Bulbous Buffer Bows: A Measure to Reduce Oil Spill in Tanker Collisions

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PREFACE

This thesis is submitted as partial fulfilment of the Danish Ph.D. degree from the Technical University of Denmark. The research has been carried out at the Department of Mechanical Engineering at the Technical University of Denmark (DTU). The work took place from February 2004 to February 2007, with Professor Dr. Preben Terndrup Pedersen as supervisor and Peter Friis-Hansen as co-supervisor. The Ph.D. study was supported financially by the Technical University of Denmark and the Japan Society of the Promotion of the Science (JSPS). The recipient is very grateful for the financial support.

Parts of the research, especially the experiment and the finite element analysis, were carried out at the National Maritime Research Institute (NMRI) in Japan as a part of the Buffer Bow Project (BBP) funded by the Japanese Ministry of Land Infrastructure and Transport (MLIT). I am really grateful to NMRI and MLIT for supporting this research.

I would like to express my deepest gratitude to my supervisor Preben Terndrup Pedersen for willingly accepting my study at DTU, and for his excellent instruction and advice throughout the entire project.

I would like to thank to my co-supervisor Peter Friis-Hansen and Professor Jørgen Juncher Jensen for their great assistance with and advice on my work. I would like to thank Professor J. Kim Paik, National Pusan University, Dr. Ge Wang, American Bureau of Shipping, Dr. Wlodek Abramowicz, Impact Design and Ou Kitamura, Mitsubishi Heavy Industry, for their helpful advice and comments on my research. I would like to express my gratitude to my former boss Mr. Endo for his instruction and advice while carrying out my research in Japan.

I would like to thank my colleagues and staffs at the Department of Mechanical Engineering for supporting and helping me while performing my study. I also thank to Dr. Erik Sonne Ravn for always helping me and my family while living in Denmark with matters such as extending visa, finding accommodation and reading Danish letters.

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Lyngby, February 2007

Yasuhira Yamada
Abstract (Danish)

Bløde Bulbstævne: En metode til at reducere olieudslip ved skibskollisioner

Abstrakt:

Formålet med denne afhandling at undersøge, hvor meget man ville kunne reducere olieudslip forårsaget af skibskollisioner ved at indføre mindre stærke bulbstævne på verdens skibe. Analysen omfatter såvel deterministiske som probabilistiske metoder.

Afhandlingen består af et sammendrag og seks publikationer.


For at undersøge, hvor effektive de bløde bovkonstruktioner er ved kollisioner, der involverer store tankskibe, er der udført numeriske simuleringer, hvor såvel de rammende bove som de ramte skibssider er modelleret ved hjælp af detaljerede finite element metoder. Resultaterne af disse deterministiske analyser demonstrerer, at de bløde bove er virkningsfulde til at reducere den kollisionsenergi, der skal til, før olietankene åbnes under kollisioner.

Ved hjælp af de udviklede simplificerede analytiske knusningsmodeller er der udført probabilistiske beregninger af effekten af at indføre bløde bulbkonstruktioner. Der er udført Monte Carlo simuleringer til at bestemme effekten på olieudslip under specielle forhold. Der er ligeledes udført en analyse af omkostningsreduktionen for oprydning efter olieudslip, hvis bløde skibsbove introduceres i et specifikt geografisk område. Denne analyse er udført ved hjælp af en metode baseret på Bayeseske Netværk.
Buffer Bulbous Bows: A Measure to Reduce Oil Spill in Collisions

Abstract:

The purpose of the present thesis is to evaluate the effectiveness in reducing oil spill during collisions if a prototype buffer bow structure is adapted on all striking ships. Deterministic as well as probabilistic analysis procedures are applied.

The thesis consists of a summary and six papers.

The study includes two series of large scale bow crushing experiments. The results of these experiments are compared to the results obtained from detailed Finite Element Analyses (FEA) and from simplified analysis tools in order investigate the basic crushing mechanisms of standard bow structures as well as the prototype buffer bow structure. It is found from this study that both the FEA and the simplified analysis procedure can be used to predict the reaction forces and the energy absorption during crushing of the two bow types against rigid walls.

In order to investigate the effectiveness of the buffer bow structure in large scale ship-ship collisions, numerical simulations has been performed using detailed finite element modelling of the interaction between different bow types and the side structure of a struck tanker. The results of this deterministic analysis demonstrate the effectiveness of buffer bows in reducing the damage of the struck ship side structure.

Using the developed simplified structural analysis tools a Monte Carlo simulation procedure and a procedure based on Bayesian Network has been applied to calculate the probabilistic oil out flow for specific scenarios and the cost associated with this oil outflow in a specific region.
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The following papers are appended to the thesis

[P1] Yamada, Y. and Endo, H.
Collapse Mechanism of the Buffer Bow Structure on Axial Crushing,

[P2] Yamada, Y. and Endo, H.
Experimental and Numerical study on the Collapse Strength of the Bulbous Bow Structures in Oblique
Collision. Marine Technology (Under Review)

[P3] Yamada, Y., Endo, H. and Pedersen, P.T.
Numerical Study on the Effect of the Buffer Bow Structure,

[P4] Yamada, Y and Pedersen, P.T.
A Benchmark Study of Procedures for Analysis of Axial Crushing of Bulbous Bows,
Marine Structures (Under Review)

[P5] Yamada, Y, Pedersen, P.T. and Friis-Hansen, P.
Risk Reducing Effect of Buffer Bow Structures on the Collision Damage of Large Oil Tankers
(To be published)

Risk Analysis of Oil Spill from Struck Oil Tankers
(To be published)
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REFERENCE
1. INTRODUCTION

1.1. Background and Motivation

A disastrous oil spill from a struck oil tanker has become one of the major problems in view of conservation of the maritime environment. The causes of oil spill incidents from oil tankers are shown in Fig. 1.2 and Table 1.1 based on a report from the International Oil Pollution Compensation Fund (IOPCF, 2005) where large oil spill incidents from ships which took place between 1970 and 2005 are reported. According to these statistical analyses, ship collisions and groundings are responsible for about half of all ship losses and are responsible for about 70% of all polluting events caused by shuttle tankers. It is seen in the figure that the most probable cause of oil spill from ships is collision. The cause of oil spills induced by grounding is also among the most probable causes and is taking a secondary place. Therefore, it can be concluded that collisions and groundings are major risks to the safety of ships. To reduce the risks to ships and the environment, it is important to minimize the probability of occurrence of collision and grounding incidents and also to reduce the consequences of such incidents. For a complete analysis of the subject, several tasks should be addressed in order to determine:

- Probability of collisions and groundings
- Damage extents to ships in a given accident scenario
- Probabilistic cargo outflow and structural residual strength and
- Design criteria for grounding and collisions

The purpose of the present study is mainly focused on an investigation of the damage in ships involved in collisions, particularly the crushing resistance exerted by the striking vessel and the damage of the striking ship bow in head-on collisions. So far double hulls have been introduced to reduce the consequences of collision and grounding events. However, it is still a fact that collision accidents involving struck double hull tankers result in oil spills. One example from the Danish waters is the collision accident involving the double hull oil tanker “Baltic Carrier” in Kadetrenden in 2001. Double hull structures have been introduced to reduce the risk of oil spill from large tankers, but it is not sufficient to avoid oil outflow when the striking ships are large and navigating at high speed.

![Fig. 1.1 Struck D/H Tanker Baltic Carrier, 2001, Denmark](image)

In order to further reduce oil spill from struck oil tankers, the concept of buffer bulbous bows has been proposed. Relatively soft buffer bows absorb part of the kinetic energy of the striking ship before penetrating the inner hull of the struck vessel and can distribute the load over a larger area of the struck ship side. The specific purpose of the present project is to verify the effectiveness of a prototype buffer bulbous bow structure in ship-ship collisions as compared with that of standard bulbous bows. The present project is part of the Buffer Bow Project (BBP) which has been initiated and financially supported by the Japanese Ministry.
of Land Infrastructure and Transport (MLIT).

**Cause of oil spills from ships in 1970-2005**

*IOPCF (2005)*

Fig. 1.2. Cause of large oil spills incidents from oil tankers compensated by IOPC Funds. Only large oil spills which cost at least 6.7 million US$ for single incident in member states are taken into account (IOPCF, 2005; IOPCF, 2006).

Table 1.1. Cause of large oil spills from oil tankers (IOPCF, 2005)

<table>
<thead>
<tr>
<th>Cause</th>
<th>Number of Incidents</th>
<th>Ratio [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collision</td>
<td>38</td>
<td>29%</td>
</tr>
<tr>
<td>Grounding</td>
<td>31</td>
<td>23%</td>
</tr>
<tr>
<td>Mishandling</td>
<td>19</td>
<td>14%</td>
</tr>
<tr>
<td>Sinking</td>
<td>15</td>
<td>11%</td>
</tr>
<tr>
<td>Discharge</td>
<td>10</td>
<td>8%</td>
</tr>
<tr>
<td>Unknown</td>
<td>7</td>
<td>5%</td>
</tr>
<tr>
<td>Breaking</td>
<td>5</td>
<td>4%</td>
</tr>
<tr>
<td>Fire</td>
<td>4</td>
<td>3%</td>
</tr>
<tr>
<td>Others</td>
<td>2</td>
<td>2%</td>
</tr>
<tr>
<td>Corrosion</td>
<td>1</td>
<td>1%</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>132</td>
<td>100%</td>
</tr>
</tbody>
</table>

*Ratio is a conditional probability given oil spill incidents

Oil spill into the sea causes environmental and economical damage to the nature, people, societies and companies in the adjacent countries. Oil spill may also cause a serious damage to the fishing and tourist industries in neighbouring countries. Fig. 1.3 shows some examples of oil spill incidents from ships. Even if not all the depicted incidents are associated with collision one can see from the photographs that oil spills in the sea cause disastrous environmental damage both offshore and onshore. It is well known that it involves much manpower and cost to completely clean up oil from contaminated seawater, rock, sand etc.
1.2. Previous studies

Experiments on structural crushing of ship bow structures have been carried out since the early 1960s. Details of these experiments are reviewed in references such as: Woisin (1979), Amdahl (1983), Jones (1979), Pedersen et al. (1993), Zhang (1999), Endo et al (2001) and Endo et al (2002). Simplified analysis methods for crushing analysis have been commonly used. The methods are based on the upper-bound theorem (energy rate balance equation), observations of accidental damage patterns, and experimental studies. Due to limitations in computer power these methods were the only possible analysis procedures until recently. Consequently, they have been widely used.


More recently bow collision analyses based on comprehensive Finite Element Analyses have been carried out. Kitamura (2000, 2002) applied the buffer bow concept to the bulbous bow of large cargo ships in the research project supported by the Japanese Association for Structural Improvement of Shipbuilding Industry (ASIS, 1997). They carried out FEM simulations assuming that a Suezmax tanker collides with a double hull VLCC with the purpose of evaluating the effectiveness of buffer bows. In this research project only one cargo oil tank of the struck tanker was modeled by elastic-plastic elements and other parts of the vessel were modeled by rigid elements, where deformation of transverse bulkheads located aft and forward of the struck cargo tank were not considered. However, the modeling of the VLCC was not satisfactory for the case of high-energy collisions due to the limitation of computer technology in those days. The relatively narrow deformable range surrounded by the relatively close rigid boundaries may cause early rupture of the side and the inner shell of the struck ship. Subsequently, Kitamura (2002) conducted a series of finite element simulations and pointed out that the effect of global-hull-girder horizontal bending moment is significant when a ship collides with the midship region of a struck ship and the size of the striking ship is the same or larger than that of the struck ship. Kitamura (2002) carried out several cases of analysis where a VLCC collides with a relatively small ship considering the global-hull-girder horizontal bending moment of the struck ship. Using a simplified model the effect of the global-hull-girder horizontal bending moment was demonstrated by Pedersen and Li (2004) which resulted in significant strains in the ship side.

1.3. The aim of the study

The main purpose of the present study is to investigate the effectiveness of prototype buffer bulbous bow structures on reduction of the risk of oil spill from struck tankers using deterministic as well as probabilistic analysis tools. In order to investigate the basic crushing mechanisms of bulbous bow structures, two series of large-scale crushing experiments ([P1], [P2]) were carried out by crushing of 7 large scale bulbous bow models against a rigid wall. One of the two series is experiments with axial crushing of the bulbous bow structures, and the other series of experiment involve bending collapse of bulbous bow structures. The results
obtained from the experiments were compared with those estimated by simplified analytical analysis methods and Finite Element Analysis (FEA) respectively. Accuracies of both the simplified analysis methods ([P1], [P4]) and the FEA methods ([P1], [P2]) were investigated in detail.

After verification of the accuracy of FEA methods for crushing of bulbous bows against a rigid wall, large-scale ship-ship collision analyses [P3] were carried out by use of FEA in collision scenarios involving tanker – tanker collisions. The effectiveness of buffer bow structures were deterministically investigated in these specific collision scenarios.

Simplified ship collision analysis tools have been developed [P5] in order to rapidly predict structural damage in given ship-ship collision scenarios. These simplified methods include prediction of deformation of the striking ship bow as well as deformation and rupture of oil cargo tanks of struck tankers.

The probabilities of rupture of cargo oil tank in struck double hull VLCC given collisions with other tankers were estimated using Monte Carlo simulations. [P5].

As a case study, risk analyses of oil spill from struck double hull VLCCs in the Great Belt and Oresund has been performed using Bayesian networks [P6]. Expected loss (risk) of oil spill from struck VLCCs when all striking ships are using buffer bow structures is compared to that assuming standard bows. The effectiveness and cost-effectiveness of the buffer bow as compared to that of the standard bow are discussed in detail.

2. DESIGN CONCEPT/PRINCIPLE OF BUFFER BULBOUS BOW STRUCTURES

The buffer bow has initially been studied by ASIS (1997) and Kitamura (2000). A concept of buffer bulbous bow is that relatively soft buffer bows absorb part of the kinetic energy of the striking ship before penetrating the inner hull of the struck vessel and can distribute the load over a larger area of the struck ship side. The basic design principle of the buffer bow is as follows:

- Low cost
- Adopt transverse stiffening system
- Minimize shell thickness
- Comply with class rule

Firstly, in order to introduce buffer bow structures into the shipping industry it is important that the initial building cost of buffer bow should be lower or equivalent with that of standard bow. Therefore, the prototype buffer bow design does not use expensive materials but is based on standard steel. Secondly, in order to reduce the crushing resistance of bow structure, which is a menace to the struck ships, a transverse stiffening system is employed instead of the usual longitudinal stiffening system. Thirdly, the plate thickness is minimized to reduce buckling strength of the bow structure. Eventually, the transverse frame spacing has to be small enough to resist the local environmental loads like slamming and anchor chain impacts. That is it has to be in accordance with the structural design rules of the classification societies. It is noted that minimum transverse frame spacing can be decided from the view point of fabrication of the ship structures. An investigation among the shipbuilding companies in Japan indicates that the minimum transverse frame spacing can be around 650mm. This will allow a worker to enter the space and weld the plates between the transverse stiffeners.
3. CRUSHING MECHANISM OF BULBOUS BOW STRUCTURES

In order to investigate the basic crushing mechanism of the buffer/standard bulbous bow structures and also to verify the accuracy of finite element analysis procedures and simplified analysis procedures, two series of quasi-static experiments have been conducted using large-scale bulbous bow models. Large-scale bulbous bow models were adopted in order to conduct the experiment as realistically as possible, and to minimize the scale effect and the effect of welding induced residual stresses. The size of the models was chosen to be almost half scale of the bulbous bow of an actual VLCC. The sizes of the test specimens were chosen considering the maximum capacity of the experimental facility, the difficulty in experimental setup and economical limitations.

One of the two series is the experiments of axial crushing of bulbous bow models against a rigid wall [P1] modelling right angle collisions. This is supposed to be one of the most severe collision scenarios for struck tankers. Other experiments involve bending collapse of bulbous bow models [P2] modelling oblique collisions. Both series of experiments were conducted in Japan based on funds from Japanese Ministry of Land Infrastructure and Transport (MLIT).

3.1. Axial Crushing of Bulbous Bow Structures [P1] [P4]

In order to analyse the axial crushing behaviour of bulbous bow structures a series of quasi-static experiments have been conducted using four kinds of bulbous bow models. All models have the same shape and dimensions of the outer shell but the stiffening system inside the outer shell is different. Three of four models had transverse stiffening system as a prototype buffer bulbous bow structures, and the forth had a longitudinal stiffening system just as a standard bulbous bow structure. The main particulars of these models are shown in Table 3.1, and the scantlings and typical sectional views of these models are shown in Fig. 3.1 and Fig. 3.2 respectively. All the models are perpendicularly collapsed by a rigid-board from the top of the bulb. The details of the experimental set up are described in [P1]. Moreover, corresponding FEA and simplified analysis have been conducted and these results are compared with the experimental results. Verification results are described in later sections. Some of the pictures from the experiments which are not shown in Ref. [P1] are shown in Fig. 3.3 through Fig. 3.7.

The following conclusions can be drawn from the study:

(1) All models collapsed progressively from the top of the bow regardless of the stiffening system.

(2) The failure of the three bulbous bow models with transverse stiffening system (buffer bows) resulted in two folds between transverse frames. The second folding tends to cause larger reaction forces than first fold presumably due to the fact that the deformed plate prevents free deformation of the second fold and that the second fold does not have enough space to fold with the minimum reaction force. However, it is noted that the number of folds between transverse frames is supposed to be affected by various factors such as the plate thickness, stiffening system, initial imperfection and so on. Further investigation is needed to derive more general conclusion about the number of folds between transverse frames.

(3) The bows with a transverse stiffening system are effective in order to decrease the maximum reaction force while still preserving moderately high energy absorption.

(4) FEA could simulate the results from the experiments well and gives good estimate of the history of reaction forces as well as the energy absorption although a slight discrepancy can be found between FEA and experiment. Relatively larger discrepancies can be found in case of longitudinally stiffened bow.

(5) Simplified analyses method gives fairly good estimate of the mean crushing force and the energy absorption.
Fig. 3.1. Drawings of the bulbous bow models.
Table 3.1 Main characteristics of the bulbous bow models.

<table>
<thead>
<tr>
<th>Model ID</th>
<th>Stiffening System</th>
<th>Outer shell Thickness [mm]</th>
<th>Type of plate intersection</th>
</tr>
</thead>
<tbody>
<tr>
<td>BC-E</td>
<td>Transverse</td>
<td>10</td>
<td>T-type</td>
</tr>
<tr>
<td>BC-F</td>
<td></td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>BC-G</td>
<td></td>
<td>10</td>
<td>X- and T-type</td>
</tr>
<tr>
<td>BC-L</td>
<td>Longitudinal</td>
<td>10</td>
<td>L- and T-type</td>
</tr>
</tbody>
</table>

Fig. 3.2. Sectional view of the bulbous bow models.
Fig. 3.3. Experimental setup of the bulbous bow model. The model BC-E is shown as an example.

Fig. 3.4. Second buckling (inward buckling) of the outer shell of the model BC-E between Fr.1 and Fr.2. Green lines represent that stiffeners or webs are attached inside the shell along the line. Yellow lines represent the mid position between adjacent transverse frames.
Fig. 3.5. Deformation of the outer shell of the model BC-E. Intrusion of the fore part of the bow into the aft part of the bow just after the first buckling (outward buckling) between Fr.4 and Fr.5. Second buckling took place after the completion of first buckling and the intrusion.

Fig. 3.6. Rupture of the outer shell of the model BC-E along the heat affected zone (HAZ), which is supposed to be caused by the fillet welding between the outer shell and the transverse ring frame.
Fig. 3.7 The deformation of the model BC-G after the collapse experiment.
Fig. 3.8. Progressive collapse of the model BC-E in axial crushing.
3.2. Bending Collapse of Bulbous Bow Structures [P2]

In order to investigate the collapse strength of the prototype-buffer/standard bulbous bow structures in case of oblique collisions, a series of quasi-static experiments have been conducted using two types of bulbous models. The outer shells of both models have the same shape and size but the stiffening systems inside the outer shells are different. One of the models (BB-D) has transverse stiffening system as a prototype buffer bulbous bow structure, and the other one (BB-E) has a longitudinal stiffening system as a standard bulbous structure. Both models were quasi-statically collapsed by a rigid board in a collision angle of about 72 degrees such that the bows were subjected to a combination of axial force and bending moment (See Fig. 3.9). The details of the experimental setup are described in [P2]. Moreover corresponding FEA and simplified analyses are conducted, and these results are compared with the experimental results. Some of the pictures from the experiments, which are not shown in Ref. [P2], are shown below.

![Experimental setup](image)

Fig. 3.9. Experimental setup.
Fig. 3.10. Deformation of the model BB-D (prototype buffer bow model with transverse stiffening system).

Fig. 3.11. Deformation of the model BB-E (standard bow model with longitudinal stiffening system).
The following conclusions and remarks can be drawn from the study:

(1) In the experiments, both bow models collapsed in overall horizontal bending mode near the root of the bulb in spite of different stiffening systems. In both cases this occurred after a small denting of the bulb tip.

(2) The maximum reaction force of the buffer bow structure (BB-D) is about 40% lower than that of the standard bow structure (BB-E). This is important in order to prevent penetration into the struck ship. Thus it is expected that buffer bow structures have a significant effect for reduction of the damage of the struck ship in case of oblique collisions.

(3) Although small discrepancies are detected between experimental results and results obtained from FEA, it can be said that the level of reaction force and total absorbed energy can be estimated with reasonable accuracy.

(4) In the analysis of the model BB-E, it is confirmed that a collapse, which is different from the mode observed in the experiment, is obtained when the effect of residual stresses induced by welding is not considered. In order to simulate the experimental collapse mode of the model BB-E the residual stresses should be taken into account in the analysis. It is also confirmed that the overall buckling mode could not be achieved by using only initial geometric imperfections. Further investigations of the effect of residual stresses on the collapse mode should be carried out for other collision angles.

(5) For horizontal bending collapse of bow structures in a collision angle of 72 degrees, the level of reaction forces and total absorbed energy are not sensitive to the collapse mode since about half of the total energy is absorbed by denting of the bow tip which has thicker outer plate thicknesses than the other parts of the outer shell.

(6) The extent of deformation at the bulb tip obtained by the present FEA is slightly larger than that observed in the experiments. This is presumably due to the effect of residual stresses induced by the metal forming process of the round bulb tip. Since the horizontal bending moment acting on the bulb is sensitive to the distance from the tip of the bow, further investigations to accurately simulate the denting mechanism at the bow tip are needed. Such investigations should consider the effect of residual stresses induced by metal forming.
4. VERIFICATION OF THE EFFECTIVENESS OF BUFFER BOWS BY LARGE-SCALE FINITE ELEMENT ANALYSES [P3]

Finite Element Analysis (FEA) is one of the most powerful tools for evaluation of the crashworthiness of ship structures as well as for simulating full scale ship collisions. Many researchers (for example ASIS, 1997; Kitamura, 2000; Kitamura, 2002; Endo et al, 2004) have performed ship-ship collision analyses using large-scale FEA. In the past the main focus has been on the crashworthiness of the sides of the struck ships such as oil tankers or nuclear ships.

In order to investigate the effectiveness of the buffer bow structures in more realistic conditions a series of large scale nonlinear 3D FEA has been conducted using the commercial software LS-DYNA. The effect of the buffer bow in collision scenarios where a large VLCC in ballast condition collides perpendicularly with the midship region of another double hull VLCC in a fully loaded condition (Fig. 4.1-Fig. 4.13) is investigated in [P3]. The details of the analysis condition are described in [P3]. From these analyses the following observations are made:

1. It is found that the fracture strain has a significant effect on rupture of the outer and the inner shell, the contact force, and the energy absorbed by the striking and the struck ship.

2. The calculations show that the results are not sensitive to the fracture strain if the fracture strain range is 10-12%. It can be said that FS=12% gives fairly reasonable results although quantitative verification is not performed.

3. It is found that the buffer bow does not completely prevent the rupture of inner hulls in case of high energy collisions, but it can delay the rupture of the inner shell to some extent.

4. For the case of a collision angle of 90 degrees without forward velocity of the struck ship, the critical striking velocity for the striking vessel with a buffer bow becomes about 77% higher than critical velocity of a vessel with a standard longitudinally stiffened bow. This indicates that a VLCC with buffer bow navigating at velocity of 11kt or lower does not penetrate the inner shell of an anchored double hull VLCC.

5. The effect of the buffer bow becomes more advantageous for right angle collisions when the struck ship has a forward velocity because early bending of the bulbous bow results in larger contact area with the struck ship side shell. It could be said that the buffer bow is effective to further reduce the risk of oil spill in cases where the struck ships have a forward velocity.

Similar investigations of a collision scenario where an Aframax tanker with a buffer bow collides perpendicularly with a fully loaded VLCC has been carried out. The experience from these numerical simulations concerning the effect of buffer bows confirms the conclusions stated above. As an illustration of the effect of the buffer bow some deformation results are shown from Fig. 4.14 through Fig. 4.18.
Fig. 4.1. Collision scenario.

Fig. 4.2. Finite Element Models of Striking and Struck Ship, $\theta=90\text{deg.}$
Fig. 4.3. Condition of Analysis (S-S’ Sectional View). Both ships are VLCC.
Fig. 4.4. Deformation of both ships at $t=1.0\text{s}$ ($V_A=0\text{kt}$, $V_B=15\text{kt}$, Standard Bow).

Fig. 4.5. Deformation of both ships at $t=1.0\text{s}$ ($V_A=0\text{kt}$, $V_B=15\text{kt}$, Buffer Bow).
Fig. 4.6. Damage of the struck ship at $t=1.0\text{s}$ ($V_A=0\text{kt}$, $V_B=15\text{kt}$, Standard Bow).

Fig. 4.7. Damage of the struck ship $t=1.0\text{s}$ ($V_A=0\text{kt}$, $V_B=15\text{kt}$, Buffer Bow).
Fig. 4.8. Deformation of the striking ship bulb $t=1.0s$ ($V_A=0$kt, $V_B=15$kt, Standard Bow).

Fig. 4.9. Deformation of the striking ship bulb $t=1.0s$ ($V_A=0$kt, $V_B=15$kt, Buffer Bow).
Fig. 4.10. Deformation ($V_A=V_B=15$kt, Standard Bow, $t=0.1s$)

Fig. 4.11. Deformation ($V_A=V_B=15$kt, Standard Bow, $t=0.5s$)
Fig. 4.12. Deformation ($V_A=V_B=15$kt, Buffer Bow, $t=0.2s$).

Fig. 4.13. Deformation ($V_A=V_B=15$kt, Buffer Bow, $t=0.5s$)
Fig. 4.14. Condition of Analysis (S-S’ Sectional View). Striking ship is Aframax, struck ship is VLCC.
Fig. 4.15. Deformation of both ships*1 in case of standard bow.  
(S-S’ sectional view in Fig. 4.1).

Fig. 4.16. Deformation of both ships*1 in case of buffer bow.  
(S-S’ sectional view in Fig. 4.1)

*1

Striking ship; Aframax Tanker (60,000DWT), $V_B = 20$kt  
Struck ship; VLCC (300,000DWT), $V_A = 0$kt
Fig. 4.17. Deformation of the Standard Bow

Fig. 4.18. Deformation of the Buffer Bow
5. SIMPLIFIED SHIP COLLISION ANALYSIS TOOL (SSCAT) [P5]

This chapter presents a simplified analyses method for prediction of structural damage to ships in a collision event.

5.1. Existing Analysis Methods and Tools

The methods for evaluating the structural consequences of a given ship collision can be classified into four categories such as

(1) Empirical formula
(2) Simplified analysis methods
(3) Non-linear FEA simulation
(4) Experiments

Both the time and cost increases as the evaluation procedure moves from (1) to (4). Among these procedures the nonlinear FEA simulation procedure is becoming a more and more practical design tool as the computer technology progresses. However, it is still very time-consuming to establish FEA models of ship structures as well as to conduct the calculations. For example, in this study, it takes about 1-2 months to build the finite element model of one ship, and it takes 4 or 5 days by using 4 parallel CPU computers to calculate one ship-ship collision scenario where about 500,000 nodes are used in total. Explicit FEA calculations do not always converge but sometimes the calculations diverge. Then restart is required after correcting the model, or by making the time steps smaller. This further increase the calculation times and there is no guarantee that the next calculation sequence converges. Therefore, the number of calculation cases which in a practical way can be carried out by FEA is rather limited.

For these reasons the FEA is at the moment only suitable for a deterministic approach, and not efficient enough to be used in a probabilistic approach such as risk and reliability analyses. Furthermore, it cannot be applied in the early stage of the ship design. Therefore, there is a high demand to establish simplified analyses tools to rapidly evaluate the crashworthiness of ship structures.

As a part of the present study simplified analyses methods, which are efficient in use and still give predictions with reasonable accuracy, have been studied. In the following section existing simplified analysis program are shortly reviewed and the content of the present method is described in more detail.

In the past the following simplified analysis methods have been developed;

- DAMAGE (MIT)
- GRACAT (DTU)
- SIMCOL (VT)

In SIMCOL (Brown, 2002a; Brown, 2002b; Brown, 2004) a Minorsky based method is used for damage evaluation. GRACAT (Friis-Hansen and Simonsen, 2002) developed at DTU is a powerful tool to evaluate the damage of the struck ships based on simplified structural analysis tools. In all these procedures the striking ship is assumed to be rigid.

There are two types of simplified collision analysis methods from the view point of treatment of external (the gross motions of the colliding structures) and internal (the structural crushing) dynamics. One is procedures which uncouples the external and the internal mechanics. The other is procedures which couples these phenomena. Brown developed a simplified analysis tool SIMCOL where the interaction of these two phenomena is taken into account. Brown conducted comparative studies between coupled methods and the uncoupled Pedersen and Zhang (1998) method. It is found from these results that total absorbed energy released for structural deformation is almost the same for two these methods.
5.2. Development of SSCAT [P5]

In this study a new Simplified Ship Collision Analysis Tool (SSCAT) has been developed expanding the structural analysis part of GRACAT by taking into account crushing deformation also of the striking ship bow. In SSCAT the external and internal dynamics are calculated independently. For the external dynamics a method proposed by Pedersen and Zhang (1998) is employed, where the energy released for structural deformation and friction is calculated taking global ship motions of both ships into account. This method can treat oblique collisions as well as perpendicular collisions. Moreover the closed formulas to estimate the structural deformation energy make the calculations fast and thus suitable for risk and reliability analysis. More details of the SSCAT are described in [P5].

In the internal dynamics structural deformations and mean crushing strength of each ship are estimated by use of rigid plastic analysis methods using super elements. Based on a benchmark study [P4], Yang & Caldwell (1988)’s method is employed in SSCAT to estimate the mean crushing strength of super elements for plate intersections such as L- T- and X- type elements.

In order to validate the accuracy of the present method, the SSCAT is applied to the previously described ship collision where a VLCC in ballast condition collides perpendicularly with the midship region of another VLCC in a fully loaded condition. The results obtained by SSCAT are compared with those obtained by FEA [P5].

From these studies it is confirmed that the SSCAT estimates the mean crushing force and energy absorption of the ship-ship collision with reasonable accuracy although some discrepancies can be seen at the fluctuation of the contact forces as function of penetration. It is found that the rupture of the side shell as well as cargo oil tank is largely dependent on the rupture strain.

The present version of the SSCAT only deals with perpendicular crushing of bow structures in the internal dynamics although the external dynamics procedure can treat oblique collisions. Therefore, the application of the present version of SSCAT is limited to perpendicular collisions. Procedure to estimate the crushing strength of the bending collapse of the bow as well as estimate increase of the contact area by the bending of bow tip is under the development. The feature to take into account the bending collapse in internal dynamics as well as the coupling of the external and internal dynamics is planned to be implemented in a future version of the SSCAT.
6. PROBABILISTIC DAMAGE ANALYSIS [P5]

The purpose of this chapter is to obtain the probability of the rupture of oil cargo tank of the struck tankers by using Monte Carlo Simulation (MCS) in the specific collision scenario. The simplified ship collision analysis tool (SSCAT) described in the previous chapter is used to evaluate the rupture of the cargo oil tank in ship-ship collision.

Tankers with 5000 dead weight tonnage or over shall be equipped with Double Hull (D/H) system according to the MARPOL convention. Statistics show that there is a slightly higher probability that ships of the same type collide with each other when the global accident statistics are used (Lutzen, 2001). Of course, in specific terminal areas one can expect that the probability that similar ship types collide against each other is quite high.

The potential risk of oil spill from large struck oil tankers such as VLCC is supposed to be relatively higher than smaller struck tankers given rupture of the cargo oil tank. Considering the disastrous risk of oil spill from large oil tankers, therefore, the assumed collision scenario is set to that a VLCC in fully loaded condition is struck by other ships of greater than 5000 DWT with bulbous bow structures.

The probability of the rupture of oil cargo tank (inner shell), $P_{f,IS}$, is estimated by carrying out Monte Carlo Simulation in the assumed collision scenario. As in previous studies (Lutzen, 2001; Brown, 2002a; Brown, 2002b; Brown, 2004) many uncertain factors such as the main particulars of the striking ships, the velocity of the striking ship and the collision point at the struck ship are generated as random variables. Three groups of striking ships are defined dependent on the minimum deadweight tonnages (DWT$_{B,min}$) of the ships in each group. First group is ships of 5,000 DWT or above (DWT$_{B,min} = 5000$). Second group is ships of 100,000 DWT or above (DWT$_{B,min} = 100,000$). Third group is ships of 200,000 DWT or above (DWT$_{B,min} = 200,000$). Details of the analyses are described in [P5].

From these analyses the probabilities of rupture of the cargo oil tank are estimated as shown in Table 6.1. Relation of the failure probability $P_{f,IS}$ depending on the minimum size of the striking ships DWT$_{B,min}$ is shown in Fig. 6.1. According to the results, the failure probability of the rupture of cargo oil tank of VLCC in case of collision with standard bow decreases about 19-24% by using buffer bow as far as present collision scenario. Moreover it can be seen from Fig. 6.1 that the failure probability $P_{f,IS}$ increases as the minimum size of the striking ship increases. This is presumably due to that the mass as well as the initial kinetic energy of the larger striking ships is larger than that of the smaller striking ships. That is, for example the collision in CASE 3 is higher energy collision than that in CASE 1. Therefore the damage of both ships as well as the failure probability of cargo oil tank in CASE 3 is supposed to be higher than those in CASE 1.

The probabilities obtained in this chapter are used in the risk analysis of oil spill from struck large oil tankers in the next chapter.
Table 6.1. Failure probability obtained by MCS. $P_{f,IS}$ denotes the failure probability of oil cargo tank (inner shell) of the struck ship. $P_{f,OS}$ denotes the failure probability of the outer shell of the struck ship. $DWT_B$ denotes the minimum dead weight tonnage of a group of striking ships.

(a) CASE 1: $DWT_{B,min} = 5000$ ($5,000 \leq DWT_B \leq 400,000$)

<table>
<thead>
<tr>
<th></th>
<th>Inner Shell</th>
<th>Outer Shell</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$N_{sim}$</td>
<td>$P_{f,IS}$</td>
</tr>
<tr>
<td>Standard Bow</td>
<td>566</td>
<td>0.66</td>
</tr>
<tr>
<td>Buffer Bow</td>
<td>1439</td>
<td>0.44</td>
</tr>
</tbody>
</table>

(b) CASE 1: $DWT_{B,min} = 100,000$ ($100,000 \leq DWT_B \leq 400,000$)

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<th>Inner Shell</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>$N_{sim}$</td>
<td>$P_{f,IS}$</td>
</tr>
<tr>
<td>Standard Bow</td>
<td>351</td>
<td>0.76</td>
</tr>
<tr>
<td>Buffer Bow</td>
<td>1013</td>
<td>0.52</td>
</tr>
</tbody>
</table>

(c) CASE 1: $DWT_{B,min} = 200,000$ ($200,000 \leq DWT_B \leq 400,000$)

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<th>Inner Shell</th>
<th>Outer Shell</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>$N_{sim}$</td>
<td>$P_{f,IS}$</td>
</tr>
<tr>
<td>Standard Bow</td>
<td>304</td>
<td>0.79</td>
</tr>
<tr>
<td>Buffer Bow</td>
<td>755</td>
<td>0.60</td>
</tr>
</tbody>
</table>

Fig. 6.1. Variation of the failure probability $P_{f,IS}$ depending on the minimum size of the striking ships.
7. RISK ANALYSIS OF OIL SPILL FROM OIL TANKERS [P6]

The purpose of this chapter is to evaluate the risk of oil spills from large struck oil tankers by using Bayesian Network. A Bayesian Network model (Fig. 7.1) and a methodology to evaluate risk of oil spill from struck tankers is established. As a case study risk analysis of oil spill is carried out in the specific collision scenarios that a VLCC is struck by another tanker in the Great Belt and Oresund. The risk of oil spill using buffer bows are compared with that using standard bows, and the reduction of the risk is investigated in detail [P6].

In order to model the cost of oil spill the total cost is broken down into constituent elements. One is the cost for clean-up oil, the other is the cost for other compensations. It is represented as:

\[ C_{\text{total}} = C_{\text{clean}} + C_{\text{other}} \]  

where \( C_{\text{total}} \), \( C_{\text{clean}} \) and \( C_{\text{other}} \) denote the total costs, costs for clean-up oil and costs for other compensation respectively. \( C_{\text{other}} \) includes several major factors such as damage to the natural resource or environment, harm to wild life, compensation for fishing industry and tourism industry, and so on. In this study \( C_{\text{other}} \) is further broken down into four elements in order to take into account various factors that affect the costs of oil spill. That is the cost for the fishing industry, the cost for the tourism industry, the cost for the environment and costs for the population living in neighbouring area.

Moreover costs for the cleaning up of the oil is expected to significantly increase once the spilled oil contaminates the shoreline due to the difficulty in cleaning and the long distance of the shoreline. Therefore, the clean-up cost is further divided into two elements. One element is the cost for cleaning offshore and the other is the cost for cleaning at the shoreline. Eventually the total cost is represented as:

\[ C_{\text{total}} = C_{\text{clean}} + C_{\text{others}} \]
\[ = (C_{\text{clean, offshore}} + C_{\text{clean, shoreline}}) + (C_{\text{fish}} + C_{\text{tourism}} + C_{\text{environment}} + C_{\text{population}}) \]  

Considering that the costs for tourism, costs for environment and costs for population are usually related to and taken place around or at shoreline, the formula can be reformed as:

\[ C_{\text{total}} = C_{\text{clean, offshore}} + C_{\text{fish}} + (C_{\text{clean, shoreline}} + C_{\text{tourism}} + C_{\text{environment}} + C_{\text{population}}) \]
\[ = C_{\text{offshore}} + C_{\text{shoreline}} + C_{\text{fish}} \]  

As a result the total cost is divided into two geometrically categorized elements (\( C_{\text{offshore}} \) and \( C_{\text{shoreline}} \)) and the costs for fishing industry (\( C_{\text{fish}} \)). One of the main factors which largely affect \( C_{\text{offshore}} \) and \( C_{\text{shoreline}} \) is the amount of spilled oil at each area. By taking \( C_{\text{offshore}} \) and \( C_{\text{shoreline}} \) in the modelling, the increase of cost of oil spill due to the arrival of spilled oil at the shoreline can be taken into account. That is, if most of the spilled oil is skimmed up within the offshore before oil reaches the shoreline, it is expected that the estimated costs can be reduced significantly.

\( C_{\text{shoreline}} \) includes not only the costs for cleaning up oil (\( C_{\text{clean, shoreline}} \)) but also the costs for other elements. However, to accurately estimate \( C_{\text{shoreline}} \) is quite complex problem because some of them are mutually dependent on each other. In this study a somewhat macroscopic way is employed. \( C_{\text{cleanup}} \) is assumed to be estimated by multiplying a scale factor to \( C_{\text{shoreline}} \) as follows:

\[ C_{\text{shoreline}} = C_{\text{clean, shoreline}} + (C_{\text{tourism}} + C_{\text{environment}} + C_{\text{population}}) \]
\[ = C_{\text{clean, shoreline}} (1 + f_{\text{tourism}} + f_{\text{environment}} + f_{\text{population}}) \]  

where

\( f_{\text{tourism}} \) : scaling factor for the cost for tourism industry.
\( f_{\text{environment}} \) : scaling factor for the cost for environment.
\( f_{\text{population}} \) : scaling factor for the cost for population living close to the contaminated shoreline area.

With this background the following Bayesian Network has been established:
The following conclusions are obtained from the study:

(1) It is estimated from the present analysis that the risk of oil spill for one single striking ship per year in case of the use of buffer bows is 28% lower than that in case of using standard bows. This corresponds to a risk reducing effect about 37,000-42,000 [US$] for one single striking ship if the lifetime of the striking ship is assumed to be 30 years. Therefore it can be said that introducing a buffer bow to a ship passing through the Great Belt and Oresund is cost-effective only if the difference between the initial shipbuilding costs and the repair costs between the standard bow and the buffer bow is smaller than 37,000-42,000 [US$/ship].

(2) Similar to the case in Great Belt and Oresund, the risk of oil spill from struck VLCCs in World Wide operation is roughly estimated by assuming that all the analysis conditions other than the probability of collision is same as those in Great Belt and Oresund. It is roughly estimated that introduction of the buffer bow has a risk reducing effect of about 48,000 [US$] for one single striking ship if the lifetime of the ship is assumed to be 30 years. Therefore it can be expected that introducing buffer bow to a ship in world wide operation is cost-effective provided that the difference between the initial shipbuilding costs and the repair costs between the standard bow and the buffer bow is smaller than 48,000 [US$/ship].
costs between a standard bow and a buffer bow is smaller than 48,000 [US$/ship] although further studies are necessary to accurately estimate the risk of oil spill from struck tankers in case of world wide trade.
8. CONCLUSION

The effects of the buffer bow on the reduction of structural damage of struck tankers as well as on the reduction of risk of oil spill from struck tankers are deterministically and probabilistically investigated by use of two series of large-scale experiments, nonlinear finite element analyses, simplified ship collision analyses, as well as risk and reliability analyses. The following conclusions can be drawn from these studies.

(1) Generally FEA could simulate the deformation pattern observed in the experiment of crushing of bulbous bows, and could estimate the history of reaction forces as well as the energy absorption with reasonable accuracy although a slight discrepancy can be found between FEA and experiments.

(2) When completing the FEA of bending collapse of a longitudinally stiffened bulbous bow it was necessary to include the effect of residual stresses induced by welding in order to obtain the same crushing mode as in the experiment. It is also found that the effect of residual stresses on the crushing force and the energy absorption was not significant. However general conclusion cannot be drawn due to the limited number of completed analyses. Further investigations taking other collision angles as well as other bulbous bow models into account are necessary to arrive at a general conclusion on the effect of the residual stress on the simulation results.

(3) The effect of the buffer bow on the reduction of the structural damage of the struck tanker is deterministically investigated in specific collision scenarios by use of non-linear FEA. It is found that the buffer bow has an effect of reducing the structural damage of the struck tankers although it does not completely prevent the rupture of a cargo oil tank in case of high energy collisions. It is also confirmed that the buffer bow has an ability to delay the rupture of the cargo oil tank to some extent, and it might prevent the rupture of the cargo oil tank.

(4) The maximum safe velocity (critical velocity) of the striking ship under which velocity the striking ship does not penetrate cargo oil tank of the struck tankers is estimated. According to the FEA results, the buffer bow has an effect to increase the maximum safe velocity by 77% as compared to that for the standard bow in the collision scenario that a VLCC in ballast condition perpendicularly collides with the midship region of another anchored VLCC in full loaded condition. This indicates that a VLCC with buffer bow navigating at velocity of 11kt or lower does not penetrate oil cargo tank D/H VLCC in the same collision scenario.

(5) A simplified ship collision analysis tool (SSCAT) was developed in this study and its accuracy was investigated by comparing with results obtained from FEA. From these investigations it can be said that the SSCAT estimate the contact force, the total absorbed energy by structural deformation with reasonable accuracy although small discrepancies can be seen. It is noted that at the moment SSCAT is only applicable for perpendicular collision due to the limitations of the internal dynamic analysis procedure. The extension for the oblique collision is under construction and will be implemented in the next version of the SSCAT.

(6) The failure probability of the rupture of cargo oil tank of the VLCC given a perpendicular collision is estimated using SSCAT in the Monte Carlo Simulation approach. Various uncertain factors such as the collision point of the struck ship, the velocity and the size of striking ships are treated as random variables and the analysis of a large number of collision scenarios is carried out. It is estimated from this analysis that the buffer bow has the effect to decrease the probability rupturing of oil cargo tank by about 19-24% as compared to the standard bow.

(7) Risk analyses of oil spill from struck VLCC in the local sea area of Great Belt and Oresund was carried out. It is estimated from this analysis that a buffer bow has a risk reducing effects about 37,000-42,000 [US$] for one single striking ship if the lifetime of the ship is assumed to be 30 years. Therefore, it can be said that introducing a buffer bow to a ship passing through Great Belt and Oresund is cost-effective only if the difference between the initial shipbuilding cost and the repair costs between the standard bow and the buffer bow is smaller than 37,000-42,000 [US$/ship].
REFERENCE


IOPCF (2006). “International Regime for Compensation for Oil Pollution Damage”


**Collapse Mechanism of the Buffer Bow Structure on Axial Crushing**

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**ABSTRACT**

The purpose of this study is to obtain experimental data on the crushing of buffer bulbous bow, especially to investigate the collapse mechanism of buffer bulbous bow structure. A series of experiments in quasi-static conditions were carried out on large-scale bulbous bow models that have conical shape and simplified cross section. Two kinds of transversely stiffened models as the buffer bow and a longitudinally stiffened model as a conventional bow were axially collapsed by rigid board. The load-displacement curve, mean crushing strength and also total energy absorption capability were investigated. Finite Element Analysis (FEA) and simplified analysis were conducted to compare with experimental results. As a result, basic collapse mechanism of buffer bulbous bow was clarified in comparison with longitudinally stiffened one. Moreover, the accuracy of simplified analysis to estimate mean crushing force was validated by comparing with experimental results.

**KEY WORDS:** Ship collision; buffer bow; collapse mechanism; mean crushing force; buckling; folding; axial crushing

**INTRODUCTION**

The adoption of double hull system in side hull of oil tanker has been recognized as an effective countermeasure to prevent a disastrous oil spill from a struck oil tanker. ASIS (1998), however, studied structural safety of oil tanker to prevent oil spill from tanker and concluded that the safety has not yet been guaranteed as a satisfactory level in the case of high-energy collision. In fact, in 2001, Double hull tanker, "Baltic carrier", struck by bulk carrier "Tern" in Baltic Sea had spilled about 2700 tons of oil, and caused disastrous environmental pollution in neighboring sea. Then further countermeasure was recognized to be necessary to reduce the risk of oil spill from struck oil tanker. The buffer bow concept, one of which is adopting a transverse stiffening system instead of longitudinal one, was proposed as the effective countermeasures to further reduce the risk of oil spill from struck tanker by ASIS (1998), Kitamura (2000).

Based on the research contribution by ASIS, Buffer bow project (BBP) has been launched out since 2001 by National Maritime Research Institute (NMRI) sponsored by Japanese Ministry of Land Infrastructure and Transport (MLIT). The objective of this project is to develop the practical design of buffer bow, and also to investigate the feasibility whether buffer bow design could be adopted for regulation. This paper describes a part of results of the project so far.

In case of ship-to-ship collision, bulbous bow and upper stem of colliding ship is likely to contact with the side structure. In view of the prevention of disastrous sea pollution, oil spill from oil tanker in full loaded condition should be avoided. For the oil tanker in full loaded condition, a colliding ship in ballast condition is supposed to be most menace because vertical location of the upper stem of striking ship is supposed to be above the deck of struck ship, and bulbous bow is likely to penetrate the vertical center of side hull of struck ship. Thus, in this study the main focus is laid on the bulbous bow of striking ship as an objective of buffer bow structure for the beginning of the project.

In order to analyze the ship-to-ship collision, FEA could be strong tool; Kitamura (2000), Kitamura (2002). However simplified analysis method is strongly needed for various collision scenarios because FEA is time consuming.

As for the simplified method, Minorsky (1959) proposed the method to estimate the absorbed energy by consideration of the volume of structure. Later a lot of simplified method to predict the mean crushing load for circular/square tube or regular plate with/without stiffener were presented by consideration of consumed energy in rigid-plastic deformation so far; Alexander (1959), Wierzbicki (1992) and so on. However, as far as axial crushing of bulbous bow structure few research was conducted; Lee (1983), Kierkegaard (1993), Ohtsubo and Suzuki (1994), Lehmann et al (1995) and so on.


In most of the simplified analysis mentioned above, energy dissipation method using rigid-plastic analysis has been adopted on the assumption of specific deformation mode. Thus to grasp deformation pattern and/or collapse mechanism is very important to develop the simplified analysis for buffer bulbous bow. The aims of
this study are not only to obtain experimental data on the crushing of
buffer bulbous bow, but also to investigate the collapse mechanism of
buffer bulbous bow structure. Those data will be intended to improve
accuracy of simplified analysis in future works.

Authors have conducted a series of quasi-static crushing tests and
numerical analysis using small-scale bow models with adopting the
transverse stiffening system. Endo and Yamada (2001) adopted the
models with simplified transverse section (circle and cruciform).
Moreover Endo and Yamada et al (2002) extended the transverse
section of the model to the more actual section (ellipse and cruciform).
Basic characteristic of buffer bow model have been investigated by
those studies. Accuracy of FEA was validated and it is found that FEA
could estimate the experimental result with reasonable accuracy
although small discrepancy are observed in the timing and fluctuation
of peak load. Moreover it is found from those studies that using thin
plate (<4mm) may cause significant weakness of crushing strength of
models due to welding induced residual stress, and that it is
appropriate to use thicker plate in order to reduce those effect.
However due to the limitation of test machine and costs few tests with
large-scale models have been conducted so far in regarding crushing
of bow. In this study, therefore, a series of quasi-static experiments in
a quasi-static condition were carried out with using large-scale
bulbous bow models that have conical shape and simplified circular
cross section. The effect of residual stress and model-scale is expected
to become relatively smaller as the model size get close to the actual
ship size.

QUASI-STATIC EXPERIMENT

Large-Scale Bulbous Bow Model

In order to investigate basic collapse mechanism of buffer bow
structure a series of quasi-static collapse experiments were carried out
using three kinds of bulbous bow models. A large-scale model was
adopted as compared with the experiments (Endo and Yamada, 2001)
to reduce the effects of residual stress in fabrication.

The model was designed as it has the conical shape and structural
arrangement is similar to the realistic ship. Fig. 1 represents the
common configuration of three models. All of the models have the
same shape although arrangement of webs and stiffeners inside shell
are different. In each models webs and stiffener were added in
addition to the common configuration. Two of the three models are
adopting transverse stiffening system as a prototype buffer bulbous
bow model. Whereas the other model were adopting longitudinal
stiffening system in order to compare with buffer bow model. Fig. 2
shows the basic cross sectional view of each model. In order to
validate simplified analysis the sectional view of the model was
simplified as its section has few intersections such as X-shape or T-
shape. Centerline web and horizontal web were stiffened at each
frame by angle plate with thickness of 7mm.

One of the peculiarities of buffer bulbous bow is adopting ring
frame as a part of the transverse stiffening system. Ring frame is
expected to be advantageous in the performance of buffer bow as it

The models were made of mild steel and tensile coupon tests were
also conducted to obtain material characteristics. The results of those
tests are shown in Table 1, where $\sigma_y$ denotes the yield stress $\sigma_u$
denotes the ultimate strength and $\varepsilon_b$ denotes the failure strain in the
gauge length of 200mm.
Experimental Method

The experiments were conducted supposing that a ship collides with another ship in right angle. As far as model test on ship collision was concerned, most of the research treated bow as rigid because it is supposed to have enough strength as compared with side hull of struck ship. However in case of collision with buffer bulbous bow, we should treat bow as elastic-plastic. Ship collision mechanics, however, is very complicated and our urgent objective is to investigate the basic collapse mechanism of buffer bulbous bow. Thus, the side hull of the struck ship was assumed to be rigid in this study. Moreover the effect of ship motion, i.e. external dynamics, was not accounted for.

Experiments were conducted by using the 30 MN universal machine at the Public Work Research Institute (PWRI) in Tsukuba-shi, Japan. The models were fixed at the bottom, and were compressed quasi-statically from the top of the bow by a rigid board as shown in Fig. 3. Displacement of the rigid board, load and bending strains of plate were measured.

ANALYSIS

Finite Element Analysis

In order to validate the accuracy of FEA and also investigate the axial crushing mechanism of bulbous bow, numerical simulation corresponding to the experiment were conducted by using LS-DYNA. In creating FE models, all the plates between frames were equally divided into ten elements in order to simulate the folding behavior of the shell. All the members including flange of ring frame are modeled by shell elements in order to accurately simulate the contact of structural members inside the bow model. As a result, the number of nodes and elements become about 20,000 in all models. Belytschko-Tsay shell elements are used for element formulation type and piecewise linear plasticity (MAT24) was used for material model. In this study initial imperfection and residual stress were not included. Strain rate effect is not taken into account because the experiment was conducted quasi-statistically. Therefore strain rate parameter for Cowper-Symonds model, & C and P, is set to 0. The loading speed is set to 2m/s after confirming the effect of loading speed as discussed later in the paper.

Simplified Analysis

In this study, the method by Endo & Yamada (2001) was applied to the bulbous bow models. In this method the model by Wang et al (1995) and the model by Lehmann (1995) were combined. Wang et al (1995) developed the formula to estimate mean crushing load for basic intersection elements, such as L-element, T-element and X-element. The mean crushing load \( P_m \) for X-shaped element was expressed as

\[
P = a_\alpha a_\beta \sigma_0 A
\]

where

\[
\alpha = \frac{1}{2} \left( \sin^2 \theta - \cos^2 \theta \right), \quad \alpha = \frac{2.66}{\beta + 0.75}, \quad \beta = \frac{b}{t}
\]

\( \sigma_0 \) is the flow stress, \( \theta \) is an inclination angle of element from the crushing direction, \( A \) is a sectional area of the element, \( b \) is a breadth of the flange, and \( t \) is a thickness of the plate.

Lehmann (1995) developed the formula to estimate mean crushing load for conical shell, which is expressed as

\[
P_n = \frac{2\pi M_\alpha \pi R / H_1 + Q_1 + 2\pi N_\alpha H_1 Q_2}{Q_3}
\]

where

\[
Q_1 = \frac{\pi \alpha_0 \cos \alpha_0 \sin \alpha_0 \tan \theta \cos \alpha_0 - (0.5 \sin \alpha_0 + \beta_0)}{2}
\]

\[
Q_2 = \frac{\pi \beta_0 \sin \alpha_0 \cos \theta + \eta \cos \theta - \sin \alpha_0 + 0.5 \cos \theta}{2}
\]

\[
Q_3 = \frac{\pi \cos \alpha_0 \cos \theta + \eta \cos \theta - \sin \alpha_0}{2}
\]

\[
M_\alpha = \sigma_0 \beta_0 / 4, \quad N_\alpha = \sigma_0 \beta_0, \quad \xi = H_1 / H_2, \quad \eta = H_2 / H_1
\]

\( \alpha_0, \beta_0 \) : initial angle of the folding element

\( H_1, H_2, H_3 \) : folding length

In this study mean crushing load for web was derived from Eq.1 and that for shell was derived from Eq.2. In calculating the mean crushing load for web, the load was reduced according to the ratio of sectional area. i.e. : \( P_n = P_n \cdot A_{web} / A_{web+shell} \) where \( A_{web} \) is a sectional area for web, and \( A_{web+shell} \) is all of the sectional area for basic elements, such as L-, T-, and X-elements. The results of simplified method were compared with that of experiments.

DISCUSSION

Loading Speed in the Analysis

In order to confirm the effect of loading speed on the accuracy of FEA additional analysis for the model BC-G in varying speed of 5m/s, 2m/s and 1m/s was conducted. Load-displacement curve for 5m/s,
2m/s and 1m/s are shown in Fig. 4, Fig. 5 and Fig. 6 respectively. The comparison of history of total absorbed energy between experiment and numerical results are shown in Fig. 7. It is found in Fig. 7 that the effect of loading speed on the total absorbed energy is almost negligible. On the other hand it is found from Fig. 4 to Fig. 6 that the shape and fluctuation of the curve in the analysis get closer to those in the experiment as the loading speed become lower, but that the difference is not so significant when the loading speed is lower than 2m/s. Moreover it is noted that the number of the peak for the case of 5m/s is quite different from that for the experiment. Those differences were confirmed to take place when the loading speed is greater than 5m/s. The main reason is supposed to be because a couple of buckling phenomena take place simultaneously in lower loading speed due to inertia effect. It is found that the numerical result for the case of 1m/s is most similar to the experimental result. However taking into account the balance of accuracy and the efficiency of calculation, loading speed of 2m/s is adopted in this study.

Load-displacement Curve and Energy Absorption Capability

Load-displacement curve for the model BC-E, BC-G and BC-L are shown in Fig. 8, Fig. 9 and Fig. 10 respectively. By comparing these curves, it is found that maximum load for BC-L is about twice higher than that for other models due to adopting longitudinal stiffening system. It could be said that ships having a bulbous bow with longitudinal stiffening system can be a great menace for struck ship in case of collision because collapse or penetration of weaker structure go ahead. In view of buffer bow design, it is recommended that maximum load should be lower so as not to penetrate the side shell of struck ship, and that energy absorption capability should be higher in order to absorb the kinetic energy of striking (own) ship. Taking into these contrary design concepts, flat load-displacement curve is preferable for the buffer bow design. Within this study, the model BC-G is supposed to have the most suitable structural arrangement for buffer bow design of these three models.

Accuracy of Finite Element Analysis and Simplified Analysis

Load-displacement curve estimated by FEA and simplified method are also show in Fig. 8 through Fig. 10 as compared with experimental results. Those figures show that mean crushing load estimated by the FEA and simplified method by Endo gives fairly good agreement with that by experiment although small discrepancies are observed in the fluctuation and timing of peak/through load especially for the model BC-L. It is assumed to be due to effect of initial imperfection, which may cause significant effect on the number of folding between frame spaces. As indicated by Ohtsubo and Suzuki (1994), several buckling or folding take place between two transverse frames if frame spacing...
is much longer than folding span like in the structure with longitudinal stiffening system. In the cases of longer frame spacing, first folding between the transverse frames is supposed to affect the collapse mode, folding span and crushing strength of the following folding of the plate. Therefore accurate estimation of the timing of peak load by FEA becomes more difficult than the case for the structure with transverse stiffening system. To raise the accuracy of FEA by considering the effect of initial imperfection and/or other cause will be open to the future work. Of the numerical result for three models, the number and timing of peak/trough for the model BC-G give the better agreement with the experiment than those for the model BC-E and BC-L. The deformations of the model BC-G calculated by FEA are shown in Fig. 19, where (a)-(d) in the both figure represent the time just after the crushing load become peak value and correspond to the (a)-(d) in Fig. 9. Bending strain of the some strain gauges (P1 and P3, see Fig. 13) on the outer shell in the experiment were compared with those in the analysis as shown in Fig. 12, where bending strain $\varepsilon_{\text{bend}}$ represent the strain calculated by following equation.

\[
\varepsilon_{\text{bend}} = \frac{\varepsilon_f - \varepsilon_b}{2}
\]

Eq. 2

where $\varepsilon_f$ denotes the strain on the front surface of the outer shell, $\varepsilon_b$ denotes the strain on the back surface of the outer shell. Thus roughly speaking, positive value of $\varepsilon_{\text{bend}}$ represent the outward buckling of the outer shell, whereas negative value of $\varepsilon_{\text{bend}}$ represent the inward buckling of the outer shell by canceling the strain component of pure compression.

In order to accurately simulate the folding mechanism coincidence of plus of minus sign is more important than the absolute value of bending strain itself. It is found in Fig. 12 that FEA could estimate the direction and magnitude of the plate folding for the model BC-G, which is adopting transverse stiffening system, with fairly good accuracy. The energy absorption capability for the model BC-G is shown in Fig. 14. It is found in Fig. 14 that almost good agreements are achieved between experiments, FEA and simplified analysis as well as load-displacement curve. It is noted that the timing of the folding is a little delaying as compared with experimental one in the early stage. It is confirmed that those tendency is also observed at the case other than model BC-G. This is supposed because buckling take place a little earlier in actual model than in present analysis due to existence of initial imperfection. Taking into account those comparison results, it could be said that FEA could estimate the load, absorbed energy, collapse mode and folding of the shell plate in the experiments with reasonable accuracy although small discrepancies are observed.

![Fig. 8 Load-displacement curve for BC-E](image)

![Fig. 9 Load-displacement curve for BC-G](image)

![Fig. 10 Load-displacement curve for BC-L](image)

![Fig. 11 Comparison of total absorbed energy (Exp.)](image)
Folding Mechanism

By observing each experiment carefully, folding mechanism of the outer shell in BC-G was investigated with the aid of the FEA result of BC-G, which represents good agreement with the experiment. The folding mechanism of the outer shell for BC-G is clarified as follows.

As contrasted with the overall buckling of the structure, buffer bow collapsed progressively from the top of the bow mainly because cross sectional area increase from the top of the bow. The outer shell collapsed almost in axis-symmetry. After an inward buckling of spherical shell in section 1, the buckling of outer shell was observed twice in each section respectively. i.e. Two folding between two transverse frames were observed as far as present models used in this study. Further study, however, is necessary to predict the number of folding between transverse frames for various plate thickness, plate width, frame space and also radius of transverse section. Two photos taken at the experiment are shown in Fig. 15 as an example. Fig. 15(a) shows the second buckling of outer shell in the section 3. Fig. 15(b) shows that section 4 is getting into the inside of the model.
This indicates that two folding occur across the ring frame, i.e. buckling just below the ring frame occurs before the folding above the ring frame finish completely. Moreover two folding element model proposed by Wierzbicki (1992) is applicable by some modification from circular tube to conical shell. As the section (i) goes downward, the first buckling in section (i+1) begins to return into the inside of the outer shell due to the tension from section i. Fig. 16 (b) shows the return of the first buckling in section (i). As the rigid board further goes down, structure above section (i) begins to get into the inside of the model as shown in Fig. 16 (c). Finally the entire crushed zone was getting into the inside of the model as shown in Fig. 16 (d). Afterwards, inward buckling (second buckling in a section) and outward buckling (first buckling in a section) occurred repeatedly.

On the basis of experimental and numerical investigation, the direction and locations of buckling for the models are summarized in Fig. 17 in order to associate the buckling of outer shell with the load-displacement curve. Moreover, Fig. 18 illustrates the detail of the folding mechanism of outer shell in a section. In Fig. 17, plastic hinge made in the folding process is represented by square mark, and alphabetical characters from (a) to (e) correspond to those in Fig. 9.

As an example, shell deformation at (c1) and (c2), which corresponds to the formation of load peak at (c) in Fig. 9 (b), is accounted for as an example based on the Fig. 17. After the folding at (b2) finished completely, the load began to increase until local buckling at (c1) occurred. When buckling at (c1) occurred, the shell at (c1) began to fold inside with the decrease of the load. Then before folding at (c1) finished completely next buckling at (c2) occurred, what we call “two folding mechanism”. This indicates that buckling strength at (c2) is weaker than folding strength at (c1). As the folding of (c1) and (c2) proceed, load decrease gradually. “Two folding mechanism” across the ring frame was observed in other section of BC-G and in BC-E, but not in BC-L. Increasing of folding strength at (c1) compared with that at (c2) was assumed to be the reason for occurrence of “two folding mechanism”. Two reasons for the increasing of folding strength at (c1) could be assumed. One is that crushed zone, which was getting into the inside of the bow, disturbed the inward folding of the outer shell at (c1) as shown in Fig. 18(e). The other is that two plates folding, BC and CF as shown in Fig. 18(d), was resisting to the load in contrast to one plate folding as shown in Fig. 18(a). Therefore, it is concluded from those observation that the shell of bulbous bow with transverse stiffening system likely to collapse with “two folding mechanism” on axial crushing. The generalization of these assumptions, however, will be conducted in the future work by FEA.
Fig. 17 Buckling direction of outer shell

Fig. 18 Folding mechanism of outer shell in a section

Effect of Outer Shell and Inner Structure
By the result of FEA, shared ratio of the load and share of energy absorption between outer shell and inner structure for the model BC-G are shown in Fig. 20 and Fig. 21 respectively, where outer shell is composed of the outside plate of the model with circular transverse section, and the inner structure is composed of all of the webs and stiffeners inside the outer shell. It is assumed that in Fig. 20 that shared ratio when displacement equal to 0 is set to 50% because magnitude of load is 0 for both outer shell and inner structure. Fig. 20 gives two important information. Firstly, the shared ratio of the load by outer shell is higher than that by inner structure. It could be said that the effect of outer shell is dominant for the axial crushing strength of the model BC-G. It is also found in Fig. 21 that more than 2/3 of the energy is absorbed by the outer shell. One of the main reasons is supposed because the plate thickness of the outer shell (10t) is a little thicker than that of the inner structure (7t). These tendencies, however, is not generalized at this stage because the material strength of shell is a little stronger than that of web. The shared ratio of load by outer shell was about 60% to 80% for the model BC-G. Secondly, the ratio by web increase/decrease when that by shell decrease/increase respectively. This means that the web was playing a complementary role of the load against the shell.

Fig. 19 Deformation of the model BC-G (FEA) ; Marks (a)-(e) correspond to those in Fig. 9; Right figure with outer shell hidden.
CONCLUSION

The prototype buffer bow, which was stiffened by ring frame, was axially collapsed with a quasi-static condition. The following conclusions are derived. (1) Buffer bow was expected to collapse progressively from top of the bow on axial crushing. (2) The results obtained by FEA and the simplified analysis have been confirmed to be reliable in estimating the performance of buffer bow structure. (3) Buckling and folding on outer shell occurred twice in a section. In most of the cases, the first buckling is “outward buckling”, and the second buckling is “inner buckling”. (4) On the basis of experiments and FEA on BC-G model, which has simple cruciform web section, “Two folding mechanism” was observed across the ring frame. (5) As far as load-displacement curve, the shell was playing a dominant role on the axial crushing of the bulbous bow. To modify the simplified method will be conducted by considering the results obtained by this study in the near future.

ACKNOWLEDGEMENTS

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REFERENCES


Kitamura, O (2002). FEM Approach to the Simulation of Collision and Grounding Damage, Marine Structures, 15, pp.403-428


Experimental and Numerical Study on the Collapse Strength of the Bulbous Bow Structure in Oblique Collision

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The purpose of this paper is to investigate the collapse strength and the mechanism of the bulbous bow structure in case of an oblique collision. In this study quasi-static experiments were conducted using two types of large-scale bulbous bow models supposing the scenario that a ship collides with another ship at oblique angle. One of the models is a prototype buffer bow adopting transverse stiffening system, and the other model is a standard bow adopting longitudinal stiffening system. Each model was collapsed by a thrusting rigid board while being subject to a combined action of compressive force and bending moment.Collapse mechanism, load-displacement curve and energy absorption capability of the buffer bow structure were investigated as compared with those of standard bow structure. Nonlinear Finite Element Analysis (FEA) corresponding to the experiments was also conducted, and it is confirmed that fairly good agreements were achieved between FEA and experiments. It is also found from these investigations that buffer bow structure is expected to be efficient to reduce the risk of oil spill especially in case of oblique collision.

1. INTRODUCTION

A double hull (D/H) design has become de facto standard for preventing oil pollution from struck oil tankers. However it is found from recent research that the crashworthiness of D/H tanker is still not improved to a satisfactory level (ASIS 1996, Kitamura 2000). It is pointed out from those researches that the relative strength of the striking bows is too high to avoid the penetration of the oil tanks of struck D/H tankers. Thus in order to further reduce the risk of oil spill, the Buffer Bow Project (BBP) was carried out by National Maritime Research Institute (NMRI) in Japan with financial support by Japanese Ministry of Land Infrastructure (MLIT). In this project the main focus is laid on the bulbous bow structure since it is most likely to make a first contact with the struck ships and to penetrate the outer shell and oil tanks (Yamada 2004).

A lot of experimental and theoretical researches have been carried out concerning crushing strength of bulbous bow structures. However most of the research focused on the axial crushing strength assuming right angle collision because it is considered to be the most severe condition for the crashworthiness of struck ships (Pedersen 1993, Kierkegaard 1993, Ohtsubo 1994, Lehman 1995, Wang & Suzuki 1995, Lutzen 2000, Endo & Yamada 2001). In BBP as well as those researches, axial crushing strength and mechanism of prototype buffer bow structure is compared with those of standard bow had been investigated by conducting a series of large-scale model tests and comparing the results with those obtained by the finite element analysis (Endo & Yamada 2000, Yamada 2003).

On the other hand, as we can see the tendency in the statistics of ship collision analysis, oblique collisions take place more often than right angle collisions. Moreover even in the case of right angle collision, it should be categorized to a kind of oblique collision when a ship collides with another ship with forward speed. Collapse mode in case of those conditions is different from that in case of right angle collision as the bow structure is subject to the combined action of compressive force and bending moment. It is important for the buffer bow structure to confirm its applicability even in those conditions. Few research, however, has been conducted so far concerning crushing strength of bow structure in an oblique collision. The authors have carried out a series of model test using large-scale bulbous bow models that are supposed to collide at an oblique angle. This kind of test had never been carried out in the laboratory due to its physical difficulties in arranging the test setup. The purpose of this paper is to investigate the collapse strength and mechanism of the bulbous bow under combination of compressive force and bending moment, and also to clarify the difference of collapse strength and mechanism between buffer bulbous bow and standard bulbous bow.

2. QUASI-STATIC EXPERIMENT

In order to investigate the collapse mechanism of the bulbous bow under horizontal bending moment, quasi-static experiments were conducted using two types of large-scale bulbous bow models (see Fig. 1 - Fig. 4).
2.1 Bow models

The bulbous bow models are made of mild steel and have an almost same size and structural arrangement. They are typical of actual fishing ship about 500 in gross tonnage. The size of the models also correspond to about half scale of large tankers although the scantling and structure in detail is slightly different. The models are fabricated in the same way as actual ships in the shipyard. Both models have the same outer shape although inner structure is different from each other. The size and schematic configuration of the models are shown in Fig. 1.

One of the models, “BB-D”, was designed as a buffer bow which was expected not to penetrate the side shell of struck ship in case of collision. The model BB-D adopted transverse stiffening system to reduce the structural members arranged in the forward direction and consequently to reduce the axial crushing strength. Moreover in order to accelerate the collapse by bending, the model BB-D adopts Low Yield Point steel (LYP100) as a material at the small area of outer shell between Fr.94 and Fr.95 instead of mild steel. LYP100 has the yield strength of about 100MPa with longer elongation capability than mild steel as shown in Fig. 5. It is supposed that, in view of local strength of stiffened plate, the area where LYP100 is used should be required thicker plate than the area where mild steel is used in order to maintain same safety level. Due to the simplicity in fabrication and welding, however, same plate thickness with surrounding plates is used for the plate of LYP100 in model BB-D. The effect of thickness increase at the area where LYP100 is used will be investigated by FEA.
On the other hand the other model, “BB-E” is designed as a standard bow adopting longitudinal stiffening system. Schematic view of inner structure for model BB-D and model BB-E are shown in Fig. 2 and Fig. 3 respectively. Material strength and failure strain obtained by a series of tensile coupon tests are shown in Fig.5 and summarized in Table 1.

Table 1 Material strength and failure strain obtained from tensile coupon test

<table>
<thead>
<tr>
<th>test specimen</th>
<th>t [mm]</th>
<th>( \sigma_y ) [MPa]</th>
<th>( \sigma_u ) [MPa]</th>
<th>( \varepsilon_f )</th>
</tr>
</thead>
<tbody>
<tr>
<td>MS-8t</td>
<td>7.8</td>
<td>290</td>
<td>432</td>
<td>0.35</td>
</tr>
<tr>
<td>MS-9t</td>
<td>8.5</td>
<td>287</td>
<td>449</td>
<td>0.35</td>
</tr>
<tr>
<td>MS-11t</td>
<td>10.9</td>
<td>302</td>
<td>440</td>
<td>0.33</td>
</tr>
<tr>
<td>LYP-9t</td>
<td>9.1</td>
<td>94</td>
<td>272</td>
<td>0.47</td>
</tr>
</tbody>
</table>

MS : mild steel, LYP : LYP100, \( \sigma_y \) : yield stress, \( \sigma_u \) : ultimate stress, \( \varepsilon_f \) : failure strain

2.2 Experimental conditions

Experiments were conducted using a test machine with the maximum loading capacity of 32MN at Seismic Resistance Experiment Center in Aichi Institute of Technology in Japan. Fig. 6 shows illustration of experimental set up, where \( \delta \) denotes the vertical displacement of rigid board, \( \mu_1 \) and \( \mu_2 \) denote the coefficient of friction between the upper rigid board and models, and the lower rigid test floor and models respectively, X and Y denotes the global coordinate system, and x and y denotes the local coordinate system. The Fr.93 section of the bow model was fixed to the rigid block with the initial collision angle of 72 deg as shown in Fig. 6.

The test was to investigate the situation of combined crushing & bending. The test machine required that no horizontal force takes place in this experiment. Therefore a set of linear roller bearings were arranged between the rigid block and the test floor so that the rigid block moved smoothly in the positive Y-direction as the upper rigid board goes down.

It was confirmed by the pre-experiment that friction coefficient of bearings is about 0.01.

The two types of bow models were collapsed quasi-statically under the vertical thrusting load driven by the upper rigid board. The X- and Y- directional loads, X-directional displacement of upper rigid board and Y-directional displacement of rigid block as well as a set of strains on the models were measured.

![Fig. 6 Schematic view of experimental condition (unit mm)](image)
2.3 Collapse mechanism

It was observed in the experiment that the overall collapse mechanism and local deformation patterns were almost the same in both models BB-D and BB-E. In both cases “overall bending mode” was observed to take place (See Fig. 7 and Fig. 32). Collapse mechanism of the bow in bending mode is as follows. At the first stage of the experiment, bow tip contacting with the rigid board began to dent locally. At the next stage buckling of outer shell at heavier compression side between Fr.94 and Fr.95 became dominant. As the rigid board went down further, the outer shell began to fold. Finally, structure between Fr.94 and Fr.95 at heavier compression side collapsed, and consequently the bow collapsed with a large overall bending deformation near the root of bow bulb (see Fig. 7).

As for local folding pattern of outer shell in heavier compression side, there were some differences between the two models. In model BB-D, only inward folding of outer shell took place (see Fig. 8). On the other hand, in model BB-E both inward and outward folding of outer shell took place at heavier compression side. It was observed that inward and outward folding in the model BB-E took place alternatively across the longitudinal frame as shown in Fig. 9, where (+) denotes outward buckling and (-) denotes inward buckling. The occurrence of inward folding is supposed to be mainly due to initial deflection induced by fillet welding between outer shell and webs. The occurrence of outward folding is supposed to take place due to the elastic stability. At the outer shell between inward folding and outward folding, some of the large crack openings were observed along the longitudinal frame (see Fig. 9 and Fig. 10). Those crack openings are supposed to take place due to shear force caused by alternating folding. Among all the plates between Fr.93 and Fr.95, only the plate between L.W.3 and L.W.4 made two cycles of buckling/folding. This is presumably due to the large curvature of the plate and excessive concentration of plates welded close to the boundary. It is supposed that the second buckling between Fr.93 and Fr.94 does not have a significant effect on the test results because it took place at the later stage of the experiment.
2.4 Collapse strength

Histories of reaction force and total absorbed energy in relation to rigid board displacement \( \delta \) for the model BB-D and the model BB-E are shown in Fig. 11 and Fig. 12 respectively. It is found in Fig. 11 that the model BB-E has about 50% higher strength than the model BB-D.

It is also found that, in case of the model BB-D, the reaction force drops down significantly just after the reaction force reaches its maximum while the reaction force decreases gradually in case of the model BB-E. This is presumably due to that the model BB-E with the longitudinally stiffened panels does not lose their resistance suddenly after maximum force take place because of existence of plastic deformation of the longitudinal stiffeners as well as the outer shell. On the other hand the resistance of the model BB-D after maximum force is mainly made by the plastic deformation of the outer shell.

It is interesting to note that in Fig. 11 average reaction force after \( \delta = 0.2 \)m for the model BB-D is about 2MN and almost keeps constant value. This is mainly caused by the plastic deformation of the outer shell.

Maximum reaction force and total absorbed energy up to the displacement of 0.5m are shown in Table 2. It is found in Table 2 that maximum reaction force of the model BB-D is about 40% lower than that of model BB-E and energy absorption capability of the model BB-D is about 42% lower than that of the model BB-E. In case of actual ship-ship collision, one side with weaker structures collapse at the contact area while the other side with stronger structure does not collapse. In view of preventing the damage on the struck ship it is one of the important performances to reduce the maximum reaction force of striking ship. Thus it is supposed that the model BB-D, which is designed as a kind of buffer bow structure, has a significant effect to reduce the threat as compared with the model BB-E, which is designed as the existing ship structure in case of oblique collision as well as right angle collision (Yamada, 2003).

![Fig. 11 Comparison of reaction force](image)

![Fig. 12 Comparison of total absorbed energy](image)

**Table 2 Comparisons of maximum reaction force and total absorbed energy**

<table>
<thead>
<tr>
<th></th>
<th>Exp.</th>
<th>FEA</th>
<th>Error [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>BB-D</td>
<td>( P_{\text{max}} ) [MN]</td>
<td>4.61</td>
<td>4.75</td>
</tr>
<tr>
<td></td>
<td>( E_{0.5} ) [MJ]</td>
<td>1.15</td>
<td>1.23</td>
</tr>
<tr>
<td>BB-E</td>
<td>( P_{\text{max}} ) [MN]</td>
<td>7.31</td>
<td>6.63</td>
</tr>
<tr>
<td></td>
<td>( E_{0.5} ) [MJ]</td>
<td>2.00</td>
<td>2.37</td>
</tr>
</tbody>
</table>

\( P_{\text{max}} \): Maximum Reaction Force  
\( E_{0.5} \): Total Absorbed Energy up to displacement is 0.5m

Longitudinal distribution of section modulus against horizontal bending moment around center line for both the model BB-D and BB-E are shown in Fig. 13. Total section modulus is divided into two components of the outer shell and the inner structure, and section modulus of the outer shell is common with both models.

![Fig. 13 Longitudinal distribution of section modulus](image)
It is found in Fig. 13 that, in the present models, section modulus of the model BB-E is slightly larger than that of the model BB-D due to the effect of inner structure. It is also found that section modulus at Fr.97 shows higher value than that at surrounding frames due to the existence of horizontal flat plate at the forward of Fr.97 as shown in Fig. 2 through Fig. 4. It is also found in Fig. 13 that against the horizontal bending moment the effect of outer shell on the section modulus is much more dominant than that of inner structure mainly due to the larger thickness and the distance from neutral axis in the outer shell than in the inner structure.

Based on the simple beam theory, approximated longitudinal distribution of bending stress, normal stress and combined total stress acting on the outer shell of heavier compression side for the model BB-D and the model BB-E are shown in Fig. 14 and Fig. 15 respectively. Axial force (F) and bending moment (M(x)) acting on each transverse section is estimated by following equations.

\[ F = P \sin \theta \]  
\[ M(x) = (x - dx)P \cos \theta \]

where \( P \) is the reaction force normal to the contact surface, \( x \) is an axial distance from top of the bow to the particular section, \( dx \) is axial shortening of the bow tip by denting, \( \theta \) is a collision angle of 72 deg. In Fig. 14 and Fig. 15, open marks represent the result of bending stress by assuming intact of bow tip. On the other hand filled marks represent the result of bending stress by assuming denting of bow tip, where \( dx \) is assumed to be 0.2m based on the observation in the experiment. As the magnitude of the reaction force \( P \) in equation (1) and (2), 1.0 MN is employed so as to calculate stress distributions in elastic range without considering buckling of the shell.

It is found in those figures that, due to the effect of combined action of normal stress and bending stress, total stress between Fr.94 and Fr.95 is the highest in both models. It is also found that the decrease of bending and total stress due to denting of bulb tip is more significant at the bulb tip than at the root of the bulb. In the experiment severe collapse of the outer shell did not take place between Fr.95 and Fr.96 but took place between Fr.94 and Fr.95. It is also supposed that the adoption of LYP100 will accelerate the deformation on outer shell of model BB-D between Fr.94 and Fr.95 in the plastic region.
simulate buckling and folding mechanisms accurately. As a result, each bow model consists of about 14000 nodes and elements. Belytschko-Tsay shell elements having 1 integration point in plane and 3 integration points through thickness (Brown, 2002) are employed. Finite element models of the bow models are shown in Fig. 1 to Fig. 3. The loading board and test floor are both modeled as rigid and frictions with the bow models are taken into account as shown in Fig. 6. Coefficients of friction are shown in Table 3. Stress-strain relationships are defined with strain hardening effect by giving piecewise linear plasticity curves, which was obtained by a series of uniaxial tensile coupon tests of the material. In this analysis strain rate effect is disregarded since the experiment was conducted quasi-statically. Material properties used in the analysis are shown in Table 3.

<table>
<thead>
<tr>
<th>Name</th>
<th>Sign</th>
<th>Value</th>
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<tbody>
<tr>
<td>Young’s Modulus [GPa]</td>
<td>E</td>
<td>206</td>
</tr>
<tr>
<td>Poisson's ratio</td>
<td>ν</td>
<td>0.3</td>
</tr>
<tr>
<td>Mass Density [ton/mm³]</td>
<td>ρ</td>
<td>7.87e-9</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion [1/deg]</td>
<td>α</td>
<td>1.21e-5</td>
</tr>
<tr>
<td>Coefficient of Friction between Model and Rigid board</td>
<td>µ₁</td>
<td>0.3</td>
</tr>
<tr>
<td>Coefficient of Friction between Model and Test floor</td>
<td>µ₂</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Some analysis parameters, which were uncertain previously, were investigated and quantified in the preliminary analysis as mentioned in section 3.1.

### 3.1 Effect of loading speed

Firstly, the effect of loading speed of the rigid board was investigated. Although the experiment was carried out quasi-statistically, some finite loading speed should be given in the analysis. Four cases of FEA were conducted with varying loading speeds of 0.5m/s, 1m/s, 2m/s and 10m/s. Histories of the reaction force obtained by FEA are shown in Fig. 16 as compared with that by experiment. It is shown in Fig. 16 that reaction force becomes larger as the loading speed increases mainly due to the inertia effect.

![Fig. 16 Effect of loading speed on the reaction force](image)

The histories of horizontal displacement of the rigid block in the case of model BB-D is shown in Fig. 17. It is found in Fig. 17 that, in the experiment, the rigid block began to move in the y-direction just after displacement of rigid board passed over 0.1m, which corresponds to the time when reaction force of the model BB-D in Fig. 16 drop down as the consequence of the buckling of outer shell in compression side. It is also found in Fig. 17 that as the loading speed becomes larger the movement of rigid block becomes smaller due to the effect of inertia. It is judged to be appropriate to employ the speed of 1.0 m/s or below in the present analysis. In this study 1.0 m/s is employed hereafter considering the efficiency of analysis time.

![Fig. 17 Effect of loading speed on the displacement-y of rigid block (BB-D)](image)
3.2 Effect of residual stress on the collapse mode

In the experiment both models collapsed in overall bending mode as shown in Fig. 7 and Fig. 32. The model BB-D collapses in the same mode as the experiment in FEA as shown in Fig. 18. However in FEA the model BB-E collapses progressively from top of the bow (defined as “Progressive Collapse Mode”) as shown in Fig. 30 and Fig. 31, which is different from the experimental results of overall bending mode (See Fig. 32). Some of the additional FEA with considering initial imperfection of the outer shell has been conducted, but it does not change the Progressive collapse mode and give the same collapse mode as experiment.

Thus it is supposed that this is mainly due to the effect of residual stress induced by welding, which decreases the buckling strength of the stiffened plate. Thus additional finite element analysis considering the effect of residual stress is conducted. Residual stress had been calculated in advance by giving thermal load at the node on the welding line so that maximum tensile residual stress at the welding line would be close to the yield stress (eg. Yao 1991). In this analysis the effect of order of the welding path are not considered, and it is assumed that thermal load given to each node are the same.

Contour of residual stress estimated by FEA is shown in Fig. 19. The z-directional distribution of x-directional residual stress at the small area along one typical frame space of the outer shell is shown in Fig. 20, where \( \sigma_x \) denotes the x-directional residual stress along the outer shell and \( \sigma_y \) denotes the yield stress of the material. The location of the small area is shown in Fig. 19 as black rectangle. The stress distribution obtained by the present analysis is close to the past researches (eg. Yao, 1991), and the estimated residual stress is judged to be reasonable.

Then finite element analysis is conducted adopting pre-estimated residual stress as the initial stress of the bow models. The results of deformation are shown in Fig. 27 through Fig. 29. It is found in those figures that the model BB-E collapses in overall bending mode and the collapse mode of the analysis with taking into account the effect of residual stress coincides with that of the experiments. Histories of reaction forces and total absorbed energies are shown in Fig. 21 and Fig. 22 respectively as compared with the result of the experiments and the former result of FEA. It is found in Fig. 21 that the reaction force in overall bending mode gradually goes down after reaction force reaches maximum value, which indicates the occurrence of overall buckling. Whereas, the reaction force in progressive collapse mode shows somewhat large fluctuation while maintaining the same force level. It is found in Fig. 22 that total absorbed energy is almost the same in both collapse modes.

In order to investigate the detail of energy absorption capability, total absorbed energy in the case of overall buckling
mode is divided into three components and they are compared. Energy ratio of outer shell top, outer shell aft and inner structures against total absorbed energy is shown in Fig. 23, where outer shell top is defined as the outer shell forward from Fr.97, and outer shell aft is defined as the rest of outer shell. It is found in Fig. 23 that 43% of the energy is absorbed by the denting of the bow tip, 23% by the outer shell aft and total 66 % by the outer shell while the rest 34 % of the energy is absorbed by the inner structure when displacement is equal to 0.6m. Thus it could be said that the effect of the outer shell at the bow top on the energy absorption capability is significant. It is mainly because the bow tip collapses more significantly than the other parts and the most of the plate at the bow tip is composed of thicker plate of 11mm. It is confirmed by the same investigation with overall buckling mode that those tendency and energy ratio for each components are almost same even in the case of progressive collapse mode. The reason why energy absorption is not significantly dependent on the collapse mode is supposed to be because about half of the energy is absorbed by the denting of bow tip that is common with both modes, and because the effect of energy absorption by other parts do not affect greatly although there is a small difference. However in view of preventing oil spill in case of collision the bow collapsing in overall bending mode is preferable because it results in larger contact area and is less likely to penetrate the side shell of the struck ship.

According to the investigation in this section, it is noted that the reaction force and total absorbed energy are not so sensitive to the collapse mode in the model BB-E. FEA result in case of overall bending mode, however, is judged as more proper for the model BB-E because they coincide with the experimental results.

3.3 Effect of low yield strength material (LYP100)

In order to investigate the effect of material LYP100 on the strength of the structures, additional FEA was conducted where the material LYP100 is replaced with the mild steel.

Deformation result of FEA is shown in Fig. 24. The collapse region of the outer shell became larger than the case using the LYP100 while keeping the overall bending mode. The result of reaction force is shown in Fig. 25 as compared with experimental results of both models and the numerical results of the model BB-D with LYP100. It is found in Fig. 25 that the effect of the buffer bows without LYP100 on the reduction of the maximum reaction force becomes about half as compared with that with LYP100.
As mentioned in 2.1, the LYP100 plate should have larger thickness than the mild steel plate according to the classification rules in order to maintain the equivalent local bending strength, \( \sigma_1 t_1^2 \). The equivalent thickness of the LYP100 plate can be calculated as:

\[
t_2 = \sqrt{\frac{\sigma_1 t_1^2}{\sigma_2}} = \sqrt{\frac{287 \cdot 8.5^2}{94}} = 14.85 \approx 15 \text{ mm}
\]

where \( \sigma_y \) and \( t \) denote yield stress and the plate thickness, subscript 1 and 2 denote mild stress and LYP100 respectively.

Then in order to investigate the effect of the plate thickness of LYP100 additional FEA with using plate thickness of 13mm and 15mm for LYP100 was conducted for model BB-D. Comparison of histories of reaction force is shown in Fig. 26. It is found from Fig. 26 that the effect of plate thickness of LYP100 on the bending strength of the bulbous bow is not so significant mainly due to the relatively small area of LYP100 plate. Therefore it could be said from the present investigation that the prototype buffer bow model using low yield point steel has a significant effect to reduce the menace in case of oblique collision even though correcting the plate thickness so as to maintain the equivalent local strength with the case using mild steel.

Fig. 24 Collapse of the BB-D model without the effect of LYP100 ( \( \delta = 0.4 \text{m} \) )

Fig. 25 Comparison of reaction force

Fig. 26 Reaction force of the model BB-D.
Fig. 27 Collapse in overall bending mode
(BB-E, FEA, top view, $\delta=0.8m$)

Fig. 28 Collapse in overall bending mode
(BB-E, FEA, side view, $\delta=0.8m$)

Fig. 29 Collapse in overall bending mode
(BB-E, FEA, side view, outer shell hidden, $\delta=0.8m$)

Fig. 30 Collapse in progressive mode
(BB-E, FEA, top view, $\delta=0.8m$)

Fig. 31 Collapse in progressive mode
(BB-E, FEA, side view, $\delta=0.8m$)

Fig. 32 BB-E model collapsing in overall bending mode
3.4 Discussion about the accuracy of finite element analysis

Histories of reaction force and total absorbed energy obtained by FEA are compared with those by experiment as shown in Fig. 11 and Fig. 12 respectively. It is found that there are small discrepancies in the curve shape of reaction force between FEA and experiment. FEA slightly overestimates the total absorbed energy in comparison with the experimental result. It is supposed to be caused by the reason that mesh size in present analysis is still large in the area of large deformation, and somewhat restricts the optimum deformation pattern.

The results of model BB-D are more similar to the experimental results than those of model BB-E. The FEA results of model BB-E has a small difference in detail as compared with experimental results on the collapse mode and fluctuation of reaction force even though the effect of welding induced residual stress is considered. It is supposed that this is because collapse behavior of structure with longitudinal stiffening system is more sensitive to residual stress and initial imperfection than that with transverse stiffening system. As for the deformation of the bulb tip, it is observed by comparing Fig. 7 and Fig. 26 that the deformation at bulb tip obtained by FEA is slightly larger than that by experiment. It is noted that the extent of the experimental deformation at bulb tip for the model BB-D and BB-E are observed to be almost same. It indicates that actual strength of bulb tip is stronger than estimation by FEA. It is supposed that structure consisting of round shell form have residual stress induced by metal forming process with/without heat input, and that residual stresses on outer surface and on the back surface may resist the denting of bow tip. Further investigation is needed in the future.

Although small discrepancies are found between experiment and FEA, it can be concluded that the reaction force and total absorbed energy can be estimated with reasonable accuracy.

4. CONCLUSION

In order to investigate the collapse strength of the prototype-buffer/standard bulbous bow structures in case of oblique collisions, quasi-static collapse experiments using two types of large-scale bulbous bow models and corresponding finite element analysis were conducted. One model (BB-D) was the prototype buffer bulbous bow with transversely stiffened, and the other model (BB-E) is the standard bow stiffened longitudinally. It is assumed in the study that the bulbous bow collides with the rigid flat side shell with the angle of 72 deg. Following conclusions are obtained by these investigations.

(1) In the experiment, both bow models collapsed in overall horizontal bending mode near the root of the bulb in spite of different stiffening system after the small denting of the bulb tip.

(2) Maximum reaction force of the buffer bow structure (BB-D) is about 40% lower than that of standard bow structure (BB-E). In view of preventing the penetration into struck ship it is one of the important performances to reduce the maximum reaction force between two ships. Thus it is expected that buffer bow structure has a significant effect to reduce the damage of the struck ship in case of oblique collisions.

(3) Although small discrepancies are detected between experiment and FEA, it can be said that for the level of reaction force and total absorbed energy, they can be estimated with reasonable accuracy.

(4) In the analysis of the model BB-E, it is confirmed that “progressive collapse mode”, which is different from the mode observed in the experiment, is obtained when the effect of residual stress induced by welding is not considered. In order to simulate the experimental collapse mode of the model BB-E, residual stress should be taken into account in the analysis. It is also confirmed that overall buckling mode could not be achieved by only using initial imperfection. Further investigation of the effect of residual stress on the collapse mode should be indispensable in collision angle other than 72 deg.

(5) As far as horizontal bending collapse of the bow structures in a collision angle of 72 deg, the level of reaction force and total absorbed energy is not sensitive to the collapse mode since about half of the total energy is absorbed by the denting of bow tip which has a thicker outer plate thickness than other part of the outer shell.

(6) The extent of deformation at the bulb tip obtained by the present FEA is slightly larger than that by the experiments. This is presumable due to the effect of residual stress induced by metal forming process of round bulb tip. Since horizontal bending moment acting on the bulb is sensitive to the distance from the tip of the bow, further investigations to accurately simulate the denting mechanism at the bow tip are needed considering the effect of residual stress induced by metal forming.

ACKNOWLEDGEMENT

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APPENDIX

Effect of time step size

Generally, finite element analysis using explicit code like LS-DYNA is time-consuming even though computer technology progresses nowadays. For example, it takes about 3 or 4 days to solve one case of analysis with 500,000 nodes of model. Thus in order to reduce the calculation time, increasing time step size is the one of the efficient way. In the LS-DYNA time step size Δt for the shell elements is defined based on the Courant condition (Hallquist, 1998) as follows.

\[
\Delta t = \frac{SF \cdot L}{E} = L \sqrt{\frac{\rho(1 - \nu^2)}{E}}
\]

where L is the smallest length of the element, E is Young’s modulus, ν Poisson’s ratio, and ρ mass density. SF is the scale factor for stability of calculation and is usually set to 0.9 as a default.

LS-DYNA has a function to use a numerical technique of “mass scaling” where time step size can be enlarged by artificially increasing material density of the elements if the inertia effect is not so significant (See detail in Hallquist, 1998). By using this function we can use larger time step size and can save the computational time. Minimum time step in the analysis is defined as an input of the absolute value of DT2MS [s] which user defines in advance.

If time step size of some elements calculated by the formula above is smaller than absolute value of DT2MS (|DT2MS|), time step size of the element is forced to be same value as |DT2MS| by increasing the density of the element. Negative value of DT2Ms is used in the present study where density is increased to only those elements whose time step would otherwise be less than that calculated by the formula above.

It is not recommended to use mass scaling function when the inertia force has a significant role during the calculation. From the viewpoint of engineering, however, mass-scaling function is worth using in order to get the solution within practical time while considering the balance of accuracy and calculation time. In this study mass-scaling function was used considering that experiment was conducted quasi-statically.

Detailed guideline for using mass scaling was not apparent and appropriate value of DT2MS is not clear. Thus in order to investigate the effect of time step size and to find the proper value of DT2MS, five cases of FEA for the model BB-D was conducted with varying DT2MS of –50µs, -20µs, -10µs, -5µs and -1µs. Histories of reaction force by FEA as compared with the experimental result are shown in Fig.A.1.

It is found from Fig.A.1 that the maximum reaction force becomes smaller as the time step (DT2MS) gets smaller, and that a convergence can be seen if |DT2MS| is same as or smaller than 10 µs. [DT2MS] larger than 10 µs bring about larger reaction force than in the experiment mainly due to the larger inertia force which resists the movements of the models.

Thus, in view of maximum reaction force, it is judged to be...
appropriate to use 10µs or less as the absolute value of DT2MS. Moreover the reaction force by FEA tends to become closer to that by experiment as the time step size gets smaller. Thus in this study –1µs was employed for all the calculation.

Fig. A.1. Effect of time step size (DT2MS) on the reaction force
Numerical Study on the Effect of Buffer Bow Structure in Ship-Ship Collision

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ABSTRACT
A disastrous oil spill from a struck oil tanker has become one of the major problems in view of conservation of maritime environment. So far double hulls (D/H) have been introduced to reduce the consequences of collision and grounding events. In order to further reduce the oil spill from struck oil tankers, the introduction of buffer bulbous bows has been proposed. Relatively soft buffer bows absorb part of the kinetic energy of the striking ship before penetrating the inner hull of the struck ship. The purpose of the present paper is to verify the effectiveness of a prototype buffer bulbous bow structure in ship-ship collisions as compared with that of standard bulbous bows. This is demonstrated by conducting a series of large-scale finite element analyses. The finite element analyses are conducted with the general-purpose nonlinear structural code “LS-DYNA”. The applied scenario is one where a very large crude oil carrier (VLCC) collides with a D/H VLCC in a laden condition. Fracture of fillet welds, elastic-plastic material properties and strain rate effects, are taken into account in the simulations. The effect of the equivalent failure strain (FS) and the forward velocity of the struck ship on the collapse mode of the bow of the striking ship are investigated. Collapse modes, contact forces and energy absorption capabilities of the buffer bows are compared with those of conventional bows.

KEY WORDS: Ship Collision; Finite Element Analysis; Oil Spill; Buffer Bulbous Bow; Critical Striking Velocity; Failure Strain

INTRODUCTION
In order to prevent oil spill from a struck tanker, the double hull system has become the de facto standard as an effective countermeasure. However, it is still a fact that collision accidents involving struck double hull tankers results in oil spills. One example is the collision accident involving the double hull oil tanker “Baltic Carrier”. Therefore there are good reasons to further minimize the consequences of ship collision accidents.

In order to reduce the risk of oil spill from struck oil tankers the Buffer Bow Project (BBP) has been carried out in NMRI sponsored by the Japanese Ministry of Land Infrastructure and Transport (MLIT). The results presented in this paper consist of a part of the results obtained within the BBP.

The background is that Kitamura et al. (1998) have suggested that the bulbous bow is regarded as a key structural part for buffer bow design and the main focus is laid on the bulbous bow structure in the BBP.

A lot of research has been conducted so far concerning numerical studies of the ship-ship collisions. Kitamura et al. (1998, 2000) applied the buffer bow concept to the bulbous bow of large cargo ships in the research project of the Association for Structural Improvement of Shipbuilding Industry (ASIS). They carried out FEM simulations assuming that a Suezmax tanker collides with a D/H VLCC with the purpose to evaluate the effectiveness of buffer bows. In this research project only one oil tank of the struck tanker was modeled by elastic-plastic elements and other parts of the ship were modeled by rigid elements, where deformation of transverse bulkheads located backward and forward of the struck oil tank were not considered. However, the modeling of the VLCC was not satisfactory for the case of high-energy collisions due to the limitation of computer technology in those days. The relatively narrow deformable range surrounded by the close rigid boundaries may cause early rupture of the side and the inner shell of the struck ship. Subsequently, Kitamura (2002) conducted a series of finite element simulations and pointed out that the effect of global-hull-girder horizontal bending moment (GHBM) is significant when a ship collides with the midship region of a struck ship and the size of the striking ship is same or larger than that of the struck ship. Kitamura (2002) carried out several cases of analysis where a VLCC collides with a relatively small ship with considering GHBM of the struck ship. Using a simplified model the effect of GHBM was demonstrated by Pedersen et al (2004) to result in significant strains in the ship side. Therefore, in this study three oil tanks at port-side and three oil tanks at starboard-side, in total six oil tanks, are modeled in order to take into account the effect of GHBM.

In order to validate the accuracy of the finite element simulation of ship collisions some large scale experiments have been conducted. ASIS (1996) conducted full-scale ship-ship collision experiments in Netherlands. Endo and Yamada (2001A), Endo et al. (2001B) and Yamada and Endo (2003) conducted a series of quasi-static axial crushing experiment with large/small-scaled bulbous bow models assuming right angle collisions. Yamada and Endo (2004) also conducted a series of quasi-static bending collapse experiment with large-scale bulbous bow models assuming oblique collision. From these experiments it is clear that the effect of welding induced residual stresses should be considered in order to raise the accuracy of the numerical simulations if small-scale models are used in the experiment.
It was also pointed out that those effects will decrease as the scale-models become close to the actual size. As a general observation the usefulness of the finite element analysis was confirmed in estimating crushing strengths and collapse mechanisms of ship collisions. Further improvements, however, are still needed.

In this study a series of finite element simulations are conducted using the general-purpose structural analysis code “LS-DYNA.” Assuming a collision scenario where a D/H VLCC is struck at mid-ship by a VLCC. Fracture of fillet weldings, elastic-plastic material properties, and strain rate effects are taken into account in the simulations. The effect of equivalent plastic strain and forward velocity of the struck ship in relation to energy absorption of the bow of the striking ship are investigated. Collapse modes, contact forces and energy absorption capabilities of the buffer bow are compared with those of similar conventional bows. The effect of critical FS on the initiation of rupture of the outer and inner shell is investigated.

**ANALYSIS METHOD**

In order to verify the effect of the buffer bow design, a series of large-scale finite element analyses are conducted using the general-purpose nonlinear structural analysis code “LS-DYNA.” That is the calculations are performed in time domain using an explicit analysis method.

**Collision Scenario**

In this study a VLCC is adopted as the striking as well as the struck ship. The reason is that the environmental damage is supposed to be most severe when oil spill happens from an accident involving a VLCC, and that same type of ship is most likely to collide with each other due to the fact that same size of tankers take the same sea routes (Brown, 2001). Two kinds of striking ship models (B4L and B4T) and one striking ship model (A15) are used in the analysis. The striking ships have the same structure and particulars except the bulbous bow. Structural designs of all the ships comply with the rules of a classification society. The details of the scantlings and drawings of the ships are shown in Endo et al (2004).

The struck ship is denoted as “ship A” and the striking ship is denoted “ship B.” The notation A or B is used to distinguish both ships in this paper. One of the striking ships “B4L” has a standard bow with longitudinal stiffening system. The other striking ship “B4T” is equipped with a prototype buffer bow with transverse stiffening system where the frame spacing is reduced to 750mm in order to reduce the outer shell thickness. This design is in accordance with classification rules. Endo et al. (2004) pointed out that the ballast condition of the striking ship is more severe for the struck ship than the laden condition. Therefore, it is assumed in this analysis that a striking ship in ballast condition collides at right angle (θ=90deg) with the midship region of a struck ship, respectively. The initial struck point is arranged to be the side structure between transverse webs of the mid-tank of the struck ship. The reason for choosing this collision point is that it is more severe for the struck ship and it is more likely to occur than a collision at a transverse web.

**Modeling of Ships and Analysis Condition**

The bulbous bow of the striking ship up to the collision bulkhead is modeled as an elastic-plastic structure while other parts aft of the collision bulkhead are modeled as a rigid structure as shown in Fig. 1. The longitudinal bulkhead. Not only the tanks at the struck side but also the tanks at the other side are modeled using elastic-plastic elements with a coarse mesh in order to take into account the effect of GHBM. The mesh size is smoothly transformed from fine to coarse in between. Every model consists of shell and beam elements. Solid elements are not employed. The number of nodes and elements used in the analysis are shown in Table 2. Most of the parts of the ships are made of mild steel, but the parts around upper deck and the lower deck of the struck ship are made of higher tensile steel. The material properties used in the analysis are shown in Table 3. A piecewise linear plasticity (MAT24) assumption was used for the material model, i.e. the uniaxial stress-strain relationships are modeled by piecewise straight lines and the strain rate effect is taken into account by adopting the Cowper-Symonds model. Primary fillet weldings, which are close to the initial struck point of the struck ship, are modeled and fracture of them are considered (See LSTC, 2003). Rupture of the side shell is modeled as element failure with the FS of 12%. The effect of FS on the results of the analyses is investigated. Initial imperfections and residual stresses are not taken into account. A six degree of freedom global ship motion model is applied to ship. The effect of seawater is taken into account as a virtual added mass for each ship and a series of 1-node spring elements are used to model the static restoring forces. Other effects of the seawater such as damping are neglected because the important part of the collision phenomena occurs during a very short time.

As for the struck ship six tanks near the mid-ship are modeled as an elastic-plastic structure and other parts are modeled using rigid elements as shown in Fig. 1. Only one central tank which is supposed to be struck at its side hull is modeled with fine mesh, and the remaining five tanks with coarse mesh. The two oil tanks located before and behind the struck tank are modeled as elastic-plastic in order to properly take into account the effect of stretching of the side hull and the longitudinal bulkhead. Not only the tanks at the struck side but also the tanks at the other side are modeled using elastic-plastic elements with a coarse mesh in order to take into account the effect of GHBM. The mesh size is smoothly transformed from fine to coarse in between.

In this study a VLCC is adopted as the striking as well as the struck ship. The reason is that the environmental damage is supposed to be most severe when oil spill happens from an accident involving a VLCC, and that same type of ship is most likely to collide with each other due to the fact that same size of tankers take the same sea routes (Brown, 2001). Two kinds of striking ship models (B4L and B4T) and one striking ship model (A15) are used in the analysis. The striking ships have the same structure and particulars except the bulbous bow. Structural designs of all the ships comply with the rules of a classification society. The details of the scantlings and drawings of the ships are shown in Endo et al (2004).

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**Table 1 Particulars of the Model Ships**

<table>
<thead>
<tr>
<th>Ship ID</th>
<th>Ship Type</th>
<th>Mass including added mass (kton)</th>
<th>L_mld x B_mld x D_mld x d_mld (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A15</td>
<td>VLCC</td>
<td>490.0</td>
<td>330 x 58 x 33 x 22</td>
</tr>
<tr>
<td>B4</td>
<td>VLCC</td>
<td>136.4</td>
<td>345 x 58 x 33 x 22</td>
</tr>
</tbody>
</table>

**Table 2 Number of Nodes and Elements**

<table>
<thead>
<tr>
<th>Ship ID</th>
<th>Number of Nodes</th>
<th>Number of Elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>A15</td>
<td>289,011</td>
<td>305,666</td>
</tr>
<tr>
<td>B4L</td>
<td>40,482</td>
<td>43,335</td>
</tr>
<tr>
<td>B4T</td>
<td>43,320</td>
<td>46,416</td>
</tr>
</tbody>
</table>

ISOPE2005-JSC-133 Yasuhira Yamada
Critical Striking Velocity

In order to validate the effect of buffer bows the measure "critical striking velocity" \( V_{B,cr} \) is proposed as one of the evaluation indexes. The speed \( V_{B,cr} \) is defined as the maximum velocity of the striking ship less than which the inner hull of the struck ship is not ruptured by the collision. Therefore, \( V_{B,cr} \) gives the safe navigational velocity for a striking ship against a specific struck ship. It is noted that \( V_{B,cr} \) depends on the mass, the structural strength and the collision scenario. The relation between the velocity \( V_{B,cr} \) and the involved masses can be derived analytically based on the conservation laws of energy (Eq.1) and the conservation laws of momentum (Eq.2) provided the collision is fully plastic, the collision angle is 90\(^\circ\) and the struck ship does not have forward velocity (\( V_{A,0} = 0 \)). With these assumptions we have:

\[
\frac{1}{2} M_B V_B^2 = \frac{1}{2} (M_A + M_B) V'^2 + E_s
\]

\[
M_B V_B = (M_A + M_B) V'
\]

where \( V' \) denotes the common velocity in the \( x \)-directional (see Fig. 1) after the collision, and \( E_s \) denotes the energy dissipated by phenomena other than ship motion, that is the energy absorbed by the deformation and fracture of the structure and the friction energy. Elimination of \( V' \) from Eqs.1-2 yields:

\[
V_s = \sqrt{\frac{2 E_s}{M_B}} \times \frac{M_A + M_B}{M_B}
\]

We can estimate the critical striking velocity \( V_{B,cr} \) by substituting \( E_{cr} \) instead of \( E_s \). Here \( E_{cr} \) is defined as the energy \( E_{cr} \) absorbed by the time \( t_{cr} \) when rupture of the inner hull occurs. In the following this energy is determined from the result of finite element analyses.

\[
V_{s,cr} = \sqrt{\frac{2 E_{cr}}{M_B}} \times \frac{M_A + M_B}{M_B}
\]

This procedure gives an equivalent value of the critical striking velocity without carrying out the collision simulation with various striking velocities. However, there may be involved some errors in evaluation of strain rate effect and friction-related energy. Preliminary calculations were carried out to verify this error and the error is recognized being small. Basically 20knots is employed as the initial velocity of the striking ship in order to give the enough velocity to be sure to rupture the inner shell and thereby determine \( E_{cr} \) from a limited number of analysis cases. It usually takes about 2days for one case analysis.

RESULTS

The Effect of the Equivalent Failure Strain (FS)

Each shell element will be removed from the calculation when the equivalent FS of the element reaches a pre-defined value (FS). It is well known that the FS is largely dependent on the element size and usually varies from 20\% - 35\% (ASIS, 1996; Kitamura, 2002; Takaoka 2004) for large-scale finite element simulation. Kitamura (2002) pointed out the importance of further validation of the absolute value of the equivalent FS. In this study the effect of FS on the analysis results are numerically investigated in the case where the model B4L collides with the model A15. Calculations have been performed with FS values of 20\%, 15\%, 12\% and 10\%. Failure Strains FS less than 20\% are used due to insufficient estimation capability of the stress concentration with present finite element model with rather coarse mesh sizes (about 500-700mm).

Fig. 2 shows the time histories of the contact force for various values of FS. The rupture initiation time of the outer shell (OS) and the inner shell (IS) is marked by circles and squares respectively. In this analysis the time of the rupture of the side shell is defined as the time when a finite element on the OS or IS is eliminated for the first time.

![Fig. 2 Time histories of contact forces](image)

Fig. 3 Energy absorption ratios for the struck ship

It is seen in Fig. 2 that the effect of FS on the contact force is significant and that the level of the contact force gets lower as the FS becomes lower. This is due to the early rupture of the outer and the inner shells of the struck ship. We can find two groups of curves in Fig. 2, one is FS=20\%, 15\%, the other is FS=12\%, 10\%. In the latter group rupture of the OS and IS occur at early stages of the collision, and the contact force drops down after the occurrence of shell rupture, whereas in the former group, the occurrence of rupture on the OS and IS occur later. As a result, the bow of the striking ship deforms slightly more for the high FS values and the contact forces do not drop so much due to the existence of large contact areas between two ships.

The ratio between the internal energy absorbed by the struck ship and the total internal energy (REA) is shown in Fig. 3 where REA is derived by the following equation.
It is seen from Fig. 5 that the dynamic scale factor against V is assumed to be the half value to the transverse frame spacing (=4.22m). Rate is estimated geometrically as shown in Fig. 6. In this figure L is and the velocity of the striking ship is shown in Fig. 5, where the strain rate and the dynamic scale factor. C and P denote Cowper-Symonds coefficient. C=40 and P=5 are widely used for mild steel. Neglecting the bending deformation the relation between the dynamic scale factor and the velocity of the striking ship is given by the Cowper-Symonds equation.

\[ R_{EA} = \frac{IEA}{IEA + IEB} \]  

where IEA and IEB denotes the internal energy absorbed by the struck ship and the striking ship, respectively. It is seen from Fig. 2 and Fig. 3 that the difference between FS=10% and 12% is rather small if it is noted that the inner side of the struck ship does not rupture if FS is greater than 20%. This fracture strain value is, however, unrealistically high against the element size of 500-700mm employed in this analysis. There may be ways to adopt finer mesh sizes, but it makes the CPU time much longer and makes the number of analysis cases much more limited in these large-scale ship-ship collision analyses. From an engineering point of view it is supposed to be rational to apply macroscopic equivalent FS.

**Tensile Test and Barba’s Law**

In the usual uniaxial static tensile coupon tests, engineering FS for gauge length of 200mm becomes about 30% or greater. Therefore, engineering FS for gauge lengths of 500-700mm is supposed to become much lower than 30% because local elongation is averaged by the large gauge length. Following an empirical formula, well known as Barba’s Law (Barba, 1880; Jernkvist et al, 2004), a good relation between engineering FS and gauge length (element size) in the uniaxial static tensile test of mild steel is:

\[ \varepsilon_{u} = \varepsilon_{u} + \frac{W_{t}}{L} \]  

where \( \varepsilon_{u} \), \( \varepsilon_{u} \), c, W, t and L denotes total engineering FS, uniform engineering FS, Barba’s constant, width of specimen, thickness of specimen, gauge length, respectively. In the right hand side of this formula, the second term represents a localized strain component induced by necking of the specimen. This formula indicates that the effect of localized necking strain on the total FS becomes very small and the total FS becomes close to \( \varepsilon_{u} \) as the gauge length gets larger. Those tendencies comply with a lot of previous works so far. Fig. 4 shows a curve of Eq.6 where \( \varepsilon_{u} \) is defined as 0.18 so that good agreement is achieved between the curve and the experimental results of tensile tests conducted in NMRI. It is found in Fig. 4 that FS becomes around 20% for the un-welded material when the gauge length varies between 500mm and 700mm. It corresponds to the true strain of about 18.23% by the following formula.

\[ \varepsilon_{true} = \ln(1 + \varepsilon_{true}) \]  

where \( \varepsilon_{true} \) and \( \varepsilon_{true} \) denote the true strain and engineering strain, respectively. Moreover, it is known that FS becomes smaller as the dynamic effect becomes greater. The following approximate formula which is the inverse of the Cowper-Symonds equation is widely used for estimating dynamic FS from static FS (Paik and Pedersen, 1996; Paik et al, 1999, Brown, 2002).

\[ \dot{\varepsilon}_f = \dot{\varepsilon}_f \left[ 1 + \left( \frac{\varepsilon_{f}}{C} \right)^2 \right] = \dot{\varepsilon}_f \cdot s \]  

where \( \dot{\varepsilon}_f \), \( \dot{\varepsilon}_f \), \( \dot{\varepsilon}_f \) and s denotes the dynamic FS, the static FS, the strain rate and the dynamic scale factor. C and P denote Cowper-Symonds coefficient. C=40 and P=5 are widely used for mild steel. Neglecting the bending deformation the relation between the dynamic scale factor and the velocity of the striking ship is shown in Fig. 5, where the strain rate is estimated geometrically as shown in Fig. 6. In this figure L is assumed to be the half value to the transverse frame spacing (=4.22m). It is seen from Fig. 5 that the dynamic scale factor against \( V_{st} \) varies from 0.6 through 0.8. Therefore it is estimated from (7) that the dynamic FS against the gauge length of 500-700mm is around 12-14%. It is noted that FS becomes close to 12% when the velocity of the striking ship is greater than 15kt and the dynamic scale factor becomes close to 0.6.

Considering those effects, FS is assumed to be 12% hereafter. It is noted that the effect of welding induced residual stresses and the decrease of shell thickness due to deterioration, which are important issues for actual ships, are not explicitly taken into account in the present analysis. It is also noted that the absolute value of FS is still open for further investigation. In the present case where the purpose is to compare the effect of buffer bow with standard bow the simple approach with an FS value of 12% seems reasonable.

**Effect of the Buffer Bow**

In order to validate the effect of introducing buffer bow structures two cases of FEA are conducted where the striking ship of B4L or B4T collides with the struck ship of A15.
Histories of the integrated buffer bow contact forces are shown in Fig. 7 and compared with those of a standard bow, where “OS Rup.” marked by circle means the rupture of the outer shell and “IS Rup.” marked by square means the rupture of the inner shell. It is found in Fig. 7 that the magnitude of the forces in the case of a buffer bow is a little larger than that of standard bow especially during the time from 1.0s to 2.0s. It is also seen in Fig. 7 that the contact force suddenly drops down just after the rupture of inner or outer shell and that the initiation of inner shell ruptures in case of a buffer bow is a little later than that of standard bow. This is because the contact force goes down when the restraint at the contact point suddenly disappears after the occurrence of shell rupture. It could be said that the buffer bow delays the initiation of inner shell rupture and the buffer bow therefore is less likely to result in oil spill in case of a collision.

Time histories of internal energy absorbed by the striking ship and by the struck ship are shown in Fig. 8 and Fig. 9, respectively. It is found in Fig. 8 that the internal energy absorbed by the striking ship (IEB) in case of a buffer bow suddenly becomes larger around a time of 0.5s and IEB in case of buffer bow is much larger than that of standard bow. On the other hand it is found in Fig. 9 that the internal energy absorbed by the struck ship (IEA) is almost the same in both cases. This indicates that the damage of the struck ship induced by a standard bow and that by a buffer bow is not greatly different in view of energy. However the damage to the side structure of the struck ship in case of a buffer bow is a little smaller than that of standard bow due to the early denting of the striking ship bulb structure and causes a rapid increase of the contact area as shown in Fig. 12 through Fig. 14. As a result the initiation of the inner hull rupture in case of a buffer bow starts much later compared to a standard bow. It is noted that the striking velocity of 20kt, which is faster than the actual navigational velocity for VLCC, is employed in this analysis due to the reason mentioned before. Therefore, the phenomena before the time about 2.0s is more important than the remaining part when considering normal navigational conditions with velocities less than 16k.

Ratios of internal energy absorbed by the struck ship and the total absorbed energy (REA) are shown in Fig. 10, and the sum of the internal and sliding energy absorbed by both ships are shown in Fig. 11. The energy ratio REA takes the value between 0 and 1, and REA value close to 1.0 indicates that the damage of the struck ship in view of energy is significant while the damage of the striking ship is very small. It is seen in Fig. 10 that the damage of the struck ship for the buffer bow case is about 30% lower in maximum (t=2.0s) than that for the standard bow. The critical striking velocities obtained from Eq.4 are listed in Table 4.
Table 4 Critical striking velocity obtained from Eq.4 and FEA

<table>
<thead>
<tr>
<th>Time of IS rupture[s]</th>
<th>Standard</th>
<th>Buffer</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.59</td>
<td>1.60</td>
<td></td>
</tr>
</tbody>
</table>

$$E_{s,cr} \quad [MJ] \quad 644 \quad 2029$$

$$V_{B,cr} \quad [kt] \quad 6.74 \quad 11.96$$

$$M_A=4.9e8 \quad [kg], \quad M_B=1.364e8 \quad [kg]$$

![Image](image1)

Fig. 12 Damage of the struck ship side (Left: Standard, Right: Buffer)

**Effect of Forward Velocity of the Struck Ship**

In most collisions the struck ship has a forward velocity. In order to investigate the effect of a forward velocity of the struck ship, FEA is conducted where the velocity of the struck ship has been varied ($V_A = 0, 1, 5, 10$ and $15kt$). It is assumed that the VLCC with standard bow (B4L) in ballast condition collides perpendicularly with the VLCC (A15) in a laden condition with the velocity ($V_B$) of $15kt$. Due to the forward velocity of the struck ship a large horizontal shear force will be expected to act on the bulbous bow of the striking ship. The simulation results show that horizontal bending deformation of the bulbous bow does not take place in all cases, that IEA is almost the same and that rupture initiation of inner shell starts later as the $V_A$ get larger. Time histories of IEB, REA and IE+SE are shown in Fig. 15 through Fig. 17 respectively, where IE+SE denotes the energy dissipated by the phenomena other than ship motion, that is, the sum of the internal energy absorbed by the both ships and the sliding (friction) energy. It is found in Fig. 15 through Fig. 17 that IEB gets larger, REA gets lower and rupture initiation of IS gets later as the velocity of the struck ship ($V_A$) becomes larger. These results indicate that the damage of the struck ship in view of energy absorption becomes lower as the velocity of the struck ship gets larger.

![Image](image2)

Fig. 13 Deformation of the striking ship bulb structure (Standard)

![Image](image3)

Fig. 14 Deformation of the striking ship bulb structure (Buffer)

Therefore, the conclusion is that collisions without forward velocity of the struck ship seem to be most severe for the struck ship in the case of right angle collisions. The reason is that the contact area on the struck ship moves backward as the struck ship moves forward before the side shell of the struck ship causes serious damage. This result may not be obvious since early rupture of the side shell could be expected to occur due to longitudinal tension stresses induced by the combination of longitudinal tension forward of the collision points and longitudinal compression aft of the collision point due to friction between the two ships. However serious rupture on the side shell is not observed in any of the cases. It is noted that we employed the dynamic friction coefficient of 0.3 (See Brown, 2002). It should also be noted that if the collision angle is larger that 90 degrees more energy has to be absorbed during the collision event (See Pedersen et al, 1998). In this case the forward speed of the struck ship may cause more damage.

![Image](image4)

Fig. 15 Histories of the energy absorbed by the striking ship

![Image](image5)

Fig. 16 Absorbed energy ratio for the striking ship (B4L; Standard Bow) with varying forward velocity of the struck ship.

![Image](image6)

Fig. 17 Absorbed internal energy by the IS rupture
Comparative FEA between the model B4L (standard) and the model B4T (buffer) with \( V_A = V_B = 15\text{kt} \) is conducted and the results are shown in Fig. 18 through Fig. 26. It is seen in Fig. 23 to Fig. 26 that the buffer bow begins to bend at \( t = 0.2\text{s} \). This results in a larger contact area with the struck ship while standard bow does not bend at all and goes on making large denting on the side shell of the struck ship. It is seen in Fig. 18 and Fig. 19 that the X-directional force and Y-directional force for the buffer bow have a remarkable drop at the time of 0.2s when the bow begins to bend. It is found by comparing Fig. 9 and Fig. 21, and Fig. 10 and Fig. 22 respectively that the difference of IEA and REA between standard bow and buffer bow is more outstanding when the struck ship has a forward velocity. This indicates that the buffer bow becomes more effective in reducing the damage of the struck ship when the struck ship has a forward velocity by making large contact area by overall bending of bulbous bow. It is also found in Fig. 22 that REA for the buffer bow becomes larger after \( t = 2.0\text{s} \). The reason is that the effect of the bulbous bow is limited to the early stages of the collision and that the latter stages of the collision is governed by the collision between backward structure of bulbous bow of the striking ship and the side shell of the struck ship.
CONCLUDING REMARKS

A series of large-scale finite element analyses has been conducted modeling a VLCC colliding with another VLCC at a right angle. It is found that the Fracture Strain (FS) has a significant effect on the rupture of outer and inner shell, the contact force and the energy absorbed by the striking and the struck ship. In order to investigate the effect of the introduction of a buffer bow on the prevention of oil spill from struck D/H VLCCs, FEA is conducted assuming a FS of 12%. This is found to give fairly reasonable results as compared with the reported collision damages. It is found that the buffer bow does not completely prevent the rupture of inner hulls in case of high energy collisions, but it can delay the rupture of the inner shell to some extent. As for the case of a collision angle of 90deg without forward velocity of the struck ship, the critical striking velocity for the buffer bow becomes about 77% higher than that for a standard longitudinally stiffened bow. This indicates that a VLCC with buffer bow navigating at velocity of 11kt or lower does not penetrate the inner shell of D/H VLCC in case of collisions with an anchored VLCC. The effect of the buffer bow become more remarkable for right angle collisions when the struck ship has a forward velocity because early bending of bulbous bow results in larger contact area with the struck ship side shell. It could be said that the buffer bow is effective to further reduce the risk of oil spill in the case of a collision with forward velocity.

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REFERENCE

http://www.sname.org/committees/tech_ops/O44/crashworthy/charter.html
A Benchmark Study of Procedures for Analysis of Axial Crushing of Bulbous Bows

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Abstract

Simplified methods to estimate mean axial crushing forces of plated structures are reviewed and applied to a series of experimental results for axial crushing of large-scale bulbous bow models. Methods based on intersection unit elements such as L, T and X type elements as well as methods based on plate unit elements are employed in the analyses. The crushing forces and the total absorbed energy obtained by the simplified analyses are compared with those obtained from large scale bulbous bow experiments. The accuracy and the applicability of these methods are discussed in detail.

KEY WORDS: ship collision, simplified method, axial crushing, effective crushing distance, bulbous bow, buffer bow, crashworthiness

1. Introduction

In order to reduce the risk of oil spill from struck oil tankers a buffer bow concept was proposed by the Association for Structural Improvement of Shipbuilding Industry (ASIS) [1] and its effectiveness was analytically and empirically investigated for several specific collision scenarios by Kitamura [2], and Endo & Yamada [3-6]. To apply the buffer bow concept it is important to be able to estimate the crushing forces and energy absorption of buffer bow structures as well as of conventional bow structures. In general the crashworthiness of ship structures against collision is one of the important matters relating to the safety of ships and crews on board. To evaluate the crashworthiness of ship structures the finite element analysis (FEA) is a powerful numerical tool. To perform a large-scale FEA, however, is still time-consuming due to the effort required when creating the finite element model as well as the numerical calculation itself. Therefore, there is a need for simplified methods to calculate the crashworthiness of ships in the early stages of the design, in the rule making process as well as for risk

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and reliability analyses. A variety of simplified formulas have been proposed in order to estimate the mean crushing forces of plated structures such as ship bows. Although these formulas are based on the same rigid - plastic material modelling procedures they are in reality different due to different assumed folding mechanisms. The purpose of this paper is a review of existing simplified analysis methods and to find the most suitable and accurate formula for estimating the crushing behaviour of bulbous bows.

The International Ship and Offshore Structures Congress (ISSC) specialist panels on structural design against collision and grounding has continuously reviewed the most recent literature and its applicability for predicting the crushing and cutting damage of ships in collision and groundings [7].

Simplified formulas for estimating axial crushing forces of prismatic plated structures have been proposed by Wierzbicki [8-9], Amdahl [10], Yang & Caldwell [11], Ohtsubo & Suzuki [12], Abramowicz [13], Wang et al [14-15], and Paik and Pedersen [16]. Lehman & Yu [17] derived analytical formulae for estimating the axial crushing strength of a conical shell modelling the outer part of bulbous bow structures. Later Paik & Wierzbicki [18] performed a comprehensive benchmark study on the application of simplified methods to a series of quasi-static crushing tests using longitudinally, transversely or orthogonally stiffened square tubes. The analytical results by Amdahl [10], Wierzbicki & Abramowicz [9], Abramowicz, Ohtsubo & Suzuki [12] and Paik & Pedersen [16] were compared with the experimental results. It was concluded in this study that the methods by Wierzbicki & Abramowicz [9] and Paik & Pedersen [16] give relatively good estimations as compared with other methods for axially compressed thin-walled prismatic structures with quite simple geometries. More recently, Zhang [19] and Endo & Yamada [3] developed a new set of simplified methods.

The above mentioned theoretical procedures for estimation of the crushing behaviour of prismatic structures are based on experimental validation using thin lightly stiffened structures measuring 2 – 4 mm. However these structures have much lower thicknesses and have much simpler geometries than found in practical bulbous bow structures on ships. It is anticipated that in more heavily stiffened structures the stiffeners and/or webs make earlier plate-to-plate contact and will prevent the deformation and folding of the outer shell. This can be expected to result in larger crushing resistance. Another source of concern is the assumption of prismatic structures. The configuration of practical bulbous bows differs from the exact prismatic shape.

In order to clarify the crushing mechanism of bulbous bow structures and to obtain realistic experimental data the first author has conducted a series of axial crushing tests using large-scale bulbous bow models, where the model is almost half scale of actual very large crude oil carriers (VLCC). These models are also representative for actual bulbous bows of ships of about 500 tons.

In the present paper theories based on intersection elements such as L, T and X type elements as well as theories based on plate unit elements are employed and applied to the axial crushing of these bulbous bow models. The results obtained by a number of different simplified analyses procedures are compared with those observed in the experiments and the applicability and accuracy of the simplified procedures are discussed in detail.
2. Simplified analysis methods

2.1 Rigid-plastic analysis

Simplified formulas for estimating axial crushing forces of prismatic plated structures have been proposed by Wierzbicki [8-9], Amdahl [10], Yang & Caldwell [11], Ohtsubo & Suzuki [12], Abramowicz [13], Wang [14-15], Paik & Pedersen [16], Zhang [19] and Endo & Yamada [3]. Lehmann & Yu [17] developed special formulas for axial crushing of conical shell structures. These formulas are derived based on the so-called “rigid-plastic analysis” where the material is assumed to be rigid-perfectly plastic. With reference to the upper bound theorem the mean crushing force can be derived by dividing the total absorbed energy by the crushing distance while assuming kinematically admissible crushing mechanism (See Jones [20]). That is,

\[ P_m = \frac{E}{2H \cdot \eta} \]  

(1)

where \( P_m \) denotes the mean crushing force, \( E \) the absorbed energy during one fold crushing distance, \( H \) is the half folding length and \( \eta \) the dimensionless effective crushing distance (to be described later). In most cases the energy is a function of the unknown variable \( H \). The value of \( H \) and the associated mean crushing force \( P \) are derived by minimizing the mean crushing force as follows.

\[ \frac{\partial P_m}{\partial H} = 0 \]  

(2)

Often \( P_m \) is a function of more than two variables describing the folding patterns such as \( I \) and \( J \). When this is the case these variables are determined from the optimality criterion:

\[ \frac{\partial P_m}{\partial H} = 0, \frac{\partial P_m}{\partial I} = 0, \frac{\partial P_m}{\partial J} = 0, \ldots \]  

(3)

2.1 Intersection Unit Method and Plate Unit Method

Here the simplified crushing analysis methods are categorized into two groups as shown in Fig. 1. These two different procedures are based on the same rigid plastic theory. The first method, called the intersection unit method (IUM), models the structure by using typical intersecting units (super folding elements) such as L, T, and X type elements as shown in Fig. 2. This method focuses on the plate intersections which usually give greater crushing resistance than the flat plates between intersections. The other method, called the plate unit method (PUM), models the structure by using individual plate units. This method was originally proposed by Paik & Pedersen [16]. One of the advantages of the PUM is that the structural model is easier to make than the model used for the IUM. This difference becomes important in case of realistic large complicated ship structures where a variety of plate thicknesses and breadths are used. In the case of IUM we need to calculate the equivalent plate thickness (\( t_{eq} \)) and the average breadth (\( b_{ave} \)) to be used in the simplified formula. In the case of PUM, on the other hand, it becomes much easier because we can directly use the thickness and the breadth data of the associated plates although the number of elements in PUM is usually higher than in IUM. It is noted that the definition of the breadth \( b \) of the elements is important. It is also noted that when applying these two methods independently to the same structure, the flange breadth \( b \) for the elements of PUM becomes twice as large as those used for IUM due to the difference of making elements mesh. Fig. 9 shows an example application of IUM and PUM to the same grillage structure with a spacing of 2b. In this case the breadth of the IUM element becomes b while that for the PUM becomes 2b. Some of these existing formulas are briefly summarized in Section 2.3.
2.2 Effective crushing distance

It is well known that a plated steel structure does not get completely squashed to zero height because of the finite folding radius and plate thickness. This means that only a limited part (60% – 85%) of the original length in load direction is collapsible. The effective crushing distance, $\delta_{\text{eff}}$, is widely used in developing simplified formula. A dimensionless effective crushing distance $\eta$ which is obtained by dividing $\delta_{\text{eff}}$ by the original length of the structure is also widely used. So far an empirical value of $\eta=2/3$ has been used by many researchers. Recently, based on a series of crushing tests, Paik et al [21] and Paik & Wierzbicki [18] proposed $\eta=0.73$ for unstiffened structures and derived an empirical formula which gives lower $\eta$ values for stiffened plates. These values are functions of the slenderness ratio $\beta (=b/t)$ of the structural elements. Therefore, we need to use different values of $\eta$ as function of the amount of stiffening and the slenderness ratios. In the following review of formulas for crushing strength, the dimensionless effective crushing distance is replaced by the symbol $\eta$ instead of using the proposed original values in order to make each formula more general.
2.3 Review of existing methods
In the following existing methods are briefly reviewed. In the formulas we have introduced:

\[ M_0 = \frac{\sigma_0 t^2}{4} \]  \hspace{1cm} (4)

\[ \sigma_0 = \frac{\sigma_y + \sigma_u}{2} \]  \hspace{1cm} (5)

where \( M_0 \) is the fully plastic plate bending moment for unit breadth, \( \sigma_0 \) the flow stress, \( \sigma_y \) the yield stress, and \( \sigma_u \) the ultimate material strength. Moreover, \( \sigma_m \) and \( \beta \) denote the mean crushing stress and the plate slenderness ratio (=\( b/t \)), respectively.

(1) Amdahl (1983) [10]
Amdahl developed the following unified closed formula for the mean axial crushing force (\( P_m \)) of plated structures:

\[ P_m = 2.42\sigma_0A \left( \frac{N_{LT} r^2}{A} \right)^{\frac{2}{3}} \left[ 0.87 + 1.27 \frac{N_X + 0.31N_T}{N_{LT}} \left( \frac{A}{N_X + 0.31N_T} \right)^{\frac{2}{3}} \right] \]  \hspace{1cm} (6)

here \( \sigma_0 \) denotes the flow stress, \( A \) is the sectional area, \( N_X \) the number of \( X \)-type elements, \( N_T \) the number of \( T \)-elements and \( N_{LT} \) the number of \( L \) and \( T \) elements. Good agreements were achieved between theoretical results and extensive experiments where a series of bow models were collapsed axially in quasi-static condition.

Based on methods derived by Wierzbicki [8-9] and Amdahl [10], Yang & Caldwell (1988) developed simplified formulas for crushing forces for \( L \)-, \( T \)- and \( X \)-type elements assuming four types of basic deformation mechanisms assuming \( \eta = 2/3 \). During the process of re-derivation of Yang and Caldwell’s results the present authors obtained slightly different crushing formulas for some element types. Most of the formulas described here are identical to the Yang & Caldwell’s formulas, but Eqs. (52), (54), (56), and (57) in the appendix are slightly different from Yang & Caldwell’s expressions. Some constant terms seem to be ignored in the Yang & Caldwell’s formulas probably based on the assumption that the effect of these terms on the entire strength is small if the slenderness ratio is large enough. However, the validity of this assumption largely depends on the structural arrangement. Therefore, in the present paper we include these constant terms. Moreover, a study of the derivations shows that the radius of the toroidal shell element, \( r \), must have been assumed to be four times the plate thickness in order to achieve the published closed formulas. That is \( r = 4t \). This assumption seems reasonable if the slenderness ratio of the plate is around 50. However, it is noted that this assumption does not always fulfill the minimization of the crushing force. The formulas derived by the present authors and presented in the appendix are expressed in the same form as in Yang & Caldwell (1988), i.e. the mean crushing forces are normalized by the fully plastic bending moment (\( M_0 \)).

The mean crushing stresses normalized by the flow stress are given below using \( H \) and/or \( r \) values which minimize the absorbed energy according to Eqs. (2) and (3):
(i) Mode 1 for L type element (Mode L1)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.52068}{\beta^{2/3}} \right) \\
H = 1.549\sqrt{b^2t}, \ r = 0.877\sqrt{b^2t}
\]

(ii) Mode 1 for T type element (Mode T1)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.1605}{\beta^{2/3}} \right) \\
H = 2.030\sqrt{b^2t}
\]

(iii) Mode 2 for T type element (Mode T2)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{0.860512 + 0.1309}{\beta^{0.5} + \beta} \right) \\
H = 1.825\sqrt{bt}
\]

(iv) Mode 3 for T type element (Mode T3)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{0.943783 + 1.67197}{\beta^{0.5} + \beta} \right) \\
H = 1.664\sqrt{bt}
\]

(v) Mode 4 for T type element (Mode T4)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.42349 + 0.19635}{\beta^{0.5} + \beta} \right) \\
H = 1.103\sqrt{bt}
\]

(vi) Mode 1 for X type element (Mode X1)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{0.81734 + 1.25397}{\beta^{0.5} + \beta} \right) \\
H = 1.922\sqrt{bt}
\]

(vii) Mode 2 for X type element (Mode X2)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.25331 + 0.25}{\beta^{0.5} + \beta} \right) \\
H = 1.253\sqrt{bt}
\]

(viii) Mode 3 for X type element (Mode X3)
\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.42349 + 0.19635}{\beta^{0.5} + \beta} \right) \\
H = 1.103\sqrt{bt}
\]
Dimensionless crushing stresses obtained by the modified Yang & Caldwell formulas for L, T and X type elements are compared with each other against the slenderness ratio of the flanges in Fig. 4. The figure shows that mode T1 produces the lowest crushing strength among the T type elements and that mode X1 produces the lowest among the X type elements while mode X3, where axisymmetrical crushing is assumed, produces the highest crushing strength. It is interesting to note that the dimensionless crushing stresses for T4 (Eq.(15)) and X3 (Eq.(21)) are the same. This is because the same crushing mechanism, “straight edge mechanism”, is assumed and the energy absorption is assumed to be proportional to the number of flanges. According to the principle of minimum force, the crushing mechanism which produces the least strength is the most probable. However, Yang & Caldwell pointed out that experimental verification is required and selected the modes T3 and X2 as representative crushing modes based on experimental results.

(3)Abramowicz (1994) [13]
Abramowicz performed comprehensive research on the crushing behaviour of T, Y and X type structural elements. He derived two theoretical formulas for T type elements by assuming Asymmetric
and Symmetric collapse modes, and derived three formulas for X type elements by assuming Natural, Mixed I and Mixed II modes. It is noted that $\eta=0.73$ is used in the original formulas:

(i) Asymmetric Mode for T type element (Mode $T_1$)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.14514}{\beta^{2/3}} \right)
\]

(ii) Symmetric Mode for T type element (Mode $T_2$)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{0.90699}{\beta^{0.5}} \right)
\]

(iii) Natural Mode for X type element (Mode $X_1$)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.11072}{\beta^{0.5}} \right)
\]

(iv) Mixed Mode I for X type element (Mode $X_2$)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.46575}{\beta^{0.533}} \right)
\]

(v) Mixed Mode II for X type element (Mode $X_3$)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.22923}{\beta^{0.555}} \right)
\]

Paik & Wierzbicki [18] made a comprehensive review of the Abramowicz’s results and derived generalized formulas by taking the average of kinematically admissible collapse modes. Below we present these formulas directly from the review work performed by Paik & Wierzbicki:

(vi) T type element (Mode $T$)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.1608}{\beta^{2/3}} \right)
\]

(vii) X type element (Mode $X$)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{0.4535 + 0.7458}{\beta^{0.5} + \beta^{0.567}} \right)
\]
Fig. 5 Comparison of dimensionless stresses using Abramowicz [13], where \( \eta = 2/3 \).

(4) Ohtsubo & Suzuki (1994) [12]
Based on Yang & Caldwell’s crushing mechanisms, Ohtsubo & Suzuki (1994) developed three additional mechanisms for an L-type element (Modes L2, L3 and L4). They also modified the energy absorption expression for the crushing mechanisms of T4, X2 and X3 elements. However, the final formulas for the mean crushing forces derived by this modified energy absorption expression are almost the same as the corresponding Yang & Caldwell’s formulas. Ohtsubo & Suzuki pointed out that it is reasonable to assume the same folding length \( H \) for all the elements in one cross section in order to preserve geometrical compatibility in neighboring elements. It means that the individual super elements in the same cross-section are not independent of each other in a complicated actual structure. Therefore, the folding length \( H \) as well as the mean crushing force is obtained numerically by minimizing the mean crushing force of the whole section depending on the structural arrangement of each case. Closed form formulas for the mean crushing forces are not explicitly described in the original paper. For this reason the present authors have derived formulas for new L-type elements by minimizing the crushing force based on the absorbed energy described in Ohtsubo & Suzuki [12] in order to perform the present
benchmark study. Thus, it should be noted that the following formulas are based on the crushing mechanisms and energy absorption expressions proposed by Ohtsubo & Suzuki, but the expression themselves are different from those in the original paper since the folding length is different. Dimensionless mean crushing stresses with the $\eta$ value as an independent parameter are here described as follows:

(i) Mode 1 for L type element (Mode $L_1$), which is identical to Eq.(7).

$$\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.521}{\beta^{2.5}} \right)$$  \hspace{1cm} (30)

(ii) Mode 2 for L type element (Mode $L_2$)

$$\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.253}{\beta^{0.5}} + \frac{0.25}{\beta} \right)$$  \hspace{1cm} (31)

(iii) Mode 3 for L type element (Mode $L_3$)

$$\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.490}{\beta^{0.5}} + \frac{0.1964}{\beta} \right)$$  \hspace{1cm} (32)

(iv) Mode 4 for L type element (Mode $L_4$)

$$\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.423}{\beta^{0.5}} + \frac{0.09817}{\beta} \right)$$  \hspace{1cm} (33)


Based on the Yang & Caldwell’s proposed crushing mechanisms, Wang (1995) developed slightly different formulas for the L, T and X type elements. Wang & Ohtsubo basically employed Yang & Caldwell’s crushing mode for the $L_1$, $T_3$ and $X_2$ elements, but modified slightly the total absorbed energy for each type of element. In Wang & Ohtsubo’s method it is assumed that the energy absorbed in horizontal plastic hinge lines for T and X elements are the same as those for L type elements although the length of the plastic hinge lines are different dependent on the type of elements. That is, $4\pi Mb$ (See Eq.(1) in Yang & Caldwell [11]). This assumption is different from other procedures such as Yang & Caldwell [11] and Ohtsubo & Suzuki [12]. In 1999 Wang & Ohtsubo developed further improved crushing formulas for T and X type elements by considering improved energy absorption expressions. As a result Wang & Ohtsubo developed new formulas for the T type element. The formulas for the X type elements are identical to Yang & Caldwell’s $X_2$ type elements. The normalized crushing stresses obtained by Wang & Ohtsubo in 1999 can be described as follows

(i) For L type element (Mode $L$)

$$\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{0.4747}{\beta^{0.5}} + \frac{2.311}{\beta} \right)$$  \hspace{1cm} (34)

$$H = 3.309\sqrt{bt}$$  \hspace{1cm} (35)
(ii) For T type element (Mode T)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{0.5791 + 1.672}{\beta^{0.5} + \beta} \right) \\
H = 2.712 \sqrt{bt}
\]  

(36)  

(37)

(iii) For X type element (Mode X)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.253 + 0.25}{\beta^{0.5} + \beta} \right) \\
H = 1.253 \sqrt{bt}
\]  

(38)  

(39)

It is interesting to note that the minimized folding length for L, T and X type elements differs considerably from each other in Wang & Ohtsubo’s formulas. For example, the folding length H for the L type element is about 3 times larger than that for the X type element. The use of a uniform folding length in structural elements and its effects on the total mean crushing force is discussed in a later section of this paper.

Wang & Ohtsubo’s original formulas are compared with Yang & Caldwell’s corresponding formula (L1, T3 X2) in Fig. 6. It is seen that Wang & Ohtsubo’s formulas for the T type element give almost the same crushing stresses as for the L type element while the formula for the X type element gives the same strength as for Yang & Caldwell’s X2 type element.
Fig. 6 Comparison of dimensionless crushing stresses for different structural elements. YC: Yang & Caldwell [11], WO: Wang & Ohtsubo [15]. $\eta=2/3$ is applied.
In bow crushing some of the strength elements have an inclination angle $\theta$ against the loading or crushing direction as shown in Fig. 7. This inclination causes a reduction of the crushing forces and a shortening of the effective crushing distance of a plate. Fig. 7 illustrates the crushing of an inclined element, where $P_m$ is the force acting on the inclined structure, $F_m$ is the component of $P_m$ parallel to the force direction, $\delta_e$ the effective crushing distance in the force direction, $\delta_{el}$ the effective crushing distance along an inclined element. Wang [14] developed the following analytically strength reduction factor $\alpha_I$, which takes into account the effect of inclination of an element as well as the shortening of the crushing distance:

$$\alpha_I = \frac{1 - \sqrt{\sin^2 \theta + (1 - \eta^2) \cos^2 \theta}}{\eta \cos \theta}$$

The variation of $\alpha_I$ against $\theta$ for different $\eta$ values is shown in Fig. 8. It is seen from Fig. 8 that the reduction factor $\alpha_I$ is not strongly influenced by $\eta$ values between $\eta=2/3$ and $\eta=0.73$. 

Fig. 8 Variation of the inclination reduction factor ($\alpha_I$) against the inclination angle ($\theta$) for different dimensionless effective crushing distances ($\eta$).
(6) Paik & Pedersen (1995) [16]

A plate unit method (PUM) was originally proposed by Paik & Pedersen [16]. Their method was based on straight edge crushing mechanisms where welding joint lines shrink axially while keeping straight during the crushing process. Amdahl’s [10] formula is employed to estimate the membrane energy. Paik & Pedersen derived two different formulas dependent on the boundary conditions of the side edges of the flange. One is the “Fix-Free (Mode I)” where one of the two side edges is fixed and the other edge free. The other is the “Fix-Fix (Mode II)” where two side edges of the plate are fixed. The mean crushing forces and the dimensionless stresses with minimized H values can be described as follows.

(i) for Fix - Free (Mode I)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.007}{\beta} + \frac{0.1334}{\beta} \right)
\]

\[ H = 0.7797\sqrt{bt} \]  

(ii) for Fix - Fix (Mode II)

\[
\frac{\sigma_m}{\sigma_0} = \frac{1}{\eta} \left( \frac{1.425}{\beta^{0.5}} + \frac{0.2672}{\beta} \right)
\]

\[ H = 0.5513\sqrt{bt} \]  

Fig. 9. Modelling of X type element by Plate Unit Method (PUM)

Fig. 9 shows the trial application of PUM (Mode2) on a prismatic plate structure, where the distance between each parallel plate element is 2b. It is seen from Fig. 9 that the X-type element (AB+CD) with the flange breadth b is equivalent to two elements of PUM Mode II (Eq.(43)) each with the breadth of 2b.

The mean crushing force in Mode II for a plate of the breadth 2b can be derived from Eq. (43) as
follows:

\[ P_n \bigg|_{b=2b} = \frac{\sigma_n}{\eta} \left( 1.425(2b)^{0.5} t^{1.5} + 0.2672t^2 \right) \]  

(45)

We can obtain equivalent dimensionless stresses for X type elements by normalizing Eq. (45) with the squash load, “\(2b\sigma_0\)”:

(iii) For equivalent X type element (Mode X)

\[ \frac{\sigma_m}{\sigma_0} = \frac{P_m \bigg|_{b=2b}}{2bt\sigma_0} = \frac{1}{\eta} \left( \frac{1.007}{\beta^{0.5}} + \frac{0.1336}{\beta} \right) \]  

(46)

Based on the simplified method for estimating the crushing force for the circular tube, see Wierzbicki et al [22], Lehmann & Yu derived expressions for the mean crushing forces for conical shell structures. In this method the circumferential membrane energy as well as the plastic bending energy is taken into account. This is different from other methods using L, T and X type elements. Moreover, two folding mechanisms, proposed by Wierzbicki et al [22], are used in this method where the active zone of plastic deformation contains two folds or buckles. It is noted that the initial inclination angle \( \alpha_0 \) before folding starts is assumed in this method. It is not analytically determined, but chosen from experiments or from FEA. Therefore, experiments or FEA are necessary in order to apply this method. Lehmann & Yu [17] obtained \( \alpha_0 \) from FEA simulations. Moreover, it is noted that this method gives the mean crushing force for the outer shell of bulbous bows with circular cross sections and the effect of stiffeners is not explicitly taken into account. A combination with other simplified methods is necessary in order to take into account the effect of internal stiffening. Lehmann & Yu combined their method with the method by Wierzbicki [8-9].

\[
P_m = \frac{22.27}{M_0} \sqrt{\frac{2RQ_1}{t}} + 2\pi Q_2 \\
\]

where \( R \) is the radius of the circular section of the conical shell, \( \theta \) the conical shell half angle, \( t \) the shell thickness. \( Q_1, Q_2 \) and \( Q_3 \) are functions of variables such as \( \theta \), initial deformation angles \( \alpha_0 \) and \( \beta_0 \), and the dimensionless ratios of folding lengths \( \xi (=H_1/H_2) \) and \( \kappa (=H_3/H_2) \). See Lehmann & Yu [17] for details.

(8) Zhang (1999) [19]

Zhang developed formulas for L, T and X type elements based on the plate unit method proposed by Paik & Pedersen [16] assuming that one of the two welding joint lines is fixed and does not deform. This condition occurs, when the distance between two adjacent intersections is large and only one intersection dents and deforms. The flow stress in plane strain condition is used in this method, i.e. the flow stress is increased about 15%. Based on Zhang’s method the present authors derived the following dimensionless stress which is slightly different from the original formula due to the introduction of the concept of effective crushing distance:

\[
\sigma_m = \frac{1}{\eta} \left( 1.64497 + \frac{0.935929}{\beta^{0.35}} \right) \\
\]

(9) Endo & Yamada (2001) [3]

Endo and Yamada developed a method especially applicable to the crushing of bulbous bow structures. They combined the method by Lehmann & Yu [7] and the method by Wang [14] such that the crushing resistance by the circular or elliptic outer shell is calculated by Lehmann’s formula for conical shells while the crushing resistance for intersection elements is calculated by Wang’s procedure.
2.4 Comparison of the reviewed procedures

The dimensionless mean crushing stress predictions as function of the slenderness ratio $\beta$ for T type elements and X type elements are shown in Fig. 10 and Fig. 11 respectively, where $\eta=2/3$ is used. The figures show that the reviewed formulas give quite different results for the same slenderness ratio, and that the mean crushing stress is sensitive to the slenderness ratio especially in the small slenderness ratio range. Since the slenderness ratios for actual ship bows lie roughly between 30 and 70, it is important to choose a suitable formula in order to accurately estimate the crushing strength.

The figures also show that

1. Yang & Caldwell’s formula (Eq.(13)) for the collapse mechanism T3, gives collapse strengths which are about the mean of all the considered T-elements. Yang & Caldwell’s formula for X2 (Eq.(19)) similarly predicts the mean strength of the X type elements. Therefore, these two formulas are used in the comparison with other formulas hereafter.

2. As for the X type element Zhang’s revised formula (Eq.(48)) gives the highest strength. On the other hand the formulas by Yang & Caldwell (X1; Eq.(17)), Abramowicz (Eq.(29)) and Paik & Pedersen (Eq.(46)) give almost the same strength prediction and constitute a lower bound for the predictions.

3. Paik & Pedersen’s equivalent formula for the X type element (Eq.(46)) gives slightly lower strength than Yang & Caldwell’s X2 formula (Eq.(19)).
Fig. 10. Comparison of the formulas for the T type element
Fig. 11. Comparison of the formulas for the X type element

(a) Zhang (1999), X type, Eq.(48)
(b) Yang & Caldwell (1988) X3 Type, Eq.(21)
(c) Wang & Ohtsubo (1999), X type, Eq.(38)
(d) Yang & Caldwell (1988) X2 type, Eq.(19)
(e) Abramowicz (1994), X type, Eq.(29)
(f) Paik & Pedersen (1995), X type, Eq.(46)
(g) Yang & Caldwell (1988) X1 Type, Eq.(17)
3. Application to the collapse strength of bulbous bow structures

3.1 Bulbous bow models

In order to perform a benchmark study and a validation, the simplified methods summarized above are applied to quasi-static axial crushing of four different designs of large-scale bulbous bow models. The models used in the experiments and their transverse sections are shown in Fig. 12 and Fig. 13. The bulbous models are made of mild steel and designed to be as close as possible to the geometry of actual ship bow structures. For simplicity, however, all the models have a circular cross section rather than elliptic cross sections and the internal structures are somewhat simplified. The models have the same shape on the outside, but have different internal structural arrangements. The models BC-E, BC-F and BC-G have transverse stiffening systems, while the model BC-L has a longitudinal stiffening system. The difference between the models BC-E and BC-F is the thickness of the outer shell.

The typical folding patterns of the outer shell of the models seen in the experiments are shown in Fig. 14 and Fig. 15. Usually two axisymmetrical folds appeared between two transverse frames. Unfortunately, the folding pattern of the plates inside the outer shell is unknown, but it is supposed that the same number of folds developed considering the continuity of the plate and the geometrical compatibility. More details of the experiments are described in Yamada & Endo [6].

Fig. 12. Scantling of the bulbous bow model. Transverse ring frames are arranged only at Fr.1 and Fr.4 in case of the model BC-L.
Fig. 13. Schematic sectional view of the bulbous bow model between adjacent transverse frames. The difference between the model BC-E and the model BC-F is the shell thickness. BC-E has a shell thickness of 10mm, which is same as the model BC-G and the model BC-L. The model BC-F has a shell thickness of 12mm. The model BC-L has 14 L-type longitudinal stiffeners.

Table 1 Material Property of the models

<table>
<thead>
<tr>
<th></th>
<th>t [mm]</th>
<th>$\sigma_y$ [MPa]</th>
<th>$\sigma_u$ [MPa]</th>
<th>$\sigma_0$ [MPa]</th>
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</thead>
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<tr>
<td>web</td>
<td>7.0</td>
<td>226</td>
<td>322</td>
<td>274</td>
</tr>
<tr>
<td>longl. stiff.</td>
<td>7.0</td>
<td>326</td>
<td>498</td>
<td>412</td>
</tr>
<tr>
<td>outer shell</td>
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<td>451</td>
<td>406</td>
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<td></td>
<td>12.0</td>
<td>302</td>
<td>451</td>
<td>377</td>
</tr>
</tbody>
</table>

Table 2
The number of unit elements used in the transverse section of each model depending on the modelling technique (IUM / PUM). Note: The number of unit elements when using a smearing out technique is not shown.

<table>
<thead>
<tr>
<th></th>
<th>L</th>
<th>T</th>
<th>X</th>
<th>Total</th>
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<tr>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BC-E/F</td>
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<td>10</td>
<td>0</td>
<td>10</td>
</tr>
<tr>
<td>BC-G</td>
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<td>4</td>
<td>1</td>
<td>5</td>
</tr>
<tr>
<td>BC-L</td>
<td>14</td>
<td>16</td>
<td>0</td>
<td>30</td>
</tr>
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</table>

<table>
<thead>
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<th></th>
<th>PUM</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>L</td>
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<td></td>
</tr>
<tr>
<td>T</td>
<td>8</td>
<td></td>
</tr>
<tr>
<td>X</td>
<td>45</td>
<td></td>
</tr>
</tbody>
</table>
Fig. 14. Number of folds on the shell of BC-G. Usually two folds appeared between two transverse frames.

Fig. 15. Typical experimental folding mechanism of the outer shell of bulbous bow model

Fr. : frame number, Sec. : section, (1-1)–(6-2) : sub-section
square : plastic hinge, \( \delta \) : displacement of rigid plate
3.2 Application of simplified methods for crushing analyses.

On the application of simplified method the following items should be noted.

(1) The mean crushing forces of longitudinal structural members are evaluated on a mid transverse section between two transverse frames while transverse ring-shaped frames are not taken into account. The sections at same vertical coordinate are used for evaluation even in case of the model BC-L with adopting the longitudinal stiffening system. Two transverse sections are used at the tip of the bow in order to capture the significant increase of sectional area. Every section is evaluated continuously in case of applying Amdahl’s method.

(2) In case of applying Yang & Caldwell’s method, the modes L1 (Eq.(7)), T3 (Eq.(13)) and X2 (Eq.(19)) are employed (See Yang & Caldwell [11]). These modes are also adopted as standard crushing modes by Ohtsubo & Suzuki [12] as well as Wang [14].

(3) As for the dimensionless effective crushing distance, $\eta=0.75$, which is obtained from the experiments, is used for the normal elements ($\theta=0\text{deg}$), whereas $\eta=0.60$ is used for the inclined elements ($\theta=18\text{deg}$) in all the examples except Amdahl’s method. The effect of an effective crushing distance is discussed later in Section 4.4.

(4) Dynamic effects such as material strain rate effect and inertia effects are not taken into account due to the quasi-static conditions during the experiments. Strain rate effects can be taken into account by the empirical method of Cowper-Symonds equation (See Paik & Wierzbicki [18] or Jones [20]).

(5) According to Eq. (40), the reduction factor $\alpha_1 = 0.87$ for inclined structural elements is used. This value corresponds to $\theta=18.62\ [\text{deg}]$ and $\eta=2/3$.

4. Results and discussion

A comparison of the experimental results and the theoretical results obtained for the crushing forces and the total absorbed energy are shown in Fig. 16 through Fig. 23

4.1 General results in the crushing force and the energy absorption

Generally Yang & Caldwell’s method (Eq. (7)) gives good estimates of the crushing strength for transversely stiffened bow models (BC-E, BC-F and BC-G). The methods by Amdahl, Wang & Ohtsubo, Paik & Pedersen and Endo & Yamada also give fairly good estimations of the crushing forces. In the case of the model BC-E/F/G, where a transverse stiffening system is used, most of the methods tend to give slightly lower crushing forces than found in the experiments. On the other hand in case of the model BC-L where a longitudinally stiffening system is used, most of the methods tend to give fairly good estimations or higher crushing forces. Most of the methods tend to slightly underestimate the mean crushing force of the first peak of the crushing force due to the effect of initial buckling phenomena. Abramowicz’s method (Eq. (28) and (29)) generally gives the lowest results among all the methods discussed in the present study and gives considerably lower results than found in the experiments. Zhang’s method (Eq. (48)), on the other hand, gives the largest crushing forces among the procedures considered in the present study and higher crushing loads than obtained experimentally in all cases. This is presumably due to an assumed crushing mechanism with fixed boundary conditions. Zhang’s formulas could be applicable to the case where only one intersection collapses at the same time such as in the early stages of the indentation of a ship side structure in a collision. Therefore it can be indicated
that Zhang’s formulas are not so well suited to predict the crushing of the present bow structures.

Fig. 16. Crushing force for the model BC-E

Fig. 17. Total absorbed energy for the model BC-E
Fig. 18. Crushing force for the model BC-F

Fig. 19. Total absorbed energy for the model BC-F
Fig. 20. Crushing force for the model BC-G

Fig. 21. Total absorbed energy for the model BC-G
Fig. 22. Crushing force for the model BC-L

Fig. 23. Total absorbed energy for the model BC-L
4.2 The effect of different X-type elements on the model BC-G

Yang & Caldwell’s method (Eq. (7)-(22)) gives relatively good predictions in the cases of the models BC-E, BC-F and BC-G. However, a slight underestimation of the crushing forces is seen in the case of the model BC-G, where T3 and X2 type elements are used. The main difference between the model BC-E/F and the model BC-G with respect to the element modelling is whether it includes an X type element or not. That is, only T type elements are used in case of the model BC-E and BC-F while both T and X type elements are used in case of test models BC-G. This indicates that an X type element results in underestimation of the experimental results and that the formula for the X type element is not suitable for the present analysis. In other words, the crushing mechanism assumed in the Yang & Caldwell’s X2 type element may be different from the crushing mechanism occurring in the experiments with the model BC-G. Ideally, we should compare the crushing mechanism in the theory and in the experiments. However, as mentioned above the actual crushing modes of the structural elements inside the outer shell of the model BC-G was not observed in the experiments.

In order to verify the effect of different X type elements on the crushing of the model BC-G, additional simplified crushing analyses of the model BC-G have been performed using T3 and X3 type elements in Yang & Caldwell’s procedure because the X3 type element is supposed to give higher crushing forces than the X2 type element as shown in Fig. 4. The comparison of crushing forces and energy absorption between case 1 (T3+X2) and case 2 (T3+X3) are shown in Fig. 24 and Fig. 25 respectively. It is seen from these figures that replacement of the X type elements does not significantly improve the results although application of the X3 type element slightly increases the crushing forces and the total absorbed energy. This modest improvement is because the sectional area for one X type element is relatively small as compared with whole cross sectional area. Another possible reason for the underestimation of the crushing forces is the effect of curvature as discussed in the following section.

![Fig. 24. Crushing force for the model BC-G](image-url)
4.3 The effect of plate curvature

Most of the methods, except the ones proposed by Zhang and Yang & Caldwell, tend to underestimate the crushing strength of the transversely stiffened bow models especially the model BC-G. This is supposed to be due to the lower number of stiffening elements in the model BC-G than in the other models as shown in Table 2. In IUM the BC-G model is divided into 5 intersection unit elements, that is 1 X type element and 4 T type elements. In this case each T type element has to include a long flange with curvature, the effect of which is not fully taken into account in the methods discussed. It is supposed that the curved plate elements will result in higher buckling strength and in higher energy absorption than flat plate elements as found in the axial crushing of circular tubes (See Lehmann & Yu [17], Jones [20], Wierzbicki [22], and Abramowicz [23-24]). Since actual bulbous bow structures may include such curved plates, the development of a simplified method to take into account the effect of the curvature of such curved plate elements is needed.

![Fig. 26. Modelling of the BC-G model by IUM (Left) and PUM (Right)](image-url)
4.4 Effective crushing distance for an inclined element

In this study, values obtained from the experiments are applied in all the analyses of the bow models (See 3.2(3)). For the present bow models, the authors have confirmed that application of \( \eta = 0.73 \) for all the elements tends to give an underestimation of the experimental results and that \( \eta = 2/3 \) tends to give results which are better correlated with the experimentally obtained crushing forces. However, the value of \( \eta \) should be discussed in more detail because it affects the results considerably. A value of \( \eta = 2/3 \) makes the estimated strength 50% higher and \( \eta = 0.73 \) makes it 37% higher than the strength obtained without considering the effective crushing distance (\( \eta = 1.0 \)). It is difficult to accurately measure the effective crushing distance in the present laboratory tests since we can almost flatten the structures with extremely high load and since the ends of the bow models are fixed in rigid foundations. That is, they will give higher resistance forces than can be expected in actual continuous bow structures. Therefore, the \( \eta \) value obtained from laboratory tests tends to be too large for estimating the actual crushing forces.

In case of crushing of structures with inclined structural elements, two kinds of dimensionless effective crushing distances can be considered. One is \( \eta_1 \) in the direction of the axial load. The other is \( \eta_2 \) in the direction of an inclined element. The dimensionless effective crushing distance \( \eta_2 \) is more relevant for calculating the mean crushing force than \( \eta_1 \) since it represents the shortening of an inclined element. Based on Fig. 7 formulas (49) and (50) can be derived for calculation of \( \eta_1 \) and \( \eta_2 \).

\[
\eta_1 = \frac{\delta_c}{L \cos \theta}
\]

\[
\eta_2 = \frac{\delta_{cl}}{L} = \frac{L - L'}{L} = 1 - \sqrt{1 - \eta_1 \cos^2 \theta (2 - \eta_1)}
\]

Fig. 27. The variation of the dimensionless effective crushing distance \( \eta_2 \) as function of the inclination angle \( \theta \) for three different values of \( \eta_1 \).

As an example we can consider the case of \( \theta = 30 \) deg and \( \eta_1 = 1.0 \). From Fig 27, it is seen that this inclined element will not be completely collapsed even if the structure is completely flattened-out in the force direction (\( \eta_1 = 1.0 \)), and that the length of the inclined element becomes half length of the original length (\( \eta_2 = 0.5 \)) in the collapsed state. In another example where \( \theta = 90 \) deg, the inclined element is not axially deformed at all (\( \eta_2 = 0 \)). Therefore, it is important to distinguish between \( \eta_1 \) and \( \eta_2 \). According to its definition \( \eta_1 \) is used for the calculation of the inclination reduction factor in Eq.(40). The dimensionless effective crushing distance \( \eta_2 \) should be used for the calculation of crushing stresses in
the formulas for the crushing forces.

As mentioned above the effective crushing distance is not accurately measured in the present bow experiments [6], but \( \eta_1 \) for the whole structure has been estimated from the final deformed shape of the bow models to be \( \eta_1 = 0.70-0.77 \). The reason why the dimensionless effective crushing distance is larger than that of prismatic structures is presumably because the deformed structural elements folds down inside the bulb shell (See Fig. 28). The corresponding value of \( \eta_2 \) for the present bow model (\( \theta = 18.62 \) deg) is 0.57-0.62 according to Eq.(50). In this study for simplicity the values \( \eta_1 = 0.75 \) and \( \eta_2 = 0.6 \) obtained from the experiments are applied in all the analyses.

In most of the past literature on crushing of structures the numerator of the absorbed energy in Eq.(1) is treated as constant even though the crushing distance in the denominator is reduced. This can be a reasonable assumption for relatively high values of \( \eta \), where the decrease of the absorbed energy in Eq.(1) can be regarded as small and therefore neglected. The reduction of energy is small because the main part of the crushing energy is absorbed in the initial stages of the folding process, when a peak of instantaneous force takes place. However, in case of lower values of \( \eta \), that is small crushing distances, the reduction in energy absorption cannot be neglected. In this case we will have a significant decrease of the absorbed energy and a shortening of the crushing distance at the same time. That is, two opposite effects of the crushing distance \( \eta \) on the mean crushing force. In this case the correction of the absorbed energy at the numerator in Eq.(1) should be calculated.

It is not always possible to obtain \( \eta \) values from the experiments, and we need to assume proper \( \eta \) values in order to estimate the crushing strengths accurately. In case of crushing of bow structures the \( \eta \) value depends on the bow tip half angle \( \theta \) as well as the density and arrangement of the internal structural elements. Fig. 29 shows the effect of different combinations of \( \eta_1 \) and \( \eta_2 \) values on the total absorbed energy of the model BC-E. Fig. 29(a) shows the case of \( \eta_1 = 0.75 \) and \( \eta_2 = 0.6 \), and Fig. 29(b) shows the case of using \( \eta_1 = \eta_2 = 2/3 \). It is found in Fig. 29 that using \( \eta_1 = \eta_2 = 2/3 \) gives slightly lower results than using \( \eta_1 = 0.75 \) and \( \eta_2 = 0.6 \), but also gives fairly good estimates especially using Yang & Caldwell’s method. The same tendencies are confirmed in the other bow models using \( \eta_1 = \eta_2 = 2/3 \). Therefore according to the present study, if empirical \( \eta \) values are not available, it might be a good solution simply to use \( \eta_1 = \eta_2 = 2/3 \) for estimation of the crushing strength of bulbous bow structures with similar \( \theta \) values and stiffeners. More experimental data for the crushing distance of complex structures are desirable due to its large effect on the crushing strength.
4.5 Equivalent plate thickness (Smearing out method)

Slightly too large force predictions are achieved in the case of the model BC-L by the Yang & Caldwell’s method. The reason for this overestimation is supposed to be due to too high prediction of the crushing strength of the relatively small longitudinal stiffeners. The concept of the equivalent plate thickness proposed by Paik et al [21] could be useful in order to take into account the effect of relatively small stiffeners. This procedure is also called smearing out technique by Lutzen [25].

Paik et al (1996) proposed the following formula for the calculation of an equivalent plate thickness.

\[
t_{eq} = t + k \frac{A_s}{b}
\]

(51)

where \( t_{eq} \) is the equivalent thickness, \( t \) is the thickness of the outer shell, \( k \) is an empirical constant usually taken to be 1.0, \( A_s \) is the sectional area of the longitudinal stiffener, \( b \) is the spacing of the stiffeners. Based on a series of experiments Paik & Pedersen conclude that the longitudinally stiffened structure could be reasonably replaced by an unstiffened plate with the equivalent plate thickness. In the present study this method is applied to the crushing of the longitudinally stiffened model BC-L in combination with Yang & Caldwell’s method. Two types of elements modelling of the bow model BC-L.
is shown in Fig. 31. It is noted that longitudinal stiffeners are arranged between Fr.2 and Fr.6. Therefore, smearing out technique is applied to the structure between Fr.2 and Fr.6. The results with or without the equivalent plate thickness in the crushing force and the total absorbed energy are compared in Fig. 33 and Fig. 34 respectively.

\[ N_L = 14, \quad N_T = 16 \]

\[ N_L = 14, \quad N_T = 2 \]

(a) without smearing out method
(b) With smearing out method

Fig. 31. Different kinds of element modelling of the model BC-L.

Example of actual bulbous section

Modelling by IUM

Fig. 32. Example illustration of the actual bulbous section and the possible modelling of realistic bulbous bow structure by IUM elements with using smearing out technique
It is seen from Fig. 33 that the mean crushing forces obtained by the smearing out method gives lower results and gives a fairly good correlation with the experimental data, especially in the latter stage of the bow crushing (displacement is larger than 0.6 m). That is, prediction of the mean crushing force is improved by using smearing out method. It is also seen in Fig. 33 that the mean crushing forces obtained by the smearing out method decreases slightly when the displacement is larger than 0.6 m. This tendency is not reasonable because the mean crushing force is supposed to increase as the transverse sectional area of the bow increases. This trend is caused by the definition of the equivalent thickness, Eq.(51). The sectional area of the longitudinal stiffeners, which is constant over the depth of the bow,
smeared out according to Eq. (51) and the result is that for a constant k value the equivalent plate thickness gets thinner as the sectional area of the outer shell increases.

It is seen from Fig. 34 that the predicted energy absorption using the smearing out method gives slightly lower results than obtained in the experiments. This discrepancy for the smeared out procedure is clearly caused by an underestimation of the mean crushing force and energy absorption at the top of the bulbous bow. An improvement in estimating the mean crushing force of the bow tip might be necessary in order to further improve the accuracy.

Even if the smearing out method gives fairly good estimates of the mean crushing force, then a slight underestimation is seen in Fig. 33. There can be two reasons for this. One is the effect of the curved plate of the outer shell in the bow as in the case of the model BC-G. As shown in Fig. 31 (b), T type elements have to cover the long curved plate when the smearing out method is used. The increase of crushing strength due to membrane stretching of the curved plates is not taken into account in the existing methods. Same tendency can be expected to take place in typical bulbous bow structures which have rather elliptical section and have large curved plates at the top and bottom as shown in Fig. 32. The other reason is the low value of the fully plastic moment calculated with equivalent plate thickness according to Eq.(51) with $k=1$. The effect of the lever of bending moment and the shift of neutral axis are not explicitly taken into account in Paik’s formula. Kierkegaard [26] analytically derived a set of formulas to estimate the fully plastic bending moment for a plate with stiffeners. The authors have confirmed that the method by Kierkegaard gives much higher plastic bending moment $M_0$ than that obtained by Paik’s method due to neglecting the effect of lateral buckling of the stiffeners. The actual value of the plastic bending moment $M_0$ is supposed to be lower than Kierkegaard’s theoretical value due to lateral and torsional buckling of webs and stiffeners subject to compression. Therefore, the relevant value of $M_0$ is supposed to lie between the values calculated by these two methods.

**Conclusions**

With the purpose of identifying suitable procedures to estimate the axial crushing strength of bulbous bow structures with internal transverse or longitudinal stiffening systems, benchmark analyses have been performed for a number of existing simplified theories. Theories based on intersection elements such as L, T and X type elements as well as theories based on plate unit elements have been analysed and applied to the axial crushing of the bulbous bow models. The results obtained by the different simplified analysis procedures are compared with those observed from large scale quasi-static bow crushing experiments. The following conclusions are derived from the present study:


2. A dimensionless effective crushing distance, $\eta_2$, for structural elements with an inclination angle to the force direction is introduced as a function of the dimensionless effective crushing distance in the force direction ($\eta_1$). It is confirmed that using $\eta_1=0.75$ obtained from the experiments and the value $\eta_2=0.6$ results in good crushing force estimates. When no experimental data is available it is also found that a good approximation is to use $\eta_1=\eta_2=2/3$.

3. Most of methods tend to give higher crushing force estimates for the longitudinally stiffened bow (the model BC-L) than obtained in the experiments. This is mainly due to the overestimation of the crushing strength of the longitudinal stiffeners. However, Yang & Caldwell’s method in combination with a smearing out method proposed by Paik et al
improves the accuracy and gives fairly good estimations of the experimental results.

(4) Most of the methods tend to give slightly lower crushing force estimates for less closely stiffened bow structures, presumably due to the effect of plate curvature. The development of a simplified method to take into account the effect of the curvature of such curved plate elements is needed. Moreover, the considered procedures tend to underestimate the mean crushing force and energy absorption at the top of the bulbous bow. An improvement in estimating the mean crushing force of the bow tip might be necessary in order to further improve the accuracy.

In this study the authors have mainly focused on the application of existing crushing formulae to the problem of quasi static bow crushing. The applied theories are not all derived for this purpose, therefore, the comparisons in this paper do not intend to fully follow the initial intension of the theoretical derivations.

Acknowledgments

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REFERENCE

APPENDIX

(1) Yang & Caldwell’s formula normalized by fully plastic moment (M₀) with η = 2/3

(i) Mode 1 for L type element (Mode L₁)

\[
\frac{P_{mL}}{M_0} = 18.25 \sqrt[3]{\frac{b}{t}}
\]

(ii) Mode 1 for T type element (Mode T₁)

\[
\frac{P_{mT1}}{M_0} = 20.89 \sqrt[3]{\frac{b}{t}}
\]

(iii) Mode 2 for T type element (Mode T₂)

\[
\frac{P_{mT2}}{M_0} = 15.49 \sqrt[3]{\frac{b}{t}} + 2.35619
\]

(iv) Mode 3 for T type element (Mode T₃)

\[
\frac{P_{mT3}}{M_0} = 16.99 \sqrt[3]{\frac{b}{t}} + 30.10
\]

(v) Mode 4 for T type element (Mode T₄)

\[
\frac{P_{mT4}}{M_0} = 25.62 \sqrt[3]{\frac{b}{t}} + 3.53
\]

(vi) Mode 1 for X type element (Mode X₁)

\[
\frac{P_{mX1}}{M_0} = 19.61 \sqrt[3]{\frac{b}{t}} + 30.1
\]

(vii) Mode 2 for X type element (Mode X₂)

\[
\frac{P_{mX2}}{M_0} = 30.08 \sqrt[3]{\frac{b}{t}} + 6.0
\]

(viii) Mode 3 for X type element (Mode X₃)

\[
\frac{P_{mX3}}{M_0} = 34.16 \sqrt[3]{\frac{b}{t}} + 4.71
\]

(2) Ohtsubo & Suzuki’s formula normalized by fully plastic moment (M₀) with η = 2/3

(i) Mode 1 for L type element (Mode L₁), which is identical to (52)

\[
\frac{P_{mL1}}{M_0} = 18.25 \sqrt[3]{\frac{b}{t}}
\]
(ii) Mode 2 for L type element (Mode L2)

\[ \frac{P_{ml2}}{M_0} = 15.04 \sqrt[4]{\frac{b}{t}} \]  

(iii) Mode 3 for L type element (Mode L3)

\[ \frac{P_{ml3}}{M_0} = 17.89 \sqrt[4]{\frac{b}{t}} + 2.35619 \]  

(iv) Mode 4 for L type element (Mode L4)

\[ \frac{P_{ml4}}{M_0} = 17.08 \sqrt[4]{\frac{b}{t}} + 1.1781 \]
Risk Reducing Effect of Buffer Bow Structures on the Collision Damage of Oil Tankers

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ABSTRACT

In order to investigate the effectiveness of buffer bulbous bow structures on prevention of oil spills in tanker collisions, ship collision analyses are performed using simplified ship collision analysis tool. The energy to be absorbed by structural deformation is calculated by considering global motions of both ships (external dynamics). Deformation of both a striking and a struck ship is independently calculated by use of the simplified methods by assuming rigid-perfectly plastic material. A Monte Carlo Simulation (MCS) has been carried out for various collision scenarios. The probability of oil spill from struck tankers when assuming all striking ships have a buffer bow is compared with that when all striking ships have a standard bulbous bow.

KEY WORDS: buffer bow, ship collision, simplified analysis, Monte Carlo simulation

1. INTRODUCTION

In order to prevent oil spills from struck tankers, the double hull system has become a de facto standard as an effective countermeasure. However, it is still a fact that collision accidents involving struck double hull tankers result in oil spills. One example is the collision involving the double hull oil tanker “Baltic Carrier”. Therefore there are good reasons to further minimize the consequences of ship collision.

In order to reduce the risk of oil spill from struck oil tankers the Buffer Bow Project (BBP) has been carried out in NMRI and the project is sponsored by the Japanese Ministry of Land

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Infrastructure and Transport (MLIT). The results presented in this paper consist of a part of the results obtained in the BBP.

As a part of the BBP, the effect of buffer bow on the protection from oil spill from D/H VLCC has been investigated (Endo and Yamada, 2004; Yamada et al, 2005) by use of large finite element analyses (FEA). These studies show that the buffer bulbous bow is effective in reducing the risk of oil spill from specific D/H VLCC in specific collision scenarios.

However it is necessary to further investigate the effect of bulbous bow structures in various ship sizes and types and collision scenarios. In this case FEA is not suitable and a more simplified approach is needed.

In this study the effectiveness of buffer bow structures on ships colliding against tankers has been investigated. Ship collision analyses have been carried out using a simplified ship collision analysis tool (SSCAT). The energy to be absorbed by the structural deformation has been calculated by considering the global motions of both ships (external dynamics). Deformation of both a striking and a struck ship is independently calculated by use of the simplified methods by assuming rigid-perfectly plastic material. A Monte Carlo Simulation (MCS) has been conducted for various collision scenarios and the probability of oil spill from struck oil tankers, when assuming that all striking ships have a buffer bow, is compared with the probability when all striking ships have a standard bulbous bow. The results are discussed in detail.
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2. COLLISION SCENARIO

In order to perform the reliability analysis of struck tankers some assumptions and collision scenarios need to be established. The collision scenario is assumed to be that of a striking ship with bulbous bow colliding perpendicularly with a double hull tanker in a fully loaded condition as shown in Fig. 6.6. As a first step of the study, large crude oil tankers are assumed to be both the struck ship (Ship A) and the striking (Ship B). The notation A and B, usually in the subscript, represents the struck ship and striking ship, respectively. The forward speeds of both ships are taken into account.

Fig. 2.1 Illustration of collision scenario (Top View).
3. ANALYSIS METHOD

3.1 Existing Analysis Methods and Tools

The methods for evaluating the structural consequences of a given ship collision can be classified into four categories such as

1. Empirical formula
2. Simplified analysis methods
3. Non-linear FEA simulation
4. Experiments

Usually the time and cost increase as the number increases from (1) to (4). Among these non-linear FEA simulation is becoming a more and more practical design tool as the computer technology progress significantly. However, it still takes a lot of time to construct FEA models as well as to carry out the calculations. For example, in this study, it takes about 1-2 months to build one ship finite element model, and it takes 4 or 5 days using 4 parallel CPU computers to calculate 1 ship-ship collision case where about 500,000 nodes are used in total. Explicit FEA calculations do not always give convergence. Then the process must be restarted after correcting the model or after reducing the time steps. Even then there is no guarantee that the next calculation sequence converges. That is, the number of calculation cases which in a practical way can be carried out by FEA is limited due to the enormous number of possible collision scenarios. FEA is at the moment only suitable for a deterministic approach and not fitting for a probabilistic approach such as risk and reliability analyses. Furthermore, it cannot be applied in the early stages of the ship design. Therefore, simplified analysis tools, which can rapidly evaluate the crashworthiness of ship structures rapidly and with reasonable accuracy, are in high demand.

Some of the simplified ship collision analysis models developed in the past are briefly reviewed in the following. A comparison of existing simplified analysis tool is summarized in Table 3.1.

(1) GRACAT (Friis-Hansen & Simonsen, 2002)
GRACAT (Software for grounding and collision analysis) developed at the Technical University of Denmark (DTU) is a comprehensive software tool dealing with risk analysis of both collision and grounding of ships. The program calculates a deterministic/probabilistic damage extent of ships as well as the probability of these events, and the risk of collision and grounding can be calculated. GRACAT solves the external dynamics and the internal dynamics independently. The external dynamics is calculated by using the method of Pedersen and Zhang (1998), which is applicable for both right angle and oblique collisions. Super elements are used to calculate structural deformation when assuming a rigid bow.

(2) SIMCOL (Brown et al, 2000; Brown, 2002a, 2002b)
SIMCOL (Simplified Collision Model) uses a time-domain coupled solution of the external dynamics and the internal mechanics similar to a method originally proposed by Hutchison (1986). The external model uses a three degree of freedom system for ship dynamics. The internal model determines forces from crushing of side and bulkhead structures using mechanisms adapted from the Rosenblatt (1975) study. The program determines the energy absorbed by the crushing and tearing of decks, bottom and stringers.

(3) DAMAGE (Abramowicz & Sinmao, 1999; Simonsen, 1999b)
DAMAGE developed at MIT solves the external dynamics and the internal dynamics independently. Lost kinetic energy to be absorbed by the internal dynamics is calculated by external dynamics, where the forward motion of the struck ship is assumed to be zero and the program is only applicable for right angle collisions. Structural deformation is calculated by use of Super Elements. The calculation of the external dynamics is based on the conservation of energy and momentum as described by Minorsky (1959).
Table 3.1 Brief comparison of existing simplified ship collision analysis tool

<table>
<thead>
<tr>
<th></th>
<th>GRACAT (DTU)</th>
<th>DAMAGE 5.0 (MIT)</th>
<th>SIMCOL 2.1 (VT)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulation</td>
<td>Displacement domain</td>
<td>Displacement domain</td>
<td>Time domain</td>
</tr>
<tr>
<td>Internal Dynamics</td>
<td>Rigid-plastic Analysis</td>
<td>Rigid-plastic Analysis</td>
<td>Minorsky(1959), McDermott / Rosenblatt method</td>
</tr>
<tr>
<td></td>
<td>With Super Element</td>
<td>With Super Element</td>
<td></td>
</tr>
<tr>
<td>Striking ship Bow</td>
<td>Rigid</td>
<td>Deformable</td>
<td>Rigid</td>
</tr>
<tr>
<td></td>
<td>3DOF[^1], any collision angle</td>
<td></td>
<td>3DOF[^1], any collision angle</td>
</tr>
<tr>
<td>Coupling External &amp; Internal</td>
<td>-</td>
<td>-</td>
<td>Yes</td>
</tr>
<tr>
<td>Oblique collision</td>
<td>-</td>
<td>-</td>
<td>Yes</td>
</tr>
<tr>
<td>Risk Analysis</td>
<td>Yes</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

[^1] 3DOF: 3-degrees of freedom (sway, surge, yaw)

3.2 Present Method
In this study a new Simplified Ship Collision Analysis Tool (SSCAT) is developed and it is expanding the structural analysis part of GRACAT. The main emphasis of the present method is on the improvement of the evaluation of the deformations and strength of the bulbous bow structures since a precise prediction of the bulbous bow damage is important in the present study. The details of SSCAT are described in the following chapter.

As described above, there are two types of simplified collision analysis methods from the viewpoint of treatment of external and internal dynamics. One is the method which uncouples external and internal mechanics. The other is a method which couples these phenomena. Brown (2002b) conducted a comparative study between a coupled method and the uncoupled method by Pedersen and Zhang (1998). Brown (2002b) found that the uncoupled method does not seem to achieve a high degree of accuracy when predicting the absorbed energy in the longitudinal direction of the struck ship. However, it is also found from these results that the total absorbed energy by structural deformation is almost the same for the two methods.

Therefore, in this study the uncoupled method is employed using the method proposed by Pedersen and Zhang (1998) for external dynamics. This method can model oblique collisions as well as right angle collisions. Moreover, the closed formulas proposed by Pedersen and Zhang make the calculation fast and thus suitable for risk and reliability analyses.
4. EXTERNAL DYNAMICS

This section describes two methods for calculating what we call Lost Kinetic Energy (LKE), which should be absorbed by structural deformation or friction. The LKE is calculated by taking the global ship motions in ship-ship collisions into account. A first method was initially derived by Minorsky (1959) and it is only applicable for perpendicular collision. The second method, developed by Pedersen and Zhang (1998), is a more general and powerful method since it is also applicable for oblique collisions. Moreover this method is applicable for the case of ships colliding with floating objects or with quays.

4.1 Simplified Method for Perpendicular Collision

If a ship collides perpendicularly with the midship region of a struck ship with no forward velocity, the closed formula to estimate the LKE can be simply derived as follows. It is assumed that both ships are mathematically modeled as one point masses. It is also assumed that both ships stick and move together at the same velocity after the collision without rotation.

In this collision scenario we can derive the following equations from the conservation law of energy and momentum (Minorsky, 1959; Yamada et al, 2005).

From conservation of momentum:

\[
M_B'V_B = (M_A' + M_B')V
\]  
(1)

From conservation of energy:

\[
E_0 = \frac{1}{2}M_B'V_B^2 = \frac{1}{2}(M_A' + M_B')V^2 + E_s
\]  
(2)

where \( M_A' \) and \( M_B' \) denote the mass of the struck and the striking ship including the virtual added mass. As coefficients of virtual added mass for sway, \( r_{sway} = 1.4 \), and surge \( r_{surge} = 1.1 \) are often used (Minorsky, 1959; Motora et al, 1969). \( V_B \) and \( V \) denote the initial velocity of the striking ship before collision and the common velocity after collision. \( E_0 \) is the initial kinetic energy of the striking ship.

By eliminating \( V \) from Eqs. (1) and (2), the lost kinetic energy which must be absorbed by phenomena other than ship motions, LKE, can easily be derived as:

\[
LKE = \frac{1}{2} M_B' V_B^2 \left( \frac{M_A'}{M_A' + M_B'} \right) = E_0 \left( \frac{M_A'}{M_A' + M_B'} \right)
\]  
(3)

The ratio between LKE and \( E_0 \) is expressed as:

\[
R_s = \frac{LKE}{E_0} = \left( \frac{M_A'}{M_A' + M_B'} \right)
\]  
(4)

If \( M_A' = 420,000 \) [ton], \( M_B' = 330,000 \) [ton] and \( V_B = 5.1 \) [m/s], \( LKE = 2445.39 \) [MJ] can be calculated. In this case \( R_s = 56 \% \) of the initial kinetic energy has to be absorbed by the structural deformations and frictions.

It is interesting to note that the energy which should be dissipated by the structural deformation is independent of the structural strength of both ships. According to the formula the collision phenomena proceeds until the deformation energy reaches LKE regardless of the rupture of outer shell and inner shell. It is interesting to note that the energy ratio \( R_s \) dissipated by structural deformation and friction is in this case determined only by the masses of the two ships (including added mass) regardless of the structural strength of both ships.
4.2 Pedersen & Zhang ‘s (1998) Method

Pedersen & Zhang (1998) developed a 2-dimensional analytical method to predict the energy which should be absorbed by structural deformation and friction (what we call “Lost Kinetic Energy (LKE)”) during the ship collision. This method gives closed form analytical formula for the solution, and therefore is extremely powerful, fast and suitable for simplified collision analyses. Another characteristic is that this method is applicable not only for perpendicular collisions but also for oblique collisions, and therefore the procedure is very flexible and applicable for generalized collision scenarios.

The method treats three degree of freedom (surge, sway and yaw motions) of both a striking and a struck ship. The pressure effect of sea water corresponding to the above three ship motions is taken into account as a virtual added mass. By solving force and moment equilibriums as well as conservation of momentum and energy, the lost kinetic energy (LKE) which should be absorbed by structural deformation and frictions can be obtained in closed form. In this method it is assumed that the collision phenomena takes place within a very short time and, therefore, the collision angle $\beta$ and the contact surface angle $\alpha$ do not change significantly during the collision (see Fig. 4.1). That is, $\alpha$ and $\beta$ are assumed to be constant. This assumption might not be completely satisfied if the collision point is far away from the center of gravity of the struck ship since yaw motion of struck ship can be anticipated. However, in this case the damage of the struck ship is supposed to be much smaller than that of the collision at midship due to the larger ship motion. Consequently the effect of this on the calculation results, especially the probability of rupture of inner shell of the struck ship, is expected to be small. Therefore this assumption is considered to be acceptable for use in the present study. It is also noted that the roll motion of the struck ship is not taken into account in this method. Yamada (2005) conducted large scale FEA and found that the effect of roll motion is not so small if a striking ship is in a ballast condition and its mass is larger than that of a struck ship. However Yamada qualitatively found that the effect of roll motion on the structural energy absorption before rupture of oil tanks is not so significant. This is because the rupture of the oil tanks usually takes place before the roll motion becomes significant if the striking ship has a sharp bulb or is relatively stronger than struck ship side. The effect of roll motion is important if the phenomena or results after the rupture of the oil tanks are important.

![Fig. 4.1 The coordinate system used for analysis of ship-ship collision](image-url)
Based on the two sets of equations of motion for the two ships the relation between impact force and relative acceleration from the view point of the striking ship in η-ξ direction can be derived as follows.

\[
\begin{align*}
\begin{bmatrix}
\ddot{\eta} \\
\ddot{\xi}
\end{bmatrix} &= \begin{bmatrix}
-K_\eta & -K_\xi \\
-D_\eta & -D_\xi
\end{bmatrix} \begin{bmatrix}
F_\eta \\
F_\xi
\end{bmatrix} \\
&= \begin{bmatrix}
D_\eta K_\eta - D_\xi K_\xi \\
D_\xi K_\eta - D_\eta K_\xi
\end{bmatrix} \begin{bmatrix}
-\dot{D}_\xi & K_\xi \\
D_\xi & -K_\xi
\end{bmatrix} \begin{bmatrix}
\ddot{\eta} \\
\ddot{\xi}
\end{bmatrix}
\end{align*}
\]  

(5)

where \(\ddot{\eta}\) and \(\ddot{\xi}\) are relative accelerations in the η and the ξ direction respectively. \(F_\eta\) and \(F_\xi\) are impact forces in the η and the ξ direction respectively. \(K_\eta, K_\xi, D_\eta\) and \(D_\xi\) are functions of several parameters of both ships such as mass and moment of inertia including added mass, collision angle and point and so on. \(K_\eta, K_\xi, D_\eta\) and \(D_\xi\) are regarded as constants.

The impact force components can be derived as;

\[
\begin{align*}
\begin{bmatrix}
F_\eta \\
F_\xi
\end{bmatrix} &= \begin{bmatrix}
-K_\eta & -K_\xi \\
-D_\eta & -D_\xi
\end{bmatrix}^{-1} \begin{bmatrix}
\ddot{\eta} \\
\ddot{\xi}
\end{bmatrix} = \frac{1}{D_\xi K_\eta - D_\eta K_\xi} \begin{bmatrix}
-D_\xi & K_\xi \\
D_\xi & -K_\xi
\end{bmatrix} \begin{bmatrix}
\ddot{\eta} \\
\ddot{\xi}
\end{bmatrix}
\end{align*}
\]  

(6)

By integrating the impact force with respect to time, the impulses in η and ξ direction can be derived as;

\[
\begin{align*}
I_\eta &= \int_0^T F_\eta dt = \frac{D_\xi \ddot{\eta}(0) - K_\xi \ddot{\xi}(0)(1 + e)}{D_\xi K_\eta - D_\eta K_\xi} \\
I_\xi &= \int_0^T F_\xi dt = \frac{K_\eta \ddot{\xi}(0)(1 + e) - D_\eta \ddot{\eta}(0)}{D_\xi K_\eta - D_\eta K_\xi}
\end{align*}
\]  

(7)  

(8)

where \(e\) denotes the coefficient of restitution defined as

\[
e = -\frac{\ddot{\xi}(T)}{\ddot{\xi}(0)}
\]  

(9)

e depends on the material property of the two colliding objects (Pedersen & Zhang, 1998). The ratio of two impulses is defined as

\[
\mu = \frac{I_\eta}{I_\xi}
\]  

(10)

If the ratio of impact impulse, \(\mu\), is lower than a critical value \(\mu_c\), the bow is assumed not to slide in η direction due to the friction force. However if \(\mu\) is higher than \(\mu_c\), that is, the sliding force \(F_\eta\) becomes larger than the maximum static friction force, then the bow slides in the η direction. In the latter case \(F_\eta\) becomes constant value equal to \(\mu_c F_\xi\) during the sliding independent of the relative velocity in η direction. Considering the uneven contact surface in collision \(\mu_c=0.7\) is used in this study. The total LKE is the sum of the \(E_\eta\) and \(E_\xi\) where:

(a) \(\mu < \mu_c\) Sticking Case

if \(K_\xi + \mu K_\eta \neq 0\) then

\[
E_\eta = \int_0^{t_{\text{max}}} F_\eta d\eta = \frac{1}{2} \frac{\mu}{K_\xi + \mu K_\eta} \dot{\eta}(0)^2
\]  

(11)

\[
E_\xi = \int_0^{t_{\text{max}}} F_\xi d\xi = \frac{1}{2} \frac{1}{D_\xi + \mu D_\eta} \dot{\xi}(0)^2
\]  

(12)

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(b) $\mu \geq \mu_c$ Sliding Case

$\mu$ is replaced by $\mu_c$ and the energy formula becomes:

$$E_\eta = \int_0^{\eta_{\max}} F_\eta d\eta = \frac{1}{2} \frac{\mu_c}{K_\eta} \left( \eta(0)^2 - \eta(T)^2 \right)$$  \hspace{1cm} (13)

$$E_\xi = \int_0^{\xi_{\max}} F_\xi d\xi = \frac{1}{2} \frac{1}{D_\xi} \left( \xi(0)^2 - \xi(T)^2 \right)$$ \hspace{1cm} (14)

4.2.1 Example Calculation 1: Perpendicular Collision

Example calculations were carried with $M_A=M_B=300$ [kton], $V_B= 5.14$ [m/s] (10 [kt]), $\beta = 90$ [deg], $\alpha=0$ [deg], $\epsilon=0.0$, $\mu_c=0.7$ while changing the forward speed of the struck ship ($V_A$) and the dimensionless longitudinal collision point from midship independently. The relation between the total lost kinetic energy and the dimensionless collision point is shown in Fig. 4.2. It is confirmed that the LKE at midship obtained by this method is identical to that obtained by the closed formulas described in Eq.(3). That is, LKE $= 2445$ [MJ]. Considering that the results are not sensitive to the $\epsilon$ value and that the bow of the striking ship tends to stick to the side shell of the struck ship, $\epsilon=0$ is employed in the present analyses. Fig. 4.1 shows that LKE is almost symmetrical around the midship section of the struck ship. As expected it is seen that LKE is highest when the collision point is at midship of the struck ship since yaw motion of the struck ship as well as its kinetic energy is supposed to be small. Therefore, in the case of perpendicular collisions, striking at the midship region of the struck ship is the most severe case for the struck ship even though the struck ship has forward speed.

![Graph showing lost kinetic energy (LKE) in relation to Dimensionless Longitudinal Collision Point (DLCP) of the struck ship collision point curve depending on the coefficient of restitution.](image)

Fig. 4.2 Lost kinetic energy (LKE) in relation to Dimensionless Longitudinal Collision Point (DLCP) of the struck ship collision point curve depending on the coefficient of restitution. $M_A=M_B=300,000$ [ton], $V_B= 10$ [knot], $\beta = 90$ [deg], $\alpha=0$[deg], $\mu_c = 0.7$, $\epsilon=0.0$. 

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5. INTERNAL DYNAMICS

5.1 Material Model

5.1.1 Flow Stress

In this study the material of the ships is assumed to be steel such as mild steel or high tensile steel which are usually used in large oil tankers. As normally done, the material is assumed to be perfectly rigid-plastic in the present ship collision analysis since the energy absorption by elastic deformation is much smaller than that by plastic deformation and thus negligible (Jones, 1989). The concept of flow stress is used to derive simplified formula for the mean crushing strength in a later section. Mathematical formula of the static flow stress, $\sigma_0$, for ductile material can be defined as follows;

$$\sigma_0 = \frac{\sigma_y + \sigma_u}{2}$$  \hspace{1cm} (15)

where $\sigma_y$ and $\sigma_u$ are the yield stress and the tensile strength of the material, respectively.

5.1.2 Strain Rate Sensitivity

Strain rate sensitivity is taken into account by use of the following Cowper-Simons formulas (Jones, 1989):

$$\sigma_0^d = \sigma_0 \left[1 + \left(\frac{\dot{\varepsilon}}{C}\right)^{\frac{1}{P}}\right]$$  \hspace{1cm} (16)

where $\sigma_0^d$ denotes dynamic flow stress. C and P are constants which can be obtained empirically. In case of mild steel $C=40.4$ [s$^{-1}$], $P=5$ are usually used (Jones, 1989). ASIS (1997) derived empirical values for high tensile steel $C=2560$, $P=5$ from the experiments. The effect of strain rate is shown in Fig. 5.1 as the variation of the strain rate value. From Fig. 5.1 it is seen that the effect of strain rate effect is larger for mild steel than for high tensile steel.

![Fig. 5.1 Strain rate effect on the flow stress. MS: mild steel (C=40, P=5), HT: high tensile steel(C=2560, P=5).](image)
5.2 Basic Collapse Mode and Simplified formulas

5.2.1 Lateral loading of side shell plate

The lateral loading of plate is assumed to be represented by a point load. Fig. 5.2 illustrates a stiffened plate subjected to a lateral point load. In the early stage of the collision the probability that the bow collides with the plate between stiffeners is supposed to be relatively higher than the probability that the bow collides just on the web. Therefore, in most cases this formula is applied from the early stage of the collision until rupture of the plate takes place. In case that the bow touches the edge of the plate (stringers or transverse webs in Fig. 5.2) the crushing strength of these structural elements are added.

![Fig. 5.2 Deformation pattern for a rectangular plate subjected to lateral point loading](image)

A simplified formula to estimate the mean indenting force of a longitudinally stiffened plate subjected to a point load was developed by Wierzbicki and Simonsen (1996) as follows.

\[
F_m = \frac{\delta A}{6\sqrt{3}} \left[ \frac{3N_{0x} + N_{0z}}{L_1L_2} + \frac{3N_{0x} + N_{0z}}{D_1D_2} \right] \quad (17)
\]

where
- \(\delta\) : indentation of the plate at the point P by the point load.
- A : area of plate (=L x D).
- \(N_{0x}\) (=\(\sigma_0t_x\)) : fully plastic membrane force per unit length in x direction.
- \(N_{0y}\) (=\(\sigma_0t_z\)) : fully plastic membrane force per unit length in y direction.
- \(\sigma_0\) : flow stress.
- \(t_x\) : plate thickness of the plate including equivalent thickness of the longitudinal stiffeners.
- \(t_z\) : plate thickness of the plate. Usually the same as the thickness of the plate itself.
- \(L_1,L_2, D_1,D_2\) : Distance from the point of load to the stringer or transverse web (see Fig. 5.2).
5.2.2 Denting of the Deck and Frames

A formula derived by Zhang (1999) is employed in order to estimate mean crushing force for in-plane plate denting:

\[ F_m = 1.623 \sigma_0 \left[ \frac{b_1 + b_2}{t^\frac{5}{3}} \right] \]

5.2.3 Axial crushing of the plate intersections

Several simplified formulas have been developed in the literature in order to estimate the mean axial crushing strength of plate intersections (Wierzbicki, 1983; Amdahl, 1983; Yang & Caldwell, 1988; Kierkegaard, 1993; Abramowicz, 1994; Wang, 1995; Paik & Pedersen, 1995; Paik et al, 1996; Paik & Wierzbicki, 1997; Zhang, 1999; Endo & Yamada, 2001). Most of the simplified methods are categorized into two groups from the viewpoint of modelling technique as illustrated in Fig. 5.4, Fig. 5.5 and Fig. 5.6. However, the basic principle of deriving the closed formulas is the same in most of the studies although slightly different crushing mechanisms are assumed. Yamada & Pedersen (2007) made a comprehensive benchmark study of the existing simplified formulas by comparing the results of simplified method with those of a series of half-scale bulb crushing experiments. In this benchmark study it was found that the methods by Amdahl (1983), Yang & Caldwell (1988), Wang (1995), Paik & Pedersen (1995) and Endo & Yamada (2001) give fairly good estimations. Among these Yang & Caldwell (1988)’s method gives relatively higher collision force predictions while other method tends to give slightly smaller predictions than found in the experiments. In this study, therefore, the methods developed by Yang & Caldwell are applied for estimating the mean axial crushing force of the plate intersections in both striking and struck ships. The mathematical formulas for mean crushing force for T-type and X-type elements are as follows:

for T-type element

\[ F_m = \frac{\sigma_0}{\eta} \left( 2.83 b^{0.5} t^{1.5} + 5.02 t^2 \right) \]  

(19)

for X-type element

\[ F_m = \frac{\sigma_0}{\eta} \left( 5.01 b^{0.5} t^{1.5} + t^2 \right) \]  

(20)

where b, t, s denote average breadth of the plate from intersection, average thickness of the plate and average flow stress, respectively.
5.2.4 Tearing of the outer shell after rupture

After rupture initiation a plate is assumed to deform in a concertina tearing mode (Simonsen, 2000). The following formula developed by Wierzbicki (1995) is employed to calculate the mean crushing force of the outer shell after rupture:

\[
F_m = 4.33 \sigma_0 \delta^3 \left( \frac{b_1 + b_2}{2} \right)^{\frac{1}{3}} + \frac{8}{3} R t
\]  \hspace{1cm} (21)

where
- \( b_1, b_2 \): distances from the point of rupture to the nearest transverse frame.
- \( R \): the specific work done by rupture. \( R = m \sigma_0 \delta_t \)
- \( m \): three dimensionality factor
- \( \delta_t \): the crack opening displacement (COD)
5.3 Plate Rupture
5.3.1 Effective Plastic Strain
The rupture of the plate and the propagation of cracks are very complicated problems. Therefore, a simplified method is used. That is, the plate causes rupture when the strain in one direction reaches a critical value, which is the rupture strain (RS). Zhang (1999) used the following formula to roughly estimate the strain of the plate in one direction where two-dimensional effects are disregarded.

$$
\varepsilon_{\text{strecthing}} = \sqrt{\frac{L^2 + \delta^2}{L}} - \frac{L}{L} = \frac{1}{L} \left( \frac{\delta}{L} \right)^2 - 1 \approx \frac{1}{2} \left( \frac{\delta}{L} \right)^2
$$

(22)

Fig. 5.7 Indentation and stretching of the side shell plate. The boundary structural members change as the penetration proceeds especially after the buckling of transverse web.
5.3.2 RTCL Criteria

In many of the simplified analyses only one directional strain is considered to determine the rupture of the shell. However in the case of collision it can be more reasonable to consider a bi-axial tension stress state if the side shell plate is subjected to lateral loading.

Tornquist (2003) and Urban (2003) have developed a fracture criterion by considering 3-dimensional stress state of ductile material, which we call stress triaxiality. To arrive at the fracture criterion Tornquist and Urban combined two triaxiality functions developed by Cockcroft & Latham (1968) and by Rice & Tracey (1969). They derived the following fracture criterion by using a damage parameter D, where fracture takes place if D reaches a critical value. Reference is left to Tornquist (2003) and Urban (2003) for the details.

\[
D = \int f(\alpha) d\varepsilon_{eq} \tag{23}
\]

Where \(\alpha = \frac{\sigma_m}{\sigma_H}\) denotes dimensionless parameter representing stress triaxiality, while \(\sigma_H\) and \(\sigma_{eq}\) denote the hydrostatic stress and the equivalent stress. \(f(\alpha)\) is the stress triaxiality function, which varies depending on the range of \(\alpha\) value as follows:

\[
f(\alpha) = \begin{cases} 
  f_{CL}(\alpha) = \frac{2 + 2\alpha \sqrt{12 - 27\alpha^2}}{3\alpha + \sqrt{12 - 27\alpha^2}} & \text{for } -\frac{1}{3} \leq \alpha < \frac{1}{3} \\
  f_{RT}(\alpha) = \exp\left(\frac{3}{2}\alpha\right) & \text{for } \alpha \geq \frac{1}{3}
\end{cases} \tag{24}
\]

In the case of lateral loading of a stiffened plate, a plane stress condition can be assumed (\(\sigma_3=0\)). The membrane effect is dominant while the bending effect is relatively small or negligible (Jones, 1989). According to the assumption of rigid-plastic material property and by neglecting the shear stress at the loading point it can be said that the two orthogonal principal stresses at the point of loading are equal to the flow stress (\(\sigma_1 = \sigma_2 = \sigma_0\)). Thus, the hydrostatic stress can be expressed as;

\[
\sigma_H = \frac{1}{3}(\sigma_1 + \sigma_2 + \sigma_3) = \frac{2\sigma_0}{3} \tag{25}
\]

The equivalent stress can be calculated by

\[
\sigma_{eq} = \sqrt{\frac{1}{2}[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]} \tag{26}
\]

Substituting \(\sigma_1 = \sigma_2 = \sigma_0\) and \(\sigma_3=0\) yields

\[
\sigma_{eq} = \sqrt{\frac{1}{2}[(\sigma_0 - \sigma_0)^2 + (\sigma_0 - 0)^2 + (0 - \sigma_0)^2]} = \sigma_0 \tag{27}
\]

The stress triaxiality in this condition is

\[
\alpha = \frac{\sigma_H}{\sigma_{eq}} = \frac{2\sigma_0}{3} \frac{1}{\sigma_0} = \frac{2}{3} \tag{28}
\]

In this case the triaxiality function \(f_{RT}\) in Eq(24) is applicable, and the following formula (Urban, 2003) can be used to estimate fracture strain \(\varepsilon_{cr}\).

\[
\varepsilon_{cr} = 1.65\varepsilon_0 \exp\left(-\frac{3}{2}\alpha\right) \tag{29}
\]

where \(\varepsilon_0\) denotes the fracture strain in a uni-axial tensile coupon test.
Substituting (28) into (29) yields:

\[ \varepsilon_{cr} = 1.65 \varepsilon_0 \exp(-1) = \frac{1.65}{e} \varepsilon_0 = 0.607 \varepsilon_0 \]  

(30)

This means that the fracture strain in a bi-axial tension state with two equal orthogonal principal stresses is 40% lower than that in a uni-axial tension state. It is noted that \( \varepsilon_0 \) is largely dependent on gauge length of the tensile test specimen according to Barba’s law. Therefore the critical damage value \( D \) for specific material is not constant and is dependent on the gauge length (or uni-axial failure strain).
6. SIMPLIFIED SHIP COLLISION ANALYSIS TOOL (SSCAT)

This chapter presents the details of a simplified ship collision analysis tool (SSCAT) which utilizes the procedures for internal and external dynamics described in the previous chapters. Mathematical modelling of both the striking and the struck ships is described, and an algorithm of contact between the bow of the striking ship and the structural members of the struck ship is described.

6.1 Modelling of the striking ship

Recently more and more ships have bulbous bows in addition to conventional upper parts of the bow. Moreover, bulbous bows have higher potential risk of penetrating the side shell of struck tankers than the upper part of the bow since the bulbous bow is often vertically located at the most weak part of the side shell structures. In this study the striking ship is assumed to have both an upper part of bow and a bulbous bow. That is, the bow is divided into stem and bulb, and each part is modelled and treated independently. The stem is modelled by a triangular pyramid following Zhang (1999).

The shape of most bulbous bows can roughly be divided into two groups. One is the blunt shaped bulbous bow (Fig. 6.1(a)) having a vertically flat tip, the other is blunt shaped bulbous bow (Fig. 6.2(a)) having round tip. However, from the viewpoint of mathematical modelling of a bulbous bow, these two groups are here assumed to be of similar shape. Two types of mathematical modelling can be considered. One is based on a spherical representation and the other is an ellipsoidal representation. Zhang (1999) and Lutzen (2001) use an ellipsoid-like modelling for a bulbous bow, which is slightly different from the ellipsoid since both the horizontal and the vertical section has parabolic shape. Zhang and Lutzen’s modelling can be categorized as an ellipsoid modelling in the sense that both modelling have elliptical shaped transverse section. In the SSCAT a bulbous bow is modelled as ellipsoid. A mathematical representation of ellipsoid is shown in Eq.(31).

\[
\frac{x^2}{R_x^2} + \frac{y^2}{R_y^2} + \frac{z^2}{R_z^2} = 1 \tag{31}
\]

where \(x\), \(y\) and \(z\) denotes the longitudinal, transverse and vertical direction of the striking ship. \(R_x\), \(R_y\) and \(R_z\) denote the half radius of the ellipse in \(x\), \(y\) and \(z\) direction respectively.

The structural arrangement of the bow structure is complicated and it varies from ship to ship. In this study various sizes of the striking ship is used for probabilistic analyses and to input thousands ships of bow data is neither efficient nor practical. Therefore, in this version the structural arrangement of webs and stiffeners inside the bulb is assumed to be symmetric against the x-y and the x-z plane. A more generalized modelling feature of non-symmetric bulb shapes and structural members will be left to future development.

Based on the structural arrangement in Fig. 6.1 and Fig. 6.2, the bulbous bow is assumed to have longitudinal/transverse frames, horizontal webs (stringers) and centre line vertical web. The effect of longitudinal stiffeners is taken into account as an equivalent thickness of the outer shell. The transverse sectional views of the bulbous bow model for a standard bow and a buffer bow are shown in Fig. 6.13.
(a) Shape of blunt shaped bulbous bow. Shape of ellipsoid in X-Z plane is shown in the right figure as a dash line.

(b) Longitudinal stiffened bulbous bow (Standard bulbous bow)

(c) Transverse stiffened bulbous bow (Buffer bulbous bow)

Fig. 6.1 Blunt shaped bulbous bow models used in FEA.
(a) Shape of sharp shaped bulbous bow. Shape of ellipsoid in X-Z plane is shown in the right figure as a dash line.

(b) Longitudinal stiffened bulbous bow (Standard bulbous bow)

(c) Transverse stiffened bulbous bow (Buffer bulbous bow)

Fig. 6.2 Sharp shaped bulbous bow models used in FEA.
6.1.1 Deformation of the bow of the striking ship

The crushing strength of a bow is calculated while assuming collision with a rigid side shell of the struck ship based on the intersecting plate unit (IPU) such as L, T and X–type elements as described in 5.2.3. An illustration of the modelling of plate intersections by IPU is shown in Fig. 6.3, where the red ellipse with the dash line shows the intersection between the outer shell and the rigid contact surface. Stiffeners inside the red ellipse are effective. Based on Paik et al (1995)’s formula, the longitudinal stiffeners are smeared out to the outer shell as an equivalent thickness. Interactions between elements are not taken into account. X type intersections are fixed to the y-z plane since the centre line web and stringer is fixed in most cases. On the other hand most of the T type elements are moving elements as the indentation of bulb increases. The details of the analysis method for calculating bow crushing strength are described in Yamada & Pedersen (2007).

![Fig. 6.3. Modelling of bulbous bow structures by intersection unit elements (sectional view on the contact surface). Left: standard bow, right: buffer bow. The red ellipse with dash line shows the intersection between the outer shell and the contact surface. Stiffeners inside the red ellipse are effective.](image1)

![Fig. 6.4 Inclination angle of the structural elements on the bulb outer shell. $\delta$: indentation of the bulbous bow.](image2)
In calculating the crushing strength of IPU along the outer shell, the effect of inclination angle is taken into account. This effect is significant. As for the inclined element on the outer shell it is necessary to calculate the inclined angle $\theta$ in order to calculate the crushing strength of the inclined elements. The inclined angle $\theta$ at $x = R_x - \delta_B$ (point c) is calculated. Based on the elliptical equation of the outer shell in x-y plane section it follows that:

$$ y = \pm R_y \sqrt{1 - \frac{x^2}{R_x^2}} $$

(32)

The tangent of the angle $\theta$ is;

$$ |\tan \theta| = \left| \frac{dy}{dx} \right| = \frac{R_y}{R_x} \frac{x}{\sqrt{R_x^2 - x^2}} $$

(33)

where $\delta_B$ denotes the indentation of the bulb. The relation between $x$ and $\delta_B$ is

$$ x = R_x - \delta_B $$

(34)

![Graph showing variation of inclination angle $\theta$ as a function of dimensionless bulb indentation.](image)

**Fig. 6.5.** Variation of inclination angle $\theta$ as a function of dimensionless bulb indentation
6.2 Modelling of the Struck Ship and its deformation

6.2.1 Modelling of the struck ship

In this study double hull oil tankers are used for the struck ships. A typical modelling of the side structure of the struck ships in this study is shown in Fig. 6.6 where side shell is stiffened with transverse web frames and stringers. The secondary structural members such as longitudinal stiffeners are smeared out as an equivalent plate thickness.

Fig. 6.6 Typical primary structural arrangement of side structure for VLCC tanker in relation to the contact point and the size of the bulbous bow of the striking ship.

- FS : transverse frame spacing
- DHS : double hull space
- IS. : inner shell (oil cargo tank / longitudinal bulkhead)
- OS. : outer shell
- STR. : horizontal stringer

6.2.2 Deformation of the side structure of the struck ship

The crushing strength of the struck ship side is calculated for each indentation $\delta_A$ increasing from 0 with equal step size $d\delta$ assuming that a rigid bulbous bow penetrates the side structure. An example of contact between two ships is shown in Fig. 6.7. The crushing strength of side structures is estimated by using the procedures described in chapter 5. The bending energy of the side shell is neglected and only membrane effects are taken into account. In the following the deformation of the side structure by a rigid ellipsoidal bulbous bow is described.
Fig. 6.7 Example display of collision point, contact area and relative draft in the SSCAT.

(1) Penetration by rigid ellipsoidal bulb

Two local coordinate systems are used in the striking (x_By_Bz_B) and the struck ship (x_Ay_Az_A). When a rigid ellipsoidal bow penetrates the struck ship with indentation δ_A, the x coordinate of cross section from the bow center O can be expressed as:

\[ x_B = R_x \cdot \delta_A \]  

(35)
From Eqs.(31) and (35) the elliptical intersection between the outer shell of the bulb and the side shell is represented in the \( x_\text{A}y_\text{A}z_\text{A} \) coordinate system as:

\[
\frac{y^2}{R_y^2} + \frac{z^2}{R_z^2} = 1 - \frac{(R_x - \delta)^2}{R_x^2} = \beta_x^2
\]  

(36)

where \( 0 \leq \beta \leq 1 \)

Therefore, the elliptical contact surface at the side shell can be described as:

\[
\left(\frac{y}{\beta_y R_y}\right)^2 + \left(\frac{z}{\beta_z R_z}\right)^2 = 1
\]  

(37)

where

\[
\beta_x = \sqrt{1 - \frac{(R_x - \delta)^2}{R_x^2}} = \sqrt{1 - \alpha_x^2} \quad (0 \leq \beta_x \leq 1)
\]  

(38)

\[
\alpha_x = \frac{(R_x - \delta_x)}{R_x} = 1 - \frac{\delta_x}{R_x} \quad (0 \leq \alpha_x \leq 1)
\]  

(39)

Here \( \beta_x R_y = \tau_{x_\text{A}} \) and \( \beta_x R_z = \tau_{z_\text{A}} \) denote the half axes of ellipse on the contact surface in the \( x_\text{A} \) direction and in the \( z_\text{A} \) direction, respectively.

By using the local coordinate system of the struck ship, the elliptical cross section at side shell can also be described as:

\[
\frac{x^2}{r_x^2} + \frac{z^2}{r_z^2} = 1
\]  

(40)

where \( r_x = \beta_x R_y \) and \( r_z = \beta_x R_z \).

(2) Contact with transverse webs and horizontal stringers of the struck ship.

With regard to the contact between the bulb and the side structure three basic situations can be considered. One is when the plate of the struck ship is subjected to lateral loading (Stage 1). The second is that in addition to the first then the struck ship is subjected to denting of either deck and/or transverse frame (Stage 2). The third is when the bulbous bow touches the plate intersection and cause crushing of X-type intersection (Stage 3).

Each stage is determined by the geometrical relations between the elliptical contact surface and the positions of stringers and transverse webs. Each plate intersection at \( (x_i, z_i) \) in \( x_\text{A}y_\text{A}z_\text{A} \) coordinate system is judged to have contact with the bow if the following formula is satisfied (stage 3).

\[
\frac{(x_i - x_0)^2}{r_x^2} + \frac{(z_i - z_0)^2}{r_z^2} \leq 1
\]  

(41)

where \( x_0 \) and \( z_0 \) denote the coordinate of the ellipse center in \( x_\text{A}y_\text{A}z_\text{A} \) coordinate system.

The crushing formulae described in 5.2 are applied to calculate the crushing force. After stage 3, it is probable that the bow contacts with several stringers, transverse webs and plate intersections. All these possible contacts are searched one by one and taken into account to estimate the total crushing strength. In case that the bulb initially collides with the mid part of the stiffened plate, lateral loading, denting and the crushing of X-type intersection is supposed to take place in this order. It may be possible that a collision takes place directly from stage 3 without going through stage 1 and stage 2. It means that a bulbous bow collides with the side structure just on the intersection of the primary structures although these probabilities are smaller than the case of colliding on the unsupported part of the stiffened plate.
Stage 1 (Lateral loading)                               Stage 2-1 (Denting of Trans. Web)

Stage 2-2 (Denting of Stringer)            Stage3 (Crushing of plate intersection)

Fig. 6.9 Geometrical contact relation between bulb and the stiffened plate.
6.3 Comparison with FEA results

A series of example calculations have been carried out to confirm the accuracy of the present method, and the results are compared with results from FEA. The test case is a VLCC in ballast condition colliding perpendicularly with another VLCC in fully loaded condition. Main parameters of both ships used in the calculation are shown in Table 6.1.

<table>
<thead>
<tr>
<th>Table 6.1. Main parameters of both ships.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Striking ship</strong>&lt;br&gt;<strong>Struck ship</strong>&lt;br&gt;(Ship B) <strong>Struck ship</strong>&lt;br&gt;(Ship A)</td>
</tr>
<tr>
<td><strong>Displacement</strong>&lt;br&gt;VLCC</td>
</tr>
<tr>
<td><strong>Initial velocity</strong>&lt;br&gt;before collision&lt;br&gt;V [m/s]</td>
</tr>
<tr>
<td><strong>Length pp</strong>&lt;br&gt;L_{pp} [m]</td>
</tr>
<tr>
<td><strong>Breadth mould</strong>&lt;br&gt;B_{m} [m]</td>
</tr>
<tr>
<td><strong>Depth mould</strong>&lt;br&gt;D_{m} [m]</td>
</tr>
<tr>
<td><strong>Draft</strong>&lt;br&gt;T [m]</td>
</tr>
<tr>
<td><strong>Bulb half radius</strong></td>
</tr>
<tr>
<td><strong>Double Hull Space</strong>&lt;br&gt;DHS [m]</td>
</tr>
<tr>
<td><strong>Loading Condition</strong></td>
</tr>
</tbody>
</table>

Penetration of the side structure by a rigid bulbous bow and the contact region on the side shell are shown in Fig. 6.10 and Fig. 6.11. The figure on the left hand side shows the sectional view at the centreline of the striking ship. The figure on the right hand side shows the sectional view at the original outer shell, where red rectangle shows the boundary of the stiffened plate subjected to lateral loading. The small circles on the side shell of the struck ship show the existence of X type Super Elements as described in 5.2.3. Each element will be activated independently if the intersection point enters inside the red ellipse showing the contact area of the bulbous bow on the side shell.

(1) Verification of crushing of the struck ship

The force-penetration and energy-penetration curves for the case where a rigid bulbous bow penetrates the side structure of the struck ship obtained by the present method are shown in Fig. 6.12 and compared with results obtained by FEA. The finite element model and the methods used in the FEA are described in Yamada et al (2005). The results of FEA are obtained by changing the material model of the original striking ship in Yamada et al (2005) from elastic-plastic to rigid. It can be seen from Fig. 6.12 that fairly good correlation is achieved between the present method and the FEA although some discrepancies can be seen especially in the contact force. The total absorbed energy by the present method has a good correlation with that obtained by FEA. It is noted that the onset of fracture of both the outer and the inner shell is largely dependent on the fracture strain which can also be seen in Yamada et al (2005).

(2) Verification of crushing of the striking ship (Standard Bow)

The force-indentation curve in the case that the bulbous bow collides perpendicularly against a rigid side structure (a rigid wall) is also calculated. Contacts areas of the bulbous bows at different indentations are shown in Fig. 6.13. Left hand figures show the contact areas for a standard bow, while right hand figures those for the buffer bow. Longitudinal stiffeners belonging to the outer shell are smeared out to the outer shell and are hidden in the figures. Red circles and squares show T-type and X-type intersection plate unit elements respectively. As far as the structural arrangements of the present bulb structure, the T-type elements are a kind of moving elements and changes position as the indentation increases while the X-type elements are located at the same position in the y-z plane. It can be seen from these figures that the T- and X-type elements are accurately identified by the present programme.
Comparisons of a force-indentation curve and an energy-indentation curve between the present method (SSCAT) and FEA are shown in Fig. 6.14. A large drop down of the force can be seen in the result of SSCAT. This drop of the force does not clearly appear in the contact force of the FEA, moreover around delta=3.0 the force by FEA is smaller than that by SSCAT. This is because the bulb in fore part is better stiffened than that in aft part in the present ship model as shown in Fig. 6.2. Some of the horizontal stringers which exist in the fore part disappear in the aft part of the bulbous bow. This causes longitudinal discontinuity of the horizontal stringers in the present ship model. This is a relatively typical structure for the bulb since fore part of the bulb is often better stiffened than aft part of the bulb in order to resist local environmental loads such as slamming and collision with floating objects. The interaction between different transverse sections is not fully taken into account in the present method.

It is also seen from the upper figure of the Fig. 6.13 that during crushing of the fore part of the bulbous bow ($\delta<3.4$m) the force obtained by present method is slightly larger than that by FEA. One of the reasons is presumably that the ellipsoidal shape of the bulbous bow assumed in the SSCAT and the transverse sectional area of the ellipsoid is larger than that of present FEA model. Another reason is that SSCAT slightly overestimates the crushing strength of the horizontal stringers. Although some discrepancies between the present method and FEA are seen, it can be said that the present method gives fairly good prediction of axial crushing of bulbous bow structures.

(3) Verification of crushing of the both ships

The combination of a crushing bulb and a penetrated struck ship is illustrated in Fig. 6.15. In calculating the combined contact force from the two independent forces, the method used by Lutzen (2001) is employed. It is assumed that the contact surface remains plane and the deformation of both ships proceeds on the condition that equilibrium of the contact forces is satisfied. That is, the ship structure with weak crushing strength collapses until its crushing strength corresponds to that of the other ship. The area surrounded by the rectangle shows the crushed zone between the two ships. The total absorbed energy is calculated by the area which is surrounded by the force curves, the horizontal axis ($F=0$) and the rectangle.

The final contact force and total absorbed energy obtained by SSCAT are shown in Fig. 6.16 and compared with results from FEA for both the standard bow and the buffer bow. The FEA results shown are for the case of a standard bow. The contact force fluctuates significantly since it involves a lot of complicated nonlinear phenomena. The present method could not capture all the fluctuations, but the mean level of the contact force is similar. Moreover a good correlation can be seen between FEA and SSCAT in the total absorbed energy. Therefore, it can be said that SSCAT can predict the force and total absorbed energy of the ship collision with reasonable accuracy although future update is necessary to further improve its accuracy.

Following improvements could be addressed in the future update.

- More precise modelling of the blunt shaped bulbous bow (Fig. 6.1).
- Stronger coupling of deformation of striking ship and struck ship.
- Coupling of internal and external dynamics.
- Bending collapse of the bulb as well as oblique collision.
Fig. 6.10. Penetration of the side structure by rigid bulbous bow PART I (SSCAT).
Fig. 6.11. Penetration of the side structure by rigid bulbous bow PART II (SSCAT).
Fig. 6.12. Force-penetration and energy absorption-penetration curves of the side structures struck by rigid bulbous bow.
Fig. 6.13. Transverse sectional views of the bulbous bows at different indentation $\delta$ in case of the collision with the rigid outer shell of the struck ship. Figures in left hand side are for standard bow, those in right hand side are for buffer bow. Filled red circles and squares show T-type and X-type intersection plate unit elements respectively which are automatically detected by the present program. The marks left to the center line are hidden due to symmetry.
Fig. 6.14. Comparison of force-indentation curves of axial crushing of a standard (longitudinally stiffened) bulbous bow into a rigid side shell based on the present method (SSCAT) and results from FEA.
Fig. 6.15. Combination of the crushing force between striking and struck ships. \( \Delta B \) denotes the indentation of the striking ship with rigid wall, and \( \Delta A \) denotes the penetration of the struck ship by rigid bow. The two forces are calculated independently.
Fig. 6.16 Combined force-$\delta$ and energy-$\delta$ curve. FEA results are the case for standard bow.

The force – delta curve and the energy - delta curve obtained by the present method compared with results obtained by FEA are shown in Fig. 6.16. It is seen from the force – delta curve that the force predicted by present method is slightly lower than that by FEA. However good correlation was achieved for the total absorbed energy. Considering the complicated phenomena of the ship collision it could be said that present method gives fairly good prediction with reasonable accuracy.
7. MONTE CARLO SIMULATION

In order to obtain the probability of rupture of the oil cargo tank of large oil tankers Monte Carlo simulations are performed using the SSCAT. The Monte Carlo method generates many random sets of the variables according to their statistical distributions and then calculates whether rupture of the oil cargo tank takes place or not.

7.1 Probabilistic Ship Database

A database containing ship characteristics for the world fleet has been obtained from LMIS (2005). The database consists of approximately 90000 ships in service all over world. The database has been analyzed to derive distribution of random variables used in the Monte Carlo simulation.

7.2 Collision Scenario and Assumption

A variety of collision scenarios can be considered, and it is difficult to take every possible collision scenario into account. Therefore in the present study the following collision scenarios are considered in order to focus on large oil spill from struck tankers.

- Considering primarily disastrous oil spill consequences, the struck ship is assumed to be an anchored Double Hull (D/H) VLCC of 300,000 DWT in fully loaded condition. Main particulars, the structural arrangement and scantling are same as the FEA model of the struck ship in Yamada et al (2005).
- As for the striking ships, three groups are defined dependent on the minimum deadweight tonnages (DWT_{B,min}) of the striking ships in each group. First group is ships of 5,000 DWT or above (DWT_{B,min} = 5,000). Second group is ships of 100,000 DWT or above (DWT_{B,min} = 100,000). Third group is ships of 200,000 DWT or above (DWT_{B,min} = 200,000).
- The striking ships in ballast condition collide perpendicularly with the side shell at positions between the transverse bulkheads of the struck ship.
- The reduction of plate thickness due to corrosion is not explicitly taken into account in the present study.
- Both ships are assumed to have zero trim.

7.3 Random variables and probabilistic data

In order to conduct Monte Carlo Simulations, random variables such as dead weight tonnage, length, depth, draft and etc are generated and used as input to the calculations. In the following the generation of each variable is described.

- Main Particulars of ships
- Double Hull Space (DHS)
- Trans. Frame Spacing
- Scantling of ships
- Rupture Strain (RS)
- Dimensionless Longitudinal Collision Point (DLCP)

7.3.1 Main Particulars of ships

In this study DWT is chosen as a main parameter of ships. DWT is assumed to follow the Beta distribution and is independently generated between DWT_{min} [ton] and DWT_{max} [ton]. Density function of the Beta distribution can be written as (ex. Lutzen, 2001):

\[
 f(x) = \frac{(x - DWT_{min})^{a-1}(DWT_{max} - x)^{b-1}}{(DWT_{max} - DWT_{min})^{a+b-1} \cdot B(\alpha, \beta)} 
\]

(42)

where

- \( f(x) \): density function of Beta distribution
- \( x \): DWT
- \( \alpha, \beta \): parameters for the Beta distribution (\( \alpha > 0, \beta > 0 \)).
- \( B(\alpha, \beta) \): the beta function
- \( DWT_{min} \): upper bound of DWT
Parameters of beta distribution are determined based on data fitting, where three sets of parameters are determined dependent on DWT min. The parameters used in the present analysis dependent on DWT min are shown in Table 7.1.

### Table 7.1. Parameter of the Beta distribution for DWT.

<table>
<thead>
<tr>
<th>CASE</th>
<th>DWT min [ton]</th>
<th>DWT max [ton]</th>
<th>α</th>
<th>β</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5,000</td>
<td>400,000</td>
<td>0.64</td>
<td>1.83</td>
</tr>
<tr>
<td>2</td>
<td>100,000</td>
<td>400,000</td>
<td>0.61</td>
<td>1.15</td>
</tr>
<tr>
<td>3</td>
<td>200,000</td>
<td>400,000</td>
<td>3.41</td>
<td>5.08</td>
</tr>
</tbody>
</table>

In addition, the following random variables are considered to be main particulars of ships in the present analysis; Displacement (M), Length between perpendiculars (L pp), breadth moulded (B m), depth moulded (D m) and draft (T). The correlation coefficients of these main particulars obtained from LMIS (2005) are shown in Table 7.2 in matrix form. Based on regression analyses of tankers over 5000 DWT in LMIS, relations of M, L pp, B m, D m, and T against DWT are shown in Fig. 7.1 respectively. It is seen from the table and the figures that DWT has strong correlation with these variables. Therefore these variables can be regarded as dependent variables of DWT. The following regression formulas are derived from the analyses:

\[
M = 1.1218 \cdot DWT + 4425.3
\]

\[
L_{pp} = 8.7171 \cdot DWT^{0.2869}
\]

\[
B_m = 0.8754 \cdot DWT^{0.332}
\]

\[
D_m = 0.5172 \cdot DWT^{0.3206}
\]

\[
T = 0.4273 \cdot DWT^{0.3085}
\]

### Table 7.2. Correlation coefficient of main particulars (Corr)

For the oil tankers over 5000 DWT.

<table>
<thead>
<tr>
<th></th>
<th>DWT</th>
<th>M</th>
<th>L pp</th>
<th>B m</th>
<th>D m</th>
<th>T</th>
</tr>
</thead>
<tbody>
<tr>
<td>DWT</td>
<td>1.00</td>
<td>1.00</td>
<td>0.93</td>
<td>0.93</td>
<td>0.92</td>
<td>0.95</td>
</tr>
<tr>
<td>M</td>
<td>1.00</td>
<td>0.93</td>
<td>0.93</td>
<td>0.93</td>
<td>0.92</td>
<td>0.95</td>
</tr>
<tr>
<td>L pp</td>
<td>1.00</td>
<td>0.96</td>
<td>0.95</td>
<td>0.95</td>
<td>0.96</td>
<td></td>
</tr>
<tr>
<td>B m</td>
<td></td>
<td>1.00</td>
<td>0.96</td>
<td>0.95</td>
<td>0.95</td>
<td></td>
</tr>
<tr>
<td>D m</td>
<td></td>
<td></td>
<td>1.00</td>
<td>0.98</td>
<td></td>
<td></td>
</tr>
<tr>
<td>T</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.00</td>
</tr>
</tbody>
</table>

In order to generate random variables of the main particulars a regression residual is necessary in addition to the regression formulas. The regression residuals is calculated in a following way.

Regression of a set of random variables \(Y = \{M, L_{pp}, B_m, D_m, T\}\) given a random variable X (DWT) can be expressed as;

\[
E[Y | X] = aX^b + c
\]

where \(a\), \(b\) and \(c\) are constants. \(E\) represents the expected mean value. The matrixes and vectors are represented in bold font, and vectors are represented with a under line.

A random vector \(Y\) can be represented as the sum of the regression \(E[Y|X]\) and its residual vector \(R\) (Ditlevsen and Madsen, 2005);

\[
Y = E[Y | X] + R
\]
Assuming \( \mathbf{R} \) can be represented by a standard normal distribution vector \( \mathbf{U} \) and arbitrary matrix \( \mathbf{T} \) as

\[
\mathbf{R} = \mathbf{TU}
\]

where according to its definition \( \mathbf{E}[\mathbf{U}] = 0, \mathbf{D}[\mathbf{U}] = \mathbf{V}[\mathbf{U}] = 1 \). \( \mathbf{D}[X] \) and \( \mathbf{V}[X] \) represents the standard deviation and variance of the random variable \( X \) respectively.

Then Eq. (45) becomes;

\[
\mathbf{Y} = \mathbf{E}[\mathbf{Y} \mid \mathbf{X}] + \mathbf{R} = \mathbf{E}[\mathbf{Y} \mid \mathbf{X}] + \mathbf{TU}
\]

The matrix \( \mathbf{T} \) has to satisfy the following equation since \( \mathbf{U} \) is a standard normal distribution;

\[
\text{Cov}(\mathbf{Y}, \mathbf{Y}^T) = \text{Cov}((\mathbf{E}[\mathbf{Y} \mid \mathbf{X}] + \mathbf{R}) \cdot (\mathbf{E}[\mathbf{Y} \mid \mathbf{X}] + \mathbf{R})^T)
\]

\[
= \text{Cov}([\mathbf{R}, \mathbf{R}^T]) = \text{Cov}([\mathbf{TU}, (\mathbf{TU})^T]) = \mathbf{T} \cdot \text{Cov}([\mathbf{U}, \mathbf{U}^T]) \cdot \mathbf{T}^T = \mathbf{T} \cdot \mathbf{V}[\mathbf{U}] \cdot \mathbf{T}^T = \mathbf{TT}^T
\]

Correlation coefficient of residual matrix and Covariance of main particulars (vector \( \mathbf{Y} \)) are numerically calculated using the data of oil tankers over 5000 DWT as shown in Table 7.3 and Table 7.4.

**Table 7.3. Correlation coefficient of residual matrix (Corr[\( \mathbf{R}, \mathbf{R}^T \)]).**

For oil tankers over 5000 DWT.

(lower triangular part of matrix is left blank due to symmetry of the matrix).

<table>
<thead>
<tr>
<th></th>
<th>M</th>
<th>Lpp</th>
<th>Bm</th>
<th>Dm</th>
<th>T</th>
</tr>
</thead>
<tbody>
<tr>
<td>M</td>
<td>1.000</td>
<td>0.085</td>
<td>0.005</td>
<td>-0.010</td>
<td>0.027</td>
</tr>
<tr>
<td>Lpp</td>
<td></td>
<td>1.000</td>
<td>-0.395</td>
<td>-0.444</td>
<td>-0.192</td>
</tr>
<tr>
<td>Bm</td>
<td></td>
<td></td>
<td>1.000</td>
<td>-0.047</td>
<td>-0.524</td>
</tr>
<tr>
<td>Dm</td>
<td></td>
<td></td>
<td></td>
<td>1.000</td>
<td>0.378</td>
</tr>
<tr>
<td>T</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.000</td>
</tr>
</tbody>
</table>

**Table 7.4 Covariance Matrix of main particulars (Cov[\( \mathbf{R}, \mathbf{R}^T \)]).**

For oil tankers over 5000 DWT.

(lower triangular part of matrix is left blank due to symmetry of the matrix).

<table>
<thead>
<tr>
<th></th>
<th>M</th>
<th>Lpp</th>
<th>Bm</th>
<th>Dm</th>
<th>T</th>
</tr>
</thead>
<tbody>
<tr>
<td>M</td>
<td>9.294E+07</td>
<td>8.758E+03</td>
<td>1.051E+02</td>
<td>-1.239E+02</td>
<td>2.217E+02</td>
</tr>
<tr>
<td>Lpp</td>
<td>1.129E+02</td>
<td>9.368E-01</td>
<td>-9.363E+00</td>
<td>-6.228E+00</td>
<td>-1.706E+00</td>
</tr>
<tr>
<td>Bm</td>
<td>4.980E+00</td>
<td>1.385E-01</td>
<td>-9.774E+00</td>
<td>-9.774E+00</td>
<td>-9.774E+00</td>
</tr>
<tr>
<td>Dm</td>
<td></td>
<td>1.742E+00</td>
<td>4.166E-01</td>
<td>4.166E-01</td>
<td>4.166E-01</td>
</tr>
<tr>
<td>T</td>
<td></td>
<td></td>
<td>6.991E-01</td>
<td>6.991E-01</td>
<td>6.991E-01</td>
</tr>
</tbody>
</table>

Since the covariance matrix is regular, matrix \( \mathbf{T} \) can be simply numerically calculated by Cholesky factorization (See Ditlevsen and Madsen, 2005). Then \( \mathbf{M}, \mathbf{L}_{pp}, \mathbf{B}_m, \mathbf{D}_m \) and \( \mathbf{T} \) given DWT are randomly generated using Eq. (45). Generated random variables using the matrix \( \mathbf{T} \) are compared with the actual data in Fig. 7.1 and Fig. 7.2. The figures show good agreements between actual data and generated data, and these data are used for the present MONTE CARLO SIMULATIONS.
Fig. 7.1 Comparison of generated random variables and actual data (against DWT).
7.3.2 Velocity of the striking ship

The distribution of the striking ship velocity in the actual accidents (see Lutzen 2001) varies significantly dependent on the size of ships. This is presumably because these distributions are obtained from limited number of actual collision data. However no strong reason for the most probable striking ship velocity for each distribution can be found.

According to these distributions it can be said that striking velocity has upper bound since a maximum velocity is limited from a service velocity of the ship or since a maximum velocity in specific sea area and strait is restricted by the local maritime authority. It can be also considered that the striking ships might decrease their velocity before collision only if watch officer is aware of the collision in advance and takes appropriate operation. Considering these circumstances the striking ship velocity is assumed to be uniformly distributed between 0kt to 16kt in the present study.
7.3.3 Double Hull Space (DHS)

Minimum Double Hull Spacing (DHS) of new oil tankers is prescribed in MARPOL 73/78 Convention, Annex1, Regulation 13F(a) as follows. This formula shows that the minimum DHS required by the international regulation lies between 1.0m and 2.0m.

\[
DHS = \text{MAX}\left(\text{MIN}\left(0.5 + \frac{DWT}{20000}, 2.0\right), 1.0 \right)
\]

(49)

According to the real ship data, however, DHS is larger than the value obtained by equation (48). Therefore, based on a limited number of actual ship data, DHS are estimated by the following regression formula with the coefficient of determination about 0.869. The DHS is set to the value obtained by Eq.(49) if the value obtained by regression is smaller than that required by Eq.(49).

\[
y = 0.1187x^{0.2658}
\]

\[
R^2 = 0.869
\]

Fig. 7.3 Approximate relation between DHS and L_{pp}

7.3.4 Transverse Frame Space

Generally transverse frame spaces are not prescribed in the structural design rules of international regulators or classification societies, and different values are used by each shipbuilding company based on the their experience or requirements by the ship owners. Therefore, in this study based on a limited number of actual typical oil tankers data transverse frame spaces are estimated as a function of L_{pp} of the ship, as shown in Fig. 7.4.

\[
y = 0.0134x + 0.9032
\]

\[
R^2 = 0.9079
\]

Fig. 7.4. Approximate relation between Trans. Frame Space and L_{pp}
7.3.5 Longitudinal Stiffener Space (LSS)
Like the transverse frame spacing also LSS differs within individual ship building companies. Therefore LSS are estimated as a function of $L_{pp}$ on data from a limited number of the data from actual typical oil tankers. In case of LSS some ships are stiffened with the almost same LSS along the height of the ship, but other ships have varying stiffener spacing. However, the variability of the LSS in the same ship is limited. Therefore, within the individual ships a constant LSS is assumed.

![Graph showing the relationship between LSS and $L_{pp}$]

$$y = 0.0009x + 0.6129$$

$R^2 = 0.6595$

Fig. 7.5. Approximate relation between Longitudinal Stiffener Spacing and $L_{pp}$

7.3.6 Partial Draft
Drafts in fully loaded condition $T_{\text{max}}$ and drafts in ballast condition $T_{\text{min}}$ can be obtained by statistical data (LMIS, 2005). However, the data for partial drafts $T_{\text{part}}$ are not obtained by statistical data. Therefore $T_{\text{part}}$ is estimated by the following formula proposed by Lutzen (2001).

$$T_{\text{part}} = T_{\text{min}} + 0.6 \cdot (T_{\text{max}} - T_{\text{min}})$$

(50)

7.3.7 Scantlings of structural members
Other scantlings such as plate thickness of outer shell, side shell and bow bottom are determined based on the DNV rules.

7.3.8 Rupture Strain (RS)
Rupture strain of the plate is dependent on many factors such as stress concentrations and triaxiality in the local region, degradation of the plate due to corrosion, and gauge length (stretching span) and so on. Therefore rupture strain is supposed to vary significantly. Kitamura (2002) pointed out that fracture strain used in many simplified analyses varies between 5 and 15%. As a value of the fracture strain for simplified analyses McDermott et al (1974) and Simonsen (2000) used a rupture strain which is about 1/3 of that obtained from uniaxial coupon tests. That is estimated about 10%. As described in 5.3.2 rupture strain further decrease by 40% in case of biaxial tension state. However it is expected that the mean vertical stress of the side shell does not reach the flow stress since the stretching span in vertical direction is longer than that in longitudinal direction. Therefore reduction of rupture strain due to biaxial tension state is supposed to lie around 60%-80%.

In order to capture the effect of the large deviation of the rupture strain, it is assumed to be a lognormally distributed independent random variable with a mean value of 8% and standard deviation of 1.6%. This 1.6% corresponds to about 20% of the mean value of 8%. According to the
characteristic of the normal distribution it is supposed that 95 % fractile of generated data of rupture strain lies between 10-20%, and this range corresponds fairly well with previous studies (McDermott et al, 1974; Simonsen, 2000) and seems to be reasonable.

### 7.3.9 Dimensionless longitudinal collision point (DLCP)

The dimensionless collision point in the longitudinal direction of the struck ship, is assumed to be uniformly distributed from 0 to 1.

### 7.3.10 Half radius of bulbous bow

Based on Zhang (1999)’s study, the half radii of the bulbous bow can be represented as:

\[
\begin{align*}
R_x &= C_x \cdot D_m \\
R_y &= C_y \cdot D_m \\
R_z &= C_z \cdot D_m
\end{align*}
\]

where \( C_x, C_y \) and \( C_z \) are ratios of half radii. \( D_m \) is the moulded depth of the ships although \( D_m \) is originally defined as bow height in Zhang (1999). These ratios are assumed to be lognormal distributed with mean and standard deviation as shown in Table 7.2. Considering the change of definition of \( D_m \) from bow height to moulded depth as well as new data, small modifications are made for mean values of \( C_y \) and \( C_z \).

<table>
<thead>
<tr>
<th>Table 7.5. Mean and standard deviation of coefficient of bulb size.</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \mu )</td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td>( C_x )</td>
</tr>
<tr>
<td>( C_y )</td>
</tr>
<tr>
<td>( C_z )</td>
</tr>
</tbody>
</table>

### 7.4 Convergence criteria of the simulation

As the number of simulations \( N_{sim} \) increases, the accuracy of the estimate increases. The accuracy of a given Monte Carlo Simulation may be estimated by calculating a coefficient of variance (COV) of the predicted failure probability, as discussed by Haldar and Mahadevan (2000):

\[
COV(P_f) = \frac{1}{P_f} \sqrt{\frac{(1-P_f)P_f}{N_{sim}}}
\]

where \( P_f \) is estimated failure probability after \( N_{sim} \) simulations. A convergence criterion of \( COV(P_f) < 0.03 \) (Collete & Incecik, 2006) is employed in the present analyses, and the calculation was stopped when the convergence criteria is satisfied. In addition to the probability of rupture of the inner shell \( (P_{f,IS}) \), that of the outer shell \( (P_{f,OS}) \) is calculated in the same run in this study. Since \( P_{f,OS} \) is larger than \( P_{f,IS} \), convergence criteria is applied to \( P_{f,IS} \). That is, \( COV(P_{f,IS}) < 0.03 \). Eventually the \( COV(P_{f,OS}) \) becomes smaller than \( COV(P_{f,IS}) \).
7.5 ANALYSES RESULTS

The probability of rupturing a cargo oil tank and the outer shell of the struck ship given a tanker collision were obtained by performing N_{sim} simulations of Monte Carlo Simulations as described in the previous sections. The results of these simulations are presented in Table 7.6, Table 7.7 and Table 7.8. The striking ships sizes differs between the three cases as described in 7.2. The definition of COV is described in 7.4.

It is seen from these tables that the failure probabilities of both inner shell and outer shell by buffer bow are lower than those by standard bow. The failure probabilities of the inner shell in the case of buffer bows are about 19-24% lower than those by standards bows. These probabilities are conditional probability given specific collision scenarios and moreover the effect of oblique collisions is not taken into account in the present analysis. However the effect of the buffer bow on the reduction of the failure probability of the cargo hold tank is evident.

It is also seen from the tables that the failure probability becomes larger as the DWT_{B,min} increases. This is due to the fact that the average mass as well as the average initial kinetic energy of the striking ships in CASE 3 are larger than those in CASE1.

Table 7.6. CASE 1: Failure probability obtained by Monte Carlo Simulations given ships (5,000 ≤DWT≤400,000) collide with VLCC perpendicularly (DWT_{B,min} = 5000).

<table>
<thead>
<tr>
<th></th>
<th>Inner Shell</th>
<th>Outer Shell</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>N_{sim}</td>
<td>P_{f,IS}</td>
</tr>
<tr>
<td>Standard Bow</td>
<td>566</td>
<td>0.66</td>
</tr>
<tr>
<td>Buffer Bow</td>
<td>1439</td>
<td>0.44</td>
</tr>
</tbody>
</table>

Table 7.7. CASE 2: Failure probability obtained by Monte Carlo Simulations given ships (100,000 ≤DWT≤400,000) collide with VLCC perpendicularly (DWT_{B,min} = 100,000).

<table>
<thead>
<tr>
<th></th>
<th>Inner Shell</th>
<th>Outer Shell</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>N_{sim}</td>
<td>P_{f,IS}</td>
</tr>
<tr>
<td>Standard Bow</td>
<td>351</td>
<td>0.76</td>
</tr>
<tr>
<td>Buffer Bow</td>
<td>1013</td>
<td>0.52</td>
</tr>
</tbody>
</table>

Table 7.8. CASE 3: Failure probability obtained by Monte Carlo Simulations given ships (200,000 ≤DWT≤400,000) collide with VLCC perpendicularly (DWT_{B,min} = 200,000).

<table>
<thead>
<tr>
<th></th>
<th>Inner Shell</th>
<th>Outer Shell</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>N_{sim}</td>
<td>P_{f,IS}</td>
</tr>
<tr>
<td>Standard Bow</td>
<td>304</td>
<td>0.79</td>
</tr>
<tr>
<td>Buffer Bow</td>
<td>755</td>
<td>0.60</td>
</tr>
</tbody>
</table>
Fig. 7.6. Variation of the failure probability $P_{f,IS}$ depending on the minimum size of the striking ships.

One of the reasons why the failure probability of the outer and inner shells is a relatively weak function of the minimum size of the population of striking ships is that the lost kinetic energy (LKE) or the energy released for crushing is only a strong function of the mass of the striking vessels when the collision point is close to midship.

Based on simple external dynamics calculations in Section 4.2, the variation of LKE and $R_s$ ($\text{LKE} / E_0$) with the size (DWT) of the striking ship and dimensionless longitudinal collision point ($0 \leq DLCP \leq 1$) are shown in Fig. 7.7. Here $E_0$ denotes the initial kinetic energy of the striking ship. The struck ship is taken to be a 300,000 DWT VLCC.

It can be seen in the upper part of figure of Fig. 7.7 that, in case of a collision point far from midship of the struck ship ($DLCP = 0.05$), LKE does not vary significantly with the mass of the striking vessel and becomes almost a constant value (400 MJ) when $\text{DWT}_B > 150,000$. This variation with striking ship mass is much larger for collision points closer to midship. This is mainly because the yaw movement of the struck ship is large if the collision point is far from the midship of the struck ship. Therefore, the effect of mass of the striking ships on the structural damage of both ships decreases when the collision point is far from the midship of the struck ship. It is noted that the distribution of damages on the two ships is largely dependent on the relative strength of the striking and the struck ship. Therefore, the extent of damage on each ship is unknown from this simple calculation. However, the relative strength of the striking ships slightly increases as the size of the striking ships becomes larger since the bow plate thickness usually increases as the length of ship becomes longer. Therefore, for the same LKE the extent of damage of the struck ship will slightly increase as the mass of the striking ship increases.
Fig. 7.7 Lost kinetic energy depending on the Size (DWT) of the striking ship and dimensionless longitudinal collision point (DLCP).
8. CONCLUSION
In order to investigate the effectiveness of the buffer bow structures on the prevention of oil spill in tanker collisions, ship collision analysis was conducted using a newly developed simplified ship collision analysis tool (SSCAT). The energy to be absorbed by structural deformation has been calculated by considering global motions of both ships (external dynamics). Structural damage of both the striking ship bow and the struck ship side shells are independently calculated by use of the simplified methods by assuming rigid-perfectly plastic material.

The simplified collision analysis procedure has been verified by comparisons with the results of Finite Element Analyses (FEA). A good correlation has been found between FEA and SSCAT results for the total absorbed energy as function of penetration. That is, the SSCAT can predict the force and total absorbed energy of the ship collision with reasonable accuracy although future update is needed to further improve its accuracy.

The following improvements could be addressed in a future update:
- More precise modelling of blunt shaped bulbous bows (Fig. 6.1).
- Stronger coupling between the deformations of the striking ship and the struck ship.
- Coupling of internal and external dynamics.
- Bending collapse of the bulb as well as oblique collision.

The simplified procedure has been applied together with Monte Carlo Simulations (MCS) for various collision scenarios and the probability of oil spill from struck oil tankers in case of the buffer bows on the striking vessels is compared with that in case of standard bulbous bows. Failure probabilities for rupture of cargo oil tank, given a collision, are calculated. The results of this sample calculation shows that the probability of rupture of cargo oil tanks on a VLCC decreases about 19 to 24 % if all meeting ships are equipped with buffer bulbous bows.
REFERENCE


MARPOL 73/78 (1997), International Maritime Organization (IMO)
RISK ANALYSIS OF OIL SPILL FROM STRUCK OIL TANKERS

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bNational Maritime Research Institute, 6-38-1, Shinkawa, Mitaka-shi, Tokyo, Japan

ABSTRACT

The purpose of this paper is to evaluate risk of oil spills from struck tankers by using Bayesian Network. A methodology to evaluate the risk of oil spill from struck tankers is established taking various factors into account such as the amount of spilled oil and type of oil which affect the cost of oil spill. As a case study risk analysis of oil spill is carried out for the specific collision scenario where a VLCC is struck by another tanker in the Great Belt and Oresund. The risk of oil spill with buffer bows on the striking vessels is compared with that of standard bows. The extent of reduction of the risk is investigated in detail.

KEY WORDS: risk analysis; oil spill; ship collision; Bayesian Network

1. INTRODUCTION

In order to prevent oil spill from a struck tanker, the double hull system has become the de facto standard as an effective countermeasure. However, it is still a fact that collision accidents involving struck double hull tankers results in oil spills. One example is the oil spill accident by ship collision involving the double hull oil tanker “Baltic Carrier” happened in Denmark in 2001. Therefore there are good reasons to further minimize the consequences of ship collision accidents. In order to reduce the risk of oil spill from struck oil tankers the Buffer Bow Project (BBP) has been carried out in NMRI supported by the Japanese Ministry of Land Infrastructure and Transport (MLIT). Bulbous bows of striking ships are the most threatening part against the struck ship from the view point of penetration of side structure of the struck ship. Therefore, the bulbous bow is regarded as a key structural part for buffer bow design and the main focus is laid on the bulbous bow structure in the BBP. Buffer bow concept is to absorb energy not only by the struck ship but also by the bulbous bow of the striking ship while the bulbous bow is preserving the enough strength to resist.

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the environmental load such as slamming and ice load. One of the measures to achieve the buffer bow concept is to adopt a transverse stiffening system instead of longitudinal one, which was proposed as the effective countermeasures to further reduce the risk of oil spill from struck tanker by ASIS (1997), Kitamura (2000).

A central objective of implementing buffer bows is to reduce the amount of oil spilled following collisions. Although the probability of occurrence of a collision event is quite small, the consequences of the resulting oil spill into the sea are often immense. When evaluating the effect of the buffer bow by its ability to reduce the consequences following collisions it is therefore important to consider both the frequency of occurrence of the unwanted events as well as the consequences that follows. Risk analysis combines the probability of occurrence of the unwanted event with the consequences that follows the occurrence. By defining risk as the expected loss (i.e. the product of the probability of occurrence and the consequences) it becomes possible to apply risk analysis to assess the beneficial effect of the buffer bulbous bow. Therefore, in order to estimate the risk of oil spill two important components are necessary. One is the consequence of oil spills, which is defined as the economical costs of oil spills in this study. The other is the probability of occurrence of oil spills from struck tankers. These two components are investigated in detail in subsequent sections and are used for the present risk analysis.

Event trees and fault trees have for decades been the commonly applied tools for conducting risk analysis. Recently, however, a Bayesian Network (BN) are becoming more and more popular and widely used in various risk analysis since its graphical user interface based on graph theory is more powerful and more user friendly than the method of event tree and fault tree. A Bayesian network is a graphical representation of the joint probability distribution of a set of random variables that effectively encodes the conditional probability assertion among the variables. The nodes in the network represent random variables and the directed links between the nodes indicates the direction of the (causal) dependency. The uncertainty modelled by the nodes can be due to inherent randomness of the variable, imperfect understanding of the domain or incomplete knowledge of the state of the domain at the time where a given task is to be performed. Modelling of complex system by Bayesian Network is much easier and more efficient than that by event tree and fault tree (Friis-Hansen, 2004).

The use of Bayesian Network for risk analysis in ship and ocean engineering is also becoming more and more popular Friis-Hansen & Pedersen (1998) made a risk analysis of conventional and solo watch keeping with Bayesian Network. Friis-Hansen et al (2000) conducted reliability analysis of upheaval buckling by using Bayesian Network. See Friis-Hansen (2000) or Zhang & Smith (2001) for other examples. In this study risk analysis is conducted by using the Bayesian Network. The modelling of the Bayesian Network and the result of the analysis are described in the following.

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2. COST OF OIL SPILL

In order to conduct a risk analysis of oil spill, it is important to estimate cost of oil spill as accurate as possible since the risk of oil spill is defined by the product between probability of oil spill and the cost of oil spill. In the last decade many studies have been carried out to investigate the various influence factors to affect the cost of oil spill (Etkin, 1999; Grey 1999; Etkin 2001; White and Molloy, 2003; McCay et al, 2004; Etkin et al, 2005). Most of these studies pointed out that type of oil, amount of spilled oil and location of oil spill are generally the most important factors influencing the cost of oil spill. Moreover it is also pointed out that response of clean up operation is also important factor, and that removal of spilled oil before spilled oil reaches the shoreline has significant influences to reduce the cost of oil spill (Etkin et al, 2005). In this section major influence factors and its effect on the cost of oil spill are shortly reviewed and investigated. These results are used to carry out the risk analysis in the later section.

2.1. Historical Data from IOPCF

In order to investigate costs of oil spill, historical data of oil spill incidents happened between 1979 and 2005 in member states of IOPCF (2005) were investigated based on the report of IOPCF (2005). Annual costs of oil spill and the amount of spilled oil from oil tankers are shown in Fig. 1, and 10 largest oil spill incidents in terms of costs of oil spill and amount of spilled oil are listed in Table 1 and Table 2 respectively. According to the statistical data the oil spill from Prestige in 2002 caused the largest costs although its amount of oil spill is on the 5th position in Table 2. It can be seen from Fig. 1 that as a global tendency the amount of spilled oil is gradually decreasing from 1991, about when double hull system was introduced to tankers. On the other hand cost of the oil spill is dependent on year by year. It is interesting that the cost of oil spill from Prestige is remarkable among others.

![Fig. 1 Costs of large oil spill and the amount of spilled oil from tankers in each year (IOPCF, 2005).](image_url)
Table 1. Largest oil spill from tankers in terms of costs (IOPCF, 2005).

<table>
<thead>
<tr>
<th>Rank</th>
<th>Ship name</th>
<th>Year</th>
<th>Cost [million US$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Prestige</td>
<td>2002</td>
<td>$1,285</td>
</tr>
<tr>
<td>2</td>
<td>Erika</td>
<td>1999</td>
<td>$256</td>
</tr>
<tr>
<td>3</td>
<td>Nakhodkha</td>
<td>1997</td>
<td>$255</td>
</tr>
<tr>
<td>4</td>
<td>Aegean Sea</td>
<td>1992</td>
<td>$114</td>
</tr>
<tr>
<td>5</td>
<td>Braer</td>
<td>1993</td>
<td>$100</td>
</tr>
<tr>
<td>6</td>
<td>Sea Empress</td>
<td>1996</td>
<td>$74</td>
</tr>
<tr>
<td>7</td>
<td>Haven</td>
<td>1991</td>
<td>$60</td>
</tr>
<tr>
<td>8</td>
<td>Pontoon300</td>
<td>1998</td>
<td>$56</td>
</tr>
<tr>
<td>9</td>
<td>Sea Prince</td>
<td>1995</td>
<td>$56</td>
</tr>
<tr>
<td>10</td>
<td>Plate Princess</td>
<td>1997</td>
<td>$47</td>
</tr>
</tbody>
</table>

Table 2. Largest oil spill from tankers in terms of amount of spilled oil (IOPCF, 2005).

<table>
<thead>
<tr>
<th>Rank</th>
<th>Ship name</th>
<th>Year</th>
<th>Amount of Spilled Oil [ton]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Haven</td>
<td>1991</td>
<td>144,000</td>
</tr>
<tr>
<td>2</td>
<td>Braer</td>
<td>1993</td>
<td>84,000</td>
</tr>
<tr>
<td>3</td>
<td>Aegean Sea</td>
<td>1992</td>
<td>73,500</td>
</tr>
<tr>
<td>4</td>
<td>Sea Empress</td>
<td>1996</td>
<td>72,360</td>
</tr>
<tr>
<td>5</td>
<td>Prestige</td>
<td>2002</td>
<td>42,820</td>
</tr>
<tr>
<td>6</td>
<td>Evoikos</td>
<td>1997</td>
<td>29,000</td>
</tr>
<tr>
<td>7</td>
<td>Erika</td>
<td>1999</td>
<td>19,900</td>
</tr>
<tr>
<td>8</td>
<td>Nakhodkha</td>
<td>1997</td>
<td>17,500</td>
</tr>
<tr>
<td>9</td>
<td>Globe</td>
<td>1981</td>
<td>16,001</td>
</tr>
<tr>
<td>10</td>
<td>Seki</td>
<td>1994</td>
<td>16,000</td>
</tr>
</tbody>
</table>
2.2. Oil weathering

The mechanical and chemical behaviour of spilled oil, so-called “Oil Weathering” process, is reviewed in order to seek the factors which significantly affect the cost of oil spill. The behaviours of spilled oil are categorized to eight main processes. The processes are illustrated in Fig. 2.

Following ITOPF (2002) the eight phase of weathering of oil are:

(1) Spreading
As soon as oil is spilled, it starts to spread out over the sea surface, initially as a single slick. The spreading speed is largely dependent on the viscosity of the spilled oil. Moreover it is also affected by the prevailing conditions such as temperature, sea currents, tidal streams and wind. Important factors are viscosity of oil, sea current, tidal streams, wind, and temperature.

(2) Evaporation — important
Lighter oils will tend to evaporate almost completely in a few days whilst little evaporation will occur from a heavy fuel oil. The initial spreading rate of oil affects evaporation since the larger the surface area, the faster light components will evaporate. Evaporation can increase due to the increased surface area of the slick, rougher seas, high wind speeds and warm temperatures.

(3) Dispersion — less important
Wave and turbulence at the sea surface cause all or part of a slick to break up into fragments and droplet of varying sizes which become mixed into the upper layers of the water column. The rate of dispersion is largely dependent on the nature of oil and the sea state, proceeding most rapidly with low viscosity oils in the presence of breaking waves. Oils that form stable water-in-oil emulsions tend to form thick lenses on the water surface that show little tendency to disperse ever with the addition of dispersant chemicals. Such oils can persist for weeks and on reaching the shoreline may eventually form hard asphalt pavements if not removed.

(4) Emulsification — important
Under the turbulence of wave on the sea surface most oil slicks will take up water inside and form water-in-oil emulsions. The emulsification is usually very viscous and more persistent than original oil and is referred as “chocolate mousse” because of its appearance (see Fig. 3). The formation of these emulsions causes the volume of pollutant to increase between three and four times. As the amount of water absorbed increase, the density of the emulsion approaches that of sea water. The formation of water-in-oil emulsion reduces the rate of other weathering process and is the main reason for the persistence of light and medium crude oils on the sea surface. Emulsification of oil will increase the cost of cleaning up because it has
semi-solid characteristic and is difficult to skim.

![Fig. 3. Appearance of highly viscous water-in-oil emulsion, so called “chocolate mousse” (ITOPF, 2002).](image)

(5) Dissolution (benzene, toluene); very few amounts — less important
The compounds in oils which are likely to dissolve also tend to evaporate rapidly typically 10 to 1000 times faster than by dissolution. Therefore the dissolution process does not make a significant contribution to the removal of oil from the sea surface.

(6) Oxidation and make tar ball — less important
Hydrocarbon can react with oxygen, which may lead to the formation of persistent tars. Oxidation is promoted by sunlight, but its progress is very slow (0.1% per day). Therefore the effect of oxidation on the dissipation of amount of the spilled oil is minor or negligible compared to that of other weathering process.

(7) Sedimentation / Sinking — less important
Very few heavier residual oils with the relative density greater than sea water (1.025) will sink in fresh or brackish water.

(8) Biodegradation — less important
A many kinds of micro-organisms or microbes in the sea water can partially or completely degrade oil to water soluble compounds very slowly and eventually to carbon dioxide and water.

Among the above-mentioned eight factors emulsification of heavy oil is supposed to significantly increase the cost of oil spill since it is difficult to skim it up from the sea due to the large viscosity. Natural evaporation of light oil will significantly reduce the amount of spilled oil as well as oil spill costs. The spreading also increase the clean-up cost especially in case of heavy fuel oil because it causes large areas of sea to be polluted. However the 2 dimensional spreading of the spilled oil at sea is a complex problem since it is affected by many factors such as sea current and wind. Therefore, in the present study the effect of emulsification and natural dissipation process are taken into account in the estimation of the cost of oil spill in the Bayesian Network.
2.3. Influencing factor (IF) on the cost of oil spill

In order to estimate the cost of the oil spill it is important to identify the major factors which influence the cost of oil spill. Many researchers focused only on the amount of spilled oil. However, not only the amount of spilled oil but also other factors affect the cost of oil spill. Etkin (1999) have investigated the factors which influence on the cost of oil spill, and pointed out that the most important factor determining cost of oil spill is a location of oil spill and type of oil in addition to the amount of spilled oil. Etkin (1999) found that clean-up costs per a ton decrease significantly with increasing amounts of oil spilled because of the costs associated with initial setup of the cleanup response, bringing in the equipment and labour as well as bringing in the experts to evaluate the situation. In this section the major influencing factor (IF) is described as follows.

(1) Amount of spilled oil

It is reasonable that most researchers related the amount of spilled oil to the cost of oil spill as a first step. Some researchers have proposed a linear relationship between the cost and the amount of spilled oil. However Friis-Hansen and Ditteven (2003) pointed out that a double logarithmic relationship is more suitable than linear relationship based on the investigation with actual reported oil spill incidents. The reasoning for this is that small increment of large spill has less worth than a small increment of little spill, or stated mathematical that the worth of the increment should be measured relative to the spill size: \( \frac{dx}{x} = c \).

Fig. 5 shows the relation between cost (C) and the amount of spilled oil (M) based on the IOPCF oil spill reports. It is found from both figures that stronger correlation is observed in the graph with double logarithm relation than that with linear relation.

---

**Fig. 4. Relation between the cost of oil spill (C) and the amount of spilled oil (M).**

```
\begin{align*}
\text{Amount of spilled oil M [Ton]} & \quad \text{Cost C [US$]} \\
0.0E+00 & 0.0E+00 \\
5.0E+04 & 5.0E+04 \\
1.0E+05 & 1.0E+05 \\
1.5E+05 & 1.5E+05 \\
\end{align*}
```

---

**Fig. 5. Relation between the cost of oil spill (C) and the amount of spilled oil (M) in double logarithm scale.**

```
\begin{align*}
\text{Log}_{10} C \text{ [US$]} & \quad \text{Log}_{10} M \text{ [Ton]} \\
0 & 0 \\
1 & 1 \\
2 & 2 \\
3 & 3 \\
4 & 4 \\
5 & 5 \\
6 & 6 \\
7 & 7 \\
\end{align*}
```

\[ y = 0.6213x + 4.6055 \]

\[ R^2 = 0.3964 \]
The following relation between the cost of oil spill and the amount of spilled oil is used

\[
\log_{10} C = a \log_{10} M + b
\]

\[
\iff \quad \log_{10} C = \log_{10} M^a + \log_{10} 10^b = \log_{10} (M^a \cdot 10^b)
\]

\[
\iff \quad C = M^a \cdot 10^b \equiv f(M)
\]

Where

- \(C\): Cost of spilled oil [US$]
- \(M\): the amount of spilled oil [Ton]
- \(a\): Regression factor (constant), representing increasing rate of cost of oil spill against the amount of spilled oil.
- \(b\): Regression factor (constant), representing initial setting up cost of oil spill.

In this study \(a\) and \(b\) are estimated as shown in Table 3 based on the mean value of the data.

<table>
<thead>
<tr>
<th>Regression Factor</th>
<th>Estimated Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>0.621</td>
</tr>
<tr>
<td>(b)</td>
<td>4.61</td>
</tr>
</tbody>
</table>

![Fig. 6. Double logarithm relation between the amount of spilled oil and the cost of oil spill (C: Cost of spilled oil, M: the amount of spilled oil).](image)
(2) Type of Oil

A variety of different types of oils are transported by ships, and the type of spilled oil might have a significant influence on the determining the cleaning up costs. When oil is more persistent and viscous, the removal will be more difficult. The composition and physical properties of oil will affect the degree of evaporation and natural dispersion, as well as the ease of removal. Lighter oils evaporate and disperse to a larger extent than heavier oils, except when water-in-oil emulsions are difficult to remove using dispersants, skimmers, and pumps, resulting in considerably higher cleanup costs from manual methods. ITOPF (2002) categorized oils into 4 groups based on its density as follows.

<table>
<thead>
<tr>
<th>Group</th>
<th>Density [ton/m³]</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>Group I</td>
<td>Less than 0.8</td>
<td>Gasoline, Kerosene</td>
</tr>
<tr>
<td>Group II</td>
<td>0.80 – 0.85</td>
<td>Gas, Oil, Abu Dhabi Crude</td>
</tr>
<tr>
<td>Group III</td>
<td>0.85- 0.95</td>
<td>Arabian Light Crude, North Sea Crude Oils</td>
</tr>
<tr>
<td>Group IV</td>
<td>Greater than 0.95</td>
<td>Heavy Fuel Oil, Venezuelan Crude Oils</td>
</tr>
</tbody>
</table>

As for the viscosity, persistency, toxicity and risk for fire and explosion, following tendencies are roughly recognized (see White et al, 2003).

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Group I (Lighter)</th>
<th>Group II, III (Medium)</th>
<th>Group IV (Heavier)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity</td>
<td>Low</td>
<td>Medium</td>
<td>High</td>
</tr>
<tr>
<td>Persistency</td>
<td>Weak</td>
<td>Medium</td>
<td>Strong</td>
</tr>
<tr>
<td>Toxicity</td>
<td>Strong</td>
<td>Medium</td>
<td>Weak</td>
</tr>
<tr>
<td>Risk for fire and explosion</td>
<td>High</td>
<td>Medium</td>
<td>Low</td>
</tr>
</tbody>
</table>

In this study taking into account that Group II and Group III have relatively similar characteristics (see Table 5) three categorise are employed in order to simplify the modelling of Bayesian Network. It is assumed in this study that the cleaning up cost can be estimated by multiplying scale factors which are derived from Etkin (1999)’s studies as shown in Table 6.

<table>
<thead>
<tr>
<th>Categories in present study</th>
<th>Corresponding IOPCF Grouping</th>
<th>Scale Factor for estimating cost of oil spill</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light oil (LO)</td>
<td>Group I</td>
<td>0.2393</td>
</tr>
<tr>
<td>Medium oil (MO)</td>
<td>Group II, III</td>
<td>1.0000</td>
</tr>
<tr>
<td>Heavy oil (HO)</td>
<td>Group IV</td>
<td>1.0146</td>
</tr>
</tbody>
</table>

The correlations between cost of oil spill and the amount of spilled oil dependent on type of oil are shown in Fig. 7 using double logarithm linear regression lines. Types of oil for some incidents are unknown due to the limited data. In order to further investigate the effect and tendency of the type of oil on oil costs Fig. 7 is divided into from Fig. 8, through Fig. 10 dependent on the type of oil. It is noted that no data is available for the light oil in IOPCF (2005). In order to compare the tendency of these 4 figures with each other, three lines based on eq. (1) are shown with the actual data with using following parameter respectively.

<table>
<thead>
<tr>
<th>Table 7. Coefficient for three lines.</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
</tr>
<tr>
<td>Upper line (red line)</td>
</tr>
<tr>
<td>Mean line (orange line)</td>
</tr>
<tr>
<td>Lower line (blue line)</td>
</tr>
</tbody>
</table>
Although the number of data especially for medium oil is limited it can be seen from these figure that most of data are located between upper limit and lower limit, and the global tendency seems to be similar. Therefore it can be said from these figure that effect of the type of oil between Heavy oil (HO) and Medium Oil (MO) on the costs of oil spill is not significant and rather small. The small difference of scale factors based on Etkin’s study in Table 6 (compare 1.0000 and 1.0146) for estimation of the cost of oil spill is supposed to be reasonable.

![Graph showing the effect of type of oil on the cost of spilled oil (IOPCF, 2005).](image-url)

Fig. 7. Effect of type of oil on the cost of spilled oil (IOPCF, 2005).
Fig. 8. Relation between cost of oil spill and the amount of spilled oil for HFO and Bitumen.

Fig. 9. Relation between cost of oil spill and the amount of spilled oil for Crude Oil.

Fig. 10. Relation between cost of oil spill and the amount of spilled oil for MO.
(3) Location
Etkin (1999) postulated that most experts agree that the most important determinant of the clean-up cost is location because location itself is a complex factor involving both geographical and political and legal consideration. In this study Baltic West is employed for the sea area where the oil spill is considered to happen and special considerations for this area such as sensitivity and distance to the location of cleaning up ship are taken into account.

(4) Emulsification of oil
As described in 2.1 the effect of the formation of emulsification on the amount of spilled oil as well as costs of oil is supposed to be large. Formation of emulsification is supposed to be dependent on the type of oil and sea condition.

(5) Weather/ Sea condition
Weather, especially the sea condition, influences significantly the dispersion of the spilled oil and oil clean-up response. If it is stormy, the spilled oil will be dispersed considerably and will spread more rapidly as compared with in calm sea. Large waves may also cause that it becomes problematic for clean-up ships to operate.

(6) Sea Currents
Sea currents significantly affect the direction and the speed of oil spreading. The spilled oil spread more rapidly as the sea currents flows faster although it also depends on the chemical properties of spilled oil. That is viscosity of oil. Lighter Oil (LO) is supposed to spread more rapidly than Heavier Oil (HO) in the same current conditions. It is noted that Lighter Oil tends to evaporate more rapidly than Heavier Oil.

(7) Shoreline oiling / Distance from shoreline
The proximity of oil spill to the shoreline is one of the most important factors impacting clean-up costs. Oil spills that impact the shoreline are considerably more expensive to clean-up than ones which can be dealt with offshore (Etkin, 1999; Etkin, 2001; White, 2003; Etkin et al, 2005). If the distance from the incident to shoreline (L) is very large, oil spill may not pollute the shoreline because some of the oil may evaporate naturally and also the emergency teams will have more time to remove the remaining oil from the sea surface before it reaches the shoreline. In addition, it may be possible to prepare an oil fence or oil boom along the shoreline in order to avoid oil impact to shoreline.

![Fig. 11. Contamination of the shoreline stones by spilled oil.](image)

(8) Oil spill response
Etkin et al (2005) studied the effect of the response method and capability on the costs of oil spill. The results show that the effect of response options and response capabilities on the costs of oil spill is very small if spill
response operations are efficiently carried out. The results also show that the rapid and efficient response operation is one of the most important factors in reducing the costs of oil spills. The effective offshore oil removal is essential for reducing the costs of oil since the contamination of the shoreline usually makes the costs of oil spills extremely high due to the difficulty in cleaning up at shoreline.

2.4. Modelling of costs of oil spill

In order to model a cost of oil spill one can break down the total costs into constituent elements. Friis- Hansen & Ditlevsen (2003) divide the total costs into two main elements. One is the cost for clean-up oil, the other is the cost for other compensations. It is represented as:

\[ C_{\text{total}} = C_{\text{clean}} + C_{\text{other}} \]  

(2)

where \( C_{\text{total}} \), \( C_{\text{clean}} \) and \( C_{\text{other}} \) denote the total costs, costs for clean-up oil and costs for other compensation respectively. \( C_{\text{other}} \) includes several major factors such as damage to the natural resource or environment, harm to wild life, compensation for fishing industry and tourism industry, and so on. In this study \( C_{\text{other}} \) is further broken down into four elements in order to take into account various factors that affect the costs of oil spill. That is the cost for the fishing industry, the cost for the tourism industry, the cost for the environment and costs for the population living in neighbouring area.

Moreover as investigated in 2.3(7) costs for the cleaning up oil is expected to significantly increase once the spilled oil contaminates the shoreline due to the difficulty in cleaning and the long distance of the shoreline. Therefore the clean-up cost is further divided into two elements. One element is the cost for cleaning offshore and the other is the cost for cleaning at the shoreline. Eventually the total cost is represented as:

\[ C_{\text{total}} = C_{\text{clean, offshore}} + C_{\text{clean, shoreline}} + (C_{\text{fish}} + C_{\text{tourism}} + C_{\text{environment}} + C_{\text{population}}) \]

(3)

Considering that the costs for tourism, costs for environment and costs for population are usually related to and taken place around or at shoreline, the formula can be reformed as:

\[ C_{\text{total}} = C_{\text{clean, offshore}} + C_{\text{fish}} + (C_{\text{clean, shoreline}} + C_{\text{tourism}} + C_{\text{environment}} + C_{\text{population}}) \]

(4)

As a result the total cost is divided into two geometrically categorized elements (\( C_{\text{offshore}} \) and \( C_{\text{shoreline}} \)) and the costs for fishing industry (\( C_{\text{fish}} \)). One of the main factors which largely affect \( C_{\text{offshore}} \) and \( C_{\text{shoreline}} \) is the amount of spilled oil at each area. By making \( C_{\text{offshore}} \) and \( C_{\text{shoreline}} \) in the modelling, increase of cost of oil spill due to the arrival of spilled oil at shoreline can be taken into account. That is, if the most of the spilled oil is skimmed up within the offshore before oil reaches the shoreline, it is expected that the estimated costs can be reduced significantly (Etkin et al, 2005).

\( C_{\text{shoreline}} \) includes not only the costs for cleaning up oil (\( C_{\text{clean, shoreline}} \)) but also the costs for other elements. However, to accurately estimate \( C_{\text{shoreline}} \) is quite complex problem because some of them are mutually dependent on each other. In this study a somewhat macroscopic way is employed. \( C_{\text{cleanup}} \) is assumed to be estimated by multiplying a scale factor to \( C_{\text{shoreline}} \) as follows:
\[ C_{\text{shoreline}} = C_{\text{clean,shoreline}} + \left( C_{\text{tourism}} + C_{\text{environment}} + C_{\text{population}} \right) \]
\[ = C_{\text{clean,shoreline}} \left( 1 + f_{\text{tourism}} + f_{\text{environment}} + f_{\text{population}} \right) \]  

where

- \( f_{\text{environment}} \): scaling factor for the cost for environment.
- \( f_{\text{tourism}} \): scaling factor for the cost for tourism industry.
- \( f_{\text{population}} \): scaling factor for the cost for population living close to the contaminated shoreline area.
3. PROBABILITY OF OIL SPILL FROM STRUCK TANKERS

3.1. Frequency and Probability of ship collision

Fig. 12 shows illustration of navigational routes and possible collision. Table 8 shows collision frequencies matrix for possible collisions between own ship and an oil tanker navigating on the route (Fig. 12). The frequency of ship collision can be calculated for each cell in the matrix considering traffic density and navigational route of specific type of ships. The total annual frequency of tanker collision $\lambda_{\text{ALL}}$ (see right bottom of the Table 8) in specific sea area (Great Belt and Oresund) was estimated by summing all the frequency in each cell. It is important to distinguish $\lambda_{\text{ALL}}$ and the frequency of ship collision for one possible striking ship navigating in specific area. The methodology to predict the frequency (probability) of ship collision in each cell is described in Friis-Hansen (2000). The collision scenario in the present study is that own tanker collide with another tanker, and the annual average frequency of collision in the present scenario can be estimated by dividing $\lambda_{\text{ALL}}$ with the number of striking ships in specific sea area.

![Fig. 12. Navigational routes and possible collision area.](image)

Table 8. The table gives a conceptual illustration of the collision frequencies matrix for two navigational routes.

<table>
<thead>
<tr>
<th>Striking Ships on Route 2</th>
<th>Tanker 1</th>
<th>Tanker 2</th>
<th>Tanker 3</th>
<th>…</th>
<th>Tanker 1 (own ship)</th>
<th>…</th>
<th>Tanker N</th>
<th>Container</th>
<th>Bulk Carrier</th>
<th>Others</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tanker A</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tanker B</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tanker C</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td>…</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tanker j</td>
<td>$\lambda_{11}$</td>
<td>$\lambda_{21}$</td>
<td>$\lambda_{31}$</td>
<td>$\lambda_{j1}$</td>
<td>$\lambda_{j2}$</td>
<td>$\lambda_{j3}$</td>
<td>$\lambda_{jN}$</td>
<td>Container</td>
<td>Bulk Carrier</td>
<td>Others</td>
<td></td>
</tr>
<tr>
<td>…</td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<td>…</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sub Total</td>
<td>$\lambda_1$</td>
<td>$\lambda_2$</td>
<td>$\lambda_3$</td>
<td>$\lambda_j$</td>
<td>$\lambda_{i1}$</td>
<td>$\lambda_{i2}$</td>
<td>$\lambda_{i3}$</td>
<td>$\lambda_{IN}$</td>
<td>Container</td>
<td>Bulk Carrier</td>
<td>Others</td>
</tr>
</tbody>
</table>
3.2. Analysis of historical data

The annual numbers of large oil spill incidents from tankers induced by both collision and those by other causes are shown in Fig. 13 based on the annual report of IOPCF (2005). The data contains oil spill incidents from tankers reported to IOPC funds for claiming compensation, which costs at least 6.7 million US$ for single incident depending on the size of tankers (see detail IOPCF, 2005). Therefore relatively small oil spill incidents and oil spills caused by other ships such as general cargo or container ships are not included. However the data are useful to investigate how much is actually paid for compensation of large oil spill incidents.

It is found from the figure that 37 incidents induced by collision have occurred during 1979 and 2005. The incidents which occurred before 1979 seem not fully reported to the IOPCF because the 1971 International Convention on the Establishment of an international Fund for Compensation for Oil Pollution Damage (1971 Fund Convention) entered into force in 1978. Therefore, in this study, the incidents before 1979 are not included in the analysis, especially in order to estimate frequency of incidents. As a result we have 37 incidents during the period of 27 years, and the average annual frequency (oil spill incidents induced by ship collision is estimated as 1.370 [times / year]. It is noted that this represents a very coarse measure of the annual frequency of having an oil spill as the estimated value neither accounts for the change in the tanker fleet at risk during the considered period, neither does it account for any implemented improvements in navigation. During the considered period the tanker world fleet has varied from 2700 to 3200 with approximately 3000 tankers in year 2000. Considering this it is seen that the probability of ship collision is relatively small. It is noted that consequence of one incident sometimes costs more than 1 billion US$ and that the risk of those incidents therefore is supposed to become relatively high.

\[ \lambda[\text{Oil spill}] = 1.37 \quad \text{[times / year]} \]  

\[ (6) \]

![Fig. 13. Annual number of oil spill incidents reported to IOPCF. (The total number of incidents is 132 during the period of 36 years)](image-url)
If probability of large oil spill from struck tankers is assumed to follow the Poisson distribution, the probability that large oil spill from struck tankers occurs \( N \) times during the time duration \( T \) can be represented as:

\[
P[N] = \frac{(\lambda T)^N}{N!} e^{-\lambda T}
\]  

(7)

where

- \( P[N] \) : Probability density function (PDF) of Poisson distribution.
- \( N \) : Number of large oil spill incidents during time period \( T \)
- \( T \) : Time period
- \( \lambda \) : Frequency of large oil spill from struck tankers incidents during time period \( T \)

The probability that the specific incidents occur at least once can be written as:

\[
P[N > 0] = 1 - P[0] = 1 - \frac{(\lambda T)^0}{0!} e^{-\lambda T} = 1 - e^{-\lambda T}
\]  

(8)

Substituting \( \lambda = 1.370 \) [times/year], \( T = 1 \) year, into Eq. (8) the probability that large oil spill from struck tankers occur at least once during one year is calculated as follows:

\[
P[N > 0] = 1 - e^{-1.370} = 0.746
\]  

(9)
It means that large oil spill from struck tankers may happen at least once in each year with the probability of about 75%. This probability is supposed to be relatively high due to considering conditional probability of large oil spills from struck tankers during specific time period of 1 year. This probability will decrease as the time period T decreases. For example if T is 1 month the probability becomes about 10%. Moreover it is not surprising because of the fact that more than 1 oil spill incident induced by collision have been occurring in most of years since 1979 as indicated in Fig. 13. It is noted that the number of data used in this study is limited to only the data reported to IOPCF. It means that incidents with relatively small oil spill exist and which are not reported to IOPCF. It may cause the probability of incidents to slightly increase, but the global tendency is not supposed to change significantly.
4. RISK ANALYSIS

4.1. Risk Identification and the loss Causation Model

This section introduces the so-called “loss causation model” that has been suggested by Bird et al (2003). In order to understand the causative factors of incidents, it is important to build up causation model. The loss causation model for oil spill incidents from ship collision is shown in Table 9. Some investigations have revealed that 80% of incidents involving ships are due to human error such as failure to take appropriate action or to perform good seamanship. It is seen that the causation of oil spill in each item level, but in the present study expected loss given collision incident is investigated because main focus is laid on making a model for estimating the cost of oil spill from struck tankers.

<table>
<thead>
<tr>
<th>Item</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loss</td>
<td>(1) Environmental damage to the adjacent sea area</td>
</tr>
<tr>
<td></td>
<td>(2) Expense for cleaning up spilled oil</td>
</tr>
<tr>
<td></td>
<td>(3) Death of wild life</td>
</tr>
<tr>
<td></td>
<td>(4) Economical damage to fishing industry</td>
</tr>
<tr>
<td></td>
<td>(5) Economical damage to tourist industry</td>
</tr>
<tr>
<td></td>
<td>(6) Fire and explosion due to oil ignition</td>
</tr>
<tr>
<td></td>
<td>(7) Loss of production</td>
</tr>
<tr>
<td>Incident</td>
<td>Oil spill from ship collision</td>
</tr>
</tbody>
</table>

### Immediate Causes

- **[Hardware]**
  - Failure in any part of CAS
  - Struck ship does not have sufficient crashworthiness
- **[Human Errors]**
  - Lack of adequate watch keeping
  - Failure to take adequate course
  - Inadequate navigation or manoeuvrability
  - Doze of a watch officer or a pilot
  - Failure to make a radio communication with the other ship
- **[Environmental Condition]**
  - Bad sea condition
  - Bad visibility (Fog, Heavy Rain, Smoke)

### Basic Causes

- **[Hardware]**
  - Inadequate Maintenance to CAS
  - Inadequate design of hulls
- **[Human Errors]**
  - Improper motivation
  - Lack of skill and knowledge
  - Lack of rest for the watch officer

### Management Control

- **[Hardware]**
  - Inspection scheme to maintain the CAS
- **[Human Errors]**
  - Lack of watch keeping training

* Collision Avoidance System such as IBS, ARPA, Radar and AIS
4.2. Modeling of Bayesian Network

Considering the various factors investigated in previous sections, the Bayesian Network for the present study has been established as shown in Fig. 15. Ellipses represent a “chance node”, which is the most general node representing the random variable with its conditional probability. In this study word “node” is defined as chance node unless otherwise denoted. Rectangles and diamonds show the decision node and utility node respectively. Decision node is the node just to make a decision and does not have conditional probability. Each node has an ID like (C1, T1, D1, and U1). IDs starting from C or T represent a chance node. IDs starting from T (T1, T2…) indicate that the random variables have a unit of time [hour]. Other chance nodes are allocated the IDs starting from C (C1, C2…). IDs starting from D or U represent the decision and utility nodes respectively.

![Bayesian Network Diagram](image)

Fig. 15. Bayesian Network to evaluate costs of an oil spill in a given area

The brown nodes are generally more or less related to oil such as type of oil, oil spill rate or oil weathering.
the amount of the spilled oil. Yellow nodes (T1, T2…) are related to the time such as the time until cleaning ship arrives or the time until oil dissipation. Two dark pink nodes are representing important probability in this model. One dark pink node shows the failure probability of rupture of cargo oil tanks given tanker collision. These probabilities are obtained depending on the structure type of bulbous bow (standard/buffer) by Yamada, Pedersen and Friis-Hansen [P5]. The other dark pink node shows the probability of occurrence of tanker collisions. Light pink nodes are related to the costs of oil spill into the sea such as costs for cleaning up oil and economical damage to the fishing and tourism industries. Light blue nodes are related to the sensitivity factor for the costs of oil spill. Rest of all nodes such as distance and wave height are assigned green color. Due to the large damage by collision the struck tanker might receive flood or considerably loose its residual strength after the collision. Consequently it might be possible that the collision might cause total loss of the ship as well as oil spill from the sunken tankers. However in the present study the sinking of ship induced by the collision damage and oil spill from the sunken ship is not taken into account. Details of some of a node or a set of nodes are described in the following.

**I. Probability of occurrence of tanker collision (C25)**

Probabilities of occurrence of ship collision for own ship with other oil tankers are estimated based on the studies by Friis-Hansen & Ditlevsen (2003). Friis-Hansen & Ditlevsen (2003) calculated the annual frequency of ship collision (\(\lambda_{\text{ALL}}\)) in Great Belt and Oresund respectively (Table 10). It is important to distinguish the frequency of ship collision in specific area (\(\lambda_{\text{ALL}}\)) and the frequency of ship collision for one possible striking ship navigating in specific area. The difference of these two frequencies is illustrated in Table 8. \(\lambda_{\text{ALL}}\) is the summation of frequencies for all possible striking ships in this sea area. In order to estimate the risk of the single striking tanker, an average frequency of tanker collision for single (own) ship is calculated by considering the number of ships navigating in the specific sea area in the following.

\(N_{\text{ships}}\) (Gross) denotes the gross number of tankers passing through the sea area per year, and therefore this number might be counting the same ship more than twice. The net number of tankers is estimated by assuming that same ship will pass through the same sea area every ten days. Eventually the net number of tankers \(N_{\text{ships}}\) (Net) is assumed to pass the same sea area 36 [times/year] and estimated as: \(N_{\text{ships}}\) (Net) = \(N_{\text{ships}}\) (Gross) / 36. The average frequency of tanker collision for single (own) ship is derived by dividing the frequency estimated by Friis-Hansen & Ditlevsen (2003) by the \(N_{\text{ships}}\) (Net).

<table>
<thead>
<tr>
<th>(N_{\text{ships}}) (Gross)(^1) [ships/year]</th>
<th>Great Belt</th>
<th>Oresund</th>
</tr>
</thead>
<tbody>
<tr>
<td>11,500</td>
<td>17,000</td>
<td></td>
</tr>
<tr>
<td>(N_{\text{ships}}) (Net) [ships/year]</td>
<td>315</td>
<td>466</td>
</tr>
<tr>
<td>Frequency for the area(^1) [1/year]</td>
<td>3.69E-03</td>
<td>6.12E-03</td>
</tr>
<tr>
<td>Frequency for one ship [1/year/ship]</td>
<td>1.17E-05</td>
<td>1.31E-05</td>
</tr>
</tbody>
</table>

\(^1\) Friis-Hansen & Ditlevsen (2003)

Based on the Eq. (7) the conditional probability tables for the node “Probability of occurrence of tanker collision” are obtained.
collision (C25)” are created as shown in Table 11, where T=1 [year] is used to estimate the annual risk of oil spill from struck tanker. This probability is used to estimate the expected loss of oil spill (C500).

Table 11. Frequency and probability of occurrence of tanker collision (C25).

<table>
<thead>
<tr>
<th></th>
<th>Great Belt</th>
<th>Oresund</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency</td>
<td>2.810E-05</td>
<td>3.176E-05</td>
</tr>
<tr>
<td>Duration</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>P[N&gt;0]</td>
<td>2.809E-05</td>
<td>3.176E-05</td>
</tr>
</tbody>
</table>

(2) Rupture of Oil Tank (C21)

This node includes the conditional probability of the rupture of oil cargo tank of the struck VLCC given collision obtained in Yamada et al (2007). The probability is dependent on the bow type (D1).

(3) Bow Type (D1)

This node is a decision node that selects the structure of the bulbous bow of the striking ship between standard bow and the buffer bow.

(4) Type of struck tanker (C32)

Oil tankers are divided into five categories dependent on the deadweight tonnage (DWT) of the tankers. A probability density distribution of the number of the tankers in worldwide is shown in Fig. 16 based on the ship database LMIS (2005), where the tankers smaller than 5,000DWT is excluded from this statistics since the main focus is laid on oil spills from large oil tankers. Although the distribution of ship type should be defined depending on each sea area a worldwide distribution can be used as an alternative if the area specific ship type distribution is not available. However in this study the risk of oil spill is estimated only from struck VLCC, and the type of the struck tanker is set to the VLCC with the probability of 100%. Therefore, conditional probability in this node is not used in the present analysis. This node would become useful if the risk of many sizes of struck tankers is focused on in a future study.
Fig. 16. Probability density of tanker fleets in World Wide depending on the size categories of tankers based on world wide data. (DWT > 5,000).

(5) Maximum amount of spilled oil from cargo oil tanks (C6, C2, C9)

Sizes of one cargo oil tank (C6) for 5 categories of oil tankers are approximated as shown in Table 12. These values are used to limit the amount of spilled oil from one cargo oil tank of corresponding struck ships.

It might be possible that two cargo oil tanks are damaged and make oil spills although this probability is supposed to be lower than that of rupture of only one cargo oil tank since transverse bulkhead is stronger than normal transverse web frame. In order to take into account the effect of rupture of two consecutive cargo oil tanks, the number of ruptured cargo oil tanks (C2) is defined and is assumed to take the value of one or two given rupture of oil tank. Moreover it is assumed that the rupture of two oil tanks take place when the center of the striking ship bow collides with the region between transverse bulkhead and the point of distance $L_{\text{damage}}$ from the transverse bulkhead in longitudinal direction as shown in Fig. 17.

Fig. 17. Illustration of oil spill from two consecutive oil cargo tanks (horizontal sectional view).
That is \( L_{cp} \leq L_{damage} \). According to the probabilistic damage study by Lutzen (2001), dimensionless damage length \((L_{cp}/L_{pp})\) for large tankers (VLCC) is estimated as 3.5%. Then it is assumed that \( L_{damage} \) is estimated as half of the dimensional damage length. Dimensionless tank length between two transverse bulkheads \((=L_{tank}/L_{pp})\) in the present VLCC is about \( 4.22\text{[m]} \times 12 \text{[spaces]} / 330\text{[m]} = 15\% \). Therefore probability that the center of striking ship is located in the region of \( L_{damage} \) between transverse bulkheads given rupture of cargo oil tank is estimated as \( 3.5/15 \approx 23\% \). The conditional probability table for the node C2 is shown in Table 13. Maximum amount of spilled oil from tankers (C9) is calculated by C2 times C6. Oil spill amount from tankers (C1) is set to C9 if the amount of spilled oil which is calculated from oil spill rate exceeds C9.

### Table 12. Approximated size of one cargo oil tank with damage hole by collision

<table>
<thead>
<tr>
<th>Approximate DWT [Ton]</th>
<th>Number of Cargo Tanks</th>
<th>Size of one tank [Ton / tank]</th>
</tr>
</thead>
<tbody>
<tr>
<td>VLCC 300,000</td>
<td>15</td>
<td>20,000</td>
</tr>
<tr>
<td>Suezmax 150,000</td>
<td>12</td>
<td>12,500</td>
</tr>
<tr>
<td>Aframax 100,000</td>
<td>14</td>
<td>7,143</td>
</tr>
<tr>
<td>Panamax 70,000</td>
<td>12</td>
<td>5,833</td>
</tr>
<tr>
<td>Handy 50,000</td>
<td>10</td>
<td>5,000</td>
</tr>
</tbody>
</table>

### Table 13. Probability of number of cargo oil tanks ruptured given rupture of cargo oil tank.

<table>
<thead>
<tr>
<th>Number of cargo oil tanks ruptured</th>
<th>Probability</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 tank</td>
<td>77%</td>
</tr>
<tr>
<td>2 tanks</td>
<td>23%</td>
</tr>
</tbody>
</table>

(6) Time to contact cleaning ship (T1).

A set of yellow and green nodes located in the upper left part of Fig. 15 relates to the estimation of the response time of the emergency team. Among these nodes T1 represents the time after oil spill takes place until the incident is reported to the cleaning ship.

(7) Time to stop oil spill (T2).

The effect of the response time and its efficiency on the costs of oil spill is very large. This node represents the time after cleaning ships arrive at the location of oil spill until the mechanical containment of oil established or oil spill is stopped. This time is supposed to vary considerably depending on the condition of oil spill, weather, sea conditions, response system, and strategy of the cleaning up process and so on. Therefore wide range of time scattering with equal probability is assumed as shown in Table 14.

### Table 14. Time to stop oil spill.

<table>
<thead>
<tr>
<th>Time [hours]</th>
<th>Probability</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 - 48</td>
<td>33.3%</td>
</tr>
<tr>
<td>48 - 168</td>
<td>33.3%</td>
</tr>
<tr>
<td>168 - 720</td>
<td>33.3%</td>
</tr>
</tbody>
</table>

(8) Time until containment of oil established (T10, C11 and C12).

T10 represents the time after oil spill happens until oil spill is stopped or the mechanical containment of the spilled oil by oil fence is established. To estimate this time, it can be necessary to consider the time to contact cleaning ship (T1), the time that the response ship arrived at the location of oil spill and the time to stop oil spill.
spill (T2). T10 is calculated as:

\[ T10 = T1 + T2 + \frac{C11}{C12}. \]

“C11/C12” is the time cleaning ships need to arrive at the location of oil spill. The node sea condition (C30) is related to the speed of the response ship given that its speed will not be high in stormy conditions. Our supposition is that with rough sea condition, the speed of response ship will be between 0 and 6 knots. The time to contact and prepare a response ship is considered to be within 24 hours after oil spill happens. This time is considered to be up to 12 hours with 50% of probability and from 12 to 24 hours with the remaining 50% of probability. This time is taken into account in the total time to arrive.

(9) Time until spilled oil reach shoreline (T5)

This node is calculated by dividing the distance from the location of oil spill to shoreline (C10) divided by the velocity of oil (C29). The velocity of oil is dependent on the type of oil (C4) since generally the heavier oil flow slowly than lighter oil.

(10) Oil type (C4)

Types of oil are divided into three categories considering its relative density, viscosity and persistency, that is, light oil, medium oil and heavy oil. The probability of these types of oil is assumed to be equal since no useful and reliable data is available at the moment.

(11) Oil spill rate from ships (C3)

Oil spill rate is the amount of oil released in the sea per hour [ton/hour]. Oil spill rate is dependent various factors such as the size of the rupture hole of the cargo oil tank, the relative location of the hole against sea level, type of oil, temperature and so on. In this study oil spill rate represents average value of oil spill rate during the entire oil spillage. Therefore, it is treated as constant in the analyses. Three ranges of oil spill rates with equal probability are considered.

<table>
<thead>
<tr>
<th>Oil spill rate [ton/hour]</th>
<th>Probability</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-50</td>
<td>33.3%</td>
</tr>
<tr>
<td>50-100</td>
<td>33.3%</td>
</tr>
<tr>
<td>100-200</td>
<td>33.3%</td>
</tr>
</tbody>
</table>

(12) Oil spill from ships (C1)

The amount of oil spilled is estimated by oil spill rate (C3) times the time until containment of oil established (T10). It is assumed that the containment of the struck ship by oil fence or boom is carried out immediately soon after the cleaning ship arrives at the location of oil spill. Moreover the maximum amount of spilled oil is limited to the capacity of one cargo oil tank (C6) of the corresponding type of struck tanker (C32).

(13) Natural dissipation of oil (T20, C71, C72 and C73)

Natural dissipation or evaporation of spilled oil as investigated in 2.1 is taken into account dependent on the type of oil. The extent of oil evaporation from the sea surface is largely dependent on the type of oil. Fig. 18 (ITOPF, 2002) shows typical time histories of amount of spilled oil dependent on the type of oil. Grouping from 1 to 4 in the figure corresponds to the categorization by ITOPF as described in 2.3(2).

It is found from the figure that oil in Group I evaporates rapidly and will completely dissipate in 7 or 8 hours.
whilst oil in Group 4 will be remaining more than 25 % in the sea even 2 year after oil spill happens. For simplification of modeling Bayesian Network, linear relationship is assumed between the time duration and the dissipation rate. Assumed dissipation rate are derived from Fig. 18 as shown in Table 16. The node C71 calculates the ratio [%] of the natural dissipation of the spilled oil. In this node, it is possible to have more than one hundred percent of dissipation. Therefore, in the next node C72, numbers greater than 100% are corrected to 100%. The ratio [%] of the remaining oil after natural dissipation is calculated in the node C73. In this node, a simple equation is made with the purpose of reducing the total quantity of spilled oil by the percentage already dissipated.

Table 16. Natural dissipation rate of oil

<table>
<thead>
<tr>
<th>Type of oil</th>
<th>Duration H [hours]</th>
<th>Dissipation Ratio R [%]</th>
<th>Dissipation Rate [%/hour] (= R/H)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light Oil (Group I)</td>
<td>192 (=8 days)</td>
<td>100</td>
<td>0.521 (=100/192)</td>
</tr>
<tr>
<td>Medium Oil (Group II, III)</td>
<td>336 (=2 weeks)</td>
<td>100</td>
<td>0.298 (100/336)</td>
</tr>
<tr>
<td>Heavy Oil (Group IV)</td>
<td>17520 (=2 years)</td>
<td>75</td>
<td>0.00428 (=75/17520)</td>
</tr>
</tbody>
</table>

The time considered in dissipation node is the result smallest time comparing time until cleaning ship arrives and time oil takes to touch the shoreline. The result of this equation is given in node “Time for dissipation”. In this case, it is considered that dissipation stops when the cleaning ship arrives or when the first part of oil reaches the shoreline.

Fig. 18. The rate of removal of oil from the sea surface according to type of oil (ITOPF, 2002)
(14) Sea Conditions (C30)
Sea condition significantly affects various factors such as cleaning up operation and formation of oil emulsification. Following three conditions are assumed with its conditional probability.

<table>
<thead>
<tr>
<th>Table 17. Sea conditions.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calm Sea</td>
</tr>
<tr>
<td>Moderate</td>
</tr>
<tr>
<td>Rough Sea</td>
</tr>
</tbody>
</table>

(15) Emulsification of oil (C27, C34)
As studied in the 2.1(4) the effect of the emulsification of the spilled oil on the amount of the spilled oil and on the costs skimming up is large. The emulsification of oil is dependent on the sea wave condition. Therefore the C27 is dependent on the sea condition (C30). It is also assumed that once the oil is judged to form emulsification all amount of the spilled oil forms the emulsification. The following conditional probabilities are assumed for the probability of formation of emulsification. The amount of oil which formed emulsification is supposed to increase its volume 3 times larger than original volume according to the investigation in 2.1.

<table>
<thead>
<tr>
<th>Table 18. Probability of emulsification dependent on the type of oil and sea condition.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sea Condition</td>
</tr>
<tr>
<td>Rough</td>
</tr>
<tr>
<td>Moderate</td>
</tr>
<tr>
<td>Calm</td>
</tr>
</tbody>
</table>

(16) Oil velocity (C29)
Oil velocity depends on the type of oil since heavier oil tends to move slowly in blocks due to its viscosity and persistency, and is not as scattered in the sea surface as lighter oil.

(17) Oil spill at shoreline after natural dissipation (C16)
Oil spill shoreline is calculated by oil spill rate considering natural dissipation (C5) [ton/hour] times the time after oil reach shoreline until oil is stopped or oil containment is established by oil fence (T7). The time for reaching shoreline is result of the time until cleaning ship arrives at the location of oil spill subtracted by the time until spilled oil reach shoreline (T5).

(18) Oil spill at offshore after natural dissipation (C23)
The amount of spilled oil at offshore is calculated by subtracting the amount of oil at shoreline (C8) from the total amount of spilled oil (C16), where natural dissipation of spilled oil is already taken into account in each node.

(19) Costs at shoreline (C301)
This node represents the costs of oil spill at shoreline and is calculated by use of the model as described in 2.4. The costs at shoreline is calculated by use of the amount of spilled oil at shoreline after natural dissipation (C31) is considered by the rate of oil considering dissipation times the time reaching shoreline. This amount must be less than the total amount of oil considering dissipation. Costs of the shoreline are based on the study by Etkin (1999) which evaluated the costs per ton spilled for heavy, medium and light oil. Hence the costs shall be weight of oil washed ashore times the costs per ton concluded. Moreover, these costs are multiplied by a factor
that will be explained in the brown rectangle.

(20) Cost for shoreline, offshore and fishing industry (C301, C302 and C303).

In this rectangle, we may notice the costs offshore. Weight of oil offshore is the result of the total oil not dissipated subtracted by oil washed ashore. Costs are this quantity multiplied by the costs per ton for each type of oil considered in this research. Another node included in this area is the fishery industry compensation. In this case, costs are only related to the total amount of oil dropped by the ship. Costs per ton were found by the costs paid by Baltic Carrier to fishery industry divided by the amount of oil spilled in that occasion.

(21) Sensitivity factor Rank of the polluted area (C700).

It is supposed that the costs of an oil spill to increase significantly in case of coast pollution in special sensitivity areas such as environment protected area. There are three special considerations to increase expenses in a polluted shoreline, such as number of hotels, hostels and campers in the area (business related to tourism), number of inhabitants in the region and number of ecological protected areas in the polluted shoreline. The factor for business in the area is up to 15% increase of the total shoreline cleaning costs if there are more than 50 establishments in the area. The factor decreases as the number of small business declines.

The second factor considers how many wildlife protection areas there are in the polluted shoreline. We consider a maximum factor of 6% if there are more than 5 protected areas within the polluted region. The last factor accounts the population living in the area. The maximum factor for population is 20% for a population with more than two hundred thousand inhabitants. Those three factors are added in order to get a unique factor to multiply the shoreline cleaning cost.

5. RESULTS AND DISCUSSION

5.1. Risk of oil spill from struck tankers in Great Belt and Oresund

Risk (expected loss) of oil spill from struck VLCC in Great Belt and Oresund obtained from the present risk analyses are shown in Table 20 and Table 21 respectively, where the type of oil is assumed to be heavy fuel oil since the struck ship is assumed to be a VLCC. The risk in case that the striking ship is using buffer bow (R_s) is compared with that in using standard bow (R_b). It is noted that R_s and R_b are risk for 1 year since annual frequency of ship collision is used to obtain these values. The risk reducing effect of a striking ship by using buffer bow, R_d, is calculated for the period of 1, 20 and 30 years considering the reduced risk during the entire life of ships. R_d for 1 year is calculated as R_d = R_s - R_b.

Undoubtedly, the risk of oil spill will change in the future. The traffic composition and number of vessels will likely increase and hence (everything else equal) increase the frequency of ship collisions. According to the LMIS the world fleet of ships is gradually increasing each year, and it is expected that increase in the number of ships can be a factor to increase frequency of ship-ship collisions. Development of the advanced navigational equipments can be another factor to avoid ship collisions, and will improve and hence decrease frequency of ship-ship collisions. Other important factors are: size and speed of vessels, condition of buoyage, and conditions of ships. It is not obvious in what direction the resulting future navigational risk picture will change. Therefore, it is assumed in the present estimation that frequency of ship collision does not change significantly in the next 20 - 30 years.

It can be seen in Table 20 and Table 21 that the buffer bow has some risk reducing effects. Quantitatively the effect of the buffer bow on reducing the risk of one striking ship for 30 years (R_d,30) is about 37,000-42,000 [US$/ship/30years]. Following standard decision theory the implementation of the buffer bow is only cost-efficient if R_d is larger than the cost of introducing the buffer bow. No information is available about the costs of changing the design from the standard bow to the buffer bow. The expense of the buffer bow design is
supposed to be almost same as that of the standard bow although quantitative analyses have been not fully performed to validate these. One of the reasons is that buffer bow in the present design concept does not need any expensive special material and only need to change the stiffening system from longitudinal system to transverse system by use of same material of steel. Moreover cost of steel in case of buffer bow is expected to be lower than that in case of standard bow due to the reduction of plate thickness of outer shell although cost for welding might increase due to the increase of transverse ring frame with relatively smaller stiffening space (650-750mm). Considering these issues, it can therefore be expected that initial costs of buffer bow is almost the same as for a standard bow. It is also noted that the repair costs for the buffer bow as compared to that for the standard bow need to be considered although such data is also not available within the present study. It can be expected that repair costs for the buffer bow is higher than that of the standard bow. However, it is disputable to directly compare this cost since repair costs are usually paid by insurance company and actual costs for repair is reflected the increase of the premium of insurance.

Considering this it can be said from the present analyses for a ship of 30 years lifetime navigating in Great Belt and/or Oresund that the buffer bow is cost-effective only if the expected additional costs for introducing bow is less than around 37,000 [US$].

Moreover it is noted that risk reducing effect of buffer bow by one ship is not fully taken into account in “risk analysis of local sea areas” since risk reducing effect in another sea area is not taken into account in these analyses although the ships might navigate many other sea areas. For example the ships passing through Great Belt and Oresund might navigate other sea areas such as Baltic West and Atlantic Sea.

### Table 19. Comparison of failure probability of outer shell (P_{fos}) and inner shell (P_{lis}), and expected oil spill amount given VLCC is struck by another ship. The maximum size of one cargo oil tank of VLCC is assumed to be 20,000 [ton] as shown in Table 12.

<table>
<thead>
<tr>
<th></th>
<th>P_{fos}</th>
<th>P_{lis}</th>
<th>E [oil spill] [ton]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard Bow</td>
<td>0.88</td>
<td>0.66</td>
<td>5,254</td>
</tr>
<tr>
<td>Buffer Bow</td>
<td>0.75</td>
<td>0.44</td>
<td>3,699</td>
</tr>
<tr>
<td>Ratio</td>
<td>85%</td>
<td>66%</td>
<td>70%</td>
</tr>
</tbody>
</table>

### Table 20. Comparison of risk of oil spill from VLCC in Great Belt.

<table>
<thead>
<tr>
<th></th>
<th>Standard Bow Rs</th>
<th>Buffer Bow Rb</th>
<th>Reduction [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Risk for striking ship [US$/year/ship]</td>
<td>4,353</td>
<td>3,120</td>
<td>28%</td>
</tr>
<tr>
<td>Reduced Risk Rd [US$/ship]</td>
<td>1,233</td>
<td>24,665</td>
<td>36,997</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>1 year</th>
<th>20 years</th>
<th>30 years</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reduced Risk Rd [US$/ship]</td>
<td>1,233</td>
<td>24,665</td>
<td>36,997</td>
</tr>
</tbody>
</table>
Table 21. Comparison of risk of oil spill from VLCC in Oresund.

<table>
<thead>
<tr>
<th></th>
<th>Standard Bow Rs</th>
<th>Buffer Bow Rb</th>
<th>Reduction [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Risk for striking ship [US$/year/ship]</td>
<td>4,926</td>
<td>3,531</td>
<td>28%</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Reduced Risk Rd [US$/ship]</th>
<th>1 year</th>
<th>20 years</th>
<th>30 years</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1,396</td>
<td>27,912</td>
<td>41,868</td>
</tr>
</tbody>
</table>

R_s: Annual risk of oil spill from struck tankers in case that striking ships are using standard bulbous bow.
R_d: Annual risk of oil spill from struck tankers in case that striking ships are using buffer bulbous bow.

5.2. Risk of oil spill from struck tankers in world wide operation

Similar to the case in Great Belt and Oresund, risk of oil spill from VLCC in World Wide operation is roughly estimated by assuming that all the analyses condition other than probability of collision is the same as those in Great Belt and Oresund. Hence, the annual frequency of tanker collision for a single ship can be estimated as:

\[
\lambda[\text{coll}] = \frac{1}{N_{\text{ships}}} \cdot \frac{\lambda[\text{oilspill}]}{P[\text{oilspill|coll}]} \tag{10}
\]

where \(\lambda[\text{oilspill}]\) is the annual frequency of oil spill from tankers in World Wide estimated from the historical data (\(= 1.37 \text{ [incidents/year]}\) as described in 3.2). \(P[\text{oilspill|coll}]\) denotes probability of oil spill from tankers given tanker collision. Considering that \(\lambda[\text{oilspill}]\) from historical data includes the oil spills from both single hull tankers and double hull tankers \(P[\text{oilspill|coll}]\) can be written as:

\[
P[\text{oilspill|coll}] = P[\text{rupture|collSHT}] \cdot P[\text{collSHT}] + P[\text{rupture|collDHT}] \cdot P[\text{collDHT}] \\ 
\approx P_{f.\text{OS}} \cdot P[\text{collSHT}] + P_{f.\text{IS}} \cdot P[\text{collDHT}] \tag{11}
\]

where

- \(P[\text{rupture|collSHT}]\) : probability of rupture of outer shell given collision with single hull tankers.
- \(P[\text{collSHT}]\) : probability of collision with single hull tankers.
- \(P[\text{rupture|collDHT}]\) : probability of rupture of inner shell given collision with double hull tankers.
- \(P[\text{collDHT}]\) : probability of collision with double hull tankers.

By assuming \(P[\text{collSHT}] = P[\text{collDHT}] = 50\%\), \(P[\text{oilspill|coll}]\) can be estimated from Table 19 as an average of \(P_{f.\text{OS}}\) and \(P_{f.\text{IS}}\) in case of using standard bow. That is \((66+88)/2 = 77\%\). Considering the approximate number of world fleet of 48,000, frequency of tanker collision for single ship in world wide can be estimated as:

\[
\lambda[\text{coll}] = \frac{1}{48000} \cdot \frac{1.37}{0.77} = 3.707 \times 10^{-5} \tag{12}
\]

It is noted that the collisions by small ships are not taken into account and therefore it is supposed that
\( P[\text{oilspill}|\text{coll}] \) is estimated higher than real value. Therefore \( \lambda[\text{coll}] \) might be estimated lower than actual value, which results in conservative or modest estimation of risk reducing effect of the buffer bow. Estimated frequency \( \lambda \) and probability is shown in Table 22. It is noted that historical data does not take into account the effect of advanced navigational equipment in the future or density of traffic. Moreover this frequency does not take into account the navigational route of each ship. Therefore this frequency is supposed to be average frequency of tanker collision in world wide.

The risk of oil spill from struck VLCC and the risk reducing effect of the buffer bow in World Wide operation are shown in Table 23. It can be seen from Table 23 that in case of world wide analyses the value of \( R_d \) increases slightly. This is presumably because the relative frequency of tanker operations world wide is larger than the relative frequency experienced in Great Belt and Oresund. The estimated collision frequency in world wide operation is extremely rough since the frequency is estimated from the historical data and does not take into account ship traffic and specific ship. However, it can be expected that the possibility that the buffer bow design becomes cost-effective for ships in world wide operation is relatively high, although more accurate calculation might be necessary in estimating frequency of tanker collision.

It is noted that the present risk analysis of oil spill from struck tankers is very sensitive to probability of ship collision. Moreover, it is also noted that estimating probabilities of tanker collisions is very complex and needs a lot of data which often may not be available.

Table 22. Estimated frequency and probability of occurrence of tanker collision in world wide.

<table>
<thead>
<tr>
<th>World Wide</th>
<th>Frequency ( \lambda[1/\text{year/ship}] )</th>
<th>P[N&gt;0]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency ( \lambda )</td>
<td>3.707E-05</td>
<td>3.707E-05</td>
</tr>
<tr>
<td>Duration ( T[\text{year}] )</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>P[N&gt;0]</td>
<td>3.707E-05</td>
<td></td>
</tr>
</tbody>
</table>

Table 23. Comparison of risk of oil spill from VLCC in World Wide.

<table>
<thead>
<tr>
<th>Standard Bow Rs</th>
<th>Buffer Bow Rb</th>
<th>Reduction [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Risk for striking ship [US$/year/ship]</td>
<td>5,743</td>
<td>4,116</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Reduced Risk Rd [US$/ship]</th>
<th>1 year</th>
<th>20 years</th>
<th>30 years</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,627</td>
<td>32,538</td>
<td>48,807</td>
<td></td>
</tr>
</tbody>
</table>

According to the ship casualty statistics, many repairs on the bulbous bow including collisions with ships, collision with floating structure, slamming, and so on, are reported. But in most cases the detail of damage extents and repair costs are not disclosed to public or even for research purpose, mainly due to the reason of ship owners confidentiality. Since especially bulbous bow contains specific know how about propulsion characteristics of ships and confidential information of shipbuilding companies, the number of accessible and available data is extremely limited practically.

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6. CONCLUSION

A methodology to evaluate the risk of oil spill from struck tankers is established using Bayesian Network. As a case study risk analysis of oil spill is carried out in the specific collision scenarios that a VLCC is struck by another tanker in the Great Belt and Oresund. The risk of oil spill when using buffer bows is compared with that using standard bows. The following conclusions are obtained from the present study.

(1) It is estimated from the present analyses that the risk of oil spill for one single striking ship per year in case of using buffer bow is 28% lower than that in case of using standard bow. This corresponds to a risk reducing effect about 37,000-42,000 [US$] for one single striking ship if the lifetime of the striking ship is assumed to be 30 years. Therefore, it can be said that introducing buffer bow to a ship passing thorough Great Belt and Oresund is cost-effective only if the initial shipbuilding costs and the repair costs between standard bow and buffer bow is smaller than 37,000-42,000 [US$/ship]. It should be mentioned that no such data are available for the present study.

(2) Similar to the case in Great Belt and Oresund, risk of oil spill from a struck VLCC in World Wide operation is roughly estimated by assuming that all the analysis conditions other than the probability of collision is the same as those in Great Belt and Oresund. It is roughly estimated that a buffer bow has a risk reducing effect of about 48,000 [US$] for one single striking ship if the lifetime of the ship is assumed to be 30 years. Therefore it can be expected that introducing buffer bow to a ship passing thorough world wide is cost-effective if the initial shipbuilding costs and the repair costs between standard bow and buffer bow is smaller than 48,000 [US$/ship] although further study is necessary to accurately estimate the risk of oil spill from struck tankers in case of world wide.

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REFERENCE


ITOPF (2002), “Fate of Marine Oil Spills”, Technical Information Paper, No.2


PhD Theses
Department of Naval Architecture and Offshore Engineering
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1961 Strøm-Tejsen, J.
Damage Stability Calculations on the Computer DASK

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Determination of the Weight Distribution of Ship Models

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A Planar Motion Mechanism

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A Phase Changer in the HyA Planar Motion Mechanism and Calculation of Phase Angle

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Anvendelse af statistiske metoder til kontrol af forskellige tilsnitmeldelses-formler og udarbejdelse af nye til bestemmelse af skibes tonnage og stabilitet

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Eksperimentel og beregningsmæssig bestemmelse af vindkræfter på skibe

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Datamatorienterede studier af planende bådes fremdrivningsforhold

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Store sideportes indflydelse på langskibs styrke

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Vibrations in Ships

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Buling af afstivede pladepaneler

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Sammenkobling af rotations-symmetriske og generelle tre-dimensionale konstruktioner i elementmetode-beregninger
<table>
<thead>
<tr>
<th>Year</th>
<th>Author</th>
<th>Title</th>
</tr>
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<tbody>
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<td>Elastic-Plastic Collapse of Long Tubes under Combined Bending and Pressure Load</td>
</tr>
<tr>
<td>1980</td>
<td>Petersen, M.J.</td>
<td>Ship Collisions</td>
</tr>
<tr>
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<tr>
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<td>Nielsen, N.J.R.</td>
<td>Structural Optimization of Ship Structures</td>
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<tr>
<td>1984</td>
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<tr>
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<td>Structural Design of Sandwich Structures</td>
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<tr>
<td>1990</td>
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<td>Structural Analysis of Marine Structures</td>
</tr>
<tr>
<td>1990</td>
<td>Wedel-Heinen, J.</td>
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</tr>
</tbody>
</table>
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Life Cycle Model for Offshore Installations for Use in Prospect Evaluation

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Analysis of Slender Marine Structures

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<th>Year</th>
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</tr>
</thead>
<tbody>
<tr>
<td>1998</td>
<td>Andersen, M.R.</td>
<td>Fatigue Crack Initiation and Growth in Ship Structures</td>
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<td>1998</td>
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<tr>
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</tbody>
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