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DYNAMIC ANALYSIS TO ESTABLISH NORMAL SHOCK AND VIBRATION ENVIRONMENTS EXPERIENCED BY RADIOACTIVE MATERIAL SHIPPING PACKAGES

Quarterly Progress Report
October 1 - December 31, 1977

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Hanford Engineering Development Laboratory

Prepared for
U. S. Nuclear Regulatory Commission

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

INTRODUCTION

The objective of this study is to determine the extent to which the shocks and vibrations experienced by radioactive material shipping packages during normal transport conditions are influenced by, or are sensitive to, various structural parameters of the transport system (i.e., package, package supports, and vehicle). The purpose of this effort is to identify those parameters that significantly affect the normal shock and vibration environments so as to provide the basis for determining the forces transmitted to radioactive material packages. Determination of these forces will provide the input data necessary for a broad range of package-tiedown structural assessments.

This is the first quarterly report on this work.

PROGRESS TO DATE

At a meeting at Nuclear Regulatory Commision headquarters in Silver Spring, MD on October 5, 1977, a proposed work plan was presented. This proposed plan was modified at this meeting to the plan shown in Figure 1. The tasks shown in Figure 1 are based on the following scope of work:

- There are several parameters that may influence the magnitude of shocks and vibrations experienced by radioactive material shipping packages. Sensitivity studies, using appropriate dynamic analysis computer codes and models, will be performed to ascertain the relative significance of these parameters. If feasible, results will be presented as sets of acceleration response spectra that illustrate the relative influence of each parameter.

- The modeling effort should be capable of evaluating longitudinal and vertical response to longitudinal inputs. Comparisons of predicted response with response observed in forthcoming rail car coupling tests should also be anticipated.

- Initial efforts should concentrate on heavy shielded casks. Specifically, for vehicle and tiedown system parameters found to significantly influence package response, a range of these values typical of existing packages should be investigated.

Each of the tasks will now be discussed and progress during the reporting period described.

1. Develop Dynamic Model

A two-dimensional, multi-degree-of-freedom model of a spent fuel shipping cask-rail car system has been developed. A sketch of the cask-rail car system modeled is shown in Figure 2, and the spring-mass model of this system is shown in Figure 3.

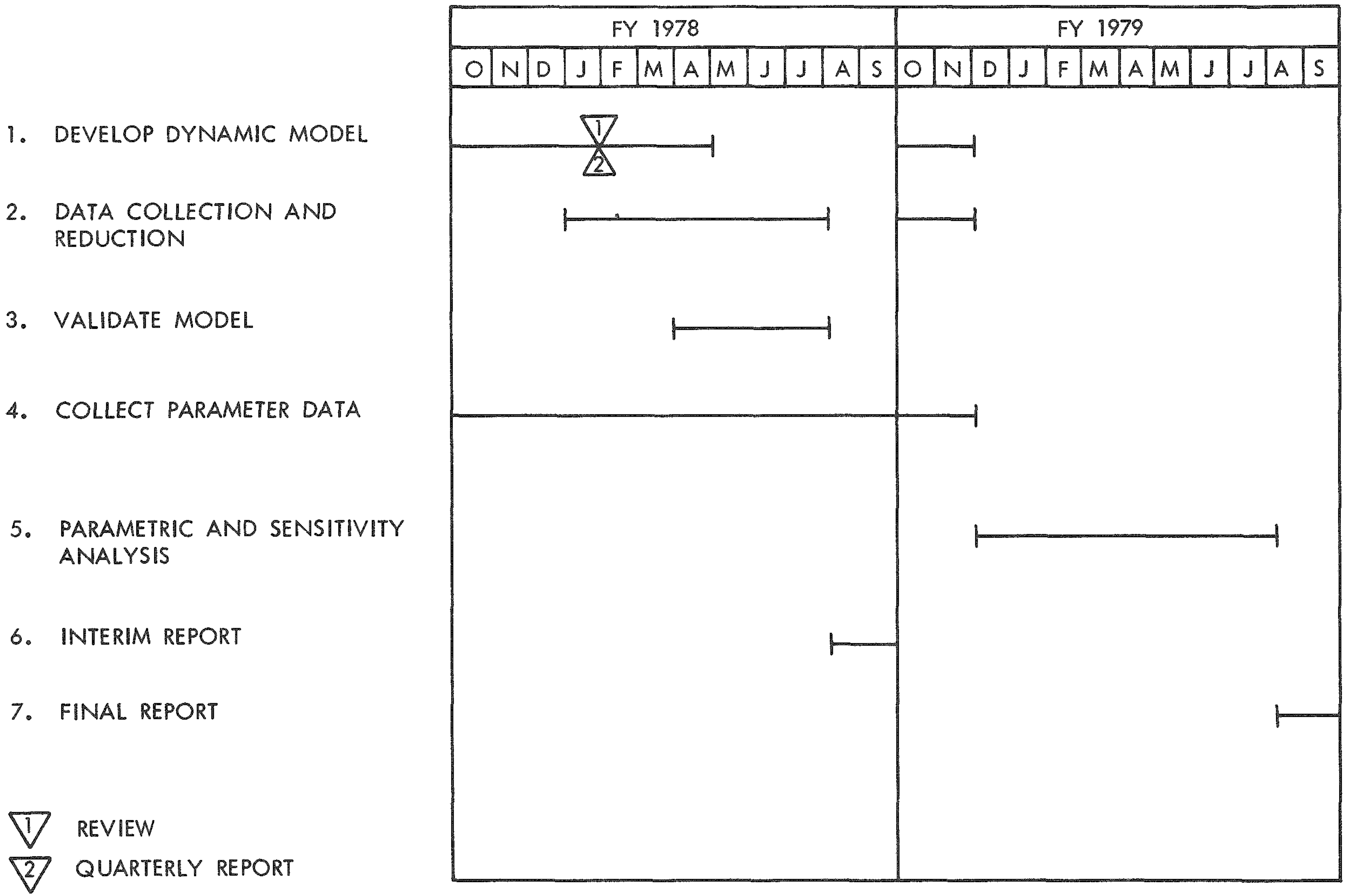

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FIGURE 1. Work Plan -- Dynamic Analysis of Radioactive Material Shipping Packages.

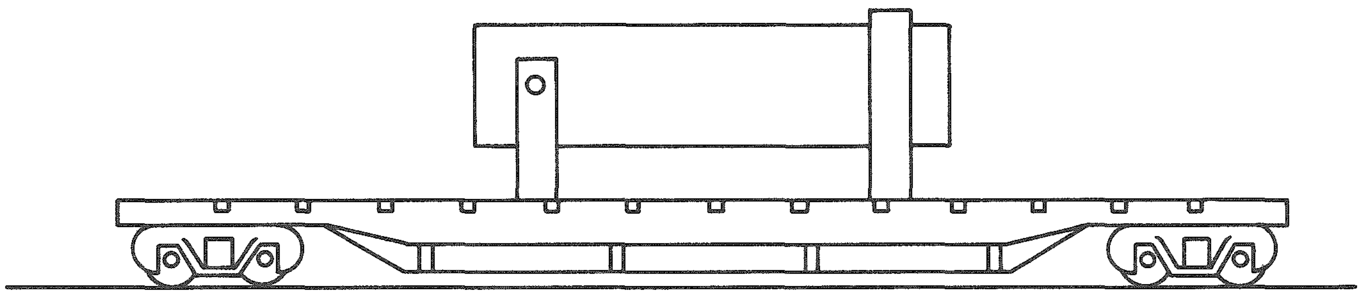

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FIGURE 2. Spent Fuel Shipping Cask-Rail Car System Modeled.

Each of the masses in the cask-rail car model of Figure 3 is free to translate horizontally (front to back) and vertically, and to rotate about its axis normal to the plane of the illustration. The system is excited by impact with one or more cars (mass M_F) at the front coupler. One possible orientation of the system after impact is shown in Figure 4, and a comparison with the initial state is illustrated in Figure 5. Figure 5 is obtained by superimposing Figures 3 and 4. (See section on Nomenclature of Terms for definition of terms used in this report.)

The model consists of nine equations of motion, one derived for each degree of freedom (generalized coordinate), and supplementary auxiliary equations. There are two general approaches that could have been used to derive the equations of motion for this dynamic system. The first is known as the <u>force-acceleration method</u> and the second is known as the <u>energy method</u>. The first method is also sometimes referred to as the <u>method of dynamic equilibrium</u>, while the second method may be referred to as the <u>Lagrange-equation method</u>. The force-acceleration method consists of analyzing the forces and the torques applied to the system and relating them to the accelerations. In the energy method one sets up the energy expressions for the system and applies Lagrange's equation to get the equations of motion. The energy or Lagrange-equation method was used for this study.

The equations of motion were derived from an energy balance (expressed in generalized coordinates) on the system. This energy balance is sometimes known as the law of virtual work, which states that the work done on the system by the external forces (virtual work) during a virtual distortion (a small change in one of the generalized coordinates) must equal the change in internal strain energy. The work done by external forces includes the work done by external loads, by inertia forces, and by damping or dissipation forces. It can be shown that the changes in the energy terms with respect to a change in a generalized coordinate may be expressed as partial derivatives, and that the energy balance may be expressed as

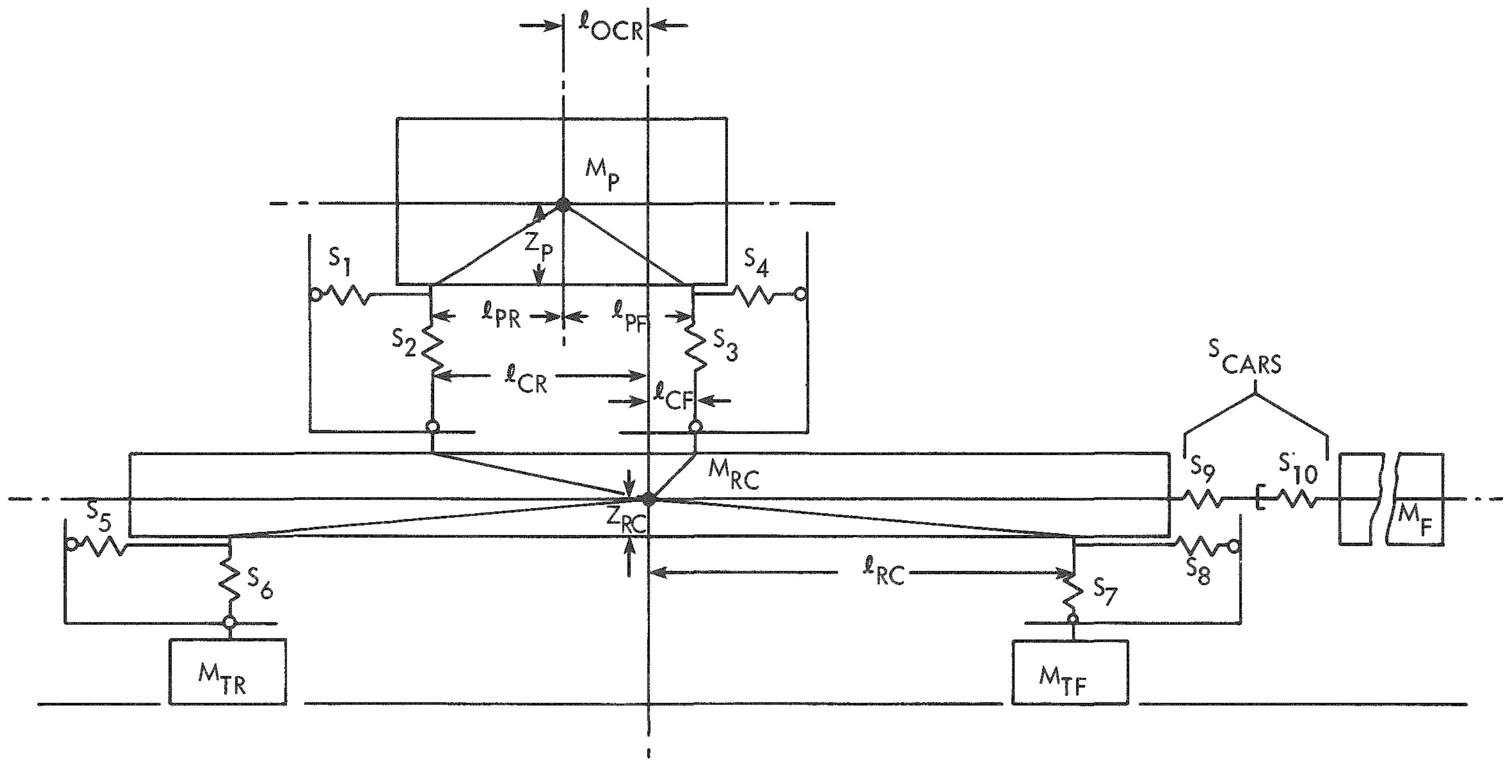

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FIGURE 3. Spring-Mass Model of Cask-Rail Car System.

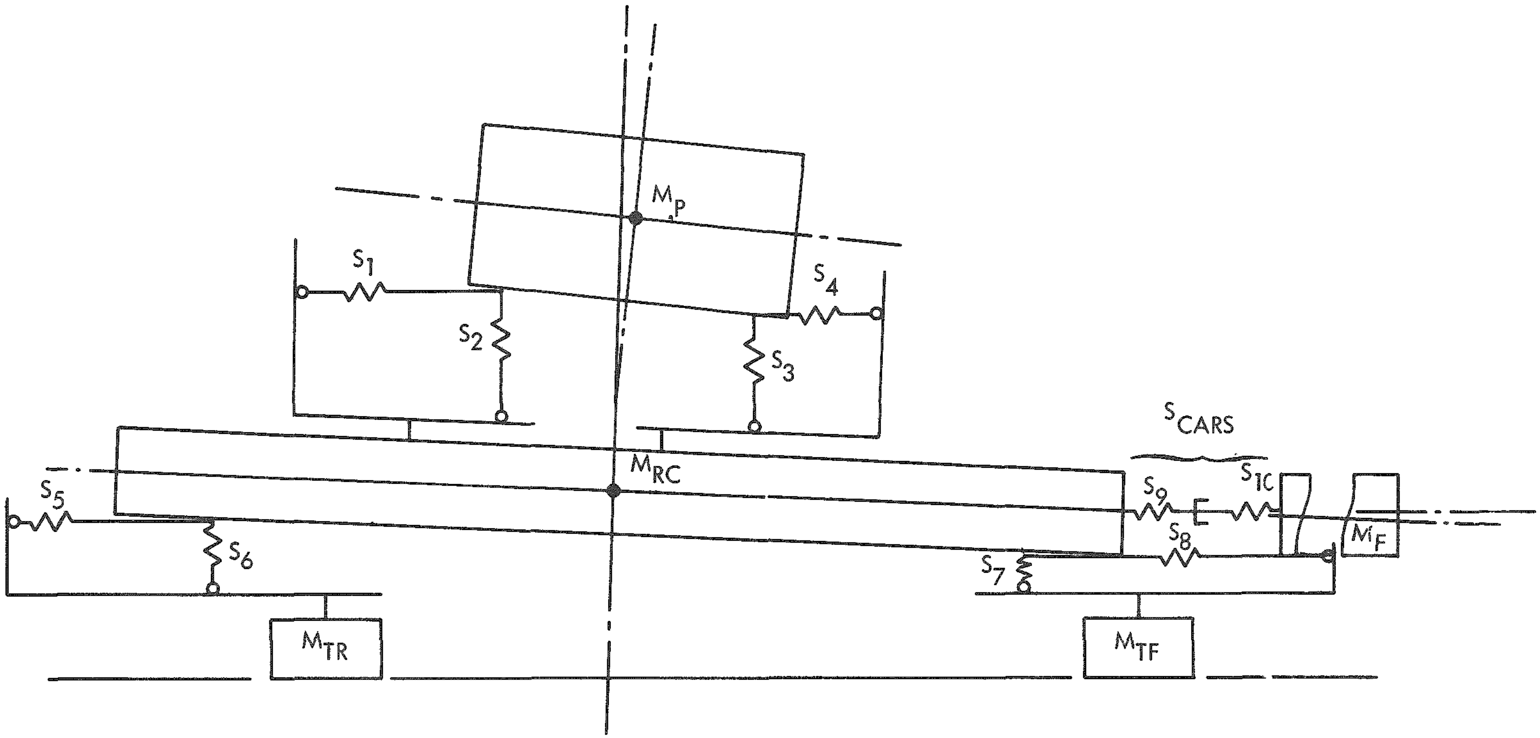

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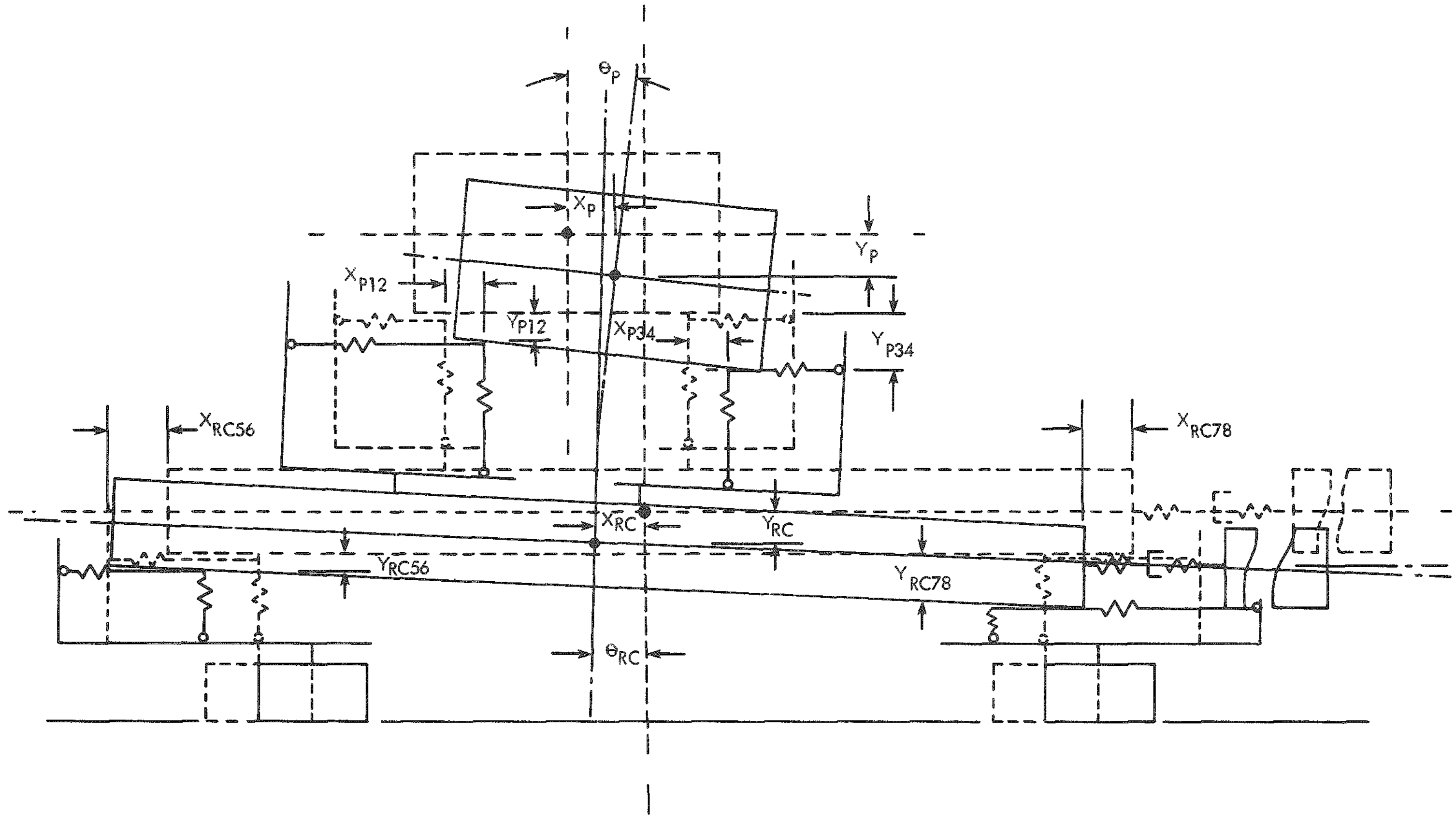
FIGURE 4. One Possible Orientation of Cask-Rail Car System After Impact.



HEDL 7803-106 5

FIGURE 5. Comparison of Orientation of Cask-Rail Car System After Impact with Initial State.

$$\frac{d}{dt}\left(\frac{\partial K}{\partial \dot{q}_i}\right) - \frac{\partial K}{\partial q_i} + \frac{\partial U}{\partial q_i} - \frac{\partial W_c}{\partial q_i} = \frac{\partial W_e}{\partial q_i} ,$$

where

t = time,

q_i = a generalized coordinate,

$\dot{q}_i$ = time rate of change of q_i,

K = kinetic energy,

U = strain energy,

W_c = work done by damping forces,

W_e = work done by external loads.

This equation is one form of Lagrange's equation. When appropriate expressions are written for K, U, W_c and W_e, all in terms of the generalized coordinates q_1, q_2,...., q_n, differentiated as indicated, and substituted into the above expression, equations of motion are obtained. There will be one equation of motion for each of the n coordinates or degrees of freedom. In all cases considered, $\frac{\partial K}{\partial q_i}$ is zero, since kinetic energy is a function of velocity rather than displacement.

The energy method was used in this study because it is a convenient and efficient process for deriving the equations of motion of the cask-rail car system. Specifically, several reasons for its selection are:

(1) It has the advantage that, for a multi-degree-of-freedom system, the equations that describe the motion of the system are simplified and reduced in number because all the internal forces that do no work will not appear in the equations.

(2) To express the results of the study as acceleration response spectra, it is necessary to relate maximum system response to system frequency. This may be accomplished by the modal method of analysis, which is considered to be an energy method because the modal equations are derived using the method outlined above. In the modal method, responses in the normal modes are determined separately, and then superimposed to provide the total response. It can be shown that, by the use of this approach, each normal mode may be treated as <u>an independent one-degree system</u>.

(3) Common practice associates matrix formulation (stiffness matrices, etc.) with the alternate method. This is not always necessary; nevertheless, it is common to set up a problem in matrix notation when using the force-acceleration method. This is not the case with the energy method, although each method produces a system of differential equations of motion that can be expressed in this form. The formulation of the equations of motion using the energy method requires more mathematical manipulation, which might be considered by some to be a disadvantage; however, in this study, this was felt to be a small price to pay to maintain a close feel for the system attributes and to be able to subdivide the equations of motion into their various energy components.

(4) The system simulation model is set up in terms of the equations of motion, which are subdivided into the various energy terms. This facilitates modification of the model at any time with a minimum of effort. This provides extreme flexibility in model construction.

Both the energy method and the force-acceleration method are only alternate methods of <u>formulating</u> the equations of motion of the cask-rail car system. They are <u>not</u> methods for <u>solving</u> the system of differential equations obtained. Because of the complexity of the system of equations and the fact that the equations are non-linear, a numerical method of integration was used in this study.

The entire model definition was written in the Advanced Continuous Simulation Language (ACSL).[1] ACSL was developed for the purpose of modeling systems described by time-dependent, non-linear differential equations and/or transfer functions. Program preparation can either be from block diagram interconnection, conventional FORTRAN statements, or a mixture of both. The ACSL program is intended to provide a simple method of representing complex mathematical models on a digital computer. Working from a system of equations describing the problem or from a block diagram, the user writes ACSL statements to describe the system under investigation. Statements describing the model need not be ordered, since the ACSL processor will sort the equations so that no values are used before they have been calculated. This operation of the language is in contrast to the usual digital programming languages like FORTRAN, where program execution depends critically on statement order.

All integration in an ACSL program is handled by a centralized integration routine. The user has a choice of four numerical integration algorithms:

(1) The Adam's-Moulton variable-step, variable-order,
(2) The Gears Stiff variable-step, variable-order,
(3) The Runge-Kutta second-order, and
(4) The Runge-Kutta fourth-order.

The Runge-Kutta fourth-order algorithm was used in the model developed for this study.

The two-dimensional, multi-degree-of-freedom model of the cask-rail car system developed during this reporting period represents <u>completion of the first phase</u> of the model development. Therefore, the model must be considered as a preliminary model. Consequently, the model has the following limitations, which will be removed during the next reporting periods:

(1) The model does not simulate bending of the rail car. This requires expansion of the system of equations to accommodate a few more degrees of freedom.

(2) The model does not simulate structural damping. The dissipation or viscous damping terms are presently included in the system of equations, but all damping coefficients are set equal to zero until it is possible to determine the critical damping of the system. However, sliding or Coulomb friction of locked wheels sliding on the tracks, programmed to always oppose the velocity of the sliding trucks, is included.

(3) No stops, slack or damping are associated with the couplers. This presents no problem since it only requires specification of coupler characteristics in the form of load-deflection curves, damping characteristics, etc., which will be programmed into the model as appropriate functions (tables).

(4) The present model provides for one or more impacting cars at the front of the cask-car only, but none at the rear. This is easily remedied by adding another degree of freedom by writing additional equations similar to those for the front.

Six coupling situations were simulated to test the model and to demonstrate its capabilities. These six situations represent two major cases, each divided into three subcases. The first major case assumes that a cask-rail car, moving at a velocity of 10 miles per hour, impacts and couples with a string of five stationary loaded flat cars with their brakes locked. The second major case assumes that a stationary cask-rail car, with its brakes locked, is coupled to a switch engine moving at a velocity of 10 miles per hour. The six cases considered are defined in Tables 1 and 2. Results obtained from the simulations are summarized in Table 3.

TABLE 1

Definition of Six Coupling Cases Simulated: Parameters

Held Constant for All Cases

1.	Weight of loaded cask or package (W_p), lb_f	140,000.(2)*
2.	Weight of one rail car truck ($W_{TR} = W_{TF}$), lb_f	7,150.(3)
3.	Weight of rail car bed (W_{RC}), lb_f	52,700.(3)
4.	Length of rail car from centerline to front or rear support, inches	259.5(3)
5.	Spring constants (stiffnesses), lb_f/inch	
	k_{SCARS}** ($k_{SRC} = k_{SF} = 2.09 \times 10^7$)(4)	1.045×10^7
	k_{s1}	1.0×10^7(4)
	k_{s2} (assumed $k_{s2} = k_{s1}$)	1.0×10^7
	k_{s3} (assumed $k_{s3} = k_{s4}$)	1.0×10^7
	k_{s4}	1.0×10^7(4)
	k_{s5} (assumed)	1.0×10^6
	k_{s6} (assumed)	1.0×10^6
	k_{s7} (assumed)	1.0×10^6
	k_{s8} (assumed)	1.0×10^6

* The numbers in parentheses are references.

** See Nomenclature of Terms for definitions.

TABLE 2

Definition of Six Coupling Cases Simulated: Parameters Varied

Parameters Varied	Cases					
	1.0			2.0		
	1.0	1.2	1.3	2.1	2.2	2.3
1. Weight of car(s) at front of cask-rail car (W_F), lb_f	177,000 (3)*	177,000	177,000	250,000 (Ave. Switch Engine Weight)	250,000	250,000
2. Number of cars at front of cask-rail car (NCARSF)	5	5	5	1	1	1
3. Brake setting at front and rear trucks on cask-car (BRAKEF,BRAKER)	OFF	OFF	OFF	ON	ON	ON
4. Brake setting of car(s) at front of cask-car (BRKIRC)	ON	ON	ON	OFF	OFF	OFF
5. Initial velocity of car(s) at front of cask-car (VXFI), inches/sec	0	0	0	176 (10 MPH)	176 (10 MPH)	176 (10 MPH)
6. Initial velocity of cask-rail car (VXRCI), inches/sec	-176	-176	-176	0	0	0
7. Distance from the rail car center-line to the rear tiedown attachment point (LCR), inches	150	100	50	150	100	50
8. Distance from the rail car center-line to the front tiedown attachment point (LCF), inches	50	100	150	50	100	150
9. Distance between vertical center-lines of the rail car and the cask (LOCR), inches	50	0	-50	50	0	-50

* References

TABLE 3

Maximum and Minimum Values of Displacements and Accelerations
Obtained from Simulation of Six Coupling Cases

Displacements and Accelerations	Cases					
	1.0			2.0		
	1.0	1.2	1.3	2.1	2.2	2.3
1. Horizontal Displacements, Inches						
XF*	0	0	0	70.12	69.85	70.12
	-12.94	-12.92	-12.94	0	0	0
XP	0	0	0	70.27	69.99	70.27
	-12.99	-12.95	-12.99	0	0	0
XRC	0	0	0	70.03	69.76	70.03
	-13.02	-12.98	-13.02	0	0	0
XTR, XTF	0	0	0	70.03	69.76	70.03
	-13.08	-13.06	-13.08	0	0	0
2. Vertical Displacements, Inches						
YP	0.0524	0	0.0597	0.0719	0	0.0619
	-0.0597	0	-0.0524	-0.0619	0	-0.0719
YRC	0.2616	0	0.2674	0.326	0	0.302
	-0.2674	0	-0.2617	-0.302	0	-0.326
YP12	0.482	0.433	0.416	0.678	0.613	0.655
	-0.555	-0.517	-0.505	-0.689	-0.668	-0.657
YP34	0.505	0.517	0.555	0.657	0.668	0.689
	-0.416	-0.433	-0.482	-0.655	-0.613	-0.678
YRC12	0.409	0.468	0.490	0.574	0.561	0.563
	-0.455	-0.506	-0.500	-0.559	-0.626	-0.621

* See Nomenclature of Terms for definition of terms.

TABLE 3 (Cont'd)

Displacements and Accelerations	Cases 1.0: 1.0	1.2	1.3	2.0: 2.1	2.2	2.3
2. Vertical Displacements (Cont'd)						
YRC34	0.500	0.506	0.455	0.621	0.626	0.559
	-0.490	-0.468	-0.409	-0.563	-0.561	-0.574
YRC56	1.502	1.313	0.973	1.858	1.624	1.206
	-1.422	-1.215	-0.888	-1.658	-1.455	-1.153
YRC78	0.888	1.215	1.422	1.153	1.455	1.658
	-0.978	-1.313	-1.501	-1.206	-1.624	-1.858
3. Angular Displacements, Radians						
THP	0.0053	0.00517	0.0053	0.00673	0.00668	0.00673
	-0.00449	-0.00433	-0.00449	-0.00638	-0.00613	-0.00638
THRC	0.0048	0.00506	0.0048	0.0059	0.00626	0.0059
	-0.00445	-0.00468	-0.00445	-0.00528	-0.0056	-0.00528
4. Horizontal Accelerations, Inches/Sec^2						
D2XF	15,300	15,300	15,300	10,400	10,300	10,400
	-13,400	-13,400	-13,400	-11,300	-11,300	-11,300
D2XP	19,500	19,500	19,500	20,300	20,300	20,300
	-11,600	-12,000	-11,600	-14,700	-16,100	-14,700
D2XRC	22,050	19,600	22,050	23,900	25,400	23,900
	-23,800	-19,900	-23,800	-29,600	-25,900	-29,600

TABLE 3 (Cont'd)

Displacements and Accelerations	Cases 1.0: 1.0	1.2	1.3	2.0: 2.1	2.2	2.3
4. Horizontal Accelerations, Inches/Sec) (Cont'd)						
D2XTR, D2XTF	28,700	29,900	28,700	23,500	23,500	23,500
	-38,800	-40,800	-38,800	-30,400	-32,900	-30,400
5. Vertical Accelerations, Inches/Sec						
D2YP	2,900	0	3,040	2,940	0	3,240
	-3,040	0	-2,900	-3,240	0	-2,940
D2YRC	12,000	0	10,900	12,000	0	11,700
	-10,900	0	-12,000	-11,700	0	-12,000
D2YP12	20,900	16,400	19,500	23,850	24,800	20,350
	-20,500	-17,160	-24,700	-23,100	-25,600	-25,350
D2YP34	24,700	17,160	20,500	25,350	25,600	23,100
	-19,400	-16,400	-20,900	-20,350	-24,800	-23,850
D2YR12	15,900	14,000	16,200	15,500	15,900	17,900
	-16,400	-14,700	-18,300	-13,200	-14,900	-18,400
D2YR34	18,300	14,700	16,400	18,400	14,900	13,200
	-16,200	-14,000	-15,900	-17,900	-15,900	-15,500
D2YR56	44,600	38,000	27,500	45,100	38,800	23,600
	-38,500	-36,000	-25,300	-43,900	-41,200	-24,700
D2YR78	25,300	36,000	38,500	24,700	41,200	43,900
	-27,500	-38,000	-44,600	-23,600	-38,800	-45,100

TABLE 3 (Cont'd)

Displacements and Accelerations	Cases					
	1.0			2.0		
	1.0	1.2	1.3	2.1	2.2	2.3
6. Angular Accelerations, Radians/Sec						
D2THP	225.9	171.6	225.9	233.8	25[illegible].2	233.8
	-182.5	-164.4	-182.5	-212.4	-247.9	-212.4
D2THRC	125.7	146.9	125.7	127.6	149.4	127.6
	-106.6	-139.7	-106.6	-123.9	-158.6	-123.9

The spring constants for the couplers between the cask-rail car and the impacting car(s), as shown in Table 1 are actually composites of spring constants of each of the rail car structures from the end of the car to its center of gravity, and of rigid couplers. The spring constants for the rail car structures were based on values reported by Magnuson.(4) Consequently, the spring constant shown in Table 1 for the connector between the cars (k_{SCARS}) is extremely stiff and represents a severe case. Also, the spring constants assumed for the rail car suspension (k_{s5} through k_{s8}) are much too high when compared to data recently collected. Data on coupler and suspension spring characteristics have been obtained, and will be used to determine correct spring constants for subsequent computations (see Task 4, Collect Parameter Data).

The results summarized in Table 3 are the maximum and minimum values of displacements and accelerations obtained over the period of time from impact until the system comes to rest at some point down the track. The model automatically keeps track of the maximum and minimum values of specified variables during the transient and records them at the end of the simulation. The model also presents results as printed output, and/or as optional printer plots and CalComp plots.

2. Data Collection and Reduction

This task is not scheduled to begin until January 1, 1978. (See Figure 1.)

3. Validate Model

This task is not scheduled to begin until April 1, 1978. (See Figure 1.)

4. Collect Parameter Data

A limited literature search was made to collect data on key parameters to be used in the cask-rail car model described under Task 1. Data collected included characteristics of bulkhead flat rail cars (i.e., dimensions, weights, etc.), data on rail car suspension systems, data on draft gear (couplers), and data on heavy shielded spent fuel shipping casks and their tiedown systems.

Dimensions, weights and other data that make up specifications for the design, fabrication and construction of a 50-ton bulkhead flat car were obtained from the Association of American Railroads (AAR).[3] These data were supplemented by drawings supplied by Savannah River Laboratories of the flat car to be used in the forthcoming coupling tests.

Load-deflection characteristics and the arrangement of springs in railcar suspension systems have been obtained from AAR specifications (References 5 and 6, respectively). The load-deflection characteristics are given for helical springs, in terms of spring diameter and number of turns. These must be related to the proper height, number and grouping for a suspension system before they can be translated into a spring constant for that particular system.

Kasbekar, et al.,[7] present a piece-wise linear load-deflection curve for an M-901E draft gear obtained from tests performed by the AAR.[8] This curve may be programmed into the model during the next development phase. Roggeveen[9] implies that a spring constant of about 6.25×10^4 lb_f/ in. may be acceptable for a draft gear in a coupling situation.

Weights, dimensions and other data on some heavy shielded spent fuel shipping casks and their tiedown systems are available in Reference 2 and in safety analysis reports for the National Lead Industries NLI 1/2, and Nuclear Fuel Services NFS-4 shipping casks.

5. Parametric and Sensitivity Analysis

This task is not scheduled to begin until December 1, 1978. (See Figure 1.)

6. Interim Report

This report is not scheduled for preparation until August 1, 1978. (See Figure 1.)

7. Final Report

This report is not scheduled for preparation until August 1, 1979. (See Figure 1.)

NOMENCLATURE OF TERMS

BRAKEF, BRAKER = Brake switches applied to the front and rear rail car trucks, respectively. When the switches are set at 1.0. the brakes at the trucks are on or locked, and when they are set at 0., the brakes are off.

BRKIRC = Brake switch applied to the car(s) at the front of the rail car. Brakes are on when BRKIRC = 1.0, and off when set at 0.

D2THP = $\frac{d^2\theta_p}{dt^2}$ = Angular acceleration of the package or cask about an axis through its center of gravity (c.g.), radians/sec^2.

D2THRC = $\frac{d^2\theta_{RC}}{dt^2}$ = Angular acceleration of the rail car about an axis through its center of gravity, radians/sec^2.

D2XF = $\frac{d^2X_F}{dt^2}$ = Horizontal acceleration of the c.g. of the car(s) (mass M_F) at the front of the rail car, inches/sec^2.

D2XP = $\frac{d^2X_p}{dt^2}$ = Horizontal acceleration of the c.g. of the cask or package (M_p), inches/sec^2.

D2XRC = $\frac{d^2X_{RC}}{dt^2}$ = Horizontal acceleration of the c.g. of the cask rail car (M_{RC}), inches/sec^2.

D2XTR, D2XTF = $\frac{d^2X_{TR}}{dt^2}$, $\frac{d^2X_{TF}}{dt^2}$ = Horizontal accelerations of the c.g.'s of the rear (M_{TR}) and front (M_{TF}) rail car trucks, respectively, inches/sec^2.

D2YP = $\frac{d^2Y_p}{dt^2}$ = Vertical acceleration of the cask or package at its c.g., inches/sec^2.

D2YP12 = $\frac{d^2Y_{P12}}{dt^2}$ = Vertical acceleration of the cask or package at the rear tiedown attachment point, inches/sec^2.

D2YP34 = $\frac{d^2Y_{P34}}{dt^2}$ = Vertical acceleration of the cask or package at the front tiedown attachment point, inches/sec^2.

D2YRC = $\frac{d^2Y_{RC}}{dt^2}$ = Vertical acceleration of the cask-rail car at its c.g., inches/sec^2.

D2YR12 = $\frac{d^2Y_{RC12}}{dt^2}$ = Vertical acceleration of the cask-rail car at the rear tiedown attachment point, inches/sec^2.

D2YR34 = $\frac{d^2Y_{RC34}}{dt^2}$ = Vertical acceleration of the cask-rail car at the front tiedown attachment point, inches/sec^2.

D2YR56 = $\frac{d^2Y_{RC56}}{dt^2}$ = Vertical acceleration of the cask-rail car at the support point at the rear truck, inches/sec^2.

D2YR78 = $\frac{d^2Y_{RC78}}{dt^2}$ = Vertical acceleration of the cask-rail car at the support point at the front truck, inches/sec^2.

K = The kinetic energy of the system, lb_f-inch.

k_{SCARS} = A combination stiffness (spring constant) for the system of springs separating the c.g.'s of the cask-rail car (M_{RC}) and the cars at the front of the cask-rail car (M_F), lb_f/inch. (The system of springs represented by k_{SCARS} includes those for the structures of the rail cars (k_{SRC} and k_{SF}) and is based on the assumption of rigid couplers.)

NOMENCLATURE OF TERMS (Cont'd)

k_{SF} = Stiffness of the structure of the car(s) (M_F) at the front of the cask-rail car, lb_f/inch.

k_{SRC} = Stiffness of the structure of the cask-rail car (M_{RC}), lb_f/inch.

k_{S1} = Stiffness of the horizontal component of the rear tie-down between the cask (M_p) and the rail car (M_{RC}), lb_f/inch.

k_{S2} = Stiffness of the vertical component of the rear tiedown between the cask (M_p) and the rail car (M_{RC}), lb_f/inch.

k_{S3} = Stiffness of the vertical component of the front tiedown between the cask (M_p) and the rail car (M_{RC}), lb_f/inch.

k_{S4} = Stiffness of the horizontal component of the front tiedown between the cask (M_p) and the rail car (M_{RC}), lb_f/inch.

k_{S5} = Stiffness of the horizontal component of the cask-rail car suspension at the rear truck, lb_f/inch.

k_{S6} = Stiffness of the vertical component of the cask-rail car suspension at the rear truck, lb_f/inch.

k_{S7} = Stiffness of the vertical component of the cask-rail car suspension at the front truck, lb_f/inch.

k_{S8} = Stiffness of the horizontal component of the cask-rail car suspension at the front truck, lb_f/inch.

l_{CF}, LCF = Horizontal distance from the vertical centerline of the cask-rail car to the front tiedown attachment point, inches.

l_{CR}, LCR = Horizontal distance from the vertical centerline of the cask-rail car to the rear tiedown attachment point, inches.

l_{OCR}, LOCR = Horizontal distance between the vertical centerlines of the cask and cask-rail car, inches.

l_{PF}, LPF = Horizontal distance from the vertical centerline of the cask to the front tiedown attachment point, inches.

l_{PR}, LPR = Horizontal distance from the vertical centerline of the cask to the rear tiedown attachment point, inches.

l_{RC}, LRC = Horizontal distance from the vertical centerline of the cask-rail car to a suspension point at a truck, inches (2*LRC = distance between suspension points.)

M_F, MF = Mass of the car(s) at the front of the cask-rail car, $\frac{lb_f - sec^2}{inch}$.

M_p, MP = Mass of the cask or package, $\frac{lb_f - sec^2}{inch}$.

M_{RC}, MRC = Mass of the cask-rail car, $\frac{lb_f - sec^2}{inch}$

M_{TF}, MTF = Mass of the front truck on the cask-rail car, $\frac{lb_f - sec^2}{inch}$

M_{TR}, MTR	=	Mass of the rear truck on the cask-rail car, $\frac{lb_f - sec^2}{inch}$
NCARSF	=	Number of cars at the front of the cask-rail car.
q_i	=	The i-th generalized coordinate.
$\dot{q}_i = \frac{dq_i}{dt}$	=	The time rate of change of the i-th generalized coordinate
S_{CARS}	=	A composite spring connecting the c.g.'s of the cask-rail car (M_{RC}) and the car(s) at the front of the cask-car (M_F). This spring is composed of springs representing the structures of M_{RC} and M_F, and is based on the assumption of rigid couplers.
S_1, S1	=	A spring representing the horizontal component of the rear tiedown between the cask (M_p) and the rail car (M_{RC}).
S_2, S2	=	A spring representing the vertical component of the rear tiedown between M_p and M_{RC}.
S_3, S3	=	A spring representing the vertical component of the front tiedown between M_p and M_{RC}.
S_4, S4	=	A spring representing the horizontal component of the front tiedown between M_p and M_{RC}.
S_5, S5	=	A spring representing the horizontal component of the cask-rail car suspension at the rear truck.

NOMENCLATURE OF TERMS (Cont'd)

S_6, S6	=	A spring representing the vertical component of the cask-rail car suspension at the rear truck.
S_7, S7	=	A spring representing the vertical component of the cask-rail car suspension at the front truck.
S_8, S8	=	A spring representing the horizontal component of the cask-rail car suspension at the front truck.
S_9, S9	=	A composite spring connecting the c.g. of the cask-rail car to the tip of its coupler.
S_{10}, S10	=	A composite spring connecting the c.g. of the car(s) (M_F) at the front of the cask-rail car to the tip of its coupler.
t, T	=	Time, seconds.
THP = θ_p	=	The angle of rotation of the X_p and Y_p axes about an axis perpendicular to the X_p - Y_p plane through the c.g. of the cask or package, radians.
THRC = θ_{RC}	=	The angle of rotation of the X_{RC} and Y_{RC} axes about an axis perpendicular to the X_{RC} - Y_{RC} plane through the c.g. of the rail car, radians.
U	=	The potential energy or internal strain energy of the system, lb_f - inch.
VXFI	=	The initial velocity of the car(s) (M_F) at the front of the cask-rail car, inches/sec.

NOMENCLATURE OF TERMS (Cont'd)

VXRCI	=	The initial velocity of the cask-rail car (M_{RC}), inches/sec.
W_C	=	The work done on the system by damping forces, lb_f - inch.
W_e	=	The work done on the system by external forces, lb_f - inch.
W_p	=	The weight of the cask or package, lb_f.
W_{RC}	=	The weight of the cask-rail car, lb_f.
W_{TF}	=	The weight of the front truck on the cask-rail car, lb_f.
W_{TR}	=	The weight of the rear truck on the cask-rail car, lb_f.
X_F, XF	=	The horizontal displacement of the c.g. of the car(s) (M_F) at the front of the cask-rail car, inches.
X_p, XP	=	Horizontal displacement of the c.g. of the cask or package, inches.
X_{p12}, XP12	=	Horizontal displacement of the cask at the rear tiedown attachment point, inches.
X_{p34}, XP34	=	Horizontal displacement of the cask at the front tie-down attachment point, inches.
X_{RC}, XRC	=	Horizontal displacement of the cask-rail car at its c.g., inches.
X_{RC56}, XRC56	=	Horizontal displacement of the cask-rail car at the support point at the rear truck, inches.

NOMENCLATURE OF TERMS (Cont'd)

X_{RC78}, XRC78 = Horizontal displacement of the cask-rail car at the support point at the front truck, inches.

X_{TF}, XTF = Horizontal displacement of the front truck on the cask-rail car, inches.

X_{TR}, XTR = Horizontal displacement of the rear truck on the cask-rail car, inches.

Y_p, YP = Vertical displacement of the c.g. of the cask or package, inches.

Y_{P12}, YP12 = Vertical displacement of the cask at the rear tiedown attachment point, inches.

Y_{P34}, YP34 = Vertical displacement of the cask at the front tiedown attachment point, inches.

Y_{RC}, YRC = Vertical displacement of the cask-rail car at its c.g., inches.

Y_{RC12}, YRC12 = Vertical displacement of the cask-rail car at the rear tiedown attachment point, inches.

Y_{RC34}, YR34 = Vertical displacement of the cask-rail car at the front tiedown attachment point, inches.

Y_{RC56}, YRC56 = Vertical displacement of the cask-rail car at the support point at the rear truck, inches.

Y_{RC78}, YRC78 = Vertical displacement of the cask-rail car at the support point at the front truck, Inches.

Z_p, ZP = Vertical distance from the horizontal centerline of the cask to its top and bottom surfaces, inches.

Z_{RC}, ZRC = Vertical distance from the horizontal centerline of the cask-rail car to its top and bottom surfaces, inches.

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