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A Comparison of Methods for Evaluating Structure During Ship Collisions*

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ABSTRACT

A comparison is provided of the results of various methods for evaluating structure during a ship-to-ship collision. The baseline vessel utilized in the analyses is a 67.4 meter in length displacement hull struck by an identical vessel traveling at speeds ranging from 10 to 30 knots. The structural response of the struck vessel and motion of both the struck and striking vessels are assessed by finite element analysis. These same results are then compared to predictions utilizing the "Tanker Structural Analysis for Minor Collisions" (TSAMC) Method, the Minorsky Method, the Haywood Collision Process, and comparison to full-scale tests. Consideration is given to the nature of structural deformation, absorbed energy, penetration, rigid body motion, and virtual mass affecting the hydrodynamic response. Insights are provided with regard to the calibration of the finite element model which was achievable through utilizing the more empirical analyses and the extent to which the finite element analysis is able to simulate the entire collision event.

INTRODUCTION

As part of an International Atomic Energy Agency (IAEA) Coordinated Research Project (CRP), Sandia National Laboratories is investigating the safety of shipments of radioactive material by ocean-going vessels [1]. The project is concerned with the potential effects of ship collisions and fires to on-board Radioactive Material (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 performance prediction methods checked by comparison with historical data, 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.

Several methods of evaluating structure during a ship-to-ship collision were investigated. The methods used include finite element analysis, the Tanker Structural Analysis for Minor Collisions (TSAMC) Method, the Minorsky Method, and the Haywood Collision Process. The results of the finite element analyses are also compared to results from full-scale tests.

FINITE ELEMENT ANALYSES

A series of analyses was performed to evaluate structure for impacts of varying severity. All of the analyses used the same ship as the vessel carrying the RAM package, a 67.4 meter long1675 Long Ton displacement hull. This vessel was struck amidships, very near to the location of a transverse bulkhead. The striking vessel was identical to the RAM carrying ship in size and mass, but the energy of the impact was increased by increasing both the mass and the velocity of the striking ship. The increase in energy was incorporated to provide results with a large

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range of damage using only one finite element model. It is realized that a vessel of this type does not have the correct geometry or size to have the mass and velocity considered in the higher energy collisions.

Finite Element 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, analysis of previously published analyses that used a loosely coupled approach [2], showed 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. Therefore, in these analyses the water is not explicitly modeled, but instead is treated as added mass to the struck ship. Following the method of Minorsky [3], added mass equal to 40% of the mass of the struck ship is used.

A top-view of the finite element model developed and element types used is shown in Figure 1. Figure 2 shows an exploded view of the center potion of the struck ship. As can be seen from the figure, the hull stiffeners and framing were not explicitly modeled, but the stiffness of these elements was smeared into the shell stiffness properties. This was done to reduce the complexity of the finite element model. The rigid elements shown in Figure 1 were used to simulate areas that were thought to not have much, if any, damage. This type of element is very computationally efficient, and its use allows for faster analysis times while maintaining correct mass distribution. The bow of the striking ship is modeled with very stiff elastic solid elements, or blocks. This forces all of the energy dissipation to occur in the struck ship, which would be the worst case for affecting the RAM package, as it will create the most damage in the struck ship. The shell elements used in the deformable part of the struck ship were modeled using a power-law hardening model, where the stress is equal to the yield stress plus a hardening coefficient times the hardening strain raised to a power. The material was assumed to be mild steel, with a static yield stress of 248 Mpa. For the strain rates expected in this analysis, mild steel has a higher yield strength than it does for static loads. Based on the expected strain rates, the yield stress used in the analysis was 372 Mpa. Mild steel has a true strain at failure of about 80%. Any element in the finite element analysis that achieves tensile strain greater than this value is deleted. In this manner tearing can be simulated, although the element size is too large to accurately capture this phenomena.

RAM transportation packages are designed to survive an impact at 13.4 m/s onto a flat essentially rigid target. This impact typically produces accelerations in the package of 40-200 Gs. Typical ship collisions have accelerations much lower than this, so the possibility of the package being damaged by the impact of the bow is minimal. The only credible way for the package to be

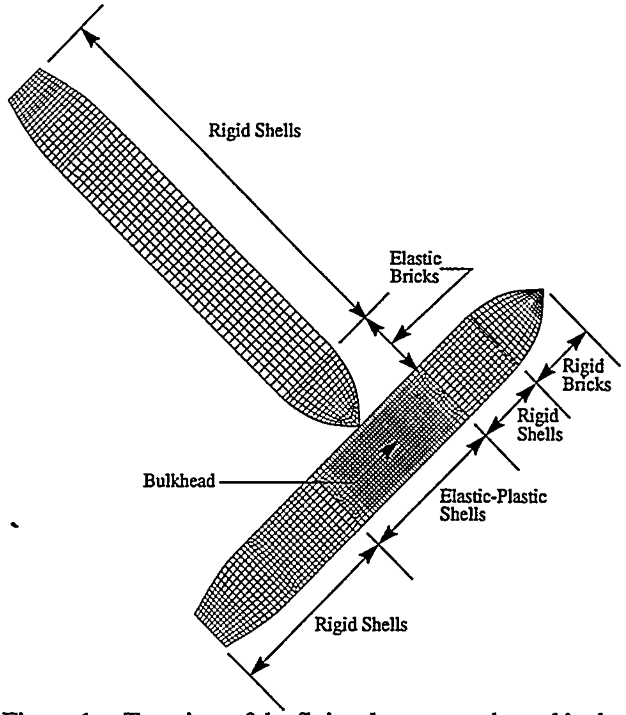


Figure 1: Top view of the finite element mesh used in the analyses.

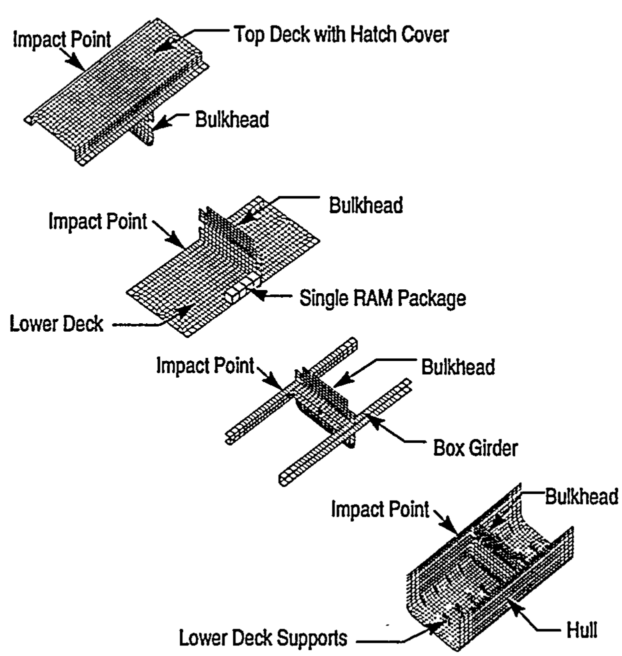


Figure 2: Detail of the finite element mesh in the center portion of the struck ship.

damaged is by the crush force exerted on it when it is pinned between the bow of the striking ship and the hull of the struck ship on the side opposite from the impact. (For some ships it would be possible for the package to be pinned against interior support structures as well, but the ship modeled did not have any of this type of structure.) The magnitude of this crush force is limited by the stiffness of the struck ship. For this reason, whenever assumptions were required there was an attempt to increase the strength of the struck ship and thereby maximize the crush force that could be exerted onto the simulated RAM transportation package. For example, the hatch covers are assumed to be rigidly attached to the top deck. While it is possible for these covers to act independently from the ship, it is believed they will likely participate in the collision process and damage. It is realized this approach will tend to minimize penetration distance.

Ten analyses were performed. Table 1 shows the variable parameters used in each analysis. The cases with an S designator have a simulation of a single RAM package in the hold and the cases with an M designator have a simulation of a series of 7 RAM packages in the hold. In all cases the RAM packages are free to move within the hold. There are no tie-downs, no gravity, and no friction acting on the packages. The M series of analyses were run after the S series because in the S series there was not sufficient penetration to subject the simulated RAM package to any crush loads. In the M series, the simulated RAM packages span 80% of the breadth of the cargo hold, and are subjected to crush loadings when the penetration distance is greater than about 2 meters. In Cases 1S and 1M the two ships are of identical mass, and the striking ship is travelling at 5.14 m/s (10 knots). Cases 2S to 4S and 2M to 4M increase the mass and velocity of the impacting ship. Cases 4S' and 4M' have the same collision properties as Cases 4S and 4M, but the tensile strain at which elements are deleted from the model during the collision process has been reduced to 20%. This case was run after the others, and was carried out because there was very little tearing noted in the collisions with 80% true strain at failure. The lower value is considered plausible because the model does not include any details, such as joints between the hull shell and its framing, that would act as locations of stress concentration and initiate tearing and because the coarseness of the finite element mesh reduces peak strains. When the tensile plastic strain in the composite shell reaches a level of 20%, it is highly probable that locations of strain concentrations will have strains high enough to initiate and propagate tearing.

Table 1: Analyses Performed

Case Designation	Striking Velocity, m/s (knots)	Mass of Striking Ship, tonnes	Tensile Failure Strain
1S, 1M	5.14 (10)	1675	80%
2S, 2M	7.73 (15)	10,050	80%
3S, 3M	12.9 (25)	16,750	80%
4S, 4M	15.6 (30)	16,750	80%
4S', 4M'	15.6 (30)	16,750	20%

Finite Element Analyses Results

A summary of the results for the S series of finite element analyses is given in Table 2. The duration in impact is the time until the total kinetic energy drops to a constant value. The penetration distance is measured from the original position of the hull, so it includes the distance the bow moves through the decks as well as any beam bending of the hull. The final velocity is the velocity of both ships at the time of maximum penetration. Recall that the model does not include any hydrodynamic forces to slow the lateral motion of the struck ship. Figure 3 shows the deformations from Case 1S. There is only a slight amount of penetration and no tearing of the hull. Figure 4 shows the deformations for Case 4S. In this analysis the penetration distance is approximately half the breadth of the ship, however, no elements reached tensile plastic true strains greater than 80%, so no tearing is indicated. It should be noted that modeling of material failure is extremely mesh dependant, and a finer mesh would indicate higher peak strains and possibly some tearing. To more accurately predict the level of damage with the coarse mesh used in this study Case 4S' with a lower failure strain was performed. The deformations for this case are shown in Figure 5, and the amount of tearing is shown in Figure 6.

Table 2: Results for Analyses with Single RAM Package

Case	Duration, sec	ΔKE, MJ	Penetration, m	Final Velocity, m/s (knots)
1S	0.27	11	0.8	2.0 (3.9)
2S	0.50	95	2.2	5.8 (11.3)
3S	0.66	381	4.2	10.1 (19.6)
4S	0.68	493	5.2	12.5 (24.3)
4S'	0.45	338	6.0	13.0 (25.3)

In the series of analyses with multiple simulated RAM packages, Case 1M still did not result in impact to the packages, but the other four cases did. To investigate how the crush force on the packages is limited by the stiffness of the ship a comparison between the crush forces generated in Case 2M and Case 4M is shown in Figure 7.

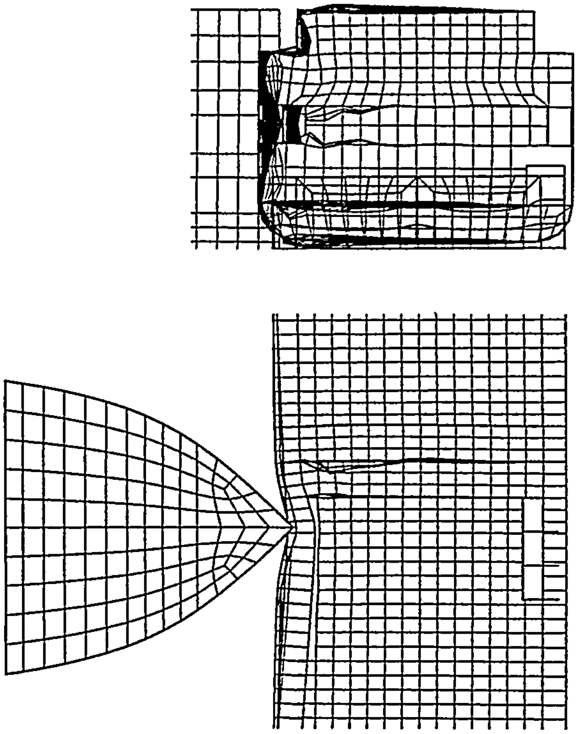


Figure 3: Deformations from Case 1S.

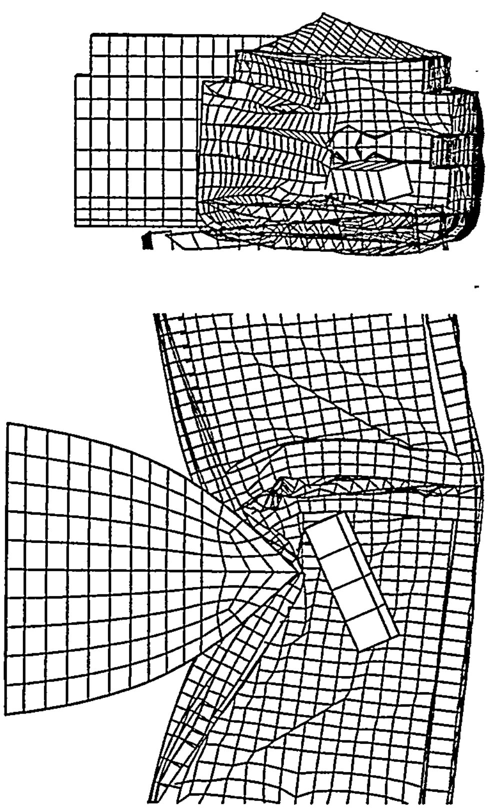


Figure 4: Deformations from Case 4S.

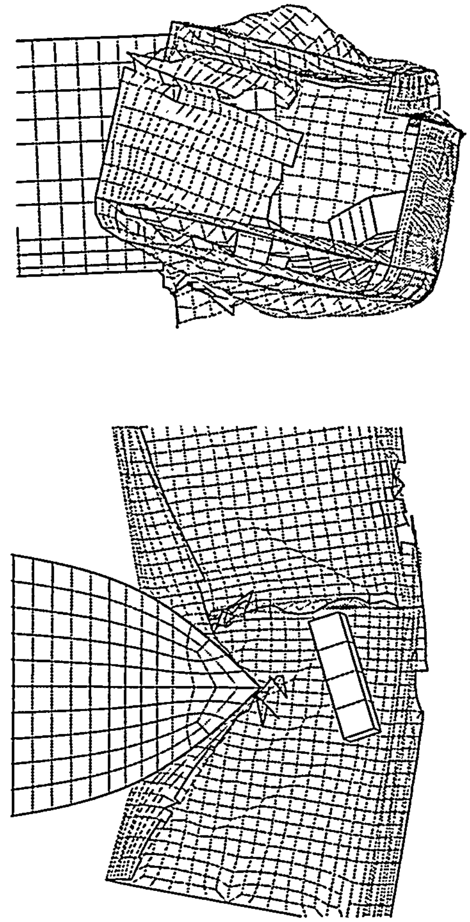


Figure 5: Deformations from Case 4S'.

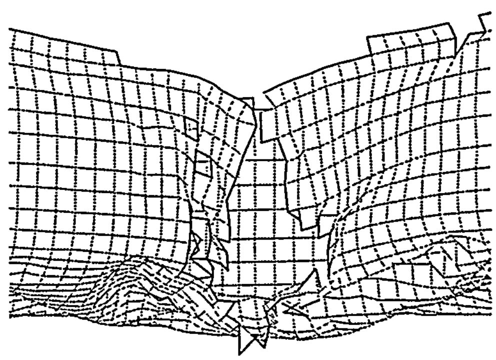
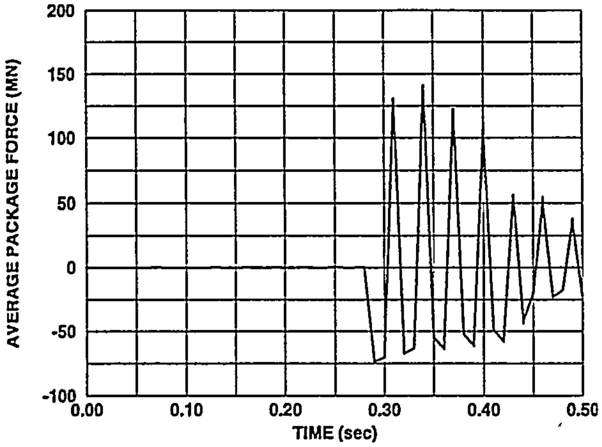
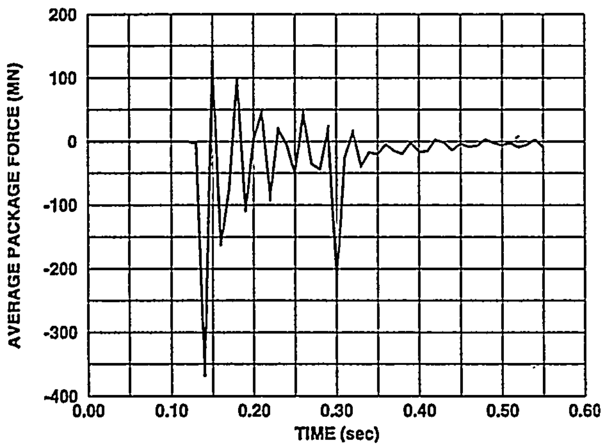


Figure 6: Tearing of the hull from Case 4S'.

The first downward spike in the two plots is the dynamic impact force when the bow strikes the packages. The upward spike following this is the force generated when the packages strike the side of the hull away from the strike (the starboard side of the ship). In Case 4M the impact force is much higher than in Case 2M, but the crush force generated by the starboard side is equal (about 140 MN) for the two cases. Figure 8 shows the deformations for Case 4M. Note how much the starboard side is pushed outward by the RAM packages. In Case 4M' there was sufficient tearing of the starboard side so that the packages were pushed through it and out of the ship.



Case 2M



Case 4M

Figure 7: Forces acting on the simulated series of 7 RAM packages.

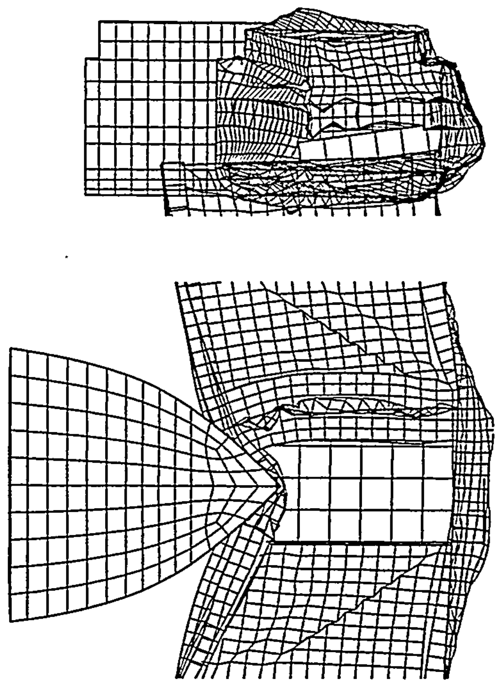


Figure 8: Deformations from Case 4M.

TSAMC METHOD

It has been determined that 70 to 90% of the energy absorbed by a ship's structure during a low energy collision results from plastic deformation of the side shell [4]. As the side shell begins to deform, large membrane stresses are built up in the plate. The Tanker Structural Analysis for Minor Collisions (TSAMC) method examines the energy required to penetrate (tear) the side shell and from this, the depth to which penetration occurs. This calculation stops when penetration occurs, however, additional penetration will still occur if more energy must be absorbed [4].

In addition to the assumptions made in the derivation of the TSAMC method, several other assumptions were made for this analysis:

1. Plastic bending energy (Ebc) of the shell plate as a longitudinal beam was neglected as it will be small in comparison to the total.
2. Transverse frames were lumped so that their spacing became 4 feet instead of 2 feet.
3. The "beam" in the longitudinal direction for the membrane tension analysis was the shell plate only between decks and transverse frames.
4. Transverse frame stiffeners are analogous to web frames in longitudinally stiffened shell plating.
5. The hatch cover at 7.19m abl was treated as a deck, with the hatch cover plating assumed to be 8mm thick.
6. The collision was assumed to occur midspan between the lumped frames.
7. The transverse bulkhead between holds was assumed to remain intact as evidenced by the finite element results.

A summary of the plastic energy calculations to rupture of the sideshell is given in Table 3.

The TSAMC analysis of the ship yields a penetration depth of 3.58 m before sideshell rupture, with a corresponding energy absorption of 49.4 MJ under the assumption that the transverse bulkhead at Frame 58B is replaced by a transverse frame. This was incorporated as the TSAMC does not address bulkhead deformation explicitly. The results, which alternately represent a collision at midbody, should therefore be considered an upper bound to penetration and will be deficient in energy

Table 3: Summary of Plastic Energy Calculations up to Rupture of Sideshell

Elements of Plastic Energy	Plastic Energy	
	in-kips	MJ
Membrane Tension in Transversely Stiffened Shell Plate (Emt/SP)	150,684	17.0
Membrane Tension Plastic Energy for Longitudinal Box Girder and Main Deck (Emt/BG-MD)	181,609	20.5
Membrane Tension Plastic Energy for Inner Bottom, Shell Bottom, and Deck with Hatch Cover (Ed) 7.19 M ABL	105,136	11.9
TOTAL PLASTIC ENERGY ABSORBED	437,429	49.4

absorption to the extent of that absorbed by the distortion of the bulkhead.

After rupture, the striking vessel will continue to penetrate the struck vessel and additional energy will be absorbed by other structure. The depth and energy stated in Table 3 represent only this transition point where the shell tears, and not necessarily the end of the penetration and energy absorption. In Case 1S, the energy of the striking vessel is less than the 49.4 MJ and it can be expected that the side shell would remain intact and the penetration would be less than that predicted by the TSAMC methodology. However, Case 2S expends nearly double the energy required to rupture the side shell but does not achieve the penetration predicted for side shell rupture.

MINORSKY METHOD

The Minorsky Method was developed by V.U. Minorsky in the late 1950's to asses the vulnerability of the reactor compartment of the N.S. SAVANNAH. It is a semi-analytical method that relates the energy of a striking ship to the volume of steel damaged in the striking ship and the struck ship, based on a linear regression of known ship accidents. This method ignores the sideshell plate in the struck ship in determining the penetration resistance factor, R_t; however, the plating is taken into account to some degree by the method's statistical nature [3].

Resistance factor and absorbed energies were calculated for the finite element predicted penetrations in accordance with the Minorsky procedure and are listed in Table 4. The resistance factor is based on the volume of structure deformed in the finite element penetration. Since the finite element model as well as the TSAMC method both use an infinitely stiff bow on the striking vessel, the resistance factor for the striking vessel was assumed to be zero; therefore the striking vessel's structure absorbs no energy. Damage extents for the calculation of resistance factors for the stricken vessel were based on the finite element penetrations and the plan of the striking bow. Also, the collision was initially assumed to occur between bulkheads with no deformation of the transverse bulkhead. This led to the columns labeled without bulkhead in the table. To investigate the amount of energy absorbed by deformation of the bulkhead, the volume of bulkhead material damaged was included in the calculation of the resistance factor. These calculations led to the columns labeled with bulkhead in the table.

Table 4: Energy Absorbed in S Series Collisions

Case	FEM Pen.	R_t w/o bulk.	R_t w/ bulk.	Minorsky Energy w/o bulkhead	Minorsky Energy w/bulkhead	FEM Energy
	m	ft^2-in	ft^2-in	MJ	MJ	MJ
1S	0.8	6.73	32.7	34	37	11
2S	2.2	33.2	128	37	47	95
3S	4.2	122	347	46	72	381
4S	5.2	190	494	54	88	493
4S'	6.0	256	629	61	103	338

The finite element analyses indicate much more energy absorption for the same resistance factor than predicted by the Minorsky method. This is true for both the calculations without taking into account the energy dissipated by deformation of the bulkhead and the calculations including this energy. When the failure strain for the finite element analyses was lowered (Case 4S'), the finite element method results are closer to those predicted by Minorsky's method, but still exhibit excess energy absorption. Assuming Minorsky's regression line is typical of the relationship between the resistance factor and the absorbed energy, the finite element analyses over predict energy absorption.

It is to be noted, however, that the Minorsky line is based on vessels with much larger resistance factors RT than in the current case. For these smaller resistance factors significant scatter of "low energy points" is noted in Minorsky's original paper [3]. It may be that the appropriateness of the line for the case in question is limited. Also note that the energy absorption predicted by the TSAMC is on the same order as the Minorsky prediction.

HAYWOOD COLLISION PROCESS

Based on a mathematical analysis of the energies absorbed in inelastic collisions of ships, F.H. Haywood [5] predicts that 70 percent of the striking ship energy will be absorbed in the structural deformation when identical vessels collide at 90 degrees. Based on this assertion, Case 1S should have absorbed 15.5 MJ in comparison to the 11 MJ predicted by the finite element analysis.

FULL SCALE COLLISION TESTS

Full scale collision test studies were conducted in 1991 [6] involving nearly identical vessels having displacements of about 60% of the Case 1S ships and similar structures colliding at about 9 knots. The tests yielded a penetration approximately 25% larger than those predicted by the finite element method. The results from the 1991 full scale tests were later duplicated in a finite element analysis that was much more detailed than the one for this study[2]. The goal of the analysis in [2] was to match the test results for damage to structure, while the goal of the finite element analysis report here was to determine an upper bound to the crush loads imparted to the stowed RAM packages.

CONCLUSIONS

The finite element analysis absorbs more energy for the same penetration when compared to the TSAMC and Minorsky. The possible reasons for these differences include:

(1) Smear of transverse stiffeners in the FEM making the sideshell effectively thicker,

(2) Deformation of the bulkhead structure in the FEM, and

(3) The large element size in the finite element analysis restricts the amount of tearing that can take place.

To a certain extent, these results are as desired in the study of the possible crush loadings on RAM packages, as underprediction of penetration distance for a given amount of imparted energy indicates the struck ship is stronger than reality. A stronger ship is able to impart larger crush loads onto the RAM package than a weaker one.

Another possible source of error is the manner in which the calculations treat the hydrodynamic forces acting during the collision. In reality the vessels will have reduced rigid body motion due to hydrodynamic sway resistance. This hydrodynamic sway resistance is likely to result in additional structural deformation and penetration during the deceleration process as the struck vessel continues to interact with the striking vessel. In the finite element analyses the struck vessel actually separates from the striking vessel at later times in the analysis. While the virtual mass assumption used in the finite element analysis has been used in other analyses of ship collisions, the 40% value for added mass may be low in comparison with some investigations. While some reports state a constant 40% in the struck vessel [3], others state up to 150% of the displacement for added mass [7]. The quantity of virtual water mass is also time dependent and not a constant [7].

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