CONF-960706--30

SAND96-1921C

STRUCTURAL ANALYSIS IN SUPPORT OF THE WATERBORNE TRANSPORT OF RADIOACTIVE MATERIALS*

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ABSTRACT

The safety of the transportation of radioactive materials by road and rail has been well studied and documented. However, the safety of waterborne transportation has received much less attention. Recent highly visible waterborne transportation campaigns have led the DOE and IAEA to focus attention on the safety of this transportation mode. In response, Sandia National Laboratories is conducting a program to establish a method to determine the safety of these shipments. As part of that program the mechanics involved in ship-to-ship collisions are being evaluated to determine the loadings imparted to radioactive material transportation packages during these collisions. This paper will report on the results of these evaluations.

INTRODUCTION

Sandia National Laboratories' (SNL) SeaRAM project (McConnell et al. 1995), which is sponsored by the U.S. Department of Energy (DOE) is studying the safety of shipping radioactive materials (RAM) by sea,. The project is concerned with the potential effects of ship collisions and fires to on-board RAM packages. Existing methodologies are being assessed to determine their adequacy to predict the effect of ship collisions and fires on RAM packages and to estimate whether or not a given accident might lead to a release of radioactivity. The eventual goal is to develop a set of validated methods, which have been checked by comparison with test data and/or detailed finite element analyses, for predicting the consequences of ship collisions and fires. These methods could then be used to provide input for overall risk assessments of RAM sea transport. The emphasis of this paper is on methods for predicting the effects of ship collisions.

A concern regarding the safety of RAM transport by sea is the possibility of another ship striking the RAM-carrying ship leading to leakage of a RAM package(s). One basis for this concern is the large amount of kinetic energy of the striking ship. Kinetic energies in excess of those for the regulatory impact test exist. This is due to the relatively large mass of some cargo ships and oil tankers, even though ship velocities are relatively small (usually less than 13.4 m/s).

The large amount of kinetic energy associated with ship collisions produces the impression that these collisions may result in greater damage to on-board radioactive material packages than the hypothetical accident sequence the packages are designed to withstand. However, the kinetic energy of the collision is not all absorbed in deformation of the radioactive material package as it is in the 9-meter drop onto an essentially rigid target. Therefore, the amount of kinetic energy in the collision is not an appropriate measure to assess the likelihood or degree of package damage. A better metric is the acceleration imposed on the packages during impact. Type B packages are designed to be leak-tight after being dropped from a height of 9 meters onto an essentially unyielding surface. Typical rigid body uniform accelerations experienced during impact are in the range of 50 to 200 G or higher. However, the highest levels of acceleration during a ship collision are less than 10 G (e.g. Lenselink and Thung1992), much less than expected for the 9-meter drop. The lower accelerations are due to the 'flexibility' of the impact surface, which is the deformable RAM-carrying ship and the bow of the striking ship. *Thus, only quasi-static, "crush" types of loading are of concern.*

Only cases in which the RAM-carrying ship is struck by another ship are considered as possible threats to RAM package integrity. Other collision scenarios in which the RAM ship strikes another ship or a rigid pier or runs aground are not believed to pose a threat to the packages since the packages are (presumably) stowed well away from the impact location.

There are two types of analyses that are necessary to determine if a given ship collision might lead to leakage from a RAM package. The first is a global analysis, devoted to the deformation of the ships during a collision, with the main output being the relative velocity of the striking ship as a function of depth of penetration into the struck (RAM-carrying) ship. The second analysis would be concerned with the "local"

*This work was supported by the U. S. Department of Energy under Contract No. DE-AC04-94AL85000.

behavior of a RAM package. The loading condition would be the bow of the striking ship on one side of the package backed by the internal structure of the struck ship or cargo on the other. The potential for damage to the package depends on the remaining velocity of the striking ship upon reaching the required depth of penetration (i.e., the package location) and the relative stiffness and strength of the striking ship bow, the RAM package, and the supporting structures in the struck ship. The work described in this paper only addresses the first of these two types of analysis. The analysis of the "local" response of the package is a topic for future research.

In the next section of this paper past research into the consequences of ship collisions, and the implications of these consequences to radioactive material package transportation will be discussed. Following this a proposed simplified method for determining the damage as a result of collision will be given. The final section will discuss the results of detailed finite element calculations to determine the response of a generic small freighter to impacts from vessels with varying mass and velocity.

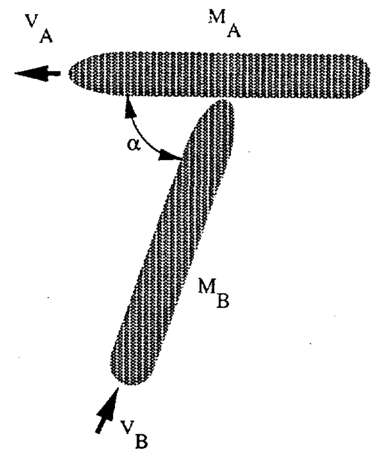


Figure 1. Ship Collision Parameters

SUMMARY OF GLOBAL SHIP COLLISION MECHANICS AND RELATED LITERATURE

Because of the complexity of the deformation processes during ship collisions, most prediction methods have been based on simplified methods for estimating the amount of damage to the respective ships. The methods are normally composed of two main steps. First, the amount of energy to be absorbed during impact must be computed. This step is sometimes referred to as the "external mechanics" part of the problem. The second step is to determine how the struck and striking ships deform in order to absorb the kinetic energy.

To simplify the ship collision mechanics, only collisions at near right angles are considered in this program. This seems to be a reasonable assumption for assessing the safety of RAM transport by sea, since transverse penetration into the RAM-carrying ship is the primary concern in a collision and such penetration will be greatest in a right angle collision.

External Mechanics

Calculation of energy to be absorbed is relatively straightforward, based on conservation of momentum and energy principles for an inelastic collision of two bodies (Minorsky 1959). First, assume that the center of gravity of the striking ship passes through that of the struck ship, such that there is no rotation of the ships during the collision. Also, assume that the angle between the striking and struck ship, α, is near 90°. The mass of the struck ship and striking ship is M_A and M_B, respectively, with initial velocities of V_A and V_B before the collision, as shown in Figure 1.

Based on conservation of momentum and kinetic energy perpendicular to the struck ship before and after the collision, the following expression can be derived for the amount of energy absorbed by deformation of the ship structures, ΔE_k:

$$\Delta E_k = \frac{1}{2}\frac{M_B(M_A + \Delta M)}{M_A + M_B + \Delta M}(V_B \sin\alpha)^2 \qquad (1)$$

As shown, ΔE_k is a function of the masses of the respective ships, the initial velocity of the striking ship, the angle between the ships just before impact, and the effective mass of water surrounding the ships that affects the collision mechanics, ΔM. The proper value of effective water mass is somewhat uncertain. Based on experiments of a ship hull vibrating in deep water, Minorsky estimated the effective mass to be 40% of the mass of the struck ship, M_A.

Internal Mechanics

It is the second step of the solution process, solving the "internal mechanics" problem, that is the most difficult. This step requires estimation of how the two ships deform in order to absorb the required amount of energy, ΔE_k. One of the earliest methods is an empirical approach developed by Minorsky in which a linear relationship was established between the amount of energy to be absorbed and the volume of material within the ships that is deformed during the collision:

$$\Delta E_k = (414.5R_T + 121{,}900) \text{ ton-knots}^2 \qquad (2)$$

R_T is known as a resistance factor, and is basically equal to the total volume of damaged structural materials in the striking and struck ships, except for the outer hull of the struck ship, which is accounted for in the constant term. The units of R_T are ft^2-in. The method for computing R_T is given in Minorsky's original paper. Minorsky studied 26 actual ship collisions, all of which involved nearly right-angle collisions. From these collisions, nine were finally used to fit a straight line between the points of ΔE_k and R_T. This line is represented by Equation 2 and is shown in Figure 2. The remaining collisions were not used since they involved relatively lower amounts of energy absorption and exhibited considerable scatter. This so-called "Minorsky Method" has been widely used and appreciated because of the simplicity that it brings to this complex problem. However, it does not account for the detailed mechanics of the collision process and, because of its empirical nature, it may not be applicable for ship designs and impact velocities that are outside the range of the parameters for which the method was developed.

There have been some attempts to check the accuracy of the Minorsky Method. These are documented in papers by (Akita et al.1972a) and others. Computations of ΔE_k and R_T based on additional ship collisions that, apparently, were not used by Minorsky have been performed (Gibbs and Cox 1961). The data from the Gibbs and Cox report and for the collision analyzed by MR&S (M. Rosenblatt & Son 1972) are shown in Figure 2, along with Minorsky's Equation 2 and the data that Minorsky used to obtain Equation 2. Note that two sets of points are enclosed within an ellipse. These points represent the same respective collisions. The only difference being the calculation of R_T by Gibbs and Cox and Minorsky.

As shown, there is considerable variance between some of this additional data and Minorsky's Equation for relatively low energy collisions. The shaded area of Figure 2 represents additional low energy ship collision data points available to Minorsky, but not used in developing Equation 2. Minorsky stated that the considerable scatter in the low energy range "undoubtedly stems from the fact that the masters of the striking vessels tend to underestimate their speed at impact." Better agreement with available ship collision data in Figure 2 can be obtained by modifying Minorsky's equation in the low energy range, as shown by the dashed lines. The proposed modified Minorsky equations are shown below:

For $0 \le \Delta E_k \le 218$ ton-knots^2:

$$\Delta E_k = 145 R_T \quad (\text{ton-knots}^2) \qquad (3a)$$

For $218 < \Delta E_k \le 744$ ton-knots^2:

$$R_T = 1500 \quad (\text{ft}^2\text{-in}) \qquad (3b)$$

For $\Delta E_k > 744$ ton-knots^2:

$$\Delta E_k = 414.5 R_T + 121{,}900 \ (\text{ton-knots}^2) \ (\text{original Minorsky Equation}) \qquad (3c)$$

Equation (3a) is taken from (Jones 1983) in which he and his colleagues developed a modified Minorsky Method for minor collisions. As shown in Figure 2, Equations 3a and 3b better represent

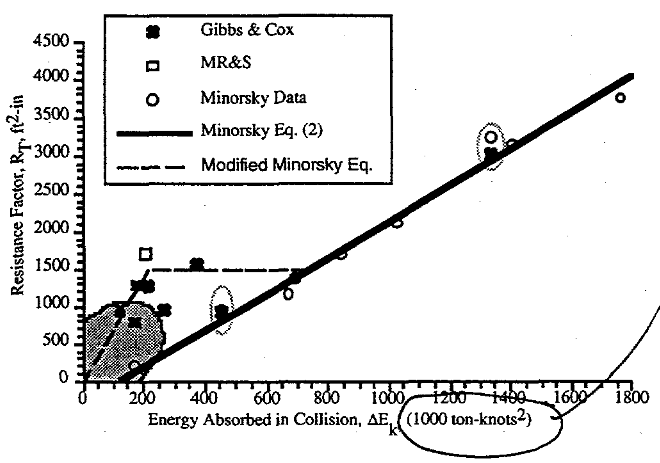


Figure 2. Comparison of Actual Ship Collision Data to Predictions from Minorsky's Equation.

the collision data for the lower energy points. Equation 3a is attractive because it begins at the origin (representing the obvious—that there is no deformed material, R_T, if no energy, ΔE_k, is absorbed) and because it traverses most of the low energy points. The physical meaning of Equation 3b is less appealing, since it indicates a constant amount of damage for increasing values of ΔE_k. However, Equation 3b does provide a more conservative estimate of damage, R_T, than Minorsky's original equation. Equation 3c is identical to Minorsky's original equation, since there seems to be good agreement with the ship collision data for these very high energy collisions. (R_T values for Equations 3a and 3b should include the hull of the struck ship using the approach described by Jones, whereas, the hull is not included in Minorsky's original equation.)

Minorsky's original work was motivated by needs to design the *Savannah*, the world's first nuclear-powered commercial ship. Protection of the nuclear reactor from collision damage was the primary concern and Minorsky's approach was employed to design the reactor protection system. During this same time period (late 1950's and 1960's), ship collision research programs were also conducted in Germany, Japan, and Italy in support of the design of nuclear powered ships. In the 1970's there was some work devoted to liquefied natural gas tanker safety in collisions; however, most of the recent and ongoing ship collision research is devoted to the safety of oil tankers involved in collisions and grounding. These programs are focused on the study of improved tanker designs to minimize the probability of oil leakage in the event of an accident.

The earlier work for nuclear-powered ships is more applicable to the present study of RAM sea transport than the more recent studies. The reason being that the nuclear-powered ship research was concerned about extremely severe collisions, since protection of the reactor, located near the middle of the ship's breadth, was its focus. Similar damage would be required to threaten on-board RAM package integrity. However, the tanker studies are primarily concerned with improving designs to resist relatively minor collisions that could rupture the oil tanks. Since it is not feasible to design tankers to resist all possible collisions, there has been little attention to the extremely severe collision scenarios.

Scale model ship collision experiments were conducted during the nuclear ship design era as described by (Akita et al. 1972a,1972b). Akita developed two sets of semi-empirical expressions for the load required for a rigid bow to penetrate the breadth of a ship's structure. The first set is for what was termed the "deformation type" of failure of the deck and the second is for the "crack type" failure. He observed that the crack type failure generally occurred when the strain underneath a bow was greater than about 30%. The crack type failure mode, which is illustrated in Figure 3, is more straightforward to use and seems to result in more conservative estimates of penetration depth.

The load-deformation (P-δ) relationship based on Figure 3 may be derived from simple statics as (Akita et al. 1972a):

$$P = 2Nq\delta\tan\theta + 2T\cos\theta \qquad (4)$$

where:

- P = collision loading from striking ship,
- δ = penetration into the struck ship,
- q = compressive reaction load per unit length on deck, $= t_d\sigma_o$,
- t_d = average deck thickness obtained by smearing deck stiffener areas over deck width,
- σ_o = effective crush stress, $n\sigma_y$,
- σ_y = deck material yield strength,

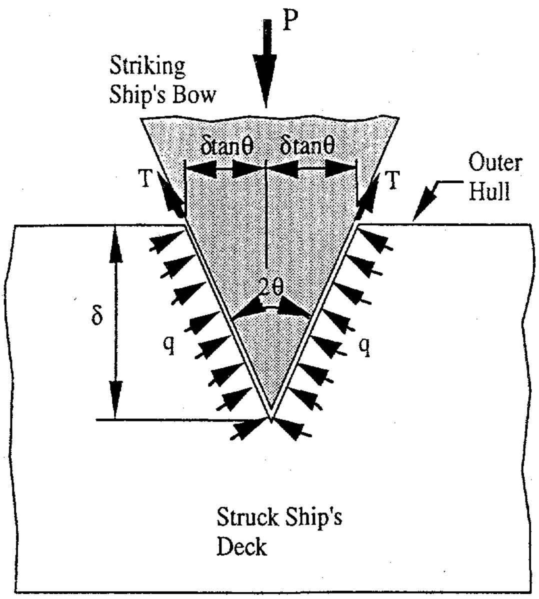


Figure 3. Deformation of Struck Ship

n = reduction factor to account for deck buckling stress as a portion of the yield stress,
N =number of deck layers,
2θ =stem angle of striking vessel, and
T = membrane strength of outer hull of struck ship.

As indicated in Equation 4, load from the striking ship is resisted by the outer hull and decks of the struck ship. Early in the collision, load is primarily resisted by the outer hull until it fails in membrane tension as it stretches between transverse supports. After hull rupture, load is resisted almost entirely by the decks. To conservatively fit his test data, Akita assumed that the deck crushed at an average stress equal to $0.8\sigma_y$, or n = 0.8 according to the above definition.

As shown in Figure 4,the energy absorbed by a struck ship for a given deformation, δ_n, is equal to the area under the P-δ curve up to δ_n. The maximum deformation for a given collision, δ_{max}, can be determined by solving for δ such that the area under the P-δ curve equals the required energy to be absorbed in a collision, ΔE_k, as computed from Equation 1.

This approach is believed to be quite conservative, since it assumes all the energy is absorbed by the struck ship and none by the striking ship. This assumption would be most valid if the striking ship's bow was effectively rigid. In order to account for energy absorbed by deformation of the striking ship's bow, one could also consider the P-δ relationship of the striking ship's bow. Several studies have been conducted to estimate this relationship, such as (Akita and Kitamura 1972b). The maximum penetration into the struck ship can be computed by the same method as described above, given that the load on both ships, P_c, will be equal at all times and by increasing the load until the combined area under both P-δ curves equals the computed value of ΔE_k. This method is qualitatively illustrated in Figure 5. The proportion of energy absorbed by the striking and struck ships depends on their relative stiffness.

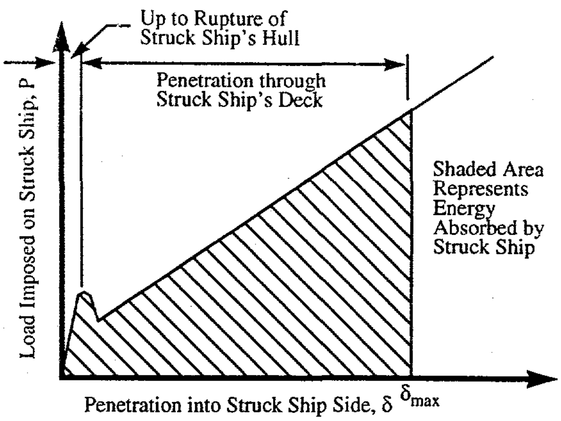


Figure 4. Typical Load-Deformation Relationship for Side of Struck Ship

Given the P-δ relationships for both the struck and striking ships, the equations of motion can be readily solved using a spring-mass formulation. A FORTRAN program, using explicit integration to solve the equations of motion, has been successfully completed. The analysis computed the collision force, velocity reduction and energy absorption, as a function of penetration and time into the collision, and total collision time and energy absorption. Since the solution time on modern PCs is only a few seconds, multiple collision scenarios, which must be considered for comprehensive risk studies, could be considered without unreasonable computing costs or time requirements.

SIMPLIFIED METHOD FOR DETERMINING SHIP COLLISION DAMAGE

Based on the literature studied to date, the P-δ approach illustrated in Figures 3 through 5 is believed to be the most appropriate approach for future use in risk assessments of the safety of waterborne RAM transport. Once software is written to fully implement the method, solutions for multiple ship collision scenarios can be obtained without requiring extensive computer costs or time.

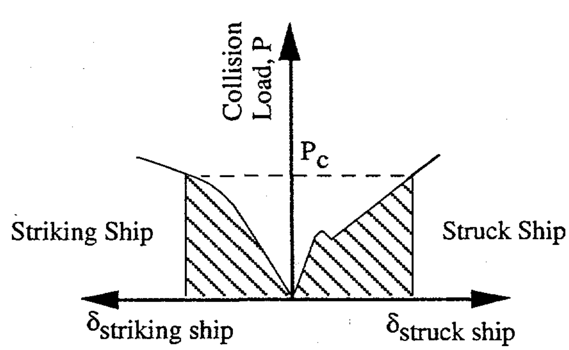


Figure 5. Qualitative View of Load Sharing Between Striking and Struck Ship

Uncertainties in the global ship collision mechanics result from the assumptions required to develop a one-dimensional P-δ approach for an actual ship collision, which is a complex three-dimensional problem. However, given the conceptual agreement with Minorsky's empirical approach and the conservative comparison with Akita's ship collision experiments, it is believed that the P-δ approach will provide reasonably good estimates for safety evaluations. Further study is needed, such as comparing the method to results from detailed finite element analyses and, if possible, to actual ship collision damage, to better quantify uncertainties in the method.

FINITE ELEMENT SIMULATIONS OF SHIP COLLISIONS

Two series of finite element simulations were performed. The first series of analyses was performed to evaluate the amount of penetration of the freighter hull by a striking ship of various masses and initial velocities. Although these analyses included a representation of a single RAM package, the package was not impacted during the collision so forces on the package could not be computed. Therefore, a second series of analyses incorporating a representation of a row of seven packages was performed to ensure direct package impact by the striking ship. Average forces on a package were evaluated for several initial velocities and masses of the striking ship.

The problem modeled is that of a small freighter with the dimensions shown in Figure 6 impacted by ships of the same mass or more.The struck freighter is assumed to have a mass of 1675 metric tons and zero initial velocity. In each series of analyses, the response of the freighter was evaluated when impacted by a striking ship of various masses travelling at various initial velocities normal to the longitudinal axis of the freighter. For all analyses the shape of the striking ship was the same as the shape of the struck freighter, with the exception that the bow of the striking ship was deeper than the bow of the freighter such that the striking ship strikes both the top deck and the bottom hull of the struck freighter. The striking bow was assumed to be vertical (zero rake angle), and all impacts were assumed to occur near the midsection of the freighter to maximize the damage incurred by the freighter. The first series of analyses (designated 1S to 4S) incorporated a simplified representation of a single RAM package initially located adjacent to the hull of the freighter opposite the striking ship. The package represented is that of a 22.7 metric tons truck cask. In the second series (designated 1M to 4M), a representation of a row of similar packages was incorporated. Specific problem parameters are given in Table 1. Case 1 represents an accident that could take place in a harbor or other congested region, Case 2 is a representation of a real accident involving a small freighter. Cases 3 and 4 are upper bounds to the velocity that ships travel and an upper bound on the mass of a ship with this narrow of a bow. These cases were modeled to ensure analyses of severe damage were included.

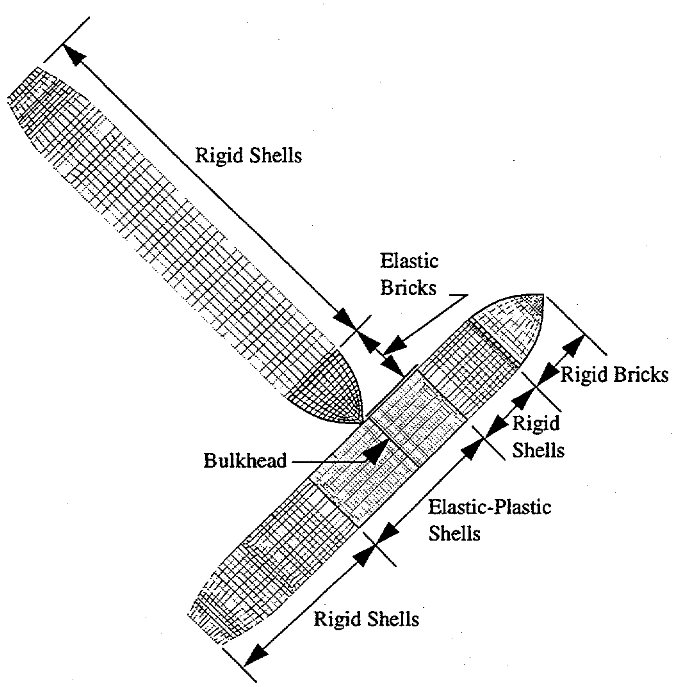


Figure 6. Finite element model.

Table 1: Problem Parameters

Analysis	Initial Velocity of Striking Ship [knots (m/s)]	Mass of Striking Ship [metric tons]
1S, 1M	10 (5.14)	1675
2S, 2M	15 (7.73)	10,050
3S, 3M	25 (12.9)	16,750
4S, 4M	30 (15.6)	16,750

Model Description

In general, modeling the collision of two ships involves a very complicated coupled problem between the response of the water and the structural deformation of the ships. During a collision, kinetic energy is dissipated in structural deformation of the ships and by motion of water. However, review of previous published analyses which used a loosely coupled approach (Lenselink and Thung 1992), showed that the amount of kinetic energy that is dissipated in structural deformation is nearly the same whether or not the water is explicitly included in the analyses (Porter 1995). Therefore, in the analyses reported here, the water is not explicitly modeled. Rather, water effects are included following the method (Minorsky 1959) in which the mass of the struck ship is assumed to be 40% greater than its actual mass.

Most of the striking ship and the forward and aft portions of the struck ship are modeled using rigid elements. Because rigid elements are computationally efficient and do not influence the critical time step, their use allows a more refined mesh of deformable elements in the areas of importance, that is, in the areas of greatest deformation. The bow of the striking ship was modeled with eight-noded brick elements assumed to be elastic so that their deformation is negligible compared to the elastic-plastic shell elements in the midship section of the struck ship. This simulates a rigid bow so that nearly all deformation energy is incurred by the struck ship. The elements in the bow of the striking ship are modeled as elastic rather than rigid in order to allow the contact algorithm to properly distribute the contact forces.

The packages were modeled with a coarse mesh using elastic rectangular prisms with four elements each, because they were used only to evaluate average forces and not to analyze deformation. The multiple package representation consists of seven packages side-by-side spanning 80% of the breadth of the freighter. In order to get a conservative estimate of the forces on the packages, they were assumed to be rigidly tied together. In all analyses performed, the packages were free to move. No tie-downs were modeled, there was no gravity, and no friction.

In both series of analyses, the major components of the deformable part of the struck ship are the outer hull, a transverse bulkhead, a lower deck that extends the entire width of the ship, a partial middle deck, and the upper deck and hatch cover. The hatch cover was assumed to be rigidly attached to the upper deck to maximize the stiffness of the struck ship and its ability to impart load to the RAM packages. All elastic-plastic shell elements were thickened to represent the smeared effects of beams and stiffeners which could not practically be modeled explicitly with shell elements due to their small size. The thickened area of the shells was the same as the area of the hull with stiffeners. This approximation changes the local behavior of the shells and may have an affect on the initiation of tearing in the shells. All shell elements are four-noded quadrilaterals using five integration points through the thickness. These shells were assumed to be constructed of mild steel similar to ASTM A36.

Rigid elements were used to capture the proper geometry but were given a very small density. The motion of each block of rigid elements is controlled through the designation of a point mass at the desired center of gravity, in this case assumed to be near the center of each ship. Similarly, the rotational motion is controlled through the use of designated mass moments of inertia. To maximize deformation, the mass moments of inertia in each of the three global directions was set large enough to prevent large rotational motion.

Results

As described above, a total of eight different ship collision scenarios were modeled. In each case, the Sandia-developed transient dynamics finite element code, PRONTO3D (Taylor and Flanagan 1989) was used running on a CRAY J90 machine. Run times averaged about 14 hours of CPU time for modeling one second of real time. However, the actual duration of impact as measured by time to maximum penetration turned out to be less than one second in all cases.

Finite element results of each scenario are described in detail below. The impact event was assumed to be over when maximum penetration was reached, or equivalently, when the kinetic energy reached a constant value, meaning that no additional energy was being dissipated by structural deformation.

Single Package Results

Results of the finite element computations for the first series of analyses with the single package representation are given in Table 2. Figure 7 contains plots of the deformation of the struck ship from three different views for the most severe impact (Case 4). Two views from the top are shown, one including the top deck and hatch cover and a second in which these have been removed so that internal damage can be seen. A view of the internal damage and the package from the stern is also included. Even in this case the striking ship only penetrated the struck ship to slightly more than half of its breadth. Therefore, during this series of analyses, the package initially located adjacent to the hull farthest from the striking ship was not directly impacted by the striking ship during the impact event.

Table 2: Results Summary for Series S

Parameter	Case			
	1S	2S	3S	4S
Impact Velocity (knots, (m/s)	10 (5.14)	15 (7.73)	25 (12.9)	30 (15.6)
Initial Kinetic Energy (MJ)	22.1	300.3	1394	2038
Duration (s)	0.27	0.50	0.66	0.68
Loss in Kinetic Energy (MJ)	11	95	381	493
Maximum Penetration (m)	0.8	2.2	4.2	5.2
Velocity of Both Ships at Time of Max. Penetration (m/s)	2.0	5.8	10.1	12.5

Multiple Package Results

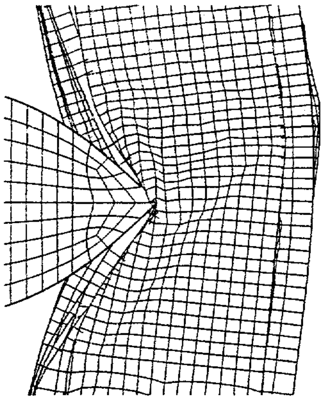
A second series of analyses was conducted in order to measure the force that a RAM package might experience during a ship collision. In these analyses, a representation of a row of packages spanning 80% of the breadth of the ship was incorporated to ensure direct impact and crushing of the packages in at least some of the analyses. Table 3 summarizes the results for duration of impact and loss in kinetic energy for this series of analyses.

Table 3: Results Summary for Series M

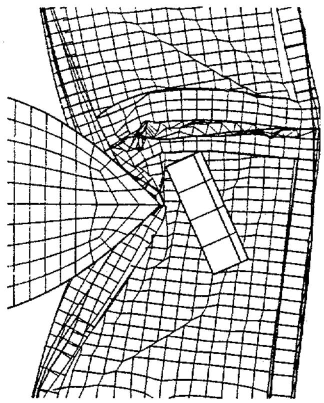
Parameter	Case			
	1M	2M	3M	4M
Duration, (s)	0.25	0.47	0.58	0.50
Loss in Kinetic Energy, (MJ)	11	81	356	369
Kinetic Energy of Striking Ship at Initial Impact with Packages, (MJ)	Did not impact.	234	1270	1880

Deformation of the freighter at maximum penetration for Case 2M is shown in Figure 8. Although it is not clear from the views shown, at the time of maximum penetration the packages were not actually in contact with either side of the freighter hull. Figure 8 also shows the time history of the average force on the packages. The initial compressive peak at 0.27 seconds was caused by the first contact of the packages with the impending ship. Initial contact with the front hull forces the packages toward the back hull.

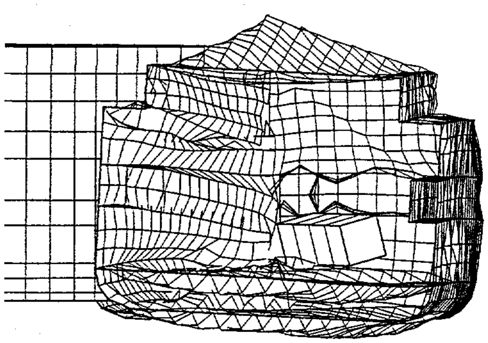
As shown in Figure 9, similar, but greater, deformation and loads are seen in Case 4M. Initial impact of the striking ship and forward hull on the packages occurs between 0.13 and 0.14 seconds, resulting in the initial peak compressive force of 370 MN. As in previous analyses, this



Top view with hatch cover in place.



Top view without hatch cover and upper deck.



View from stern.

Figure 7. Maximum deformation for Case 4S (v = 30 knots, mass = 16,750 tonnes, t = 0.68 seconds).

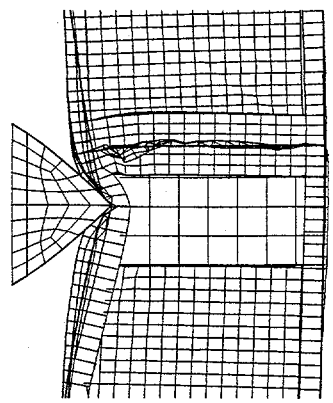
is not a crushing force because the packages are not in contact with the back wall at this time. By 0.16 seconds, the packages have impacted the back hull, and at 0.22 seconds they are in contact with both sides of the hull simultaneously. By 0.30 seconds the packages have rotated so that the back end impacts the lower side of the middle decks.

Recommendations From Finite Element Analyses

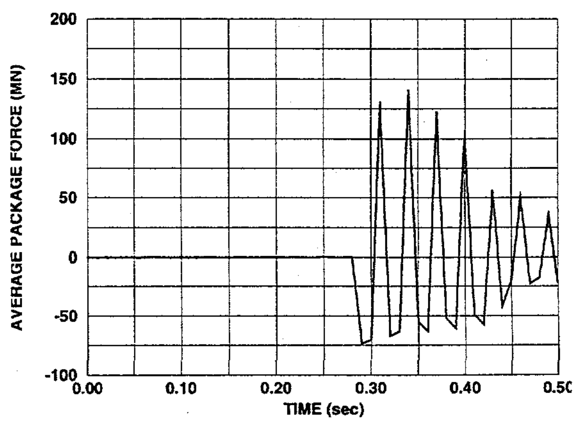
Eight different scenarios of two ships colliding have been analyzed using transient dynamic finite element computations for three-dimensional structures. The first four analyses included a representation of a single truck cask initially placed adjacent to the hull opposite the striking ship. These analyses were used to compute the penetration of the striking ship into the hull of a common small freighter. None of these analyses resulted in the impact or crushing of the included package. The maximum penetration computed was 5.2 m, a distance only slightly greater than one-half of the breadth of the freighter. However, even with this amount of penetration, the hull of the struck ship indicated only minor tearing. Because the prediction of tearing is mesh dependent, more computations should be done with a finer mesh in the area of greatest plastic strain.

Each analysis in the second series included a representation of seven casks lying side-by-side spanning 80% of the breadth of the freighter to ensure impact and crushing of the packages in at least some of the analyses. Results show that the greatest force on the packages occurs at the initial impact with the forward side of the hull as the striking ship penetrates. Crushing forces that occur later during the collision are much less. These impact forces are likely less than would be seen during a regulatory drop test because the impact occurs at a lower velocity and the bow of the striking ship is not rigid.

The amount of penetration seen in these analyses is less than the amount predicted using simplified calculations, such as the Minorsky method, and the degree of tearing is less than is typically seen in this type of impact. Some of the reasons for these results are the fact the impact point on the struck ship is very near to the transverse bulkhead and partial between- decks. This is the stiffest region of the struck ship for side impacts. Also, the artificial stiffening of the shell elements to

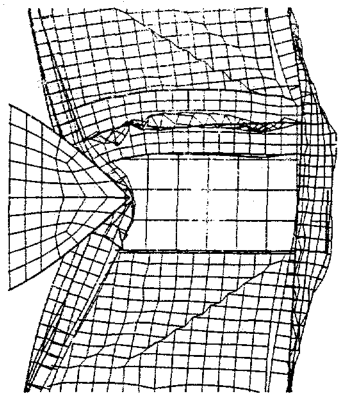


Top view with hatch cover and upper deck removed.

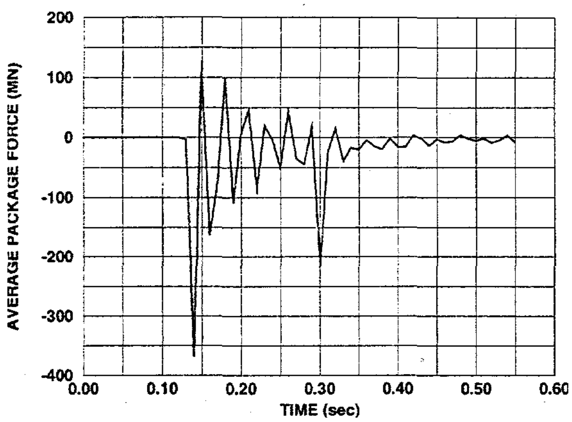


Average forces acting on the simulated RAM packages.

Figure 8. Maximum deformation and average force on the simulated radioactive material packages for Case 2M (v = 15 knots, mass = 10,050 tonnes, t = 0.47 seconds).



Top view with hatch cover
and upper deck removed.



Average forces acting on the
simulated RAM packages.

Figure 9. Maximum deformation and average force on the simulated radioactive material packages for Case 4M (v = 30 knots, mass = 16,750 tonnes, t = 0.50 seconds).

eliminate the need to model the beam stiffeners makes these elements more resistant to tearing. In all of these models the hatch covers were assumed to be rigidly attached to the top deck. This assumption causes the struck ship to be stiffer than it would be if the hatch covers were allowed to slip off the top deck. The final source of limited tearing is the mesh size. A coarser mesh distributes localized strains over a larger area, thereby reducing the average strain in the element and delaying the onset of tearing. It is likely the stiffening of the ship caused by these factors does not decrease the crush forces seen by the simulated radioactive material packages because these factors make the back hull of the struck ship stiffer as well. So even though the penetration distance and tearing of the forward portion of the ship are underestimated, the forces acting on the package are probably conservative.

CONCLUSIONS

The mechanics of collisions between two ships has been studied. For this type of collision to have the potential to damage on-board radioactive material transportation packages three things must occur. First, the collision must be severe enough so the bow of the striking ship penetrates to the location of the package. Then, the striking ship must have sufficient residual velocity to penetrate further into the ship, as the initial collision between the bow and the package will be less severe than the regulatory impact of the package onto an unyielding target. Finally, the residual velocity of the striking ship must push the package against something that is strong enough to crush it. In the finite element analyses it was seen that the strength of the hull on the opposite side of the ship modelled limited the magnitude of the crush force that could be applied. It is possible, however, to postulate scenarios where other cargo in the hold can distribute the force over a sufficiently large portion of the hull that crushing of the package may occur. Detailed finite element analyses of the package subjected to crush forces of this magnitude will be performed to assess the amount of damage to the package and the potential for radioactive material release.

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