11th International Symposium on Practical Design of Ships and Other Floating Structures
Rio de Janeiro, RJ, Brazil

Investigation of an Actual Collision Incident between a Tanker and a Bulk Carrier

AbuBakar, A.[1], Dow, R.[1], Tigkas, I.G.[2], Samuelides, M.S.[2], Spyrou, K.J.[2],

[1] School of Marine Science and Technology, Newcastle University, England, UK

[2] School of Naval Architecture and Marine Engineering, National Technical University of Athens, Greece.

Introduction

In 2001 a 34,365 dwt bulk carrier collided with a 37,000dwt tanker carrying 33.000 Tons of Heavy Fuel Oil in the Baltic. During the incident the bow of the bulk carrier penetrated largely a ballast and a cargo tank whereas the bow of the bulk carrier suffered considerable damage. The damage was described in (THE OFFICE OF THE MARITIME ADMINISTRATOR, 2002) as follows:

"The bow of the TERN penetrated approximately 5 meters into the double hull of the BALTIC CARRIER on the starboard side and holed the side shell plating between frames 43 and 60. The starboard #5 wing ballast tank and the #6 starboard cargo tank on the BALTIC CARRIER were opened vertically from the main deck to a point well below the waterline. The double bottom tanks located below the damaged ballast and cargo tanks remained intact. Damage to the TERN involved the bulwark, stem, and bow plating on both sides of the hull in way of the forepeak tank and deck storeroom, and included the collision bulkhead. The shank of the port anchor on the TERN was broken and the flukes were missing as a result of the impact from the collision"

The present work attempts to simulate a collision of two ships having the same particular dimensions as the ships involve in the incident and following the same collision course. Data needed for the analysis such as the relative positions of the vessels prior to the incident and the steering of the vessels that resulted the collision were extracted as far as it was possible from (THE OFFICE OF THE MARITIME ADMINISTRATOR, 2002) The ultimate goal is on one hand to test available tools for collision simulation including tools for the simulation of the collision course and the actual collision, and on the other to investigate the sensitivity of the consequences to actions taken a few minutes before the incident by the ship masters. It is to be understood that since not all data needed for the simulation were available, the authors had to make assumptions that were mostly related to the structure of the collided ships.

Simulation Methodoligy

The simulation consists of two stages:

The first stage aimed to reproduce the geometry of the impact using DYNASIM a real time ship simulator. The code is designed to simulate multi-ship dynamics in restricted water, in the presence of waves, wind, currents and obstacles. General topography, channel configuration, and environmental conditions can be input by the user. The simulator can be used to generate multiple ship tracks to evaluate harbor safety from a system perspective. It can also be used for continual training of masters and pilots to minimize human error, and by harbor designers and port managers for structures and waterway design, modification, improvement and implementation of safety measures.

The second stage simulates the structural response of the two ships. The simulation uses the Finite Element ABAQUS explicit code to investigate the effect of the collision between the bow of the striking ship and the full beam of the struck ship. Simulations are carried out to investigate the effect of both a lateral collision and also to simulate the actual collision conditions. For the striking ship alternative models for the bow shape are investigated. The progressive failure of the side shell is investigated considering both, the effect of damage due to plastic deformation during collision, and damage evolution including material rupture. The results presented include the crushing force as a function of time; the energies involved in plastic deformation and a comparison between the predicted resultant damage levels and the actual damage caused during the incident.

Manoeuvring Prior To Collision

Accident history

A brief history of the accident is outlined in order to understand the situation and realise the kind of manoeuvres that the two ships performed prior the accident. In this way, the manoeuvres of both ships that need to be simulated are described.

Accordingly, when firstly addressing the tanker ship, it can be said that she was steaming with 13knots at a manoeuvring mode (105 rpm) and at 23:33 hrs, she changed her course from 244° to 237°. Meanwhile, the bulk carrier was maintaining a 057° course with a speed of 10.5 knots and she was planning a port-to-port pass. Likewise, the tanker also was also planning a port-to-port pass and therefore at 24:00hrs, she altered her course to 236°. Due to the presence of a relatively strong wind (15m/s SE), she seemed unable to keep her course steady and at 00:12hrs she started drifting to port. When the distance between the two ships was 1.2 miles, the Master of the tanker ship order at "hard starboard rudder", however, the rudder remained amidships due to a steering gear failure and the ship continued turning to port. At 00:13hrs, the second steering system was switch on, but the ship's course was already about 205° and the distance separating the two ships about 0.75 miles. Consequently, the Master decided that they had no time to turn starboard, so he ordered "hard" to port in order to pass ahead of the bulk carrier. The bulk carrier attempted likewise to avoid the collision and her Master ordered "hard" to port. Unfortunately, the distance between the two ships had already been critically reduced to 0.3 miles. Therefore, before actually turning, at 00:15hrs, the bow of the bulk carrier breached the starboard side of the tanker with an angle of attack of about 50° at the frames 40-68.

Manoeuvring simulation

In this study at a first stage, it is attempted to simulate the manoeuvring behaviour of both ships that lead to the investigated collision accident. The reason of attempting to reproduce the ships' motion responses is twofold; it is firstly of prime importance to accurately define both the relative speed of the collision and its associated incident angle and secondly it is interesting to notice if and how a quicker action of the rudder would have averted this unfavourable incident.

Computing the velocity and the angle of incident that the two vessels had during the impact is essential, in order to proceed to the second stage of this effort where the structural response of the two ships is assessed. Accordingly, for reproducing the manoeuvres of the two ships, we use the commercial software DYNASIM (Chahine et al, 2006) that employs a nonlinear mathematical model in 4-DOF. This software has the capability to simulate multi-ship dynamics in restricted water, in the presence of waves, wind, currents, and obstacles. It is supported by the vendor that the simulator can be used for evaluating harbour safety from a system perspective, for designing waterways and for continual training of masters and pilots in order to minimise the human error.

The equations of motion for surge, sway, roll and yaw can accordingly be written as shown below (Cheng et al, 2002):

$$\begin{aligned} &m\left(\dot{u} - rv - x_g r^2\right) = X \\ &m\left(\dot{v} + ru + x_g \dot{r}\right) = Y \\ &I_x \dot{p} = K \\ &I_z \dot{r} + m x_g \left(\dot{v} + ru\right) = N \end{aligned} \tag{1}$$

Where u, v are the surge and sway velocities in a body axis system, r, p are the yaw and roll angular velocities once more in a body axis system, m is the ship mass, x_g is the longitudinal distance from amidships of ship's centre of gravity, I_x, are the roll and yaw moments of inertia and X, Y, K and N are the external forces and moments acting upon the ship in surge, sway, roll and yaw respectively.

When these loads are expressed in a modular form, they can be separated in hydrodynamic, rudder, propeller, wind, waves and current forces and moments. The hydrodynamic loads for both ships in the investigated scenario, were calculated separately intrinsically in the software by a standard *Taylor* approximation procedure utilising semi-empirical still water manoeuvring derivatives (Kijima, 1990). The telegraph settings on the other hand of both ships were used to match to each revolution rate of the propeller, the corresponding attained ship speed in calm water. Detailed characteristics of the propeller and the rudder were inserted in order to enable the program's capability to compute the corresponding associated loads. Since also during the collision it was reported that the sea was relatively calm, the sea current velocity was insignificant and only wind velocity was relatively strong (approx 15m/s), the effects of waves and currents were neglected. Nevertheless, the influence of wind was noteworthy upon the manoeuvring behaviour of the ships, especially for the tanker that was reporting problems to maintain her course. It was thus taken into account by using Martin's (1980) and Lamb's (1932) simple but practical methodologies, to model wind loads and thenceforth accurately reflect the experienced environmental conditions that partly contributed to this unfavourable incident. More details regarding advanced techniques for assessing the directional stability of ships in strong wind, can be found by in Spyrou, Tigkas and Chatzis (2007).

Once the ships' characteristics were entered, then the ship's initial conditions such as their course and speed and the realised environmental conditions such as the wind direction and velocity were set appropriately in order to run a simulation. At appropriate time intervals according to the realised scenario, the rudder settings were altered and the tracks of both ships were recorded. From these recorded tracks, it can be observed that the actual collision was successfully simulated. By monitoring in addition both speeds of the ships, it can be calculated that the impact speed or else the relative speed of collision was 8.4m/s and the angle of incident of those two ships approximately 50°.

Sensitivity analysis

Additional scenarios were also examined in order to investigate whether a quicker "hard" to port rudder action by either of the two vessels, would have been sufficient to avert their collision. These scenarios comprised both a 1 and 2 minutes earlier rudder deflection of the two ships, keeping all the other parameters of the original scenario unaltered. The tracks of both ships for the realised scenario and for the hypothetical scenarios are all consolidated in Fig 1. From this figure, it becomes evident that the accident would have been evaded if at least one of the two ships was able to perform a "hard" to port manoeuvre at least one minute earlier.

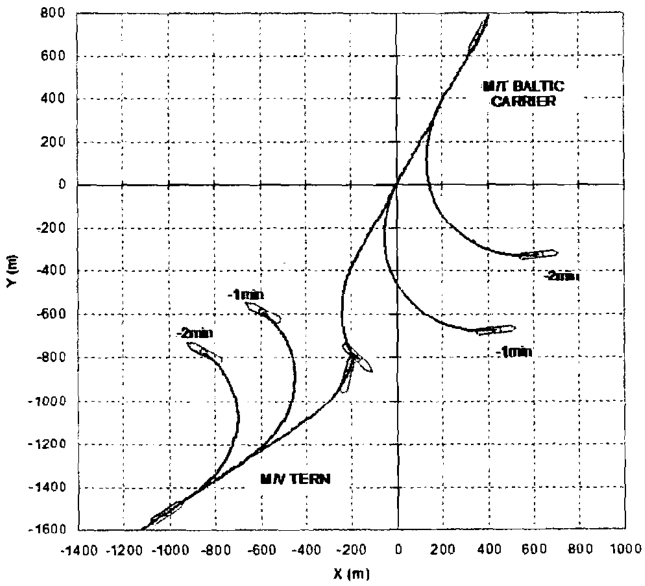

Fig. 1 Diagram indicating the collision scenario and alternative scenarios that would have been occurred if the rudder action of each vessel had taken place 1min or 2min earlier.

Simulation Of Structural Response

Finite element simulations

The use of finite element codes for the simulation of collisions has been a challenge since the early 80ies. Almost thirty years ago Chang et al (1980) and Valsgård & Jorgensen (1983) published work including the use of FE codes for the assessment of the collision behaviour of ships. At that time, even if computers were slower, pre- and post- processors did not have any similarities with the respective tools that exist today and CPU and storage much more expensive than today, researchers identified the advantages of finite element technique over analytical methods: FE codes may simulate the interaction of structural elements, users do not need to anticipate the collapse mode of the individual elements, complicated geometries and contact between structural elements may be modelled, material models that simulate more realistically the actual material behaviour may be used, and large strain and large displacement formulations are possible. With the sharp increase of computing capacities, the reduction of CPU and storage cost, and the continuous development of the finite element techniques, FE codes, and in particular codes that employ the explicit integration scheme, have become the necessary element for the assessment of the behaviour of any ships in a collision incident. Further finite element analysis is used today to validate simplified methods, which are preferred when large number of collision simulations is needed, for example in the case of a risk based collision investigation (Hutchinson et al 1986; Otto et al 2002).

However the use of FE codes for the simulation of impacts between structures with complicated geometries, which result in large strains and rupture, remains a challenge and there are not yet any established and widely accepted guidelines for the representation of the phenomenon. Factors that affect the quality of the results that are produced are:

- *Knowledge, experience and skills of the user:* By choosing the type of elements, the material model and the mesh size, the user has a control on the structural modes of response and consequently on the results. Further there are cases where the results do not show any clear convergence by reducing the mesh.
- *Lack of basic knowledge:* Although there has been much experimental and numerical research on initiation and propagation of rupture there is still a lack of knowledge in this area and to a realistic representation of the phenomenon within a collision simulation.
- *Inadequate verification of FE simulations:* The means to verify the simulations of collision incidents by finite element are incomplete. During the last decades, considerable number of measurements of impacts on structural components has been performed, but there is a lack of data of actual collisions which could be used for a comprehensive verification. In order to fill this gap a number of large scale collision and grounding tests were performed in the nineties (Carlebur 1995; Peschmann 2001), which produced results that were compared with results obtained by finite element.

+The important aspects of collision simulations with finite elements are: the rupture criterion; the mesh size and the global response of the hull. It is noted that the first two aspects are closely related, i.e. a realistic modelling of rupture depends on the mesh size. Further aspects of finite element modelling in general are the boundary conditions, if ship motions are not included in the analysis, and ship motions in the sea and the modelling of the striking bow.

Material Properties

The Materials used in this analysis are mild steel (S235JR-EN10025) and high strength steel (S355NH-EN10210) the material properties are describe in Table 1.

Table 1: The properties of steel taken from (Alsos and Amdhal 2009)

Material types	K (MPa)	n	ε_{plat}	ε_f	σ_y (MPa)
S235-A	740	0.24	-	0.35	285
S235-B	760	0.225	0.015	0.35	340
S355-C	830	0.18	0.01	0.28	390

The material is assumed to be isotropic and to exhibit strain hardening properties in accordance with the true stress-strain relationship approximated by the equation below, where K and n are material parameters proposed by (Alsos, Amdahl et al. 2009).

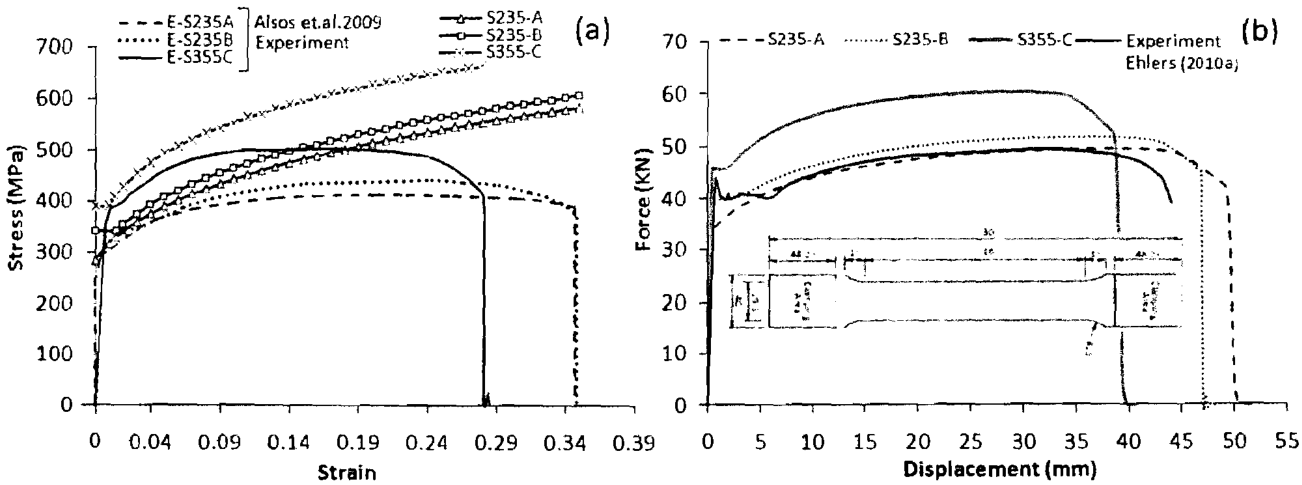

Fig.2 : (a) The stress-strain curve. (b) The tensile test force-displacement curve.

$$\sigma = \begin{cases} \sigma_y & if\ \varepsilon \leq \varepsilon_{plat} \\ K(\varepsilon + \varepsilon_0)^n\ \dot{\varepsilon}^m & otherwise \end{cases}$$

and

$$\varepsilon_0 = \left(\frac{\sigma_y}{K}\right)^{1/n} - \varepsilon_{plat}$$

Where ε_{plat} is the plateau strain.

These equations apply only until the onset of necking. A comparison of the engineering stress-strain curves (E-S235A, E-S235B, E-S355) and true stress-strain curves (S235A, S235B, S355) are shown in Fig.2a. In the true stress–true strain curve also known as a flow curve the curve increases continuously up to fracture. Mild steel and high tensile steel tensile test results for force-displacement using dog-bone specimens as described in (Ehlers, 2010a) are shown in Figure 2b. Theoretical comparisons were carried out using a 4.4mm mesh size and FLD material failure model as discussed in section 4.3. The results give good correlation between (Ehlers, 2010a) and S235JR-EN10025

Material Failure

The material failure model used is based on forming limit diagram (FLD) method which is a concept introduced by (Keeler and Backofen 1964) to determine the amount of deformation that a material can withstand prior to the onset of necking instability.

The maximum strains that sheet material can sustain prior to the onset of necking are referred to as the forming limit strains as described in the ABAQUS documentation.

Considering the forming limit strains as rate independent effects in the FLD method, details of which can be found in Jie, Cheng et al. (2009) the following relationships are used:

$$\varepsilon_1 = \begin{cases} \dfrac{n}{(1 + r_\varepsilon)} & if\ r_\varepsilon \leq 0 \\ \dfrac{3r_\varepsilon^2 + (2 + r_\varepsilon)^2 n}{2(2 + r_\varepsilon)(1 + r_\varepsilon + r_\varepsilon^2)} & if\ r_\varepsilon > 0 \end{cases}$$

where: $r_\varepsilon = \frac{\varepsilon_2}{\varepsilon_1}$ is strain ratio, $r_\varepsilon = 0$ for plain strain, $r_\varepsilon = -0.5$ for simple tension and $r_\varepsilon = 1$ biaxial tension which is the basis for localized necking failure.

Mesh convergence studies were carried out, for a range of different mesh sizes, aligned with element characteristic length as discusses in ABAQUS documentation. For all of the simulations carried out the friction coefficient was set at 0.3 and the displacement at failure considered to be $\varepsilon_u L$. Where ε_u is ultimate strain, approximately $0.5\varepsilon_f$; ε_f is fracture strain and L is characteristic element length. In the post necking regime the element characteristic size has a significant influence on the accuracy of the

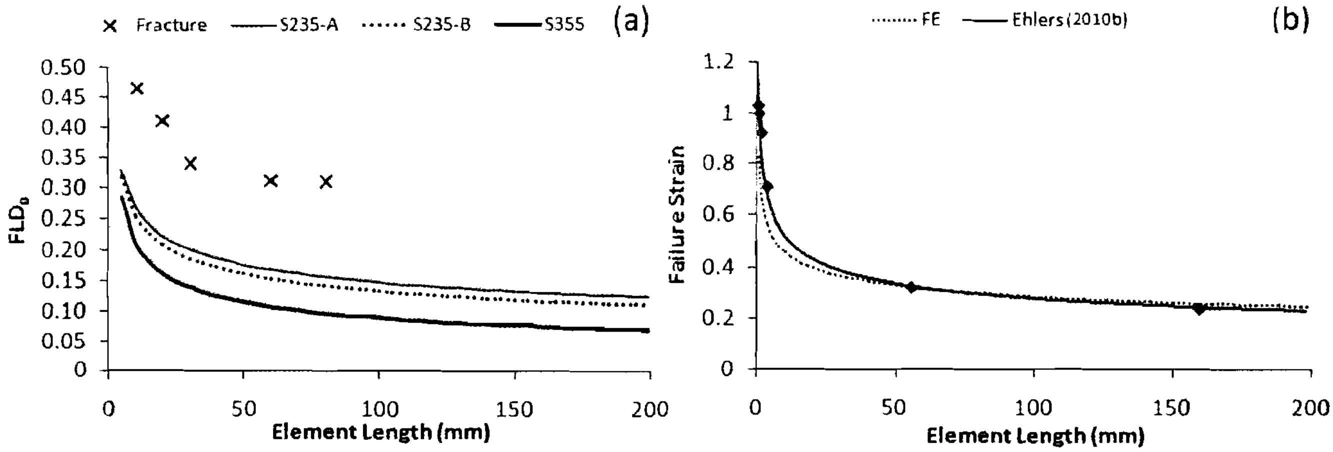

Fig.3 : (a) The scaling forming limit Diagram at onset necking versus Element Length, (b) The failure strain versus Element length

results. For shell and 2D elements, L is square root of the integration area and for 3D elements, L is the cube root of the integration of volume.

The results obtained were plotted as element length against FLD_0, where FLD_0 is the local necking point which intercepts at y-axis when r=1. The results are shown in Fig. 3a and b, where Fig. 3a is FLD_0 curve and Fig. 3b shows the comparison of the failure strain trend line with (Ehlers, 2010b) which gives a good degree of correlation.

Structure geometry

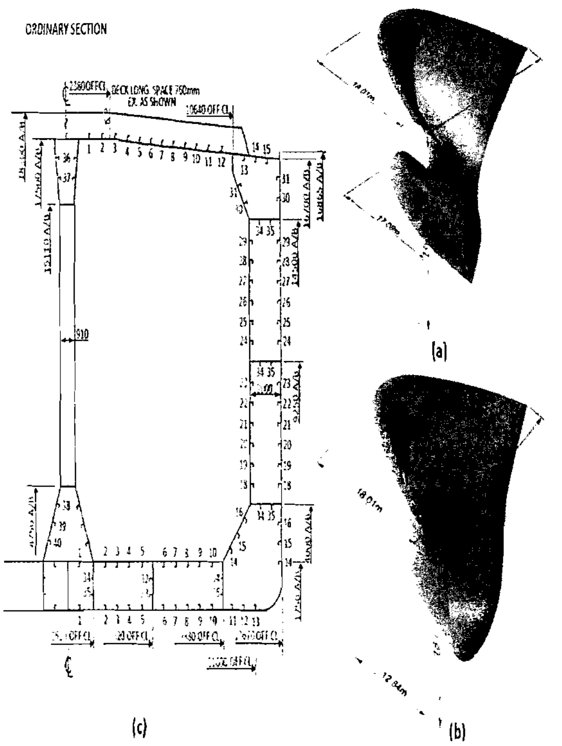

Fig. 4 : (a) Bulbous bow (b) Normal bow and (c) Mid-ship section details (not in scale)

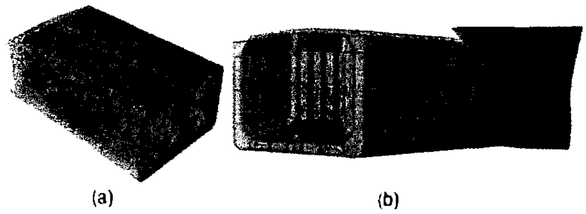

Fig.5 : The boundary condition (red marks)

The geometry of the struck ship (double hulled tanker) and the geometry of the two alternative bow shapes used in the analysis are shown in figure 4 (a), (b) and (c) respectively.

Simulation of structural response

The structural model chosen to represent the struck ship was that of a complete compartment plus half a compartment either side to remove the influence of boundary condition effects from the area of interest.

Mesh sizes of 60mm and 80mm were chosen for the simulations carried out. The struck ship was assumed to be at rest with the striking bow having a relative speed of 10m/s and assumed to decelerate to an absolute stop when maximum penetration was achieved. The actual collision between the two vessels was reported to have occurred at an angle of approximately 50 degrees, for this study collision angles of 50 and 90 degrees were investigated. The analysis utilised a structured quadrilateral dominated mesh for both fine mesh and coarse mesh regions and an unstructured mesh for the transition region.

The boundary conditions on the FE model were set as ENCASTRE (fully fixed) for both ends of compartment see Fig.5a and b. The impact point was set at a main transverse frame in the centre of the compartment for both collision scenarios considered using a rigid body representation of the bow.

Simulation Results

The lateral penetration and resultant force of rigid normal and bulbous bows, obtained from the FE simulations, are shown in figures 6(a) and (b) respectively. In figure 6a, the force on the bulbous bow BX, BY and BZ for 60mm and 80mm mesh size show good agreement which the scaling of FLD_0, this gives a level of confidence that the theoretical modelling of failure strain and characteristic element length can be successfully applied to a large structure.

For the normal bow (NR-80), figure 6 shows the vertical stem start to penetrate the outer shell at point I (3.67m, 59MN); at point II (4.52m, 106MN) it shows the outer shell onset of rupture.

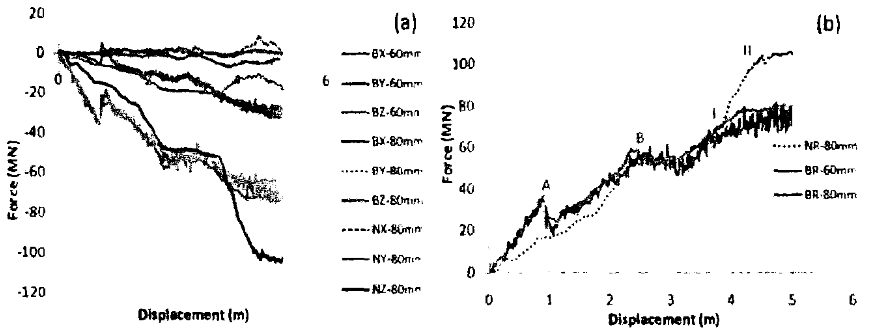

Fig.6 : (a) Force - displacement of lateral penetration (b) resultant force - displacement of lateral penetration

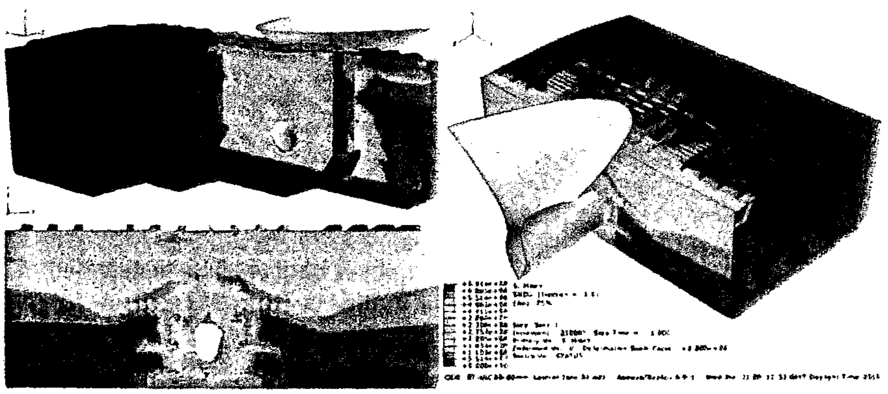

Fig.7 : The lateral penetration of rigid body of bulbous bow penetrated to double side shell of Baltic tanker.

For bulbous bow (BR-60mm and BR-80mm) the different mesh refinements give close results, where outer shell rupture is predicted at point A (0.97m, 25.88MN) and inner shell rupture at point B (2.7m, 56.12 MN).

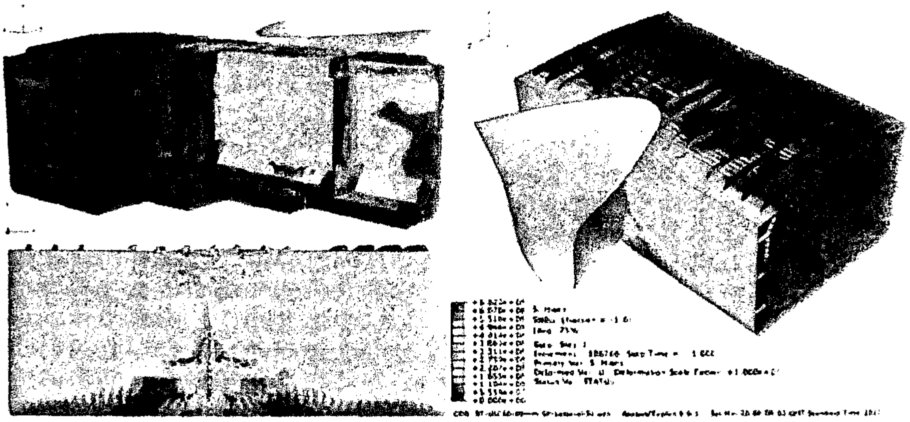

Fig.8 : The lateral penetration of rigid body of normal bow penetrated to double side shell of Baltic tanker..

Figures 7 and 8, show the different deformation shapes produced by the penetration of a bulbous bow and a normal bow. The bulbous bow produces more severe damage compared to normal bow at same collision speed. The collision of the bulbous bow causes rupture of the inner hull which is not apparent during the collision with the normal bow.

The simulation with the normal bow produces a stress distribution, as shown in Figure 8. Figure 7 shows the same stress distribution for the bulbous bow. The bulbous bow stress distribution shows higher stress concentrations, as would be expected, than the normal bow. These results reinforce the idea that bulbous bows should be designed to absorb energy during a collision event to reduce the resulting levels of damage to the struck ship.

The Finite Element results for the forces produced during the 50 degree collision angle simulation are shown in Figure 9. Figure 9(a) shows the individual components of force with the resultant force being shown in Figure 9(b). Results are presented in these figures for both the normal and bulbous bow simulations

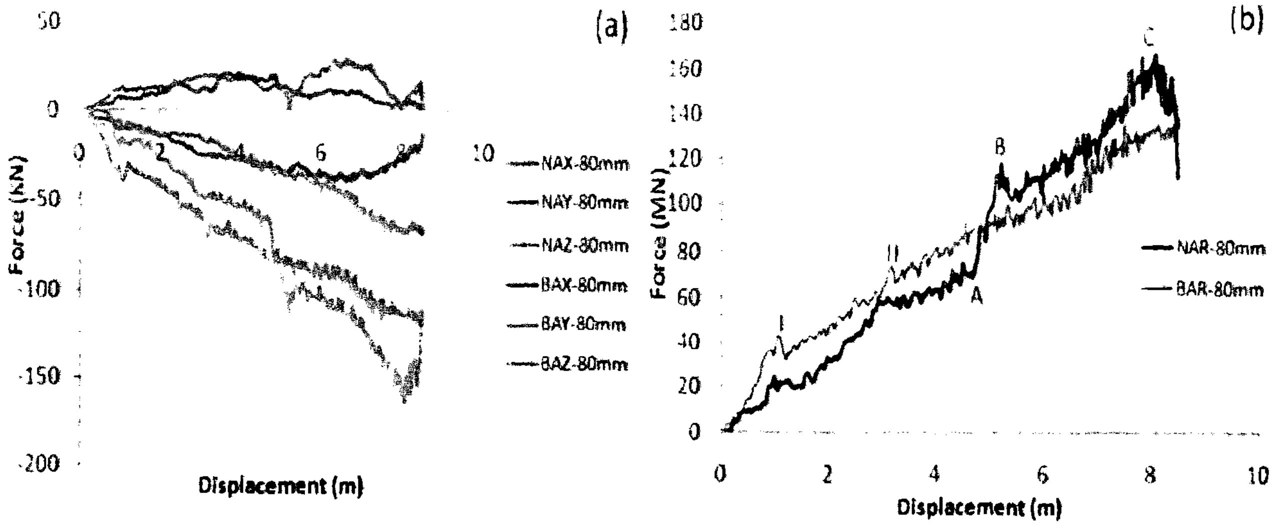

Fig.9 : (a) Force - displacement of 50⁰ collision angle. (b) Resultant force - displacement of 50⁰ collision angle.

These simulations attempt to replicate the actual collision incident of bulk carrier Tern with the Baltic oil tanker. These simulations were carried out to produce a penetration depth of 8.4m for a striking angle of 50^0 to side shell which is equivalent to a damage depth of 6.5m. This should provide damage levels equivalent to the actual damage suffered by the Baltic tanker.

Figure 9a, shows the magnitude of the force acting during penetration for both bow forms in X, Y & Z direction, these results can be compared with the levels of damage shown in Figure 9b. The results for the bulbous bow show the outer shell and inner shell rupture at point I (1.1m, 42.5MN) and point II (3.2m, 73.14MN) respectively.

The curve of resultant force for the normal bow starts to increase at point A (4.7m, 76.85MN) when the flat vertical stem comes in contact with the side shell, with the rupture of outer shell and inner shell occur at point B (5.23m, 112MN) and point C (8.16m, 556MJ), respectively.

Figures 10 and 11, show the severe levels of damage to both the outer shell and inner shell that occurs for both bow shapes. This is in line with the levels of damage that occurred during the actual collision incident. The damage levels produced from the simulation appear to be less severe than the actual damage due to the reduced level of penetration of inner shell and the unknown value of stem angle of the striking ship. The bigger the stem angle the more severe the level of damage that will occur during the collision when the bow is modelled as being rigid.

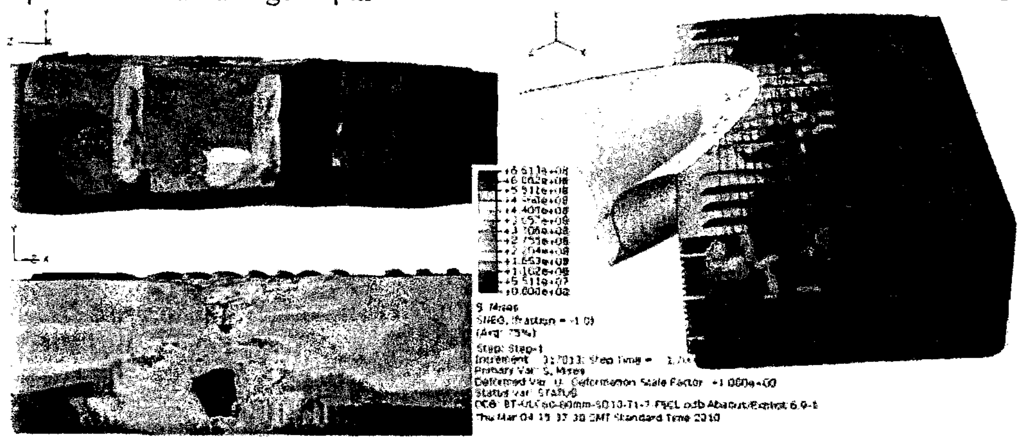

Fig.10 : The rigid body of bulbous bow penetrated to double side shell of Baltic tanker at 50^0 collision angle.

These parameters are probably not the main contributors to the damage during collision of rigid body bow, others such as beam, depth and bulbous bow shape will have a significant influence on the levels of damage and the onset of rupture.

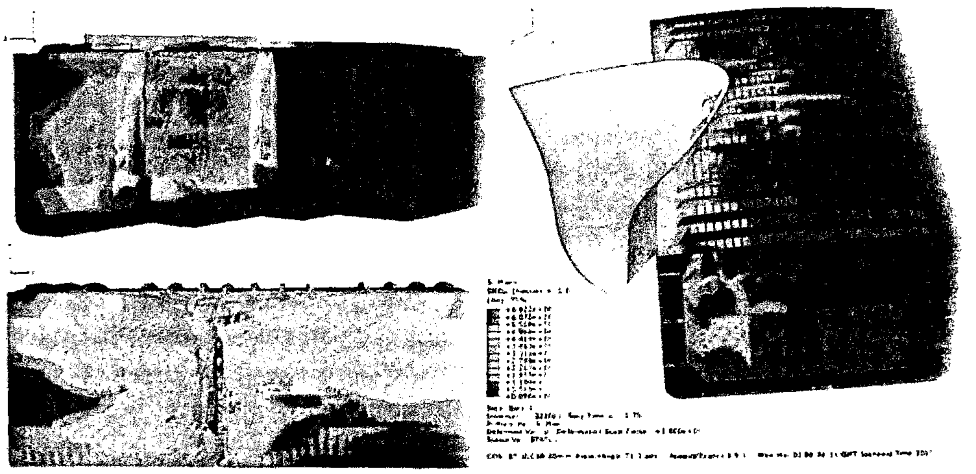

Fig.11 : The rigid body of normal bow penetrated to double side shell of Baltic tanker at 50⁰ collision angle.

The energy dissipated during both the 90 and 50 degree collision scenarios are presented in figure 12a and 12b respectively. The right-angle scenario always demonstrates larger levels of energy to rupture for both outer shell and inner shell.

The graphs for the 90 degree collision scenario for both normal and bulbous bow penetration energy, figure 12a, for both mesh sizes BL-60mm and BL-80mm produce very similar results. The outer shell and inner shell of the struck ship ruptured at 17.7MJ and 85.7MJ, respectively during the collision with the bulbous bow. The outer shell of the struck ship ruptured at 184MJ with no rupture of the inner shell during collision with the normal bow. The collision penetration energy peak at 230 MJ for all meshes and both bow shapes using the same weight/displacement input parameters.

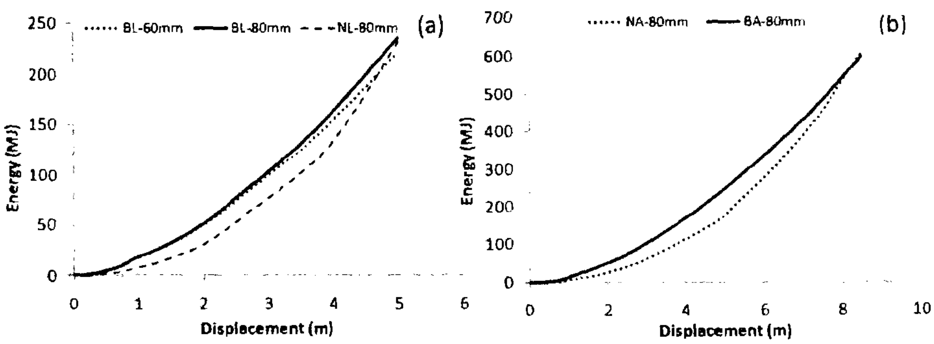

Fig.12 : (a) The lateral penetration energy-displacement (b) The 50⁰ collision angle penetration energy-displacement.

The results for the 50 degree collision angle simulation are presented in figure 12b, the rupture of outer shell and inner shell occur for both types of bow collision simulations. Outer shell and inner shell ruptured at 202MJ, 556MMJ for the normal bow shape and 18.5MJ, 125MJ for the bulbous bow shape, respectively. The collision energy peaked at same point for both collision scenarios with a peak of 600MJ.

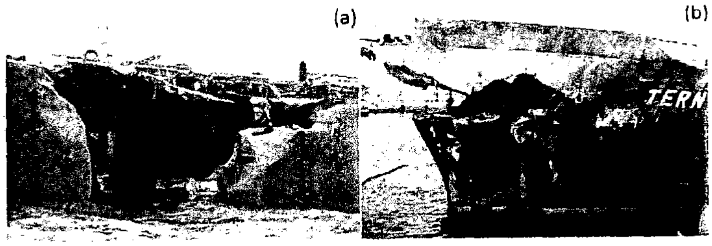

Fig. 13: The collision damage of (a) Baltic Carrier and (b) Bulk Carrier Tern

Simplified Analysis

As a quick check on the energies involved in this collision the empirical equations developed by Minorsky have been used to evaluate the energy absorbed during the collision event.

For the actual collision angle of 50 degrees the energy absorbed by deformation of ship structure are compared to (Minorsky, 1959) using the empirical expression;

$$E = \frac{0.5M_2(M_1 + \Delta M)}{M_2 + M_1 + \Delta M}(vsin\alpha)^2$$

where E losses energy; M_1 mass of struck ship; M_2 mass of striking ship; v initial speed of striking ship; α the angle of impact between ships and ΔM the added mass of sway motion estimated by (Minorksy, 1959) to be about 40% of mass of striking ship. The energy losses calculated are 601MJ for Minorsky's method and 596MJ for the simulation result respectively. These results demonstrate a good degree of correlation with each other even though the collision damage of actual ship as shown in Figure 13a, appears to be far greater than simulation damage as shown in Figure 10 and 11.

The absorbed energy for the 90 degree collision for both simulations settles approximately at 232MJ compared to 1024MJ using Minorksy's simplified empirical expression. This significant difference between the results is mainly due to the assumption that the collision damage depth will be the same for the 90 degree case as in the 50 degree scenario, in the direction of the collision, this is clearly an incorrect assumption. For the 90 degree collision scenario we have no real data available about the penetration damage, duration of collision penetration, and initial speed of collision. This therefore is a purely theoretical study carried out in ABAQUS which assumes damage of the same magnitude as the 50 degree collision scenario which is obviously unrealistic.

The actual collision investigations are based on data about the angle of collision, taken from the incident report (THE OFFICE OF THE MARITIME ADMINISTRATOR, 2002). Comparison between the two scenarios is difficult due to the number of unknown variables in this analysis.

Discussion

The analyses were carried out using both normal and bulbous bow shapes due to lack of information on the bulk carrier Tern structure details.

The material failure model was validated using tensile test data presented in section 4.2, which were then modelled using refined element meshes, the results of which are presented in section 4.3 for simplified structure and different penetration damage scenarios and compared the material strain to (Ehlers, 2009a). Section 4.5 extended to complex structure using 60mm and 80mm mesh elements which also show good correlation.

In all cases considered the energy of the collision for the normal bow was larger than that for the bulbous bow during both outer shell and inner shell rupture. The energy settled at the same peak value during the collision simulation for both bow types using the same input parameter for weight of displacement.

The differences in the actual damage and the predicted damage can be as a result of the assumptions being made about the shape of the bow of the striking ship; the modelling of the bow as being rigid and the condition of the structure of both the struck and striking ships at the time of the collision. Also the assumption of idealised boundary conditions for the finite element model can affect the results produced.

Further work is still being carried out to simulate deformable bow and reanalyse the collision scenarios.

Conclusion

The paper presents the simulation of an actual ship-ship collision. The simulation was performed in two stages: a) the simulation of the path that the ships followed before collision to define the collision geometry, i.e. relative speed and location of impact, and b) the simulation of the structural behaviour of the struck ship. For the present analysis the bow of the striking ship was assumed to be rigid. Further studies will remove this assumption in order to provide adequate results for a comprehensive comparison of the results of the simulation with the actual observations.

The result of the analysis carried out are interesting and give a good insight into the collision event., While very difficult to validate with the actual collision event due to lack of detailed information. The results of complex ship structure collision analysis are presented making a number of assumptions about the structure of the ships involved in the collision and the details of the collision.

In general, during the simulation of the collision we have attempted to analyse the worst scenario for the struck ship by assuming that the bow of the striking ship was rigid. In the future this analysis will be improved and extended to modelling both vessels as deformable bodies during collision event.

References

Alsos, H. S., Amdahl, J. (2009). On the resistance to penetration of stiffened plates, Part I - Experiments. International Journal of Impact Engineering. 36(6): 799-807.

Alsos, H. S., Amdahl, J., Hopperstad, O. S. (2009)."On the resistance to penetration of stiffened plates, Part II: Numerical analysis." International Journal of Impact Engineering 36(7): 875-887.

Chahine G.L. , Wu X., Choi J.-K. (2006): DYNASIM User Manual, DYNAFLOW INC., Jessup, USA.

Cheng, J.-Y., Chahine, G.L., and Hsiao, C-T. (2002): A Computational Tool for Simulating Hydrodynamics Behavior of Multiple Vessels in a Harbor, Technical Report 2M1011-NOAA-1, DYNAFLOW INC., Jessup, USA.

Ehlers, S (2010a). " Strain and stress relation until fracrture for finite element simulations of thin circular plate.Thin-walled Structure, 48, 1-9.

Ehlers, S. (2010b). "A procedure to optimize ship side structures for crashworthiness. Engineering for the maritime environment". Journal of Engineering for the Maritime Environment. 224:1-11.

Jic, M., C. H. Cheng, et al. (2009). "Forming limit diagrams of strain-rate-dependent sheet metals." International Journal of Mechanical Sciences 51(4): 269-275.

Keeler, S. P. and W. A. Backofen (1964). "Plastic Instability and Fracture in Sheets Stretched over Rigid Punches." ASM Transactions Quarterly 56: 25–48.

Kijima Katsuto (1990): On a Numerical Simulation for Predicting of Ship Maneuvering Performance, Proceedings 19th ITTC, 2.

Lamb H. (1932): Hydrodynamics, Dover.

Martin L.L. (1980): Ship Maneuvering in Wind, SNAME Transactions, 88, 257-281.

Minorsky, V.U., 1959. "An analysis of ship collision with reference to protection of nuclear power ships". Journal of Ship Research 3 (2), 1-4.

Spyrou K.J., Tigkas I. and Chatzis A. (2007): Dynamics of a Ship Steering in Wind Revisited, Journal of Ship Research, 51, 2, 160-173.

THE REPUBLIC OF THE MARSHALL ISLANDS, DECISION OF THE MARITIME ADMINISTRATOR REPUBLIC OF THE MARSHALL ISLANDS and the Report of Investigation in the Matter of the Collision between the Tankship BALTIC CARRIER (O.N. 1430) and the Bulk Carrier TERN in the Baltic Sea on 29 March 2001, PUBLISHED BY THE OFFICE OF THE MARITIME ADMINISTRATOR 20 May 2002,