Collision with a Crushable Bow

by

Peter W. Gooding

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Signature of Author..... ....
Department of Ocean Engineering and
the Department of Mechanical Engineering
05 May 2000

Certified by ....
Tomasz Wierzbicki
Professor of Applied Mechanics
Thesis Supervisor

Certified by ... .
Frank McClintock
Professor of Mechanical Engineering
Thesis Reader

Accepted by ....
Professor Ain A. Sonin
Chairperson, Department Committee on Graduate Studies
Department of Mechanical Engineering

Accepted by . ...
Professor Nicholas Patrikalakis
Chairperson, Department Committee on Graduate Studies
Department of Ocean Engineering

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Abstract

Oil tanker collisions and groundings pose the potential for large spills. Currently, the International Maritime Organization and national governments have design standards for tankers that do not account for a vessels' crashworthiness. By using crashworthiness, a vessel can be optimally designed for both weather and extreme loading situations.

Minorsky (1959) examined the problem of collision damage on a vessel side wall. Using previous collision data, Minorsky generated a simple damage volume to kinetic energy relationship for the safety of these reactors. Since then, many researchers have provided solutions for the extent of damage on a tanker based on an assumption of a rigid bow collision. As a result, the damage extent to the side of a vessel is over-estimated. By adding a crushable bow, the final result will allow for a more optimally designed vessel. This paper provides a simple closed form solution for modeling the collision resistance of the bulbous bow portions of tankers. Later, this closed form solution can be combined with side collision solutions to optimally design a tanker for extreme loading conditions.

Thesis Supervisor: Tomasz Wierzbicki
Title: Professor of Applied Mechanics

Thesis Reader: Frank McClintock
Title: Professor of Mechanical Engineering

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Chapter 1

Introduction

Figure 1-1: Tanker Collision

The purpose of this research is to provide a simple closed form solution for modelling the collision resistance of the bulbous bow portion of tankers. In solving this problem, two collision scenarios are evaluated separately (uncoupled) and combined (coupled) to determine the overall damage to each vessel. Two uncoupled scenarios are shown in figure (1-2). The first uncoupled scenario is a rigid bow colliding with the side of a tanker, already done in the computer program DAMAGE (Damage Assessment of Grounding and Collision Events), version 4. The second scenario is a crushable bow colliding with a rigid wall, the subject of this research.

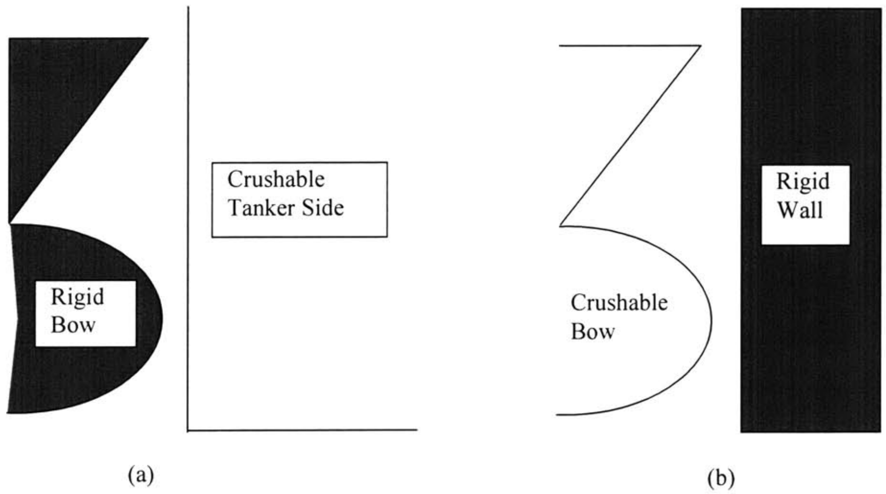

Figure 1-2: Uncoupled Collisions (a) rigid bow with tanker side wall (b) bow with rigid wall

As seen in figure (1-3), the bow of a tanker consists of two portions, the upper rake portion and the lower bulbous bow portion. The purpose of this research is to develop a simple closed form solution for the force-displacement relationship of the bulbous bow. With this relationship, the DAMAGE model can be enhanced to allow for a crushable bow.

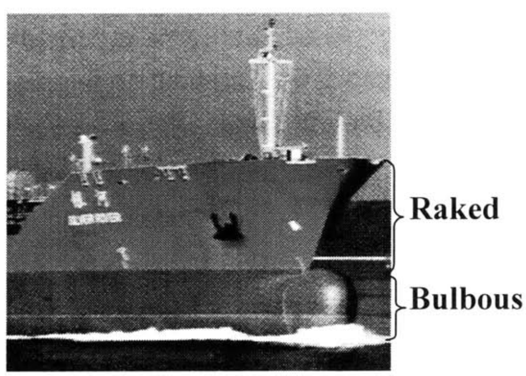

Figure 1-3: A Tanker Bow Consisting of Two Distinct Parts

1.1 Background

Oil tanker collisions and groundings pose the potential for large spills. Currently, the International Maritime Organization and national governments have design standards for tankers that do not account for a vessels' crashworthiness. By using crashworthiness, a vessel can be optimally designed for both weather and extreme loading situations.

Minorsky (1959) examined the problem of collision damage on a vessel side wall. His initial formulation was developed for the safety of ships powered by a nuclear reactor. Using previous collision data, Minorsky generated a simple damage volume to kinetic energy relationship for the safety of these reactors. Since then, many researchers have provided solutions for the extent of damage on a tanker based on an assumption of a rigid bow collision. Such a formulation does not account for the work consumed by the bow deformation. As a result, the damage to the side of a struck vessel is over-estimated. By adding a crushable bow, the final result will predict less damage on the side wall of a tanker and a better overall force-deflection relationship.

Figure 1-4: Typical Bulbous Bow

1.2 History

The crushing mechanics of tanker bows have been studied by a number of researchers including Wierzbicki et. al. (1982), Chang, et al. (1980), Yang and Caldwell (1988), Pederson, et al. (1993), Wang, et al. (1995), and Paik, et. al. (1998). Chang (1980) used methods developed

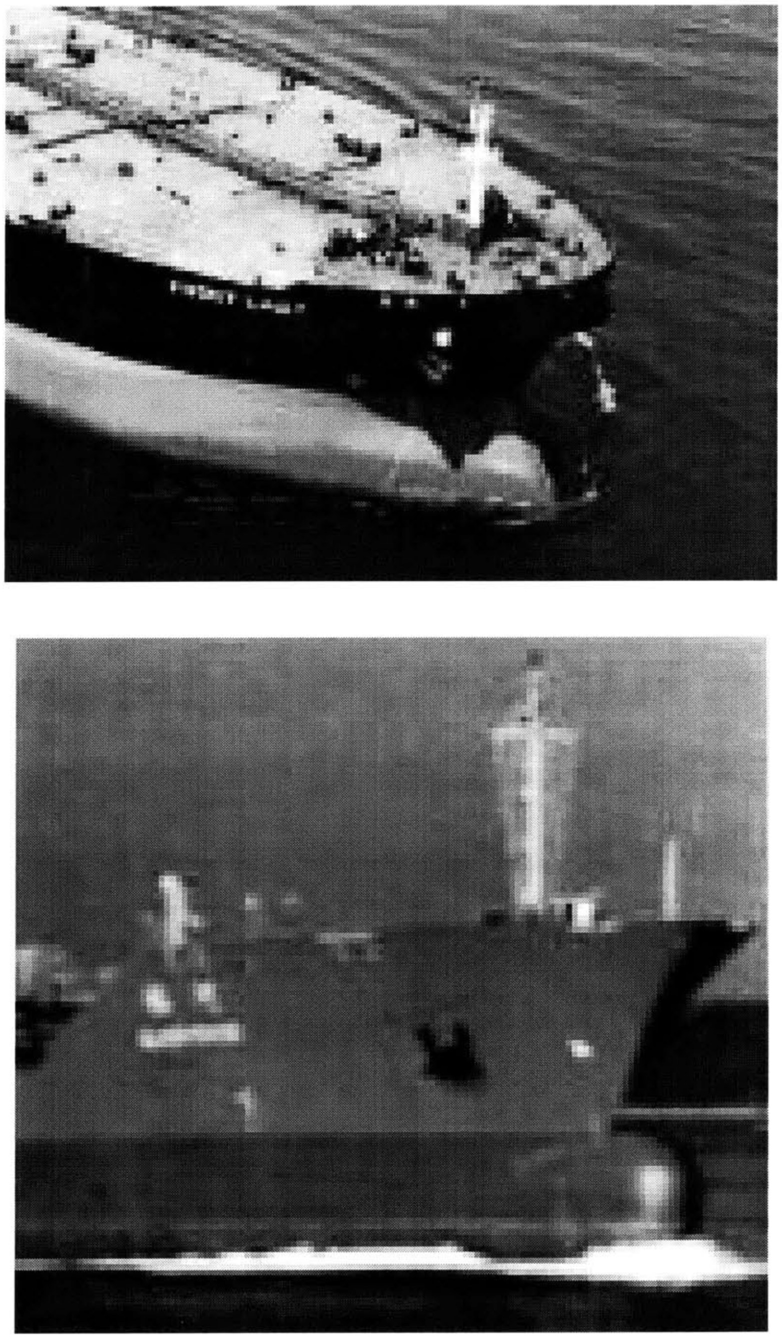

Figure 1-5: More Typical Bulbous Bows

for the aircraft and automotive industries and relates them to the shipping industry. Chang used the results of model tests conducted by Woisin (1976) and generated equations to describe these deformations. These equations were a function of the material properties and vessel speed. However, this solution is similar to Minorsky, in which it only relies on the speed of the vessel and approximate shape to determine the extent of damage. Chang improved Minorsky's equation by adding a striking ship bow factor. This striking bow factor reduces the forces in collision and is a factor of the bow's plate thickness. This reduction factor does not account for any variation in bow shape.

Yang and Caldwell (1988) developed a bow deflection equation based on the calculation of bending and membrane work of ship plating. This simple equation uses work formulas developed by Abramowicz (1983) and Wierzbicki (1983). However, their deflection calculation relates the overall deflection of the bow to the bow deflection wavelength (H) without optimizing this distance. The equation accounts for the plate thickness and width but it ignores the involvement of longitudinal stiffeners. The derived solution is applicable only to square columns and boxes. Thus it is more suited for the raked portion of the bow than the bulbous bow.

Pederson (1993) evaluates the Yang and Caldwell method along with Amadhl method for the bow deformation in collisions. Pederson's papers uses these force-deformation relationships to determine the force application to bridges and piers from these striking ships. Pederson identified five typical bow structures that he uses to compare various force-deflection relationships of bows. From these different equations, Pederson developed a relationship between vessel size and collision forces placed on a stationary structure. Wang (1995) proposed a one term approximation for the bulbous bow crushing force, which is a function of the total cross sectional area of the bow. This is an approximation that does not allow for variations in cross sectional shape or location and size of the bow's girders and stiffeners. Paik (1998) developed a program, BOWCOL, to calculate the mean crushing force of a bow. This program is applicable for head-on collisions. It uses a thickness smearing method to account for various stiffeners. The Paik solution allows to a limited extent for variation in girder and stiffener sizing and spacing.

Wakeland (1980) discussed the importance of an adequately designed bow for collision. In the 1975 collision of *Sea Witch/ Esso Brussels*, there was 23 fatalities and 23 million dollars in damage. Minorsky and Chang showed that the bow of the *Sea Witch* was stronger than necessary, causing a deeper penetration into the side of the *Esso Brussels*. Besides determining the extent of damage to the struck vessel, an optimally designed bow will be able to properly respond in this type of collision.

Chapter 2

Statement of Problem

2.1 Problem

The behavior of two ships colliding has been modelled in the computer program DAMAGE. This program can determine the extent of damage for a tanker collision. The main assumption is that the striking ship has a rigid bow and all of the loss in energy is absorbed by the deformation of the struck ship. However, actual collisions show that the bow damage can account for as much as fifty percent of the work done on the structures. Damage to the struck ship is reduced by the deformation of the striking ship's bow.

2.2 Procedure

The collision problem is divided into two scenarios. The first is a rigid bow colliding with the side of a tanker. These calculations have already been done in the program DAMAGE. The second scenario is a crushable bow colliding with a rigid wall. The two scenarios are combined to evaluate the force-deflection relationship between the colliding vessels. This is done by maintaining force equality at the interface of two ships until their relative velocity is zero.

The bow can be divided into two sections as shown in figure (1-3). The raked portion of the bow has been studied by Kim (1999). For the bulbous portion, an approximate method suggested by other problems will be used here. The first method is thickness smearing in which all of the stiffeners and girders will be converted to an equivalent outside thickness of the

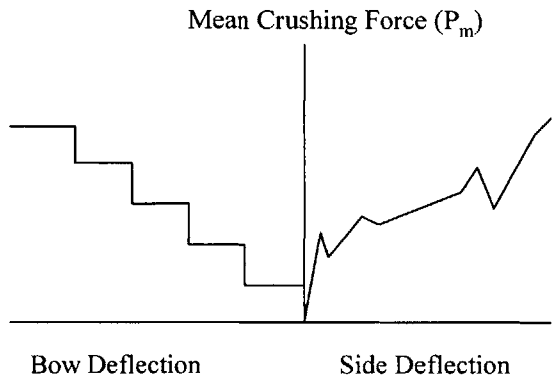

Figure 2-1: Combined Force-Deflection Curve

bulbous plating. This will increase the thickness of the plate and simplify the force-deflection calculations to one equation. The other method will be to use the intersecting unit (plate) method. Wierzbicki et al. (1982) and later Abramowicz (1994) developed crushed formulas for angle, "T", and cruciform shapes to calculate the crushing strength of an object. These two methods will be compared to help judge their validities.

2.3 Force Optimization

Throughout these equations, the formulation depends on minimizing the mean crushing force. By minimizing the mean crushing force, the solution will give the least amount of energy necessary to cause plastic deformation. The force is minimized by adjusting the folding wavelength (H). The folding wavelength is the folding distance to which the structure will deform with the least amount of load applied. In figure (2-2), a plot of the mean crushing force (P_m) divided by the optimal crushing force (P_{opt}) is plotted versus the folding distance (H) divide by the optimal folding distance (H_{opt}). This graph shows that if the bow is to crush in the easiest mode, it will crush at the lowest required force. The theorem of Least Upper Bound (Chakrabatry, 1987) is tentatively applied since the geometry is continuously changing.

However, not all deformations occur at this point. By adding the transverse frame spacing, the actual folding wavelength may not be optimum. This deflection will increase the force required to cause the structure to deflect over this wavelength. Thus, the intersecting method provides a lower bound solution, using the minimum force-deflection relationship of the bulbous bow.

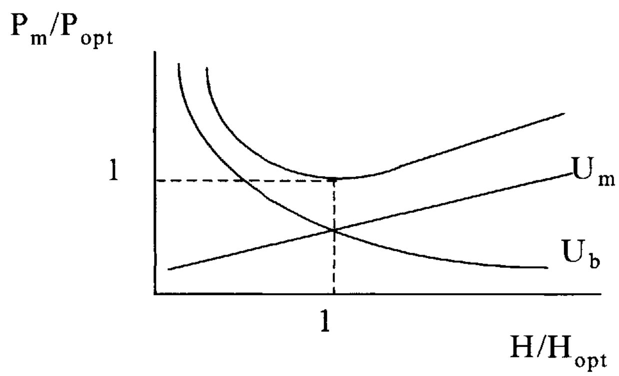

Figure 2-2: Optimal mean crushing force

Chapter 3

Equation Formulation

3.1 Kinetic Energy Calculations and External Dynamics

In order to calculate the amount of penetration that occurs during a collision, the amount of change in kinetic energy must be known. This change is calculated from momentum conservation of the two ships prior to collision and afterwards. This problem was first formulated by Minorsky (1952) and later by Simonsen (1999) for the program DAMAGE. It is assumed that the ships collide at right angles and that the struck ship has no initial velocity.

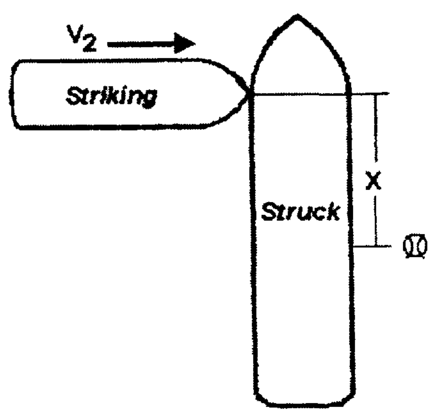

Figure 3-1: Ship-Ship Interaction

The parameters that are used in the equations are:

M_1	Mass of the struck ship
$\alpha_{y,1}$	Added mass coefficient for the sway motion of the struck ship
$r_{i,1}$	Radius of inertia of the struck ship (yaw motion)
$x_{1,LCG}$	Longitudinal location of the center of gravity of the struck ship
$x_{1,imp}$	Longitudinal Location of the impact on the struck ship
M_2	Mass of the striking ship
V_2	Velocity of the striking ship
$\alpha_{x,2}$	Added mass coefficient for the surge motion of the striking ship
$\alpha_{zz,1}$	Added mass coefficient for the yaw motion of the striking ship

First, calculate the apparent inertia quantities of the ships. They are

$$M_{2,x} = M_2(1 + \alpha_{x,2}) \tag{3.1}$$

$$M_{1,y} = M_1(1 + \alpha_{y,1}) \tag{3.2}$$

The apparent moment of inertia for the stuck ship around its center of gravity is

$$I_{zz} = M_1 r_{i,1}^2 (1 + \alpha_{zz,1}) - M_{1,y} x_{1,LCG}^2 \tag{3.3}$$

The longitudinal distance from the point of impact to the center of gravity of the struck ship is

$$x_1 = x_{1,imp} - x_{1,LCG} \tag{3.4}$$

From conservation of momentum, the velocities after impact of the striking ship, struck ship at the center of gravity, and the yaw rate of the struck ship are

$$\begin{aligned} V_2^a &= V_2 \frac{1 + M_{1,y}x_1^2/I_{zz}}{(1 + M_{1,y}x_1^2/I_{zz} + M_{1,y}/M_{2,x})} \\ V_1^a &= V_2 \frac{1}{(1 + M_{1,y}x_1^2/I_{zz} + M_{1,y}/M_{2,x})} \\ \omega_1^a &= V_2 \frac{M_{1,y}x_1^2/I_{zz}}{(1 + M_{1,y}x_1^2/I_{zz} + M_{1,y}/M_{2,x})} \end{aligned} \tag{3.5}$$

The sway velocity of the struck ship at the amidships section is

$$V_{2,MS}^a = V_2^a - \omega_1^a x_{1,LCG} \tag{3.6}$$

The total kinetic energy after impact is then

$$E_{kin,coup}^a = \frac{1}{2}M_{2,x}(V_2^a)^2 + \frac{1}{2}M_{1,y}(V_1^a)^2 + \frac{1}{2}I_{1,zz}(\omega_1^a)^2 \tag{3.7}$$

If an uncoupled dynamic solution is desired, the struck ship is assumed to be stationary, so the kinetic energy after impact is zero. Then, to calculate the work done on the struck ship and striking ship, the kinetic energy absorbed is calculated using equation (3.8).

$$\Delta E_k = \frac{1}{2}M_{2,x}(V_2^a)^2 - E_{kin}^a \tag{3.8}$$

This kinetic energy is available for plastic deformation of striking and struck ships. It is assumed that the amount of kinetic energy is large enough to cause plastic deformation and there is no elastic reactions.

However, if a simple equation for the change in kinetic energy, relying on fewer parameters, is desired, the following method could be used. This relationship does not account for rotational energy developed from an impact away from the center of gravity of the stuck ship or the added mass terms of the two vessels. The momentum before the collision is

$$M_2 V_2 \tag{3.9}$$

and after the collision is

$$(M_1 + M_2)V \tag{3.10}$$

from which the new velocity of both ships stuck together is

$$V = V_2 \frac{M_2}{M_1 + M_2} \tag{3.11}$$

Thus, the change in kinetic energy is

$$\Delta E_k = \frac{1}{2}M_2V_2^2 - \frac{1}{2}\frac{(M_2V_2)^2}{M_1+M_2} \tag{3.12}$$

$$\Delta E_k = \frac{1}{2}V_2^2 M_2 \frac{M_1}{M_1+M_2} \tag{3.13}$$

3.2 Applications of the Principle of Virtual Velocity

In order to compare the work done on the ships with the final damaged displacement, a relationship between force and displacement must be established. The principle of virtual work

$$P\dot{\delta} = \dot{U}_b + \dot{U}_m \tag{3.14}$$

where the external force-displacement of the structure is equal to the work done in bending and stretching, U_b and U_m. Because the crushing force acts over a time t_1 and the corresponding folding wavelength $2H$, the equation (3.14) can be integrated in time from 0 to t_1 (or from 0 to $2H$).

$$\int_0^{t_1} P\dot{\delta}dt = \int_0^{t_1}(\dot{U}_b + \dot{U}_m)dt \tag{3.15}$$

Transforming the left hand side

$$\int_0^{t_1} P\dot{\delta}dt = \int_0^{2H} P(\delta)d\delta = 2H\{\frac{1}{2H}\int_0^{2H} P(\delta)d\delta\} \tag{3.16}$$

From the mathematical definition

$$P_m = \frac{1}{2H}\int_0^{2H} P(\delta)d\delta \tag{3.17}$$

is the mean crushing force over the range (0,2H). Finally

$$\int_0^{t_1} P\dot{\delta}dt = P_m \cdot 2H \tag{3.18}$$

The mean crushing force (P_m) is

$$P_m \cdot 2H = U_b + U_m \tag{3.19}$$

where

$$U_b = \int \dot{U_b}\, dt \tag{3.20}$$

$$U_m = \int \dot{U_m}\, dt \tag{3.21}$$

3.3 Bow Damage Calculations with Intersecting Unit Method

3.3.1 Prior Research

Abramowicz (1994) developed an intersecting unit method for analyzing angle, "T", and cruciform elements. The crushing of a bulbous bow consist of several folding layers. As each section reaches its critical crushing force, the bow folds over this wavelength. The distance folded (H) is determined by minimizing the work. The actual folding of a bulbous bow against a tanker side wall can be seen in figure (3-2). As seen in the figure, there is no fracture. As a result of the speeds of tanker collisions and the transition temperatures of steel, the work of fracture is ignored.

Figure 3-2: Actual Bow Collision with a Tanker Side Wall

3.3.2 Force-Deflection Derivation

When the bulbous bow is deforming, two things are occurring. First, an annulus of the structure is bending from zero degrees until it is on average flattened at 90 degrees. Second, as the structure is bending, it is also stretching to keep the continuity between the deforming flanges. This can best be seen in figure (3-3). In order to calculate the force-displacement

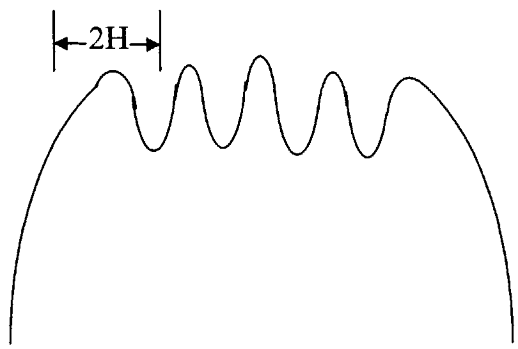

Figure 3-3: Intersecting Unit Deflection Method

relationship for the intersecting unit method, the plastic work is calculated as that for membrane stretching (U_m) and that for membrane bending (U_b), neglecting the interaction on the yield locus (Chakrabarty, 1987). These processes can be evaluated separately and then combined to determine the total force. The paper model in figure (3-4) shows the deflection process of two intersecting units. The open area represents the section of membrane stretching.

Now, to calculate the membrane work associated with the stretching occurring on the side of a plate, use the following derivation. In this formulation, it is assumed to only look at the deformation in stretching while ignoring the bending work. The membrane work is associated with the energy required to keep the angled section together. This angled section is divided into four segments. One segments looks like figure (3-5).

The membrane work rate is defined as

$$\dot{U}_m \stackrel{df}{=} \int_S N_{\alpha\beta}\, \dot{\varepsilon}_{\alpha\beta}\, dS \tag{3.22}$$

The following assumptions are made with respect to the membrane stretching energy. The

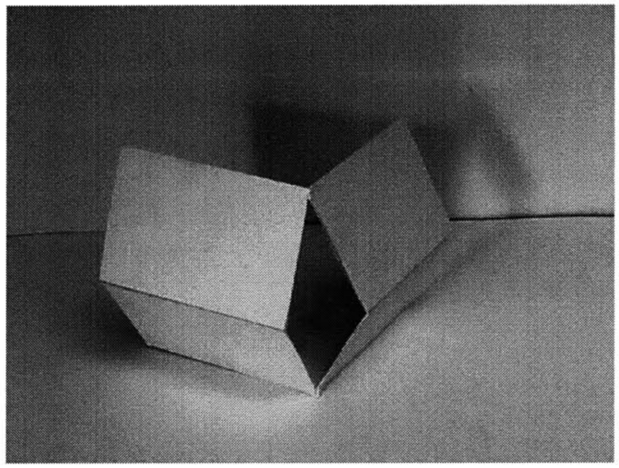

Figure 3-4: Paper Model of an Angle Flange

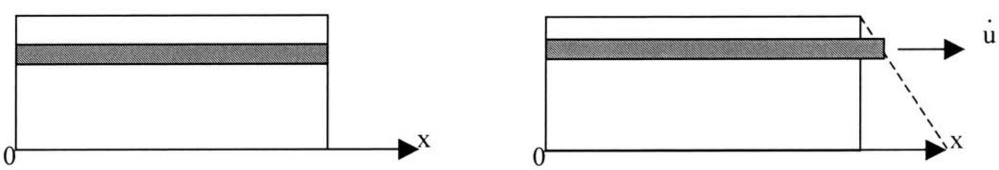

Figure 3-5: Membrane Resistance

membrane stretching only occurs in the axial direction. The membrane force (N) is a function of the materials flow stress (σ_o) and thickness. The only stretching that is occurring during this transformation is in the x-direction (figure (3-5)). The material in the y-direction is assumed to be inextensible.

$$N_{\alpha\beta} = t \cdot \sigma_{\alpha\beta} \tag{3.23}$$

$$N_{\alpha\beta} = \begin{bmatrix} N_o & 0 \\ 0 & 0 \end{bmatrix} \tag{3.24}$$

$$\dot{\varepsilon_{\alpha\beta}} = \begin{bmatrix} \dot{\varepsilon} & 0 \\ 0 & 0 \end{bmatrix} \tag{3.25}$$

$$\dot{\varepsilon}_{xx} = \frac{d\,\dot{u}}{dx},\ \mathrm{N}_o = \sigma_o t \tag{3.26}$$

This stretching occurs over the entire flange. To evaluate one flange, the energy for stretching is determined by integrating over the change in lengths.

$$\dot{U}_m = 2\int_0^H \int_0^b N_o \frac{d\,\dot{u}}{dx} dxdy = 2\int_0^H N_o\, \dot{u}\,(y)dy \tag{3.27}$$

$$\text{where } \dot{u}\,(y) = \dot{u}\,(x=b,y) - \dot{u}\,(x=0,y) \tag{3.28}$$

Integrating equation (3.27) in time one gets

$$U_m = 2N_o \int_0^H u(y)dy \tag{3.29}$$

Using figure (3-6), the stretching function is expressed as

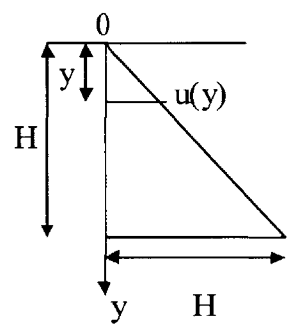

Figure 3-6: Membrane Stretching Function

$$u(y) = \frac{y}{H} H = y \tag{3.30}$$

And finally, the membrane stretching work for one flange is

$$U_m = 2N_o \int_0^H ydy = N_o H^2 \tag{3.31}$$

The next step is to calculate the work associated with bending one flange. The bending work is summed up for each of angled elements. The work required to bend this flange is a

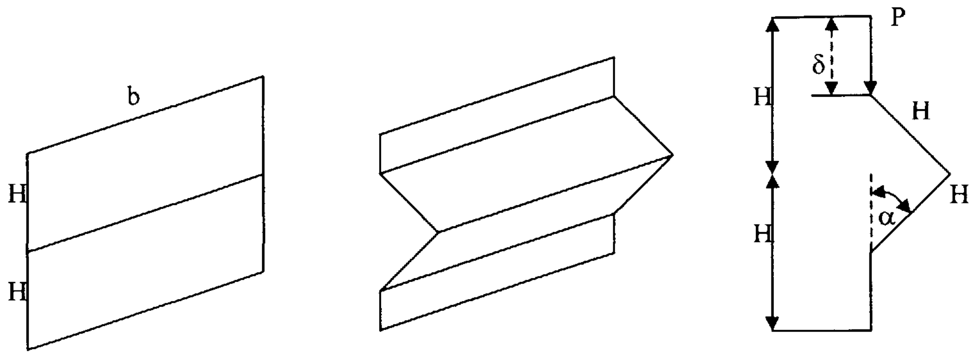

Figure 3-7: Bending Resistance

function of the material's plastic bending moment (M_o), angle bent (θ) and width bent (b)

$$\dot{U_b}=\sum M_o\,\dot{\theta_i}\,b_i = 4M_o\,\dot{\theta}\,b \tag{3.32}$$

and the fully plastic bending moment is

$$M_0 = \frac{\sigma_o t^2}{4} \tag{3.33}$$

To determine the bending work for the flange folding, equation (3.32) is integrated in time.

$$U_b = \int_0^{t_1} \dot{U_b}\,dt \tag{3.34}$$

Assuming a quasi-static response, equation (3.34) becomes

$$U_b = 4M_o b \int_0^{\frac{\pi}{2}} d\theta \tag{3.35}$$

$$\begin{aligned} U_b &= 4M_o b\frac{\pi}{2} && (3.36)\\ U_b &= 2\pi M_o b && (3.37) \end{aligned}$$

Using the principle of virtual velocities and equation (3.19), the mean crushing force for a single flange is

$$P_m \cdot 2H = 2M_o\frac{H^2}{t} + 2\pi M_o b \tag{3.38}$$

$$\frac{P_m}{M_o} = \frac{H}{t} + \pi\frac{b}{H} \tag{3.39}$$

As seen in figure(3-8), the model assumes that each fold collapses on itself. However, actual deformation is coil-like. Since the deflection is not 100 percent, there is less crushing force needed to deflect the structure. From experiments, Wierzbicki (1993) showed the mean crushing force is decreased by 25 percent to represent actual material behavior. The folding distance

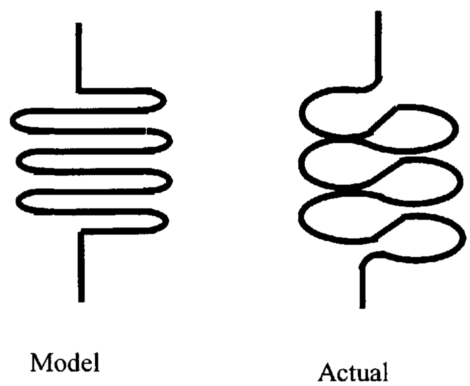

Figure 3-8: Effective Crush Distance

(H) for each incremental load (P) is then found from equation (3.39). If a whole structure is desired, the bending and stretching work or each element are summed together.

$$P_m = M_o(\sum\frac{H}{t} + \pi\sum\frac{b}{H}) \tag{3.40}$$

In order to do this summing, the structure is cut into conceptual pieces forming either angles, "T's", or cruciforms. For the bulbous bow, the structure might be cut like figure (3-9). The plate stiffeners are smeared into the outer thickness of the plate while the longitudinal girders are considered as angled flanges connecting to the outside plate. The thickness smearing is accomplished using equation (3.64).

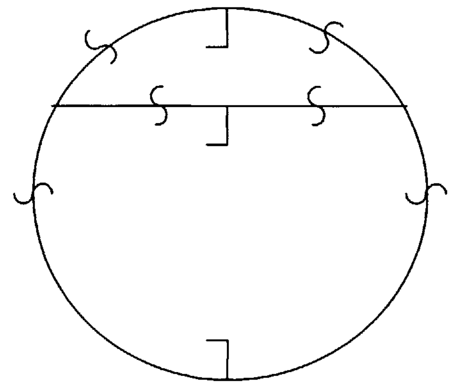

Figure 3-9: Example of Bow Slices

3.4 Bow Damage Calculations with Smearing Method

3.4.1 Prior Research

Several articles have been written about the study of energy absorbing characteristics of thin walled axi-symmetrical and hemispherical structures. The damage to the bow can be modelled as large deformations of rigid-plastic, spherical shells being compressed between rigid plates. Updike (1972) developed a sequence of limit loads and a force-displacement relationship for a simple geometry and yield condition. This relationship was valid for displacements of a few shell thicknesses to 1/10 the shell radius. Wierzbicki (1982) used similar geometries and rigid-plastic deformation to develop a force-displacement relationship based on the bending and membrane energy for plate deflections up to half the shell radius. In order to develop a simple, analytical method for evaluating the crushing force and work on a the bulbous bow, these two methods will be utilized.

3.4.2 Force-Deflection Derivation

The bulbous bow will be modeled as a spherical shell of radius R. The shell will be loaded by a moving rigid plate with the base of the shell fixed to a rigid plate. The shell will form a dimple which is an inversion of the original bulbous bow curvature. In this deflected state, the shell surface remains rigid while the point of inversion creates two circumferential plastic hinges. The curvature of the plastic hinge increases outward as the plate continues to deflect.

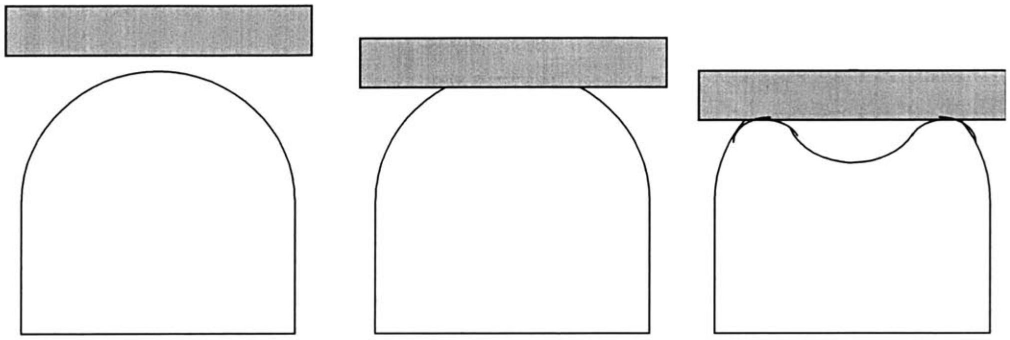

Figure 3-10: Bow Deflection Model

The derivation of this model was done by Lenhardt (1998) for a composite structure. Since the bulbous bow has an identical strength in both compression and tension, the equations are modified slightly.

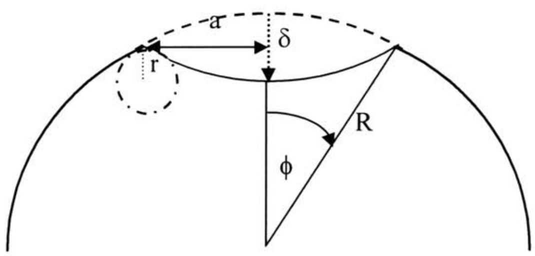

Figure 3-11: Geometric Bow Parameters

First, the radius to the center of the toroidal surface,and the displacement of the moving rigid plate δ can be expressed in terms of the shell radius R and the deformation angle ϕ as

$$a = R\sin\phi \tag{3.41}$$

$$\delta = R(1-\cos\phi) \tag{3.42}$$

Through differentiating equations (3.41) and (3.42), the radial and axial velocities of the toroidal section and rigid plate are

$$\dot{a} = \frac{da}{dt} = R\cos\phi\frac{d\phi}{dt} \tag{3.43}$$

$$\dot{\delta} = \frac{d\delta}{dt} = R\sin\phi\frac{d\phi}{dt} \tag{3.44}$$

By approximating cos ϕ with the first two terms of its Taylor expansion and rearranging equation (3.42), an equation relating the deformation angle to the rigid plate displacement can be deduced.

$$\phi = \sqrt{\frac{2\delta}{R}} \tag{3.45}$$

The toroidal surfaced area A and the average plastic hinge length L can be calculated from

$$L = 2\pi a \tag{3.46}$$

$$A = 4\pi r a\phi \tag{3.47}$$

The radial and rotational velocities of the plastic hinge can be taken as

$$V = R\frac{d\phi}{dt} \tag{3.48}$$

$$\frac{d\theta}{dt} = \frac{V}{r} = \frac{R}{r}\frac{d\phi}{dt} \tag{3.49}$$

The expressions for the membrane force N and fully plastic bending moment M are as follows

$$N = \sigma_o t \tag{3.50}$$

$$M = \frac{\sigma_o t^2}{4} \tag{3.51}$$

The flow stress σ_o is the average stress related between the yield and tensile strength of a material. The principle of virtual velocities will be used to relate the rate at which work is being done on the system with the rate at which it is being done on elements within the system. Starting with the rate of energy being introduced, the left hand side of equation (3.14) can be written as

$$P\frac{d\delta}{dt} = PR\sin\phi\frac{d\phi}{dt} \tag{3.52}$$

The rate of energy being absorbed through bending will be absorbed along two locations of the plastic hinge. This can be quantified as follows

$$\frac{dU_b}{dt} = \sum M_i \frac{d\theta}{dt} L_i \tag{3.53}$$

Equation (3.53) is further simplified with the following assumptions

· the magnitudes of the bending moment and the rate of angular rotation along two plastic hinges are equal.

· the total length of the hinge $\sum L_i$ as $2L$.

$$\frac{dU_b}{dt} = 4\pi R^2 \sin\phi \frac{M}{r} \frac{d\phi}{dt} \tag{3.54}$$

The rate of membrane energy dissipated on the plastically deforming zone can be written as

$$\frac{dU_m}{dt} = NA\frac{d\varepsilon}{dt} \tag{3.55}$$

where $\frac{d\varepsilon}{dt}$ is the instantaneous rate of strain experienced by the plastic hinge. As a simplification, the average strain rate, $\frac{d\varepsilon_{av}}{dt}$, will be used. The average strain rate is

$$\frac{d\varepsilon_{av}}{dt} = \phi\frac{d\phi}{dt} \tag{3.56}$$

Thus, equation (3.55) becomes

$$\frac{dU_m}{dt} = 4\pi r N R \sin\phi \phi^2 \frac{d\phi}{dt} \tag{3.57}$$

New expressions have been formulated for equation (3.14). By substituting the new equations (3.52),(3.54), and (3.57) the instantaneous crushing force (P) is as follows

$$P = 4\pi(\frac{RM + Nr^2\phi^2}{r}) \tag{3.58}$$

The mean crushing force ($\bar{P}$) can be calculated and minimized with respect to the toroidal radius to obtain an optimal toroidal radius.

$$\bar{P} = \frac{P}{2\pi M} = 2(\frac{R}{r} + r\frac{N}{M}\phi^2) \tag{3.59}$$

$$\frac{1}{r_{opt}} = \phi\sqrt{\frac{N}{RM}} \tag{3.60}$$

Now, the dimensionless mean crushing force can be written in terms of displacement and shell thickness as

$$\bar{P}_{opt} = 8\sqrt{\frac{2\delta}{t}} \tag{3.61}$$

Finally, the optimal instantaneous crushing force becomes

$$P_{opt} = 4\pi\sqrt{2}\sigma_o t^{\frac{3}{2}}\delta^{\frac{1}{2}} \tag{3.62}$$

This solution is only valid with the optimal mean force is less than the mean force for a cylinder. Since the bulbous bow will incrementally deform in the method which requires the least amount of load, equation (3.62) needs to be compared with the results of a cylinder under compression. Wierzbicki (1993) developed a formula that relates the diameter D and thickness t of a cylinder to the optimal mean crushing force. The deflection model of this process can be seen in figure (3-12). The formula for the optimal mean crushing force of a cylinder is as

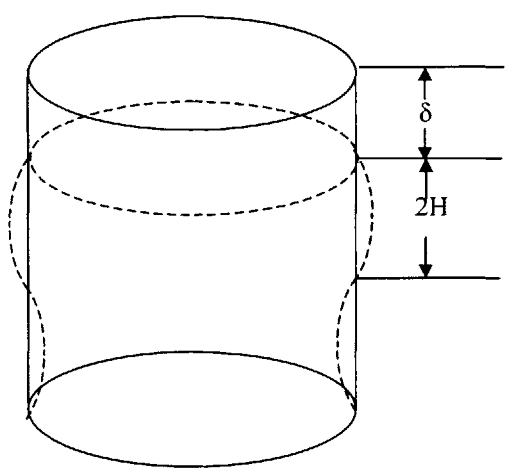

Figure 3-12: Deflection Model for Cylindrical Structure

follows

$$P_{opt} = 8.81\sigma_o D^{\frac{1}{2}} t^{\frac{3}{2}} \tag{3.63}$$

Now, with the optimal instantaneous force as a function of displacement and thickness, the overall thickness of the stiffened plate must be determined. This uses the thickness smearing method. The equivalent thickness is

$$t_{eq} = t_{plate} + K_l \frac{\sum t_i b_i}{b_{plate}} \tag{3.64}$$

where t_{plate} is the thickness of the outer plating, t_i is the thickness of the inner plate, stiffeners, and girders, b_i is the length of the stiffeners,b_{plate} is the outer plating length, and K_l is the coefficient accounting for the influence of welding methods and stiffener buckling, which normally is taken to be less than 1.0.

Chapter 4

Analysis

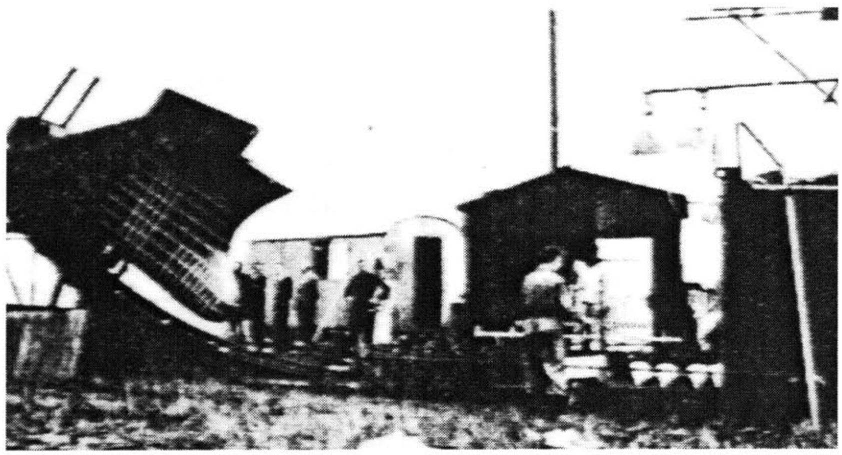

Figure 4-1: Actual Bulbous Bow Crushing into Tanker Side Wall

4.1 Kinetic Energy and Work Comparison

Knowing the relationship of the bow calculations, the next step is to relate the energy calculations for crushing the bow to the kinetic energy lost in the collision. This is done by integrating the mean crushing force of the ship over the distance deflected until it equals the kinetic energy loss. This is done in a step method and each step is assumed to be quasi-static. Since the velocities of the ships are small, inertia effects are neglected. Thus equation (4.1) provides the amount of bow and side deflection that each ship will endure.

$$\int_0^{\delta} P(\delta) d\delta = \Delta E_k \tag{4.1}$$

4.2 Force-Deflection Evaluation

Pederson (1992) used various bows to calculate the force-deflection relationship for pier and bridge studies. In these calculations, the bow was evaluated using two different equations that did not optimize the folding distance. Pederson generated five bows. In order to compare the intersecting unit method and thickness smearing method to other work, the Pederson bows are evaluated here. The first bow is a 150,000 dwt tanker with a bow profile as shown in appendix A.

The mean crushing force for a 150,000 dwt tanker was calculated using both the smearing formulation and intersecting method formulation. The bow was sliced at each folding distance H and the crushing force was calculated. Once slice looked like figure (4-2).

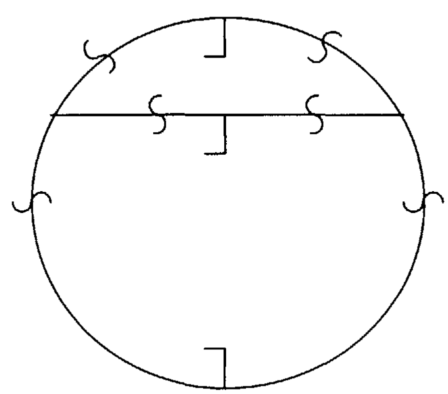

Figure 4-2: 150,000 dwt Tanker Bow Slices

The force displacement relationship for this bow was calculated using both methods. When the two methods were compared, the intersecting unit method was a lot lower than the thickness smearing method. This was due to the bow slices on the lower half. Since the width, b, of the circular section was not the same size at the longitudinal girder, the force was under estimated. Therefore, the lower section had to be cut again and converted to angle flanges in order to account for these widths. The modified bow is shown in figure (4-3).

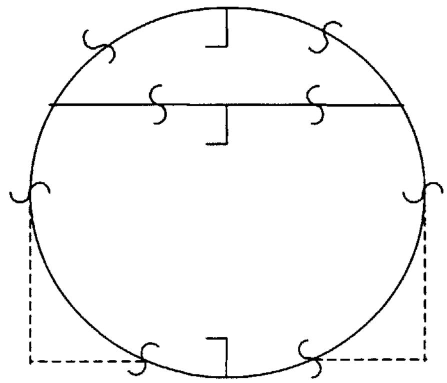

Figure 4-3: Adjusted Bow Slices

The force versus displacement results are plotted in figure (4-4). Both results give very similar force-deflection curves. The work necessary to deflect the bulbous bow section 5.5 m are nearly identical. These values also match those calculated by Pederson(1996) using Yang and Caldwell's formula.

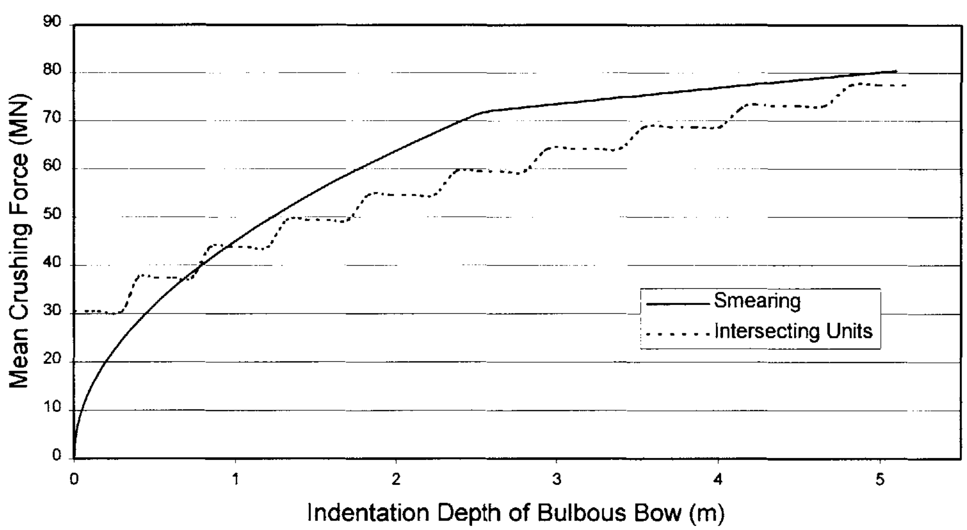

Figure 4-4: 150,000 dwt Tanker Force versus Deflection

The bulbous bow sections of a 40,000 dwt tanker was also calculated using these equations.

The 40,000 dwt tanker's bow profile is found in figure (B-1). These results are plotted in figure (4-5). The results also provided a simple closed form solution to the force deflection problem of the bulbous bow colliding with a rigid wall.

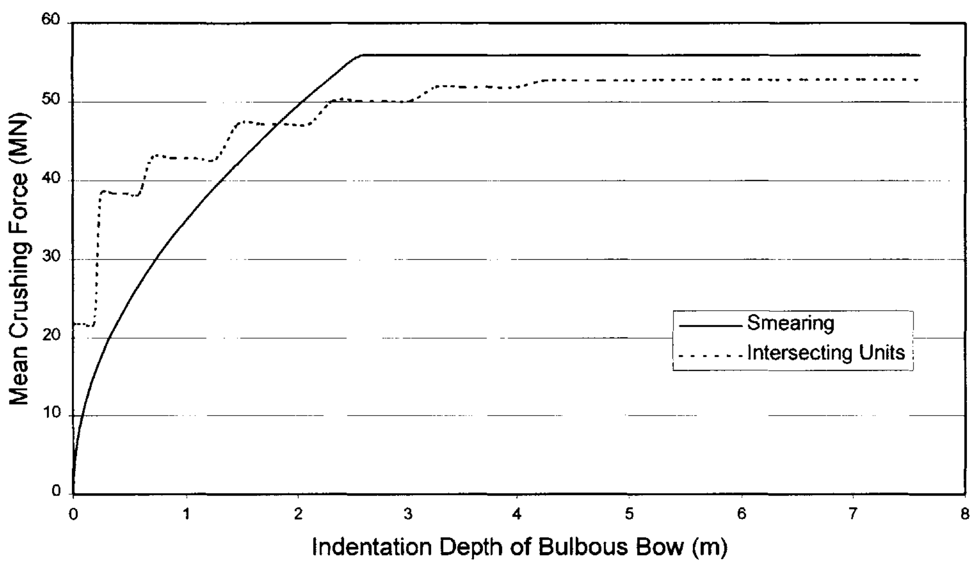

Figure 4-5: 40,000 dwt Tanker Force versus Deflection

In looking at the equation for thickness smearing, the intersecting unit method and thickness smearing method provided the same overall work done on the structures when coefficient for stiffener and welding method (K_l) was set to 0.73. This K_l factors accounts for eccentricities in the structures performance from weld failure and structural alignment. By adding this factor, it reduces the overall strength of the bow. In both the 40,000 and 150,000 dwt tankers, the results showed similar force-deflection results. Since both results assume that the bow will deflect with the lowest load, the results rely on the bulbous bow deforming before any after portion of the bow. As seen in the model tests (figure (3-2)), this correlates with the models.

Chapter 5

Conclusion and Recommendations

5.1 Collision with a Crushable Bow

The equation formulation provides two different methods for determining the closed form solution for the force-deflection relationship. Both methods rely on little computational requirements but provide a lower bound solution for the mean crushing force. The intersecting unit method allows for a more optimal bow design. This method allows the deck plating and girder locations to be adjusted to find the optimal bow design for a collision.

5.2 Recommendations

With these equations, the computer program DAMAGE can now be updated with a crushable bow. Kim (1999) produced the equation formulation for the raked portion of the bow. These equations can then be modified after testing with real collision scenarios. From the combination of the crushable bow and side equations, a more accurate force-deflection curve is generated for a given collision scenario. With this more accurate solution, the side walls and bows of tankers can be optimally designed.

The results of the crushable bow only contain mathematical comparisons. To improve on the overall formulation, actual bow tests should be compared. Woisin (1972) completed several of these tests. In addition, some finite element method validation would also improve the solution. In order to improve the computer program DAMAGE, actual collisions should be used.

The interface between two vessels is assumed to be a right angle collision. Further research into the bow-side wall interface will allow for an even more accurate solution. This is due to tearing occurring from the friction between two vessels. Neither the thickness smearing nor the intersecting unit method account for this type of scenario. Also, both methods assume that the bow is colliding with a rigid wall, where in actuality the wall is deforming during the collision. The overall solution does provide an easy method to calculate the force-deflection relationship between a bulbous bow and a rigid wall.

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Appendix A

150,000 dwt Tanker Bow

The following are the dimensions of the 150,000 dwt tanker bow

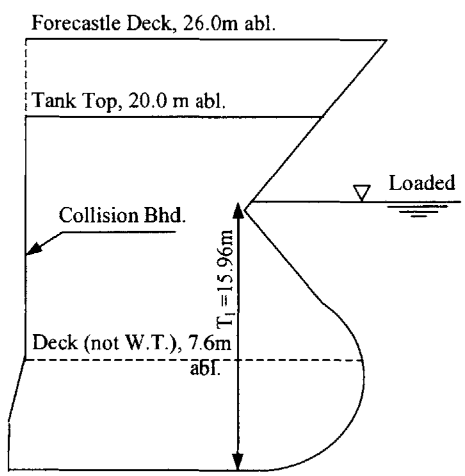

Figure A-1: 150,000 dwt Tanker Bow

length between perpendiculars (also applied as rule length):	274.00 m
breadth molded:	47.00 m
depth molded:	21.60 m
depth to forecastle deck:	26.00 mm
maximum draft:	15.96 m
displacement loaded:	174,850 tonnes
maximum service speed:	7.7 m/s (15 kts)

The bulbous bow is stiffened longitudinally . The transverse frames supporting the longitudinals have a spacing of 3.2 m. The structural data are as follows:

Material	
yield stress for plates and stiffeners (σ_y)	315.0 MPa
ratio between ultimate and yield stress (σ_u/σ_y)	1.6
Bottom:	
plate thickness	18.0 mm
longitudinals, spacing 0.8 m	L450 x 150 x 12/16
CL-girder	L2500 x 400 x 15/25
Side shell:	
plate thickness, side up to 8.1 m abl:	18.0 mm
plate thickness, side shell between 8.1 and 17.0 m abl:	33.0 mm
plate thickness, side shell between 17.0 and 26.0 m abl:	16.0 mm
longitudinals between 1.2 and 6.8 m abl, spacing 0.8 m:	L450 x 150 x 12/16
longitudinals between 8.4 and 16.2 m abl, spacing 0.6m:	L400 x 100 x 19/19
longitudinals between 17.4 and 24.8 m abl, spacing 0.8m:	L350 x 100 x 12/17
Deck (not watertight) 7.7 m abl:	
plate thickness:	13.0 mm
longitudinals, spacing 0.8 m:	L250 x 90 x 12/16
CL-girder	L1400 x 250 x 12/25

Appendix B

40,000 dwt Tanker Bow

The following are the dimensions of the 40,000 dwt tanker bow:

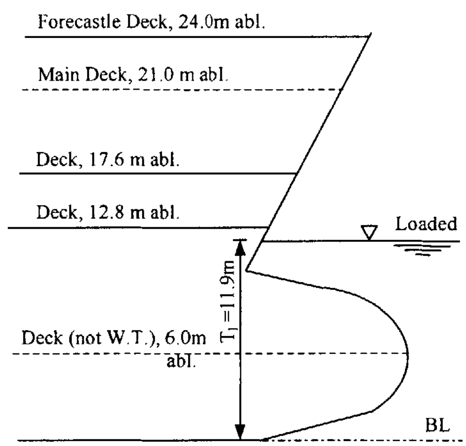

Figure B-1: 40,000 dwt Tanker Bow

length between perpendiculars (also applied as rule length):	211.50 m
breadth molded:	32.20 m
depth molded:	21.00 m
depth to forecastle deck:	24.00 mm
maximum draft:	11.90 m
displacement loaded:	54,000 tonnes
maximum service speed:	11.3 m/s

The bulbous bow is stiffened longitudinally . The transverse frames supporting the longitudinals have a spacing of 2.4 m. The structural data are as follows:

Material	
yield stress for plates and stiffeners (σ_y)	235.0 MPa
ratio between ultimate and yield stress (σ_u/σ_y)	1.9
Bottom:	
plate thickness	19.0 mm
longitudinals, spacing 0.8 m	L250 x 90 x 12/16
CL-girder	L1900 x 250 x 15/25
Side shell:	
plate thickness, side up to 6.1 m abl:	17.0 mm
plate thickness, side shell between 6.1 and 12.3 m abl:	35.0 mm
plate thickness, side shell between 12.3 and 21.0 m abl:	14.0 mm
longitudinals below 5.2 m and above 13.2 m abl, spacing 0.8 m:	L250 x 90 x 10/15
longitudinals between 6.6 and 12.0 m abl, spacing 0.6m:	L250 x 90 x 12/16
Deck (not watertight) 6.0 m abl:	
plate thickness:	11.0 mm
longitudinals, spacing 0.8 m:	FL150 x 12