; approximately 5, 7.5, and 10 mph. Full data will be taken for all six runs. Accelerometer and strain gage data, and car velocity measurements will be stored continuously on magnetic tape. High speed movies will be taken starting just prior to impact.

Data Analysis-Oscillographic playback will be used to generate photographic records of all data versus time. Velocity prior to impact, velocities after impact, maximum accelerations in longitudinal, lateral and vertical directions, and axial forces in the draft gear will be determined for each test from these data.

6.2 Test Plan 2 - Effectiveness of Increased Vertical Force Capacity Anticlimber

Purpose-Determine the increase in crash speed without override due to a modified anticlimber with a greater vertical force capacity.

Test Specimen-This test will be conducted using a representative transit car and a fixed abutment. An existing car will be procured on a loan (or rent) basis. The anticlimber on this car will be temporarily strengthened to ensure no permanent damage to the car. The coupler and draft gear will be removed. An existing abutment will be modified or a new abutment will be constructed with suitable framing to attach the test anticlimber structures. Two test anticlimber structures will be procured. One will be a close duplicate of a typical anticlimber structure and the second will have the anticlimber modified to have the greater vertical force capacity.

Instrumentation-Accelerometers will be mounted rigidly to the car frame near its c.g. Three accelerometers will be used; one for longitudinal, one for lateral and one for vertical motions. The longitudinal accelerometer will have a full-scale capacity of 20 g and will be capable of measuring accelerations in the frequency range from DC to 200 Hz. The other two accelerometers will have full-scale capacities of 5 g and will be capable of measuring accelerations in the frequency range from DC to 200 Hz with less than 2 percent cross axis sensitivity as mounted.

Strain gages will be mounted on the abutment frame in suitable locations and will be connected in suitable bridges to measure longitudinal and vertical forces. The arrangements will be such to cancel out all forces and moments other than the force being measured. The full-scale force will be 1,000,000 lb for the horizontal gage setup and 100,000 lb for the vertical gage.

Since only average velocity before impact is required, car velocity will be measured by using a counter to time rotations of a car wheel.

A tape unit with sufficient capacity to store all data will be mounted in the car. The tape unit will be shock mounted to ensure no damage to the unit during impact.

At least two high-speed movie cameras with sufficient floodlights will be used to record the test pictorially.

Test Procedure-Prior to the actual test, the IITRAIN computer code (or a similar analytical procedure) will be used to obtain a set of obtainable initial conditions which will make override likely at approximately 15 mph for the standard anticlimber. The speed at which override will occur for the modified anticlimber will also be obtained with this procedure.

With the standard anticlimber in place the car will be brought to the speed at which the analytic procedure predicted override. The proper initial conditions will be given to the car prior to impact. Full data will be taken and stored on magnetic tape. High speed movies will be taken starting just prior to impact. After the test any damage to the test setup will be repaired. Should override have occurred, the test will be rerun with the impact velocity reduced by 1 mph. This process will be continued until override does not occur. If, however, override does not occur in the initial test, the impact velocity will be increased by 1 mph and the testing will be continued until override occurs.

After the limiting impact velocity for the standard anticlimber has been determined, the same procedure will be used for the modified anticlimber.

Data Analysis-Oscillographic playback will be used to generate photographic records of all data versus time. Velocity prior to impact; maximum accelerations in the longitudinal, lateral and vertical directions; and maximum horizontal and vertical anticlimber forces will be recorded for each test run.

6.3 Test Plan 3 - Effectiveness of Modified End Sill Structure with Controlled Longitudinal Force Characteristics

Purpose-Determine the decrease in car body accelerations due to a 20 mph impact for a modified end sill structure with controlled longitudinal force characteristics.

Test Specimen-This test will be conducted using a retired or scrapped car body and a fixed abutment. The car will be sufficiently restored for the test purpose. The end sill structure at one end of the car will be retained while the structure at the other end will be modified to give it the desired controlled longitudinal crush force characteristics. Couplers and draft gear will be removed. An existing abutment will be modified or a new abutment will be constructed with suitable framing to attach the end sill structures. Two end sill structures will be procured. These will closely match the end sill structures on the test car.

Instrumentation-Accelerometers will be mounted rigidly to the car frame near its c.g. Three accelerometers will be used; one for longitudinal, one for lateral and one for vertical motions. The longitudinal accelerometer will have a full-scale capacity of 20 g and will be capable of measuring accelerations in the frequency range from DC to 200 Hz. The other two accelerometers will have full-scale capacities of 5 g and will be capable of measuring accelerations in the frequency range from DC to 200 Hz with less than 2 percent cross axis sensitivity as mounted.

Strain gages will be mounted on the abutment frame in suitable locations and will be connected in suitable bridges to measure longitudinal and vertical forces. The arrangements will be such to cancel out all forces and moments other than the force being measured. The full-scale force will be 1,000,000 lb for the horizontal gage setup and 100,000 lb for the vertical gage setup.

Since only average velocity before impact is required, car velocity will be measured by using a computer to time rotation of a car wheel.

A tape unit with sufficient capacity to store all data will be mounted in the car. The tape unit will be shock mounted to ensure no damage during the impact.

At least two high-speed movie cameras with sufficient floodlights will be used to record the test pictorially.

Test Procedure-The test will be conducted on straight level track. The standard end of car structure will be mounted on the fixed abutment and the test car will be oriented so that its unmodified end will impact the abutment. The car will be brought to 20 mph using either a ramp or an auxiliary pusher car. Full data will be taken and stored on magnetic tape. High speed movies will be taken starting just prior to impact.

After the test with the standard end structure is completed, the test car will be oriented so that its modified end will impact the abutment. Some repair of the damaged end may be necessary before the second test is run. The modified end sill structure will be mounted to the abutment and the second test will proceed in the same manner as the first.

Data Analysis-Oscillographic playback will be used to generate photographic records of all data versus time. Velocity prior to impact; maximum accelerations in the longitudinal, lateral and vertical directions; and maximum horizontal and vertical anticlimber forces will be recorded for each test run.

7. REFERENCES

1. Widmayer, E., Tanner, A.E., Klump, R., "Crashworthiness Analysis of the UMTA State-of-the-Art Cars," Boeing Vertol Company Final Report D174-10034-1 (DOT-TSC-791-3), June 1975.

2. Cassidy, R. J., Romeo, D. J., "Assessment of Crashworthiness of Existing Urban Rail Vehicles," Calspan Corporation Final Report ZL-5368-K-1, June 1975.

3. Raidt, J. B. (Principal Investigator), "A Preliminary Study of Vertical Motions During Impact," Pullman-Standard Research Project 38-1853, Final Report, August 1972.

4. Hohenemser, K. H., Diboll, W. B., Yin, S. K., Szabo, B. A., "Computer Simulation of Tank Car Head Puncture Mechanisms Classification Yard Accidents," School of Engr and Appl Sci, Washington University Preliminary Report 64274 (FRA-ORD&D-75-23), February 1975.

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