Analysis of Accidental Iceberg Impacts with Membrane LNG Carriers

by

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Abstract

The topic of this report is accidental iceberg collision with the bow shoulder area of a 150000ton membrane-type LNG carrier. It is expected that the traffic of LNG-carriers will increase in the arctic areas in the years to come, and ice collisions are considered to be a real hazard.

A finite element model of the ship side have been developed, where the area in question is the foremost tank of the ship. The collision point is assumed to be in the forward part of the tank, close to the collision bulkhead. However in the FE-analyses the collision point is moved to the centre of the tank, to avoid interference from the boundary conditions.

A maximum size criterion for icebergs is used, where it is assumed that all icebergs with less than 2 meter sailing height are hazardous. This means that the ship is assumed to hit these iceberg at its cruising speed of 19.5 knots. 2 meter sail height corresponds to 5 meter radius for a spherical iceberg.

The steel material used is based on power-law formulations. This allows the material model to take into account increase in stress due to reduction in cross-section area.

The ice material uses the elliptical Tsai-Wu yield surface. Elements are deleted when they violate the fracture criterion.

The FE-analyses are carried out to find the critical energy levels. The analyses were done with spherically shaped icebergs, with radii of 1,2,5 and 10 meter. Failure of the ship structure is based on preventing the gas from leaking out due to fracture of the membranes in the cargo containment system. The criterion which is used is 700mm deflection of the inner hull in the middle of the tank. The internal energy for the ship structure when the failure criterion is reached is found to be approximately 70MJ.

The actual collision energy levels are found using a 3D external mechanics model. The results indicate that the critical energy levels are not reached. The levels for 1 and 2 meter radius icebergs are negligible. For the 5 meter radius the level is 8MJ, or 11% of the critical value. For the 10 meter iceberg the margin is quite small, with 57MJ or 81% of critical value. The critical size is found to be 11 meter radius, although it is very conservative since icebergs larger than 5 meter radius will probably be detected before impact.

In previous studies of ship-iceberg impacts, inner hull accelerations are found to be very high, up to 2000g. There has also been concern about accelerations causing problems far from the impact zone. The accelerations have been investigated on several location on the inner hull, and none show higher levels than 200g. Displacements corresponding to these accelerations show no rapid jumps or fluctuations in the curves. This should indicate that the accelerations cause no serious problems for the membrane and are more of a numerical issue than a real life problem.

Scope of work

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Analysis of accidental iceberg impacts with membrane tank LNG carriers

The expected increase of exploitation of gas fields in Northern regions will precipitate the development of arctic LNG shipping. LNG ships carry huge amount of energy and it is vital that these ships possess adequate resistance to ice actions, so as to keep the risk of catastrophic events sufficiently low. A potentially severe risk is associated collision small icebergs (bergy bits/growlers). Large icebergs are very likely to be observed by radars installed aboard the ship or by airborne radars, but smaller icebergs may avoid detection. This implies that rare (accidental) events cannot be disregarded and must be considered in the design.

For accidental iceberg collisions use of ship classification design rules may yield overly conservative design. The rules are typically based upon elastic or plastic bending failure modes of stiffeners and plates implying small deformations. For accidental/abnormal iceberg impacts some degree of damage to the structure (side shell/frames) may be accepted, but the integrity of the cargo tank should impaired, causing gas leakage to the environment and possible ignition. In the Accidental Limit State the resistance may be assessed by non-linear methods of analysis; the structure may undergo yielding, buckling and large permanent deformations on member and sub-structure level. This can only be assessed accurately if both the ship and the ice are modeled, and the interaction between the two structures is accounted for.

When dealing with numerical simulations of ship-ice interaction, there is always a search for the best suitable models for such investigations. Generally, it could be categorized as Discrete Element modelling (DE), e.g. Matlock (1971), Matlock (1969), Daley (1990), Sayed (1997) and Finite Element modelling (FE), e.g. Varsta (1983), Xiao (1991), Derradji-Aouat (2005), Gagnon (2007), Grtner (2008). Nevertheless, it is not possible to find a conventional method to simulate the ice behaviour due to complicated ice properties, which mainly depend on temperature, salinity and strain rate. A PhD-student at department of marine technology Zhenhui Liu - approaches the ship-iceberg collision problem numerically by using the Tsai-Wu based ice material model by an Explicit FE code. In the study, the Tsai-Wu material model turned out to be a promising candidate for calculating the ice impact loads.

Recently the Gas Transport /Technigas (GTT) cargo containment system (CCS) the membrane tank - has become popular. The membrane tank consists of a cryogenic liner directly supported by the ship inner hull. The primary and secondary insulation system consists of 0.7 mm nickelsteel alloy carried by prefabricated plywood boxes filled with expanded perlite.

It has been maintained that the support of membrane tank directly on the inner hull as well as the smaller space between the cargo tank and the side shell make these concepts more vulnerable to iceberg collisions than e.g carriers with spherical tanks. Further, in some Korean numerical studies very large accelerations have been reported (up to 2000g). The findings are not discussed in detail, but the magnitude of the accelerations have caused some concern that iceberg collisions may cause failure in the cargo containment system far away from the contact area, in addition the local hull damage. If this should be correct, it represents a significant drawback of membrane tanks.

The purpose of the work is to study the behavior and resistance of Arctic LNG carriers with the membrane tank system subjected to accidental impacts. The work is proposed carried out in the following steps:

1) Determine relevant impact scenarios: among others impact geometry, speed of vessel and iceberg, size and shape of iceberg. Analyze the external mechanics by means of the 3D method developed by Zhenhui Liu. For given ships/platform and ice berg shape parameters establish the fraction of kinetic energy that has to be dissipated as train energy for a range of ice and structure sizes. .

2) Establish a finite element model of an LNG fore ship side structure and the iceberg for the selected impact scenario(s). The finite element model for the ship and the ice shall be sufficiently fine to capture the governing deformation mechanisms of the ice, but still meet requirements with respect to acceptable CPU consumption. The kinematical/boundary conditions adopted for the study shall be discussed with respect to physical relevance

3) Perform integrated analysis of internal mechanics.. The damage and energy dissipation in the ship and the iceberg shall be documented. The pressure distribution and energy dissipation during crushing shall be identified and if available be compared with relevant information (tests, measure-

ments etc.).

4) The displacements and acceleration levels at critical locations shall be discussed and checked against acceptance criteria established in pt 1. Perform sensitivity analysis where iceberg material parameters are varied. Estimate the critical collision energy for exceeding acceptance criteria. If needed suggests strengthening measures of the ship side structure.

5) Conclusions and recommendations for further work

Literature studies of specific topics relevant to the thesis work may be included.

The work scope may prove to be larger than initially anticipated. Subject to approval from the supervisors, topics may be deleted from the list above or reduced in extent.

In the thesis the candidate shall present his personal contribution to the resolution of problems within the scope of the thesis work.

Theories and conclusions should be based on mathematical derivations and/or logic reasoning identifying the various steps in the deduction.

The candidate should utilise the existing possibilities for obtaining relevant literature.

Thesis format
The thesis should be organised in a rational manner to give a clear exposition of results, assessments, and conclusions. The text should be brief and to the point, with a clear language. Telegraphic language should be avoided.

The thesis shall contain the following elements: A text defining the scope, preface, list of contents, summary, main body of thesis, conclusions with recommendations for further work, list of symbols and acronyms, references and (optional) appendices. All figures, tables and equations shall be numerated.

The supervisors may require that the candidate, in an early stage of the work, presents a written plan for the completion of the work. The plan should include a budget for the use of computer and laboratory resources which will be charged to the department. Overruns shall be reported to the supervisors.

The original contribution of the candidate and material taken from other sources shall be clearly defined. Work from other sources shall be properly referenced using an acknowledged referencing system.

The report shall be submitted in two copies:

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Drawings and/or computer prints which cannot be bound should be organised in a separate folder. The report shall also be submitted in pdf format along with essential input files for computer analysis, spreadsheets, Matlab files etc in digital format.

Ownership
NTNU has according to the present rules the ownership of the thesis. Any use of the thesis has to be approved by NTNU (or external partner when this applies). The department has the right to use the thesis as if the work was carried out by a NTNU employee, if nothing else has been agreed in advance.

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Deadline: June 11, 2011

Trondheim, January 17, 2011

Jørgen Amdahl

Preface

This report is the result of my master thesis in Marin Structural Engineering at NTNU.

The work on the thesis has been interesting but also challenging. The FE-modelling has been very long and somewhat tedious at times. Due to this time consuming process, no actual analyses were run until well over halfway in the semester. Looking back, further simplifications should perhaps have been made to reduce the modelling time.

I would like to thank Professor Jørgen Amdahl, who has been an enthusiastic and helpful supervisor during the course of this thesis. I would also like to thank Ph.D candidate Zhenhui Liu who helped me with modelling and analysis setup. Liu continued to be of assistance also after he left the Marin Technology Center midway in the semester for a job in the industry. In addition, I appreciate the assistance from Håvard Nyseth at DNV who helped with providing the drawings of the ship, and input in the beginning of the modelling process.

Trondheim, June 14 2011

Andreas Berling

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Nomenclature

α	Waterline angle
β'	Normal frame angle
$\ddot{s}_i$	Relative acceleration
ϵ	Strain
ϵ_{eq}	Equivalent strain
ϵ_{plat}	Plateau strain
$\hat{R}_{ii}(i = x, y, z)$	Radius of gyration
μ_n	Normal friction factor
μ_n	Tangential friction factor
ν	Poisson's ratio
$\overline{m}$	Equivalent mass
σ_m	Hydrostatic stress
σ_Y	(Initial) yield stress
σ_{eq}	Equivalent stress
ε^p_{eq}	Equivalent plastic strain
ε_f	Failure strain
dp	Impulse
dv_i	Incremental velocity
E_0	Maximum energy in plastic impact scenario
E_i	Energy dissipation
f	Force
m	Mass
v^0_1	Relative velocity after impact

v_i^t Relative velocity before impact

B Bulk modulus

D Accumulated damage

E Young's modulus

G Shear modulus

K Power law parameter

n Power law exponent

T Ratio between hydrostatic stress and equivalent stress

Acronyms

GTT Gaztransport-Technigaz

CCS Cargo Containment System

DNV Det Norske Veritas

LNG Liquified Natural Gas

LNGC Liquified Natural Gas Carrier

NTNU Norwegian University of Science and Technology

1 Introduction

In the recent years there has been a large increase in activity in the arctic area. Global warming is leading to larger ice-free areas and for longer parts of the year than only a few years ago. This makes it easier to reach the vast reserves of petroleum in the Arctic ocean. With the newly signed agreement between Russia and Norway that defines the border between the two countries in the Barents sea, even larger areas for possible production of oil and gas are opening up. This will in turn lead to an increase in traffic of LNG-carriers in the arctic area. These LNG carriers are loaded with thousands of tons of natural gas. In theory the energy stored on board a fully loaded LNG-carrier even surpasses some nuclear detonations. This indicates that the consequences of a large collision followed by gas leakage can be very serious. In addition, exposure to the super-cooled gas can cause problems for the structural integrity of the ship itself.

When sailing far to the north in the arctic ocean there is always a threat from drifting icebergs. Large icebergs will probably be detected well in time before impact to change course, while smaller iceberg can actually be more dangerous. If the height above the water line is small they can be very difficult to detect. This is particularly the case at night or in bad weather where radar is the most effective warning system. Still, with a low sailing height the icebergs can have a mass of hundreds or even several thousand tons. Being unable to detect the iceberg, the ship can collide without having the time to slow down or take evasive action.

Several reports have been written on iceberg collisions with membrane-type LNG carriers. The main concern from these reports are high levels of acceleration in the inner hull of the ship. In membrane-type cargo containment system the tanks are directly supported by the hull of the ship. This makes it probably more vulnerable to collisions than the competing cargo containment system, spherical tanks. Here, the tanks are more of independent and self-supporting structures inside the ship.

There have also been written a master thesis on ice impact with a membrane type LNG carrier. The thesis focuses on a mid-ship side impact, but does not consider bow collisions. The bow collision scenario is probably the most realistic and also most dangerous case for ship-iceberg collisions.

In this thesis analyses of the external and internal mechanics of ship-iceberg collisions will be carried out. The area on focus is the bow shoulder of a membrane type LNG carrier. The general focus will be on strain energy dissipation in the ship and iceberg, and accelerations in the inner hull. Internal mechanics will be investigated by integrated finite element analyses. At the end of the report results found by means of 3D external collision mechanics will be compared to the results from the FE-analyses.

Outline of thesis

The report is divided into the following chapters.

1. Introduction
2. Collision scenarios with discussions around point of impact, velocity of ship and iceberg, iceberg size and survival criterion of the ship.
3. Theory behind the ice and steel material models applied in the simulations.
4. Theoretical background for 3D external mechanics model.
5. Description of important software.
6. Description and discussions around the finite element modelling process of ship and iceberg.
7. Description, results and discussions of FE-analyses.
8. Results of external mechanics model applied to the ship-iceberg collisions.
9. Conclusion and discussion of results.
10. Recommendations for further work.

2 Collision scenarios

Point of impact

The worst place for an iceberg to hit the foreship of the LNGC is expected to be close to the forward collision bulkhead of the forward tank, tank 1. In this area the waterline angle is relatively large, hence the colliding body is not deflected off too easily and the normal forces on the ship can be large. Since the model is just a small part of the entire ship, it is important to define proper boundary conditions. If the collision took place very close to such boundaries, the results would not be realistic since most of the collision energy would be absorbed at the boundaries. The geometry in the middle of the tank is assumed to represent the actual collision area, since both frames and stiffeners are quite similar along the tank.

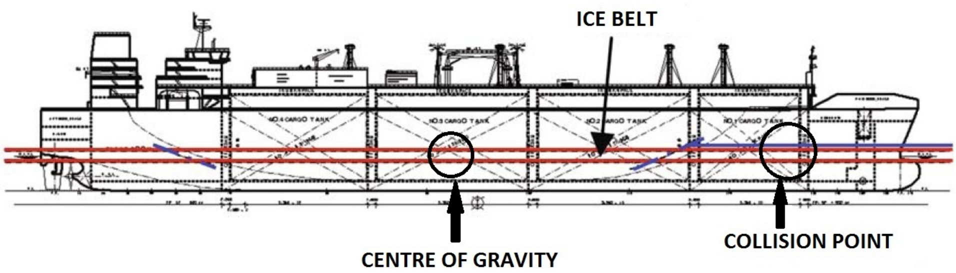

Figure 1: Collision point,[1]

Iceberg velocity

Velocity of ship and iceberg is important parameters in collision analyses. In [8], [9] and [1] a drift speed of either 2 or 1.3 knots is applied to the icebergs. Zhenhui Liu, [10], on the other hand assumes that the iceberg is stationary. This is not entirely realistic for most real life situations. But an important factor for iceberg motion at open sea is current. This current will also affect the ship to some extent in the same direction as the iceberg. This means that the relative velocity will be smaller than the actual current velocity. In addition, the velocity of the ship will in most cases be the governing velocity factor and the drift speed of the iceberg is less important. In this thesis the iceberg is assumed stationary in all analyses.

Ship velocity

According to several articles, such as [8] the cruising speed of a 150000ton LNGC is 19.5 knots or approximately 10m/s. In ice infested waters the cruising speed will of course be reduced, since collision with large icebergs can be very dangerous. However, this thesis focuses on smaller icebergs that may not be detected at all before impact, so called growlers and bergy bits. A common rule is that all icebergs with sail height less than 2 meters may not be detected before impact. Thus any icebergs below this level are considered hazardous. This indicates that a cruising speed of 19.5 knots should be realistic for accidental iceberg collision purposes.

Iceberg shape and size

The shape of icebergs depend on many parameters such as from where it originates, age in open water and the sea states it encounters. Therefore simplifications with respect to the shape must be made when modelling the iceberg. Regarding this, the recommendations by DNV,[6], is three different shaped iceberg. A sphere, a cone and a cube, where the dimensions are based on a maximum sail height of 2 meters. In several reports such as,[8],[1] and [11], the cubic shape of 20mx20mx20m is applied. This is the absolute worst case since the mass is very high (7200 tons), while the sail height is no more than 2 meters. Yet, a perfect cube with a totally flat impact surface will never be encountered in real life. According to both Håvard Nyseth at DNV, and Prof. Jørgen Amdahl a curved surface will simulate real life conditions better than a flat. The impact force will also be more concentrated than for a flat surface. Due to this and the fact that Zhenhui Liu has already developed FEM-models of icebergs at the shape of spheres, the sphere shape is chosen for all analyses in this thesis.

Ship survival criterion

The survival limit for a membrane-type LNG carrier, is in many ways governed by deformation and eventually rupture of the invar membranes that line the membrane tanks. Leakage of gas could have disastrous consequences. Invar is a nickel steel alloy with very low coefficient of thermal expansion. In [1], a risk analysis has been carried out to find an ultimate capacity criterion for these membrane and thus for the ship. The invar membrane has very good properties when it comes to deformation in elongation. The elongation limit is given from the manufacturer, GTT, as 40mm/m, which in theory could amount to more than 6000mm lateral deflection in the centre of a tank. This is not very realistic since the surrounding steel structure would collapse well before this limit is reached. In any case elongation is not suitable as a survival limit since the limiting conditions are stresses at welded joints and anchor points for the membrane. In the report three different types of conditions are assumed to be possible failure conditions:

- Tension failure of the membrane due to inner hull deformation.
- Failure of the membrane due to inner hull deformation, based on an actual LNGC grounding case.
- Failure of inner hull causing leakage of ballast water into the insulation space which again can damage the membranes.

The final criterion is based on a combination of the three conditions. 70cm of inner hull deflection in the middle of the tank is used as survival criterion.

3 Material Properties

3.1 Ice Properties relevant for FEM

Yield surface

Ice modelling for FEM simulations is not very well established. The crushing of ice is important when modelling for ship-iceberg impact simulations. There has been done simulations based on so-called crushable foam, but this model lacks a proper physical explanation. In addition ice cracks and damage are not considered.

Icberg ice can be assumed isotropic with a reasonable accuracy, and stress-strain relationships can be found from yield surface formulation and the selected flow rule. The surface is in a tri-axial state, since the particles in the contact area are confined from the surrounding particles. A widely used yield surface is the so-called Tsai-Wu surface, but which has not yet been implemented in ship-iceberg collision scenarios. In [2], an elliptical yield surface is assumed to describe the iceberg. The Tsai-Wu yield surface can be rewritten in the following way into an elliptical surface:

$$f(p, J_2) = J_2 - (a_0 + a_1 p + a_2 p^2) = 0 \tag{1}$$

$p = \frac{\sigma_{kk}}{3}$ is the hydrostatic pressure

Temperature profile

According to Løset(1993) there is a temperature gradient from the surface to the core of the iceberg. The change in strength due to this effect is:

$$f(p, J_2) = J_2 - (a_0(T) + a_1(T)p + a_2(T)p^2) = 0 \tag{2}$$

Strain rate effects

According to Gagnon and Gammon (1995) the strength of the iceberg increases with increasing strain rates. Later experiments show that the strength decreases again for very high strain rates. Numerical simulations have shown that ship-iceberg impacts involve high strain rates. There is little experimental data on strain rate, and the strain rate has not been incorporated into the current model. A yield envelope representative for the high strain rates is used. This should be a realistic model in high velocity impacts where the ice behaves in a brittle manner.

Failure criterion

The Tsai-Wu yield surface has been made as a subroutine in LS-DYNA. In Figure 2, the Tsai-Wu yield surface is plotted with different data sources and fitting methods in order to approach the experimental data sets.

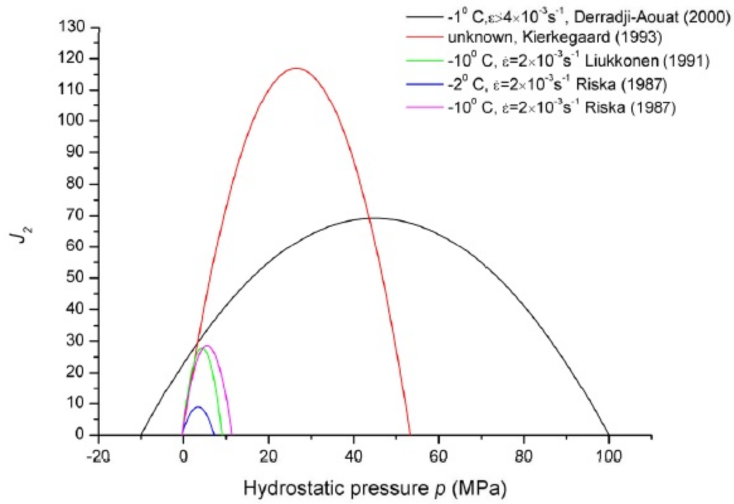

Figure 2: Tsai-Wu yield surface, [2]

An important point to consider is how long the stress trajectory travels along the yield surface before failure occurs. The following is an empirical failure criterion.

$$\varepsilon_{eq}^{p} = \sqrt{\frac{2}{3}\varepsilon_{ij}^{p}\varepsilon_{ij}^{p}} \tag{3}$$

$$\varepsilon_f = \varepsilon_0 + (\frac{p}{p_2} - 0.5)^2 \tag{4}$$

ε_{eq}^{p} is equivalent plastic strain and ε_f is the failure strain. The failure criterion is an empirical equation, based on effective plastic strain and hydrostatic pressure. Failure is simulated by using erosion, where elements violating the failure criterion are deleted. However, this method does not simulate brittle failure very well. Therefore the ice is assumed to be elastic-perfect plastic. A quasi-brittle material is proposed and used in the simulations.

The difference in strength between tensile and compression states is described by using a cut-off pressure. If $\epsilon_{eq}^{p} > \epsilon_f$ or the pressure is not greater than this cut-off pressure, erosion is activated.

The ice material model used by Zhenhui Liu in [2] has the following parameters:

Initial failure strain	0.01	[-]
Density	900	$[kg/m^3]$
Elastic modulus	9.5	[GPa]
Poisson ratio	0.3	[-]
Tensile cut-off pressure	-2	[MPa]
Strain rate	$> 10^{-3}$	$[s^{-1}]$

Table 1: Ice material properties

3.2 Steel material properties

The steel materials used in the simulations are all based on power-law formulations which represent the equivalent stress-strain relationship of the material. As opposed to the engineering stress-strain relationship, the power-law formulation follows the true stress-strain and takes into account the increase in stress due to reduction in cross section area.

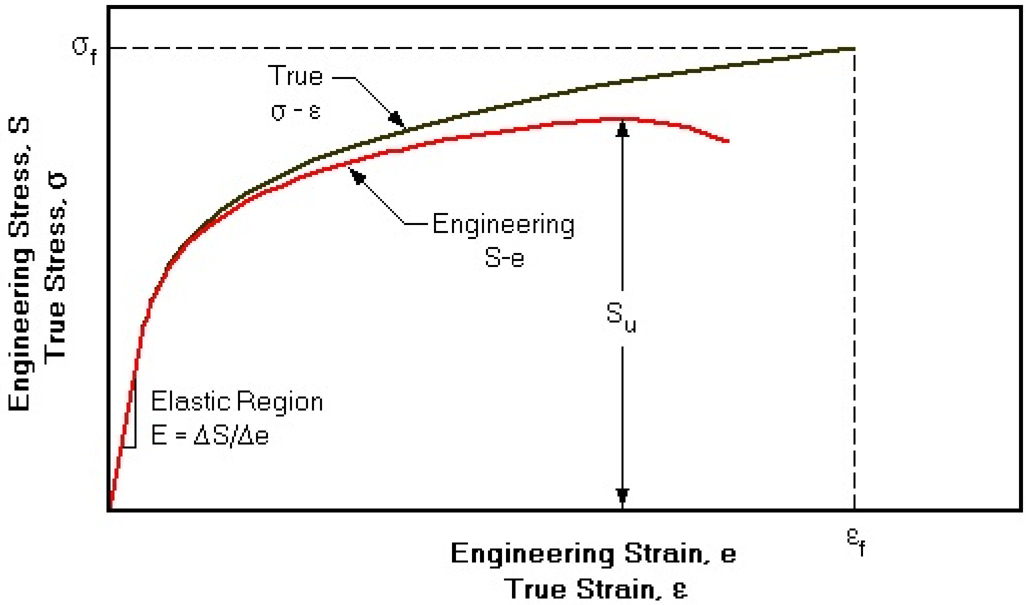

Figure 3: Engineering vs true stress-strain curve

The equivalent stress-strain relationship is defined in equation 5 and 6

$$\sigma_{eq} = \begin{cases} \sigma_Y & \text{if } \epsilon_{eq} \leq \epsilon_{plat} \\ K(\epsilon_{eq} + \epsilon_0)^n & \text{otherwise} \end{cases} \tag{5}$$

$$\epsilon_0 = \left(\frac{\sigma_Y}{K}\right) - \epsilon_{plat} \tag{6}$$

where $\epsilon_p lat$ is the equivalent plastic strain at the plateau exit, and σ_Y is the initial yield stress.

Specimen	Material type	σ_Y [MPa]	K [MPa]	n	ε_{plat}
Plate US, 1-FB and 2-FB	S235JR EN10025	285	740	0.24	—
Plate 1-HP	S235JR EN10025	340	750	0.20	—
Plate 2-HP	S235JR EN10025	260	640	0.22	0.003
Flat bar stiffeners	S235JR EN10025	340	760	0.225	0.015
Bulb stiffeners	S355NH EN10210	390	830	0.18	0.01
Frame	S355NH EN10210	390	830	0.18	0.01

Figure 4: Steel materials,[3]

The steel material is based on Alsos,[3]. After a discussion with Jørgen Amdahl the circled materials in Figure 4 were chosen. In Figure 5 and 6, equivalent stress is plotted with equivalent strain running from 0 to 1. The initial yield stress, σ_Y, the power-law parameters, K and n, and the plateau strain, ϵ_{plat} were changed using trial and error to find the appropriate values for the actual steel on the ship. The upper, blue curves are the original parameters from Figure 4, while the lower, red curves are the new material. An important point is that the yield stress given on ship drawings is more of a guaranteed level than what is actual on the ship. However to be on the safe side, the specified yield stress of 235 and 315 MPa respectively, is used all over. It is found that the yield stress has little influence on the shape of the curves, while the K and n factors have more influence. In table 2 the chosen values for the material model are given.

	Mild steel	**High strength steel**
$\rho[kg/m^3]$	7850	7890
E[GPa]	210	210
Poisson's ratio	0.3	0.3
$\sigma_Y[MPa]$	235	315
K[MPa]	700	770
n	0.24	0.18
ϵ_{plat}	-	0.005

Table 2: Steel material properties

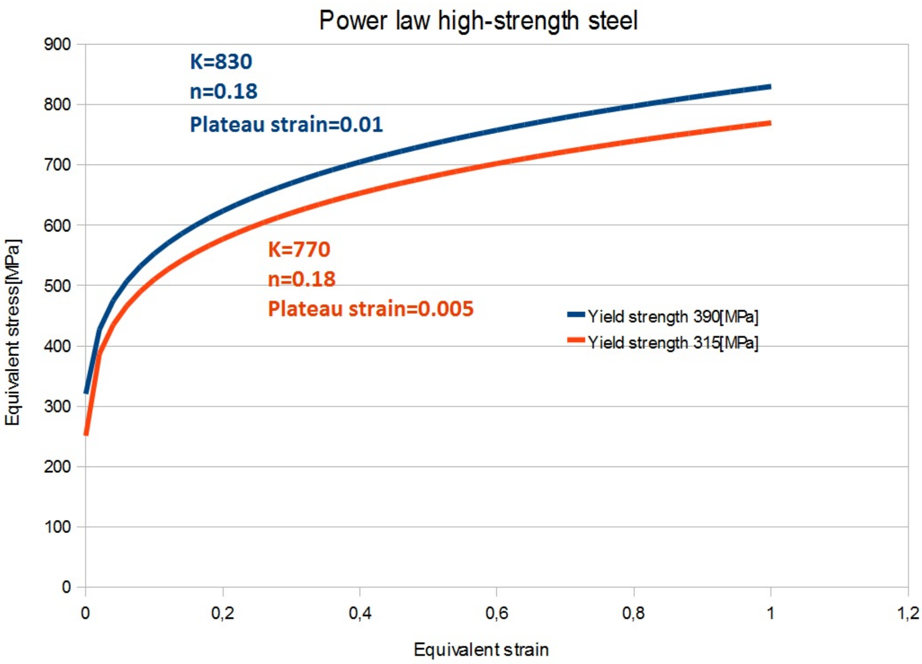

Figure 5: Power law curves, High strength steel

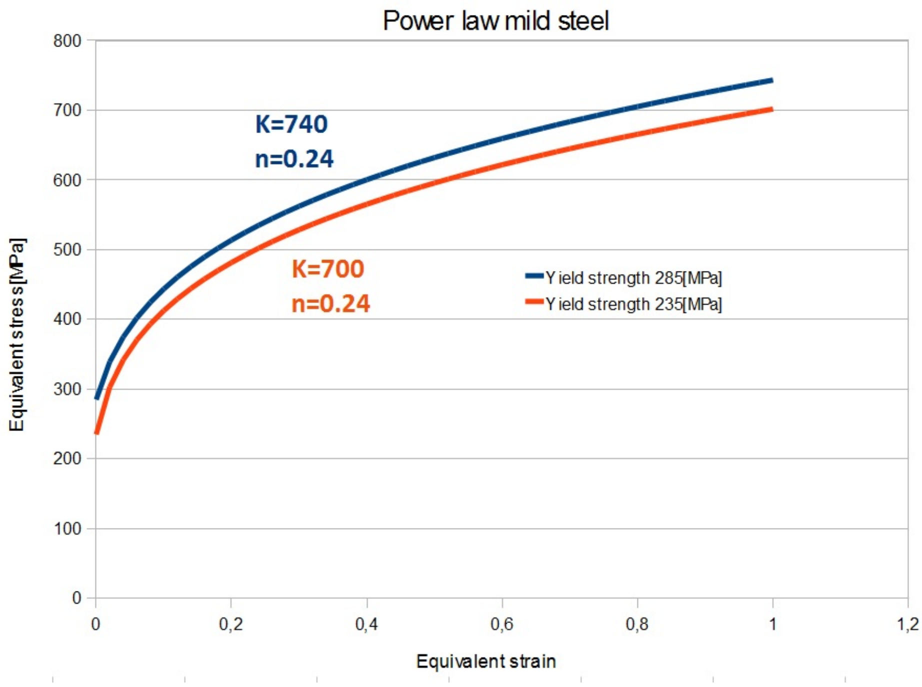

Figure 6: Power law curves, Mild steel

3.2.1 Steel material fracture criterion

The so-called RTCL-criterion is implemented in some simulations to account for possible fracture of the steel during collision. This is based on the work of Rice-Tracey and Cockcroft-Latham, and can be expressed by the so-called evolution rule. The following equations are taken from Alsos, [3].

$$\dot{D} = \begin{cases} 0 & \text{if } T < -1/3 \\ \frac{\sigma_1}{\sigma_{eq}}\dot{\epsilon}_{eq} & \text{if} - 1/3 \leq T < 1/3 \\ exp(\frac{3T-1}{2})\dot{\epsilon}_{eq} & \text{otherwise} \end{cases} \tag{7}$$

T is the ratio between the hydrostatic stress σ_m and the equivalent stress σ_{eq}

Fracture is initiated when the accumulated damage reaches a critical level. When T=0.33, meaning that we have proportional loading in uni-axial tension, the damage evolution, $\dot{D}$, is matched by the rate of equivalent plastic strain, $\dot{\epsilon}_{eq}$. This is convenient since the critical damage, D_{cr} is most easily found from uni-axial tensile test. From this a normalised damage criterion is expressed, where failure will occur when D reaches the value of one.

$$D = \frac{1}{\epsilon_{cr}} \int \dot{D}dt \tag{8}$$

Through thickness crack growth is simulated as a loss of resistance and stiffness in each through-thickness integration point. The elements are removed when all integration satisfies $D > D_{cr}$.

The code for the key file is taken from Martin Storheim's work,[12]. It has the same values as the previous steel material including some new parameters

	Mild steel	**High strength steel**
$\rho[kg/m^3]$	7850	7890
E[GPa]	210	210
G[GPa]	80.77	80.77
B[GPa]	175	175
Poisson's ratio	0.3	0.3
$\sigma_Y[MPa]$	235	315
K[MPa]	700	770
n	0.24	0.18
ϵ_{plat}	-	0.005

Table 3: RTCL-criterion steel material properties

The shear modulus G, represents the response of a material to shear. It can be defined as $G = \frac{E}{2(1+\nu)}$ For regular steel the value is $G = \frac{E}{2.6} = 80.77GPa$

The bulk modulus, B, is a material's ability to resist uniform compression. It can be defined as

$$B = \frac{p}{\epsilon_{kk}} = \frac{hydrostatic\ pressure}{change\ in\ volume}. \tag{9}$$

B is related to E and G in the following way $K = \frac{GE}{9G-3E}$ For regular steel $B = \frac{E}{1.2} = 175GPa$

4 Collision mechanics

Collision mechanics are governed by kinetic energy of the impacting bodies. During the impact there is a reduction in total kinetic energy, and this difference in energy has to be dissipated as strain energy in the ship and the iceberg. The strain energy can be seen in the force-deformation relationships, as the area under the curves. Figure 7 is taken from NORSOK N-004, [4], and is about collision between ship and offshore installations, yet the principle is the same for ship-iceberg collisions.

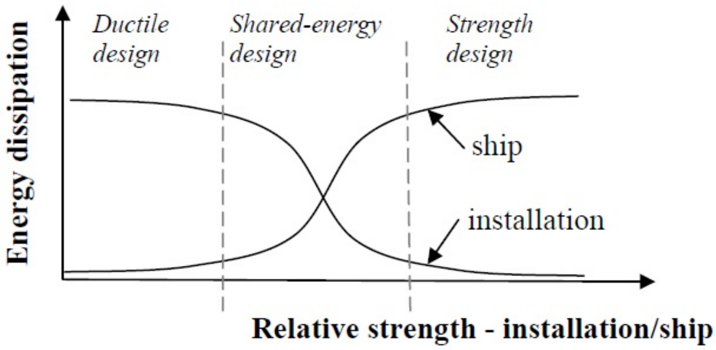

Figure 7: Relative strength, [4]

- **Strength design:** The ship suffers only minor deformations, i.e. the iceberg dissipates the largest part of the energy
- **Ductility design:** the ship experiences large plastic deformations and dissipates most of the collision energy.
- **Shared energy design:** both ship and iceberg contributes significantly to the energy dissipation.

Strength design and ductility design are less complex approaches compared to shared energy design. In the shared energy method the magnitude and distribution of the forces depend upon the deformation of both structures, consequently making the analysis more complex. The ductility and shared energy methods are the most common approaches. These two methods will be investigated in this thesis by integrated FEM-analyses.

In the accidental limit state the collision analyses may be divided into two uncoupled processes, external and internal mechanics. The external case consider rigid body motions and the total magnitude of the strain energy dissipation. Internal mechanics focus on the distribution of the strain energy in the colliding bodies. The internal mechanics will be investigated by integrated FE-analyses.

4.1 External mechanics model

Zhenhui Liu and Jørgen Amdahl,[5], have developed a new formulation for the external mechanics for ship/ship and ship/iceberg collisions. The result is a closed form solution for 3D, 6DOF collisions scenarios. It is first of all based on Stronge's external mechanics impact model. In this model Stronge makes a few basic assumptions:

- The impact duration is short and the impact force is large, so all other external forces are neglected

- The deformations are limited to a small area within the contact surface

A local coordinate system is introduced to derive the equations of motion.

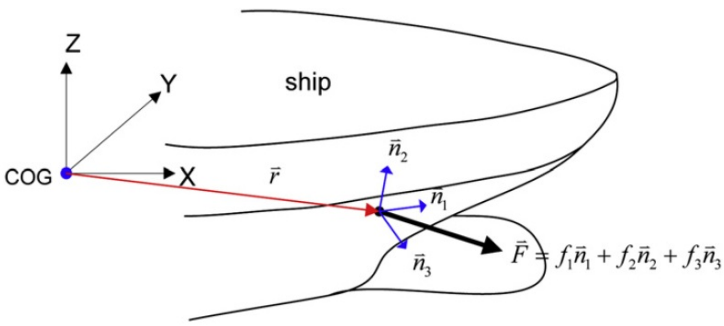

Figure 8: Collision point geometry and local coordinate system, [5]

Since the ship motion are generally described in the global coordinate system, it is necessary to transform between the local and global coordinate system. This matrix is determined by the geometry of the outer hull in the collision. These angles are defined according to DNV,[13].

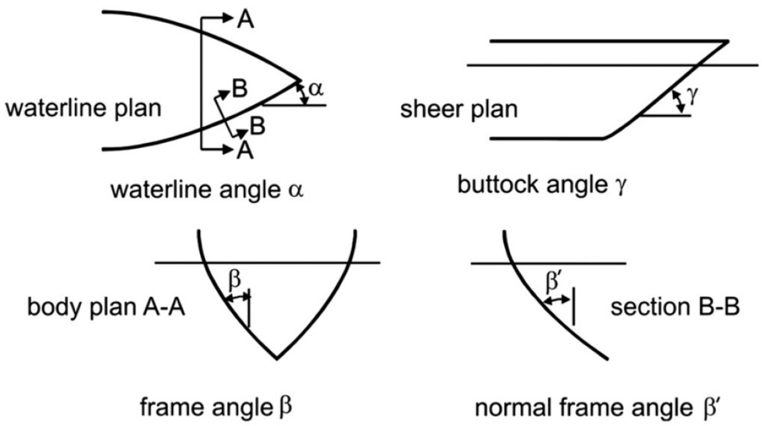

Figure 9: Hull angles, [6]

The transformation matrix from global to local coordinate system is given as

$$T_{lg} = \begin{bmatrix} cos(\alpha) & -sin(\alpha) & 0 \\ -sin(\alpha)sin(\beta') & -cos(\alpha)sin(\beta') & -cos(\beta') \\ sin(\alpha)cos(\beta') & cos(\alpha)cos(\beta') & -sin(\beta') \end{bmatrix} \tag{10}$$

The final expression in Stronge's method is

$$dv_i = m_{ij}^{-1} dp_j \tag{11}$$

where dv_i is the incremental velocity in each direction, m_{ij}^{-1} is the inverse of the equivalent mass, and dp_j is the impulse. This impulse in each direction, $\overrightarrow{n_i}(i = 1, 2, 3)$, is given by

$$dp_i = f_i dt \tag{12}$$

where f_i is the ith component of the interacting force acting on the infinitesimally small deforming element in the n_i direction. The relative acceleration $\ddot{s_i} = \frac{dv_i}{dt}$ in each direction is introduced. After integrating over the impact duration and introducing an equivalent mass variable $\frac{1}{\overline{m}} = m_{ij}^{-1} \frac{f_j}{f_i}$, the dissipated energy is found to be

$$E_i = \frac{1}{2} abs(\overline{m_i} \Delta v_i^2) \tag{13}$$

where

$$\Delta v_i^2 = (v_i^t)^2 - (v_1^0)^2 \tag{14}$$

v_i^t *and* v_1^0 are the relative velocities before and after the collision respectively, thus Δv_i^2 is the change of the squared relative velocities.

Two friction factors, normal friction, μ_n, and tangential friction μ_T, are introduced in order to solve the energy equation. Depending on the ratio of impulse between normal direction and tangential direction we will either have a sliding case or a sticking case. These cases each have a set of energy equation, $E_1,\ E_2,\ and\ E_3$, where $E_{TOT} = E_1 + E_2 + E_3$.

Sticking case

$$E_1 = \frac{1}{2}abs(-\overline{m_1}(v_1^0)^2) \tag{15}$$

$$E_2 = \frac{1}{2}abs(-\overline{m_2}(v_2^0)^2) \tag{16}$$

$$E_3 = \frac{1}{2}abs(-\overline{m_3}(v_3^0)^2) \tag{17}$$

Sliding case

$$E_1 = \frac{1}{2}abs(\overline{m}_1 dv_1(dv_1 + 2v_1^0)^2) \tag{18}$$

$$E_2 = \frac{1}{2}abs(\overline{m}_2 dv_2(dv_2 + 2v_2^0)^2) \tag{19}$$

$$E_3 = \frac{1}{2}abs(\overline{m}_3(e^2 - 1)(v_3^0)^2) \tag{20}$$

For 3D analysis of ship-iceberg the energy dissipation is typically compared with the maximum possible energy dissipation, E_0, which occurs if we have a central, plastic impact. This means that the iceberg sticks to the ship after impact and the velocity after impact is the same for both ship and iceberg.

$$E_0 = \frac{1}{2}M_{ice}\hat{v}_{ship}^2 \frac{(1 - \frac{\hat{v}_{ice}}{\hat{v}_{ship}})^2}{1 + \frac{M_{ice}}{M_{ship}}} \tag{21}$$

where $\hat{v}_{ship}$ and $\hat{v}_{ice}$ are velocities under the global coordinate system of ship and iceberg respectively. M_{ship} includes added mass.

Zhenhui Liu has developed a MATLAB-script where his mechanics model has been implemented. Input parameters in this script is among other things the gyration radius for the iceberg. In this thesis the icebergs take the shape of spheres, and thus the radius of gyration is for all directions:

$$\hat{R}_{xx}^2 = \hat{R}_{yy}^2 = \hat{R}_{zz}^2 = \frac{I}{m} = \frac{2}{5}r^2 \tag{22}$$

5 Software

5.1 MSC.Patran

Patran is according to MSC Software the world's most widely used pre-/post processor for FEM applications. It provides a tool for modelling, including meshing and analysis setup for several FE-codes, such as MSC.Nastran, Marc, Abaqus, ANSYS and LS-DYNA. The latter is used in this thesis.

5.2 LS-PrePost

LS-PrePost is an advanced and custom made pre-/post processor for LS-DYNA, designed to provide the following main functionalities:

- Support for all LS-DYNA keywords
- Visualization of LS-DYNA models
- LS-DYNA model creation and editing
- Advanced post-processing, including states result animation, fringe component plotting, and XY history plotting.

5.3 LS-DYNA

LS-DYNA is a general purpose finite element code for analysing large deformations in static and dynamic response of structures. The main solution methodology is based on explicit time integration. A modification of the central difference time integration is applied.

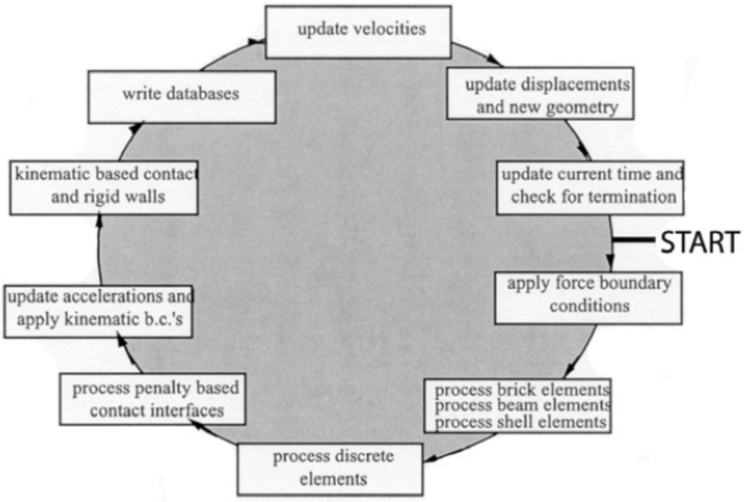

Figure 10: The time integration loop in LS-DYNA, [7]

5.3.1 Time step size

The time step size is important for several reasons. First of all the stability of a explicit dynamic finite element solver depends on a small enough time step size. The critical time step has to be smaller than the time it takes a pressure wave, travelling at the speed of sound, to pass through the smallest element in the model. The contact algorithms also depend on small time steps to be stable.

On the other hand the time step size is an important factor when it comes to the computational time of an analysis. Thus there must always be a compromise between small enough elements and a reasonable computational time.

In general a new time step size is determined by taking the minimum value over all elements

$$\Delta t^{n+1} = a\Delta min\{\Delta t_1, \Delta t_2, \Delta t_3, ..., \Delta t_N\} \tag{23}$$

Where N is the number of elements. The scale factor, a is set to 0.9 or smaller, and is a safety factor to ensure a small enough time step.

For SHELL elements the critical time step size is given by:

$$\Delta t_e = \frac{L_s}{c} \tag{24}$$

where L_S is the characteristic length and c is the speed of sound The default definition of L_S, which is used in this thesis, is given by the following.

$$L_s = \frac{(1+\beta)A_s}{max(L_1, L_2, L_3, (1-\beta)L_4)} \tag{25}$$

Where $\beta = 0$ for quadrilateral elements and 1 for triangular elements, A_s is the area, and $L_i (i = 1, 2, 3, 4)$ is the sides defining the element.

For SOLID elements the critical time step size is:

$$\Delta t_e = \frac{L_e}{\{[Q + (Q^2 + c^2)^{1/2}]\}} \tag{26}$$

where Q is a function of the bulk viscosity, L_e is the characteristic length and c is the so-called adiabatic sound speed.

6 Modelling

6.1 Ship modelling

Due to copyright issues the drawings of the ship could not be e-mailed. This made is necessary to visit the Advanced Structural Analysis section at DNV, Høvik. The most important drawings were copied manually, including transverse frames, stringers, shell expansion for inner and outer skin and stiffener dimensions.

The main part of the modelling was done using Msc.Patran. Håvard Nyseth at DNV provided a curve-model of the outer skin of a similar LNGC, which was very helpful in the beginning of the modelling process. The model was scaled up 25 times, so the total length matched the drawings. Still, the modelling and meshing process proved to be very time consuming since the geometry in the bow shoulder area is not regular and repeatable. Some simplifications were made to reduce the complexity of the geometry and hence reduce the modelling time. Most of these simplifications made the structure more conservative, e.g some stiffener dimensions were reduced instead of increased compared to the real ship. Only large holes in the frames were considered, and to further simplify all rounding of corners in these holes were neglected. The possible stress concentrations caused by these sharp corners are only interesting for fatigue purposes, and should not influence the results in a collision study. Smaller manholes and cut-out were neglected, and the superstructure over the tank was left out completely. The superstructure is assumed to contribute to a very small degree to the total stiffness of the ship side. Where stiffeners pass through frames, the nodes along the web are fixed to the frame while flanges are free to move. This should represent the real geometry at these locations, were the web is typically welded to the frame.

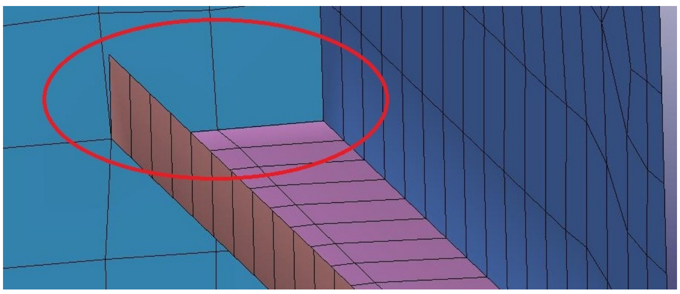

Figure 11: Stiffener passing through frame

The model has been meshed using 4 node quadrilateral elements. The mesh size for the ship structure was set to 200mm, which is larger than the 150mmx150mm that Zhenhui Liu recommends. However the improvement in results for a smaller element size will probably not make up for a large increase in computational time. The area of a 150x150mm element is just over half that of a 200mmx200mm, and thus the number of shell elements would almost double. In addition the characteristic length and the critical time step will be reduced to 3/4 of the original value. The combination of these factors would yield a large increase in the computational time. In [14], Stine Aas Myhre even use 250mm Quad4 shell elements for very similar analyses, which seem to give reasonable results.

The geometry was divided into to many smaller surfaces to help with the meshing process and the assignment of properties such as thickness and material models. The meshing itself was done using the paver function, which is especially effective for irregular geometry. For more regular rectangular surfaces it still creates a uniform mesh. With the paver function most elements on adjacent stiffeners and plates were automatically assigned the same nodes. This reduced the time needed to modify the mesh drastically. Still, the complexity of the geometry made it necessary to move nodes around to attach stiffeners to frames and so on. The finished model was exported from MSC.Patran into LS-PrePost. Boundary conditions, contact definitions, material properties and simulation parameters were all defined using LS-PrePost.

Three contact properties are defined according to Zhenhui Liu:

∗CONTACT_ERODING_SURFACE_TO_SURFACE
for contact between ice and ship, which allows for ice elements to be deleted to simulate crushing

∗CONTACT_AUTOMATIC_SINGLE_SURFACE
is used for contact within the ship itself. When deformations become large, parts of the ship that are not originally connected may make contact with each other.

∗CONTACT_ERODING_SINGLE_SURFACE
is used for contact within the ice material.

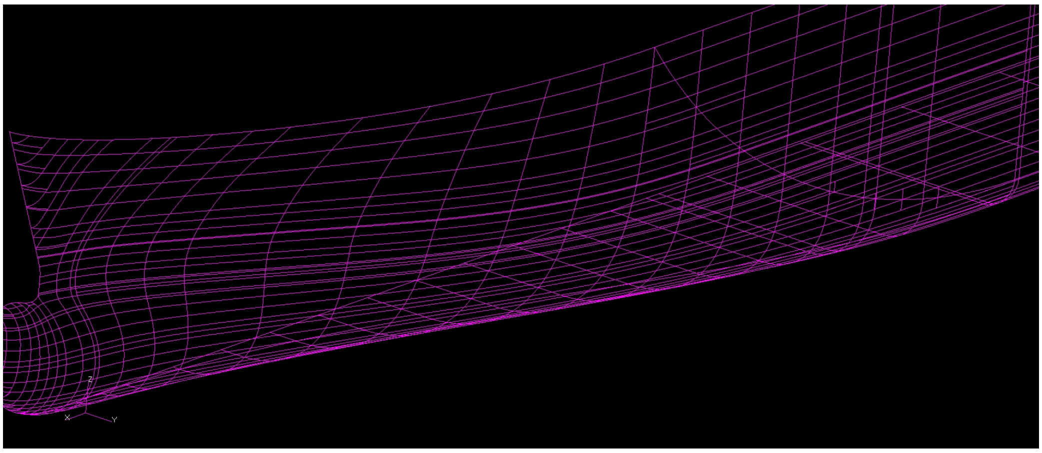

Figure 12: Curve model

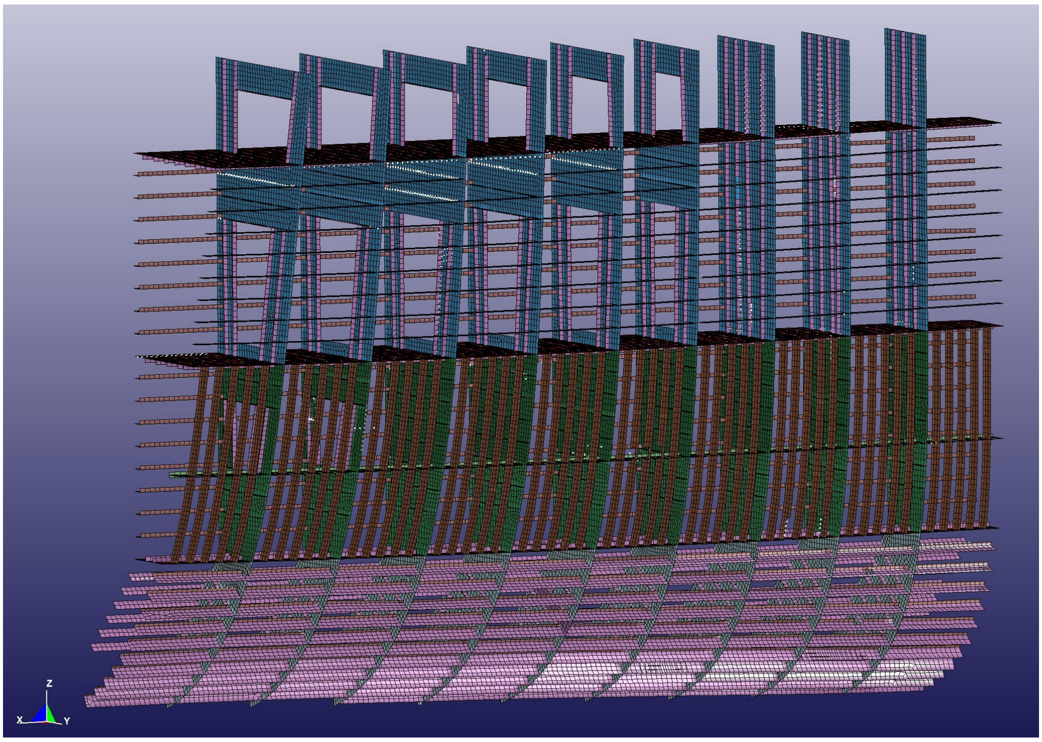

Figure 13: Finished model in LS-PrePost, plating is removed

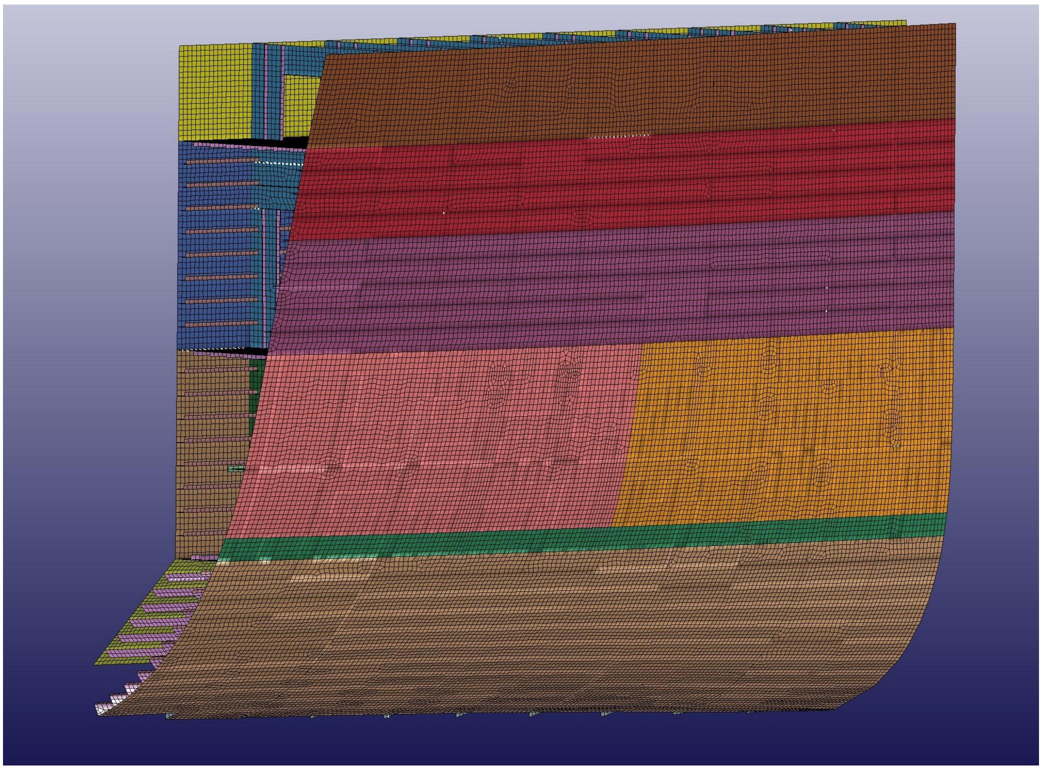

Figure 14: Finished model in LS-PrePost

Boundary conditions

The model is fixed in all directions of translation at each end of the stringers. Both inner and outer plates are fixed in translation in the x-direction along the ship. This is to simulate the continuity of the plates, and to take into account membrane stresses when the plate is deforming. The x-direction is not entirely parallel to the skin in the forward part of the model, but assumed to be a good approximation. At the lower edge of the model, parallel to the centre line there are no boundary conditions. This is surely not entirely realistic since there will be influence from the surrounding structure and the double bottom is probably very stiff. On the other hand the collision force will most probably be transferred out from the collision zone longitudinally in the stringers, and not so much into the double bottom. The forces that do go into the double bottom will eventually be taken by the hull girder as shear. Thus the shear stresses have to go down to zero at some point towards the centre line. Therefore it is not completely unrealistic, although conservative, to apply no boundary conditions in this area.

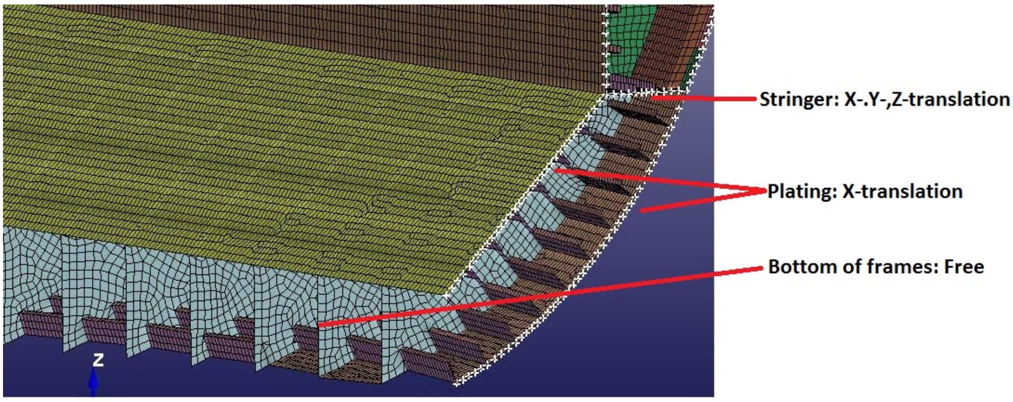

Figure 15: Boundary conditions

6.2 Iceberg modelling

The iceberg models in this thesis were all provided by Zhenhui Liu. To reduce the total modelling time, the collision scenarios considered are limited to these models. All models have a spherical shape, and there are in effect two models. To provide different sized icebergs they have been scaled up or down. The model in Figure 16 has originally a 2 meter radius with an elements size of 50mmx50mmx50mm. According to Zhenhui Liu,[2], this elements size is the best compromise between computational time and accuracy of the results. The front is assigned with Liu's ice material, while the disc at the rear is rigid. A scaled down version with 1 meter radius is also used. The elements size is reduced to 25x25x25mm and time step size is thus reduced. The model in Figure 17 is a 5 meter radius model, were the front part has ice material, while the back part is rigid. The elements size is also here 50x50x50mm. Another version is a scaled up model with 10 meter radius where the length element edges is doubled to 100x100x100mm. The rigid material helps reducing the computational time since no CPU effort is spent on these elements.

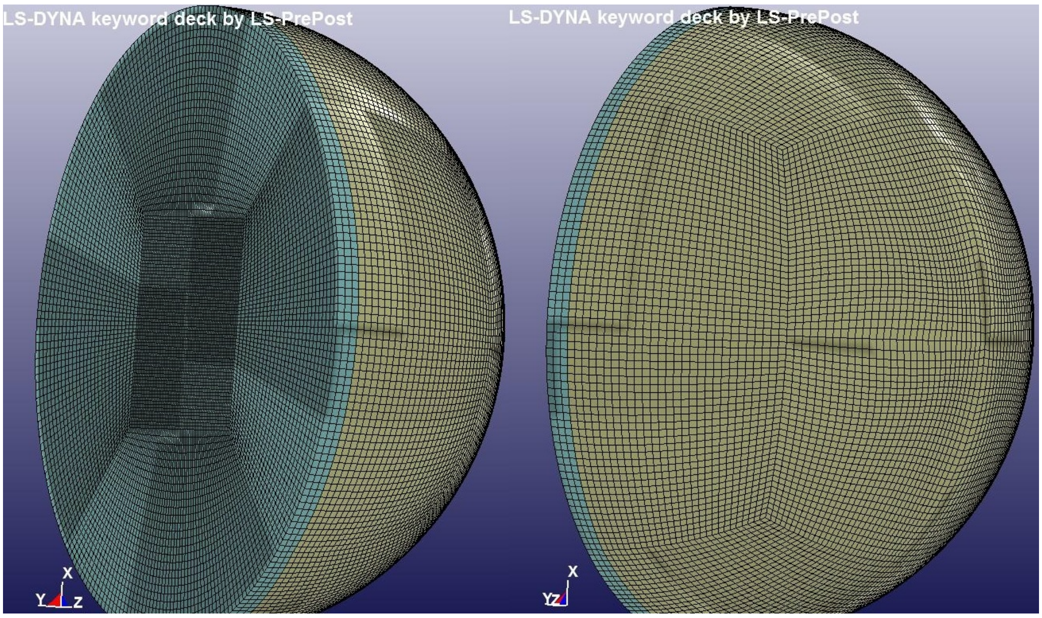

Figure 16: 2m iceberg

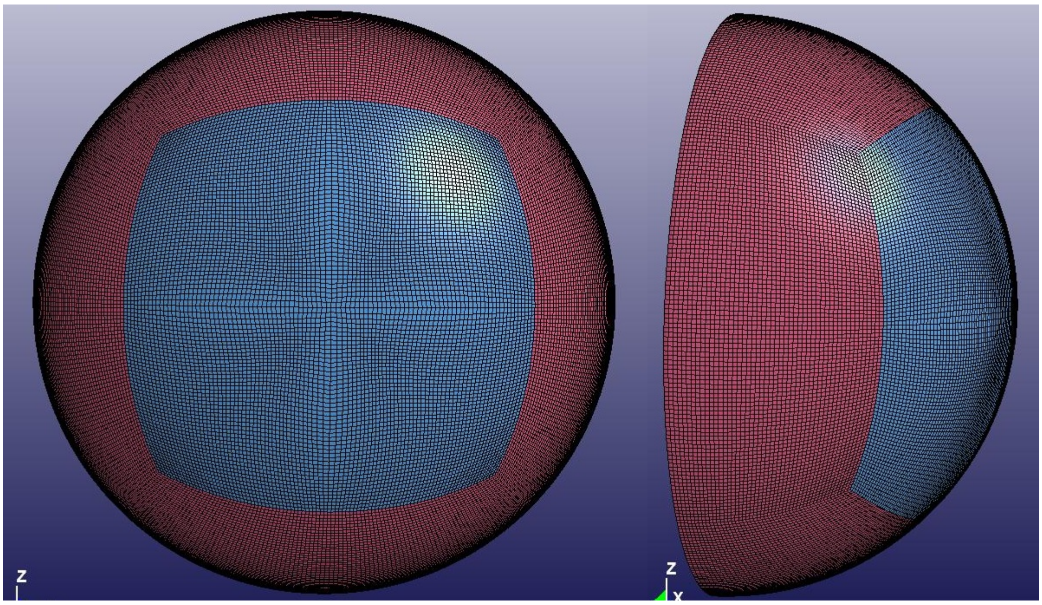

Figure 17: 5m iceberg

7 Collision analysis

Collision point

The point of impact for all analyses can be seen in Figure 18, where parts of the outer plating have been removed for visual purposes. It was decided after a discussion with Jørgen Amdahl that the impact should take place on the middle frame and in the middle of the ice-strengthened waterline area. This is somewhat under-conservative since the frames are stiff components, but for the larger icebergs which span several frames it will have little influence on the results.

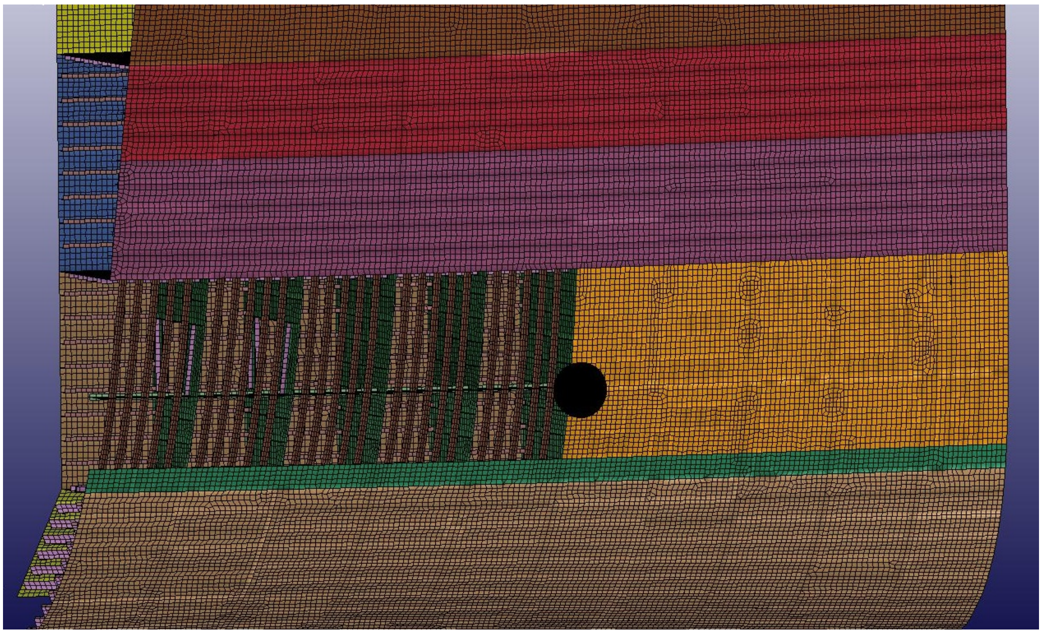

Figure 18: Point of impact

Prescribed displacement

In LS-DYNA the relative motion between ship and iceberg is defined using prescribed displacement. The rigid part at the rear of the iceberg model, is set to push into the ship at a constant speed of 5m/s. The simulation time for most cases is 0.5 s, and the actual displacement is then 2500 mm. The iceberg is not allowed to rotate or bounce off the ship side. The simulations will therefore show the deformations and energy dissipation if the iceberg would be pushed this far into the ship side.

Force-deformation curves

Pressure-area curves are not found. According to Zhenhui Liu it is not straight forward to find these curves, and after a discussion with Liu and Prof. Jørgen Amdahl, it was decided not to use time on this. The focus should instead be on strain energy and force-deformation relationships. These force-deformation curves are based on the assumption that the constantly changing front elements of the iceberg always stay in contact with the ship side. The time history of a node on the outer skin in the middle of the impact zone is plotted using the DATABASE_NODOUT-function in LS-DYNA. The ship deformation curve follows the sideways displacement of this point. In order to find how the iceberg is crushed, the same node on the ship side is compared with the movement of the rigid part pushing the iceberg into the ship side. Both curves are plotted against the contact force, which is found using the DATABASE_RCFORC function. By plotting displacement against force, it is easy to see the relative collision energy for ship and iceberg, which is the area under the curves.

7.1 2m radius iceberg

7.1.1 2m, Rigid material

The first analysis is done with a spherical iceberg shape with 2 m radius. The material is rigid, which should be on the conservative side since no collision energy is used to deform the iceberg. The iceberg is given a prescribed displacement of 4500mm during 0.9 s into the ship side. This is equivalent to a constant velocity of 5 m/s. The direction is normal to the centre line and not normal to the ship side at that point, which probably would be more realistic.

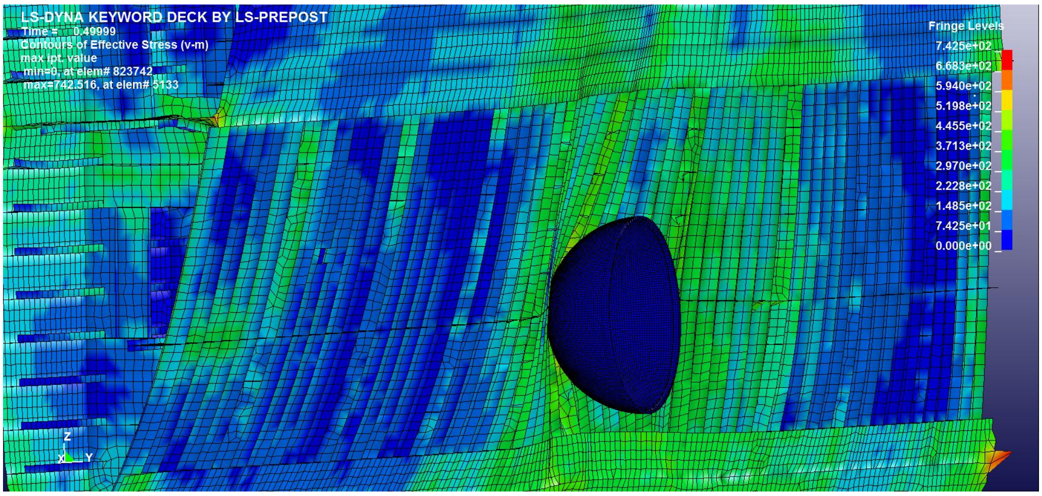

Figure 19: Collision with rigid iceberg, 0.5s, von Mises stress

The deformation zone extends over several frames and strain energy dissipation is distributed over many parts in the ship side. As seen in Figure 20, the frames prove to be relatively weak, and tend to buckle as one in stead of between their horizontal stiffeners. The dimensions of these flat bar stiffeners are a mere 175mmx12mm. Due to this extensive buckling, the plate is pushed on over a large area and the pressure in the impact zone is not large enough for the iceberg to penetrate the plate at an early stage. The failure criterion for the LNG containment system of 700mm deformation of the inner hull is reached after around 0.45 seconds. The internal energy at this time is approximately 70MJ.

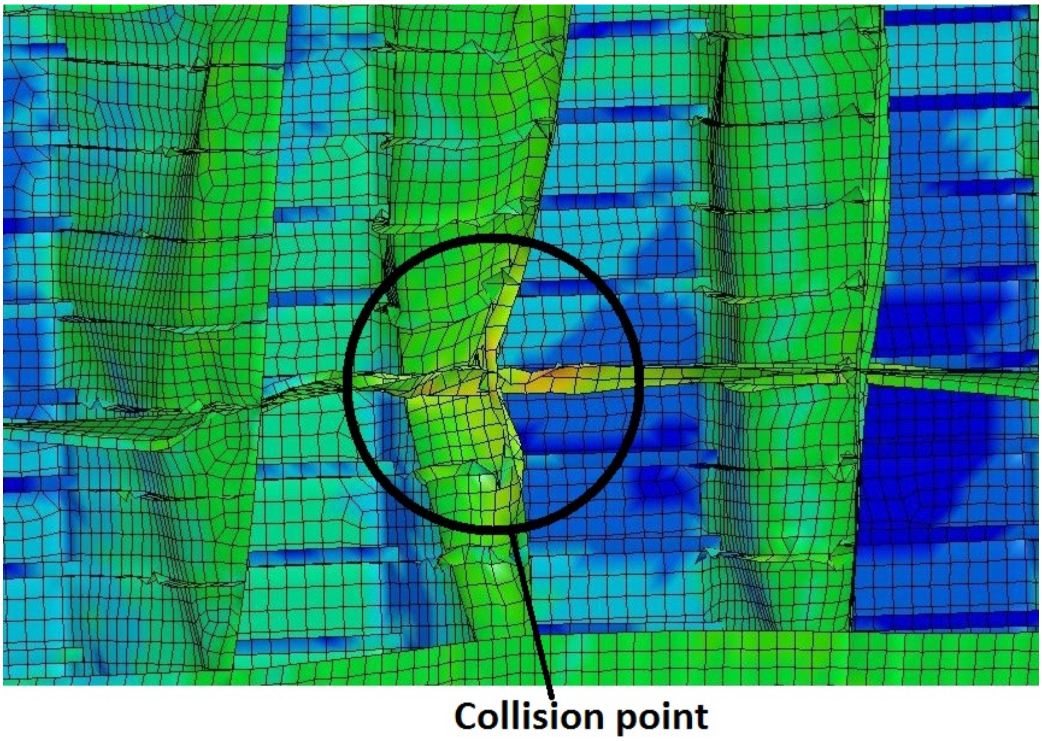

Figure 20: Buckling of frames, 0.5s, von Mises stress

In this analysis the model of the ship had some minor flaws in the mesh. Certain elements were for some reason deleted, and there were parts that should have shared nodes that did not. On later analyses most of these flaws are corrected, although there still are some errors, but none should be critical for the accuracy of the results.

7.1.2 2m, Ice material

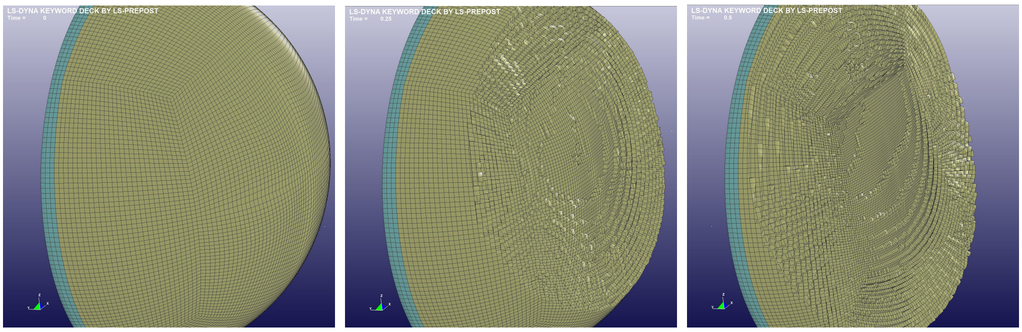

Figure 21: 2m iceberg, stages of collision: 0s, 0.25s, 0.5s

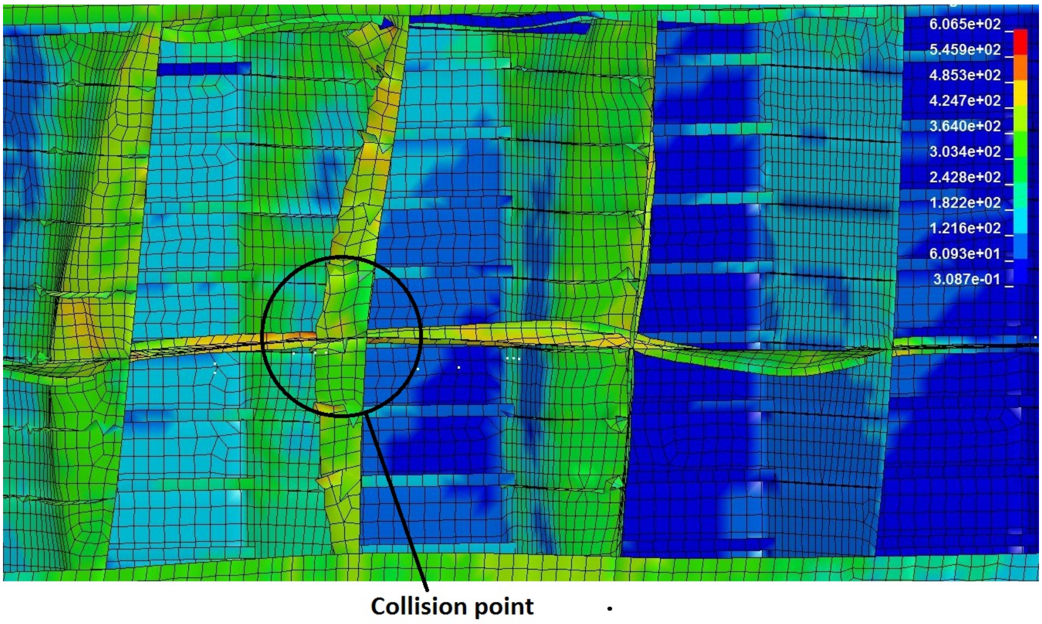

Figure 22: 2m collision, 0.5s, von Mises stress

There is a significant difference in the deformation of the ship side when the ice material is applied to the model of the iceberg. The buckling of the frames is not as extensive, while the majority of the elements assigned with the ice material are eroded off during the 0.5s simulation. Thus there is no point in increasing the simulation time, even though the 700mm failure criterion is not reached. The inner plate is only pushed in about 225mm. Although the deformations are within survival limits, they are not insignificant. If the 2 meter hemisphere is part of a larger iceberg with a large mass, this simulation can still be relevant.

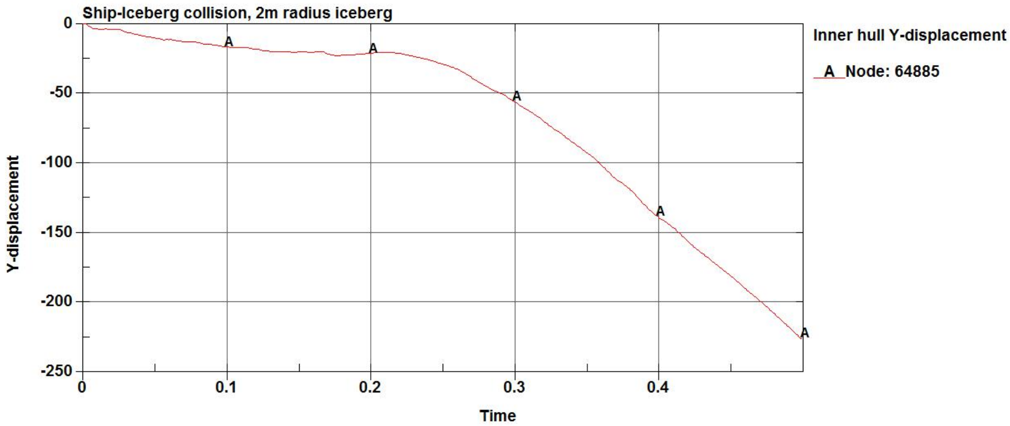

Figure 23: 2m iceberg, Displacement of centre of inner hull

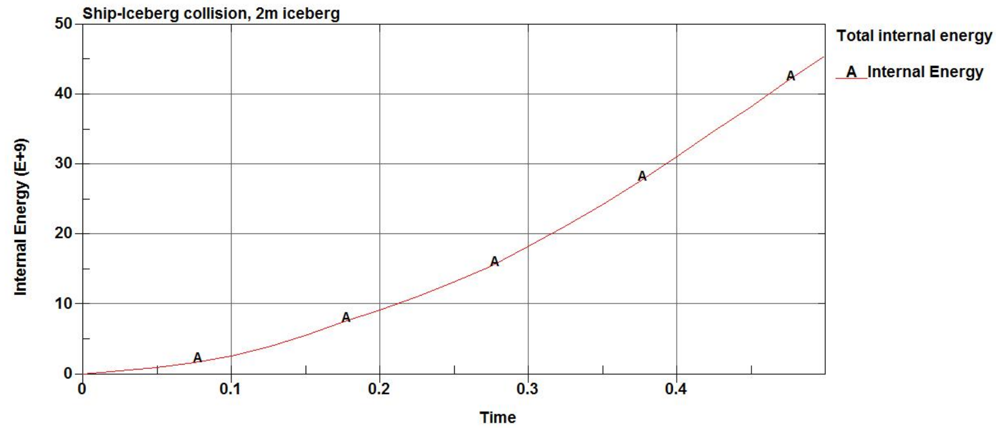

Figure 24: 2m iceberg, Total internal energy in ship and ice

The crushing of the ice is significant and can also be seen on the force vs. displacement curves, Figure 25, for the ice and ship. Up to a point the ship resists deformation almost entirely. However at a contact force of around 17MN the transverse frame directly behind the point of impact starts to buckle, and the deformation increases with almost no increase in force. When the deformation reaches 450mm the curve shows a rapid increase in the contact force. This seems to be due to membrane forces in the plates caused by the lateral deflection. These forces will in turn increase the lateral resistance of the plates. Another contributing factor can be the neighbouring frames which begin to carry more load after the centre frame has buckled.

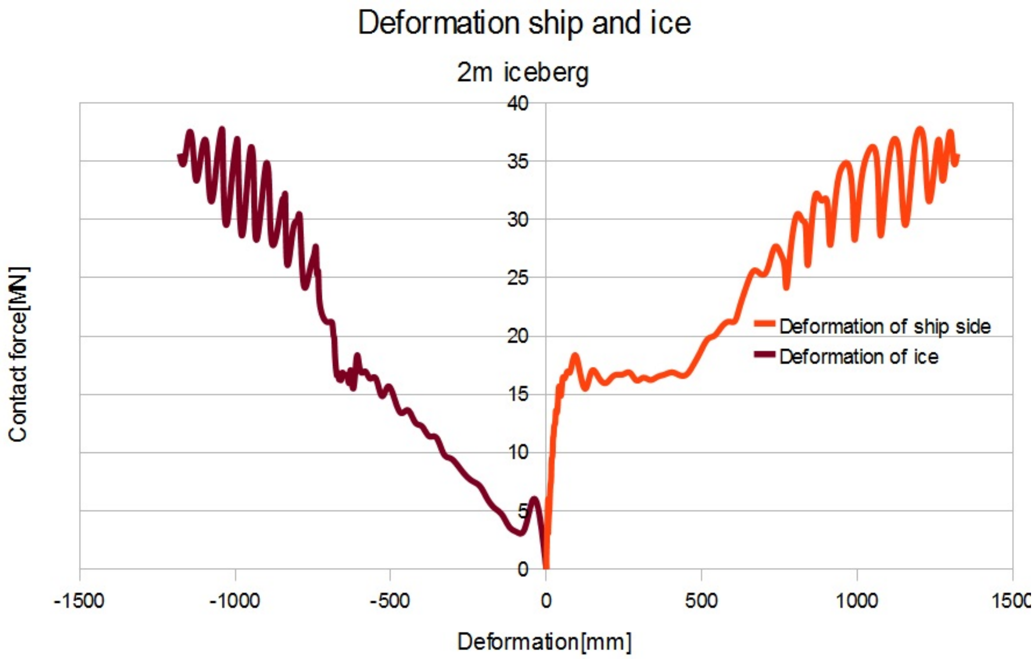

Figure 25: Force vs deformation ship and iceberg

7.2 1m radius iceberg

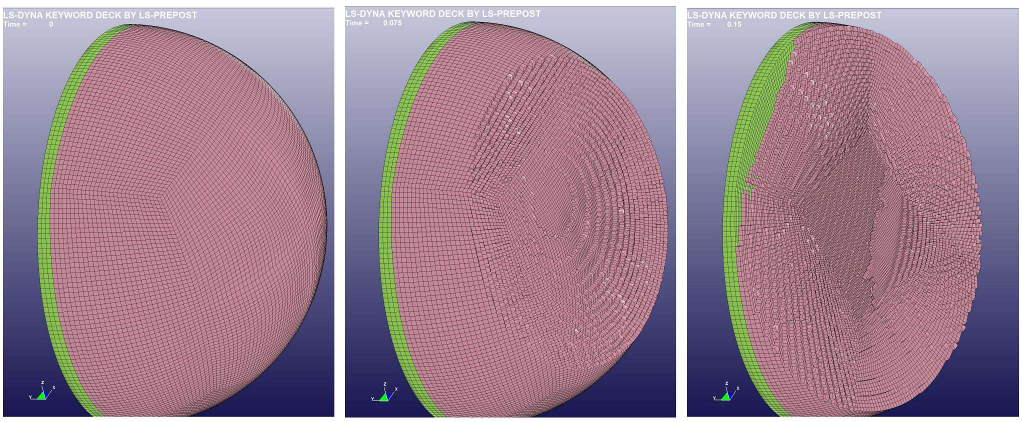

Figure 26: 1m iceberg, stages of collision: 0s, 0.075s, 0.15s

The 1 meter radius iceberg is so small that the diameter is smaller than the frame spacing(3360mm). This means that it could in theory penetrate the plating between frames and reach far into the ship with limited strain energy being dissipated in the ship. This scenario can first of all be thought of as part of an uneven surface of a bigger iceberg, since a free-floating iceberg of 1 meter radius is unlikely to cause any significant damage to the ship. The key point in this analysis is hence to see to what extent the iceberg is crushed. The simulation time is set to only 0.2 seconds, due to very long computational time. The 1 meter iceberg model is scaled down from the 2 meter model, and thus the element size is decreased. This causes the time step size to decrease according to equation 26. The result is a computational time in excess of 40 hours. The prescribed displacement-parameters are changed to keep the velocity at 5m/s, and the distance is 1000mm.

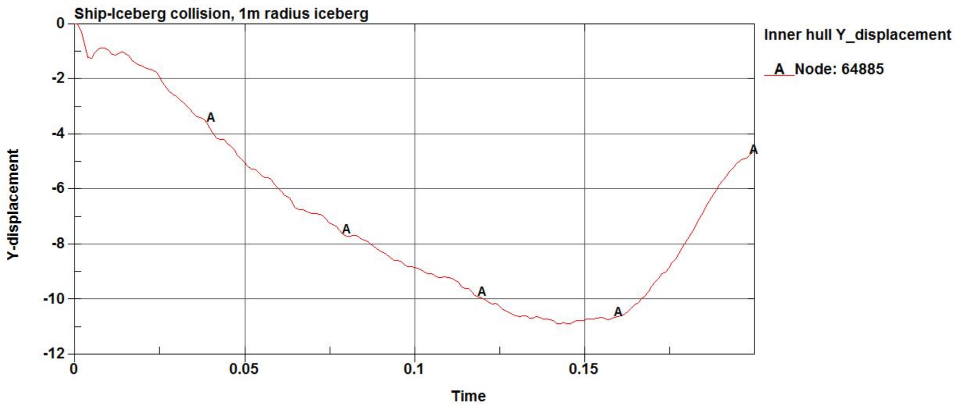

Figure 27: 1m iceberg, Displacement of centre of inner hull

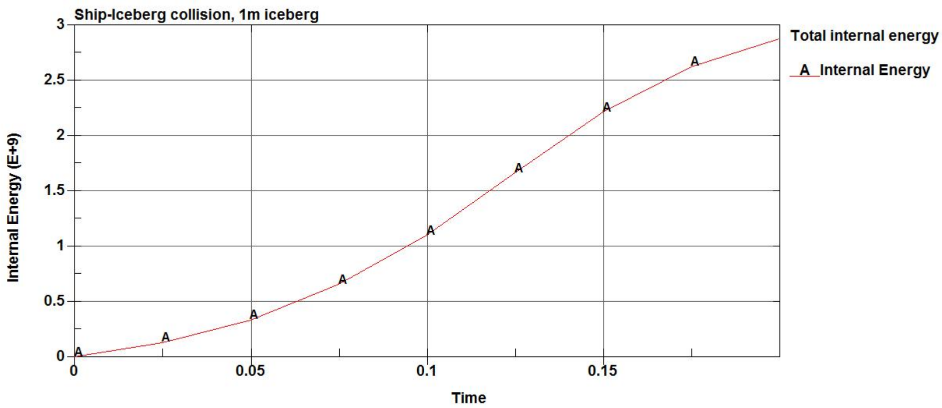

Figure 28: 1m iceberg, Total internal energy in ship and ice

And as seen in the plots and results, there is virtually no visible deformation in the ship structure. The ship side is pushed in only 15mm while the ice is being completely crushed, even though the simulation time is short. The analysis is somewhat un-conservative since the impact point is directly on a frame. In any case the deformations in the ship side are so small that it would make little difference to move the point of impact to a mid-frame position.

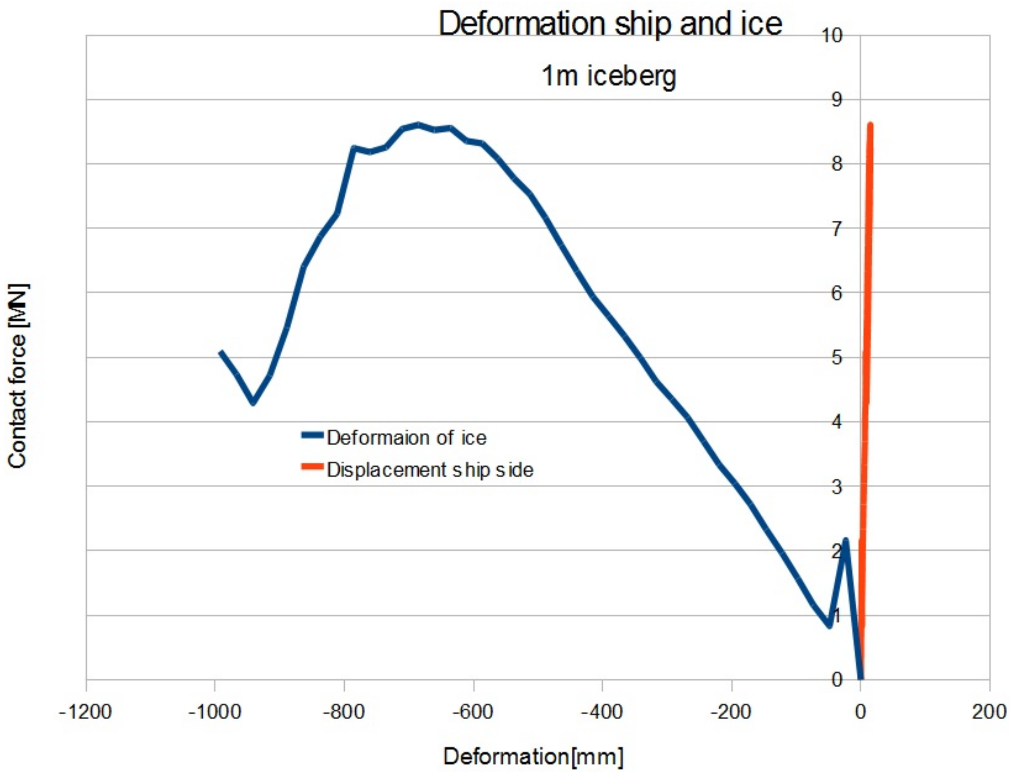

Figure 29: Force vs deformation ship and iceberg

7.3 5m radius iceberg

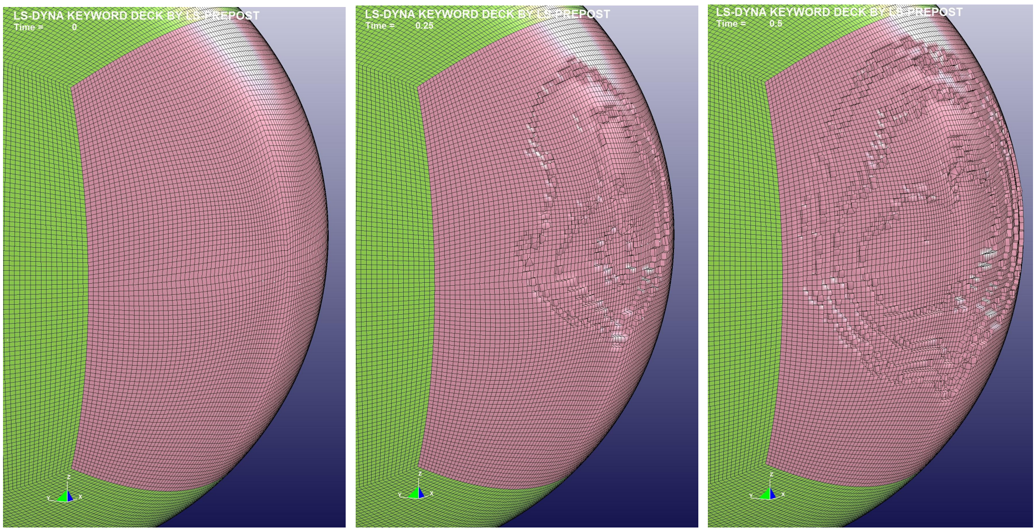

Figure 30: 5m iceberg, stages of collision: 0s, 0.25s, 0.5s

This iceberg corresponds to DNV's recommendation in [6] for a 2 m sail height. From the animation of the impact can be seen that the erosion of ice-elements is not as extensive as for the smaller icebergs(Figure 30). The main cause of this is probably that the larger contact area reduces local pressure and thus the crushing of ice. There is some erosion at the rear of the ice material, in the contact area between the rigid material and ice material. Still, compared to the erosion on the ice/ship interface this effect is not large. The part of the model with ice material is relatively small, and the boundaries with the rigid part have influence on the results. The number of elements with ice material is an important factor for computational time. Due to this the model size should be limited as much as possible, without compromising the accuracy of the results. If the shape of the iceberg was more blunt and the ship was even stiffer, the back-erosion would probably be more severe and possibly be more of a problem.

The deformation in the ship side is much more severe than for the smaller iceberg. During the 0.5s of simulation the 700mm criterion for the inner plating is barely reached. The critical total strain energy level is approximately 72MJ. From the force-deformation curve it is quite clear that most of the energy is dissipated in the ship structure. As with the 2m iceberg, the deformation of the ice appears to be consistent during the simulation. The ship on the other hand has at first very small deformations while the contact force builds up. Then there is a jump in deformation which corresponds to when the centre frame starts to buckle. However membrane forces quickly begin to take effect as the force starts climbing rapidly again. As seen in Figure 31 the frames are being severely deformed towards the end of the simulation. The frames closest to the impact zone are buckling and folding, and the stringers also show buckling patterns. This reduces the load-carrying capability of these structures to a large extent. But even in this post-buckling state stiffened plates can carry more loads. This is due to in-plane membrane forces that take effect in the transverse direction.

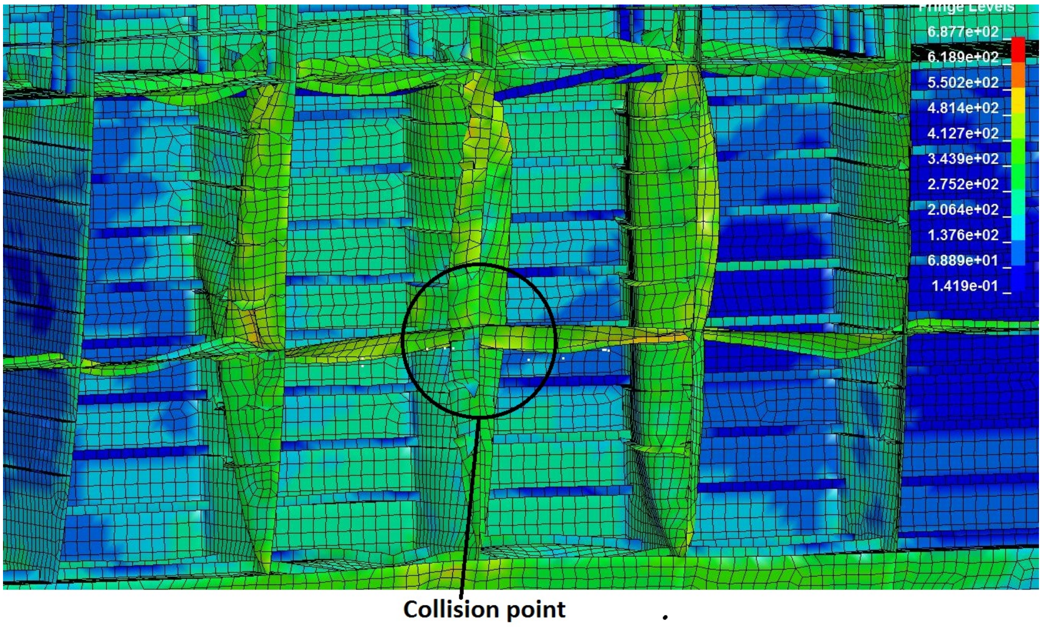

Figure 31: 5m iceberg collision, impact zone, 0.5s, von Mises stress

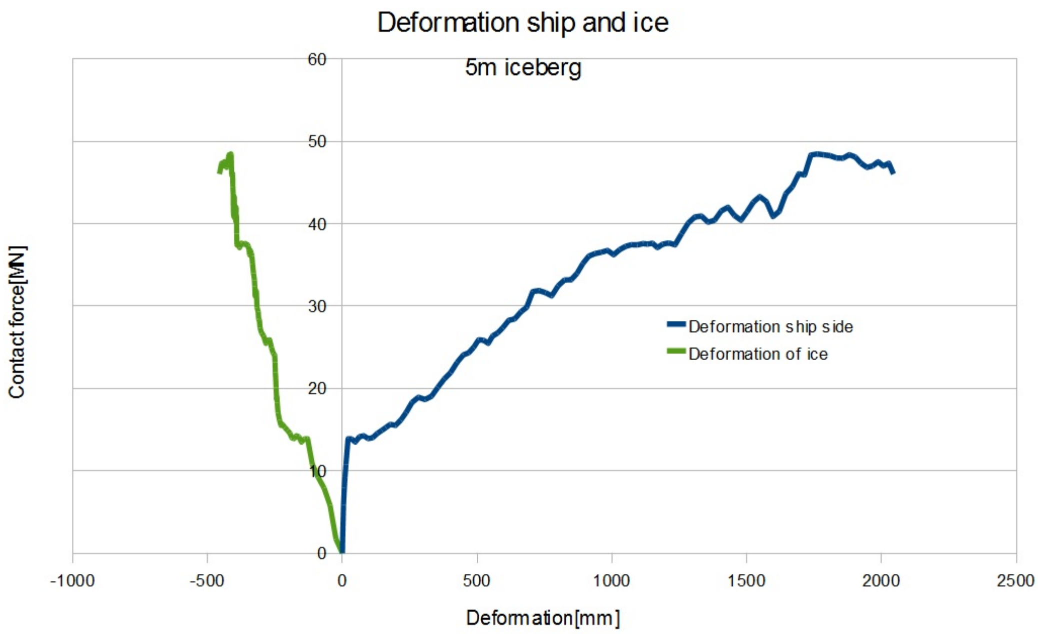

Figure 32: Force vs deformation ship and iceberg

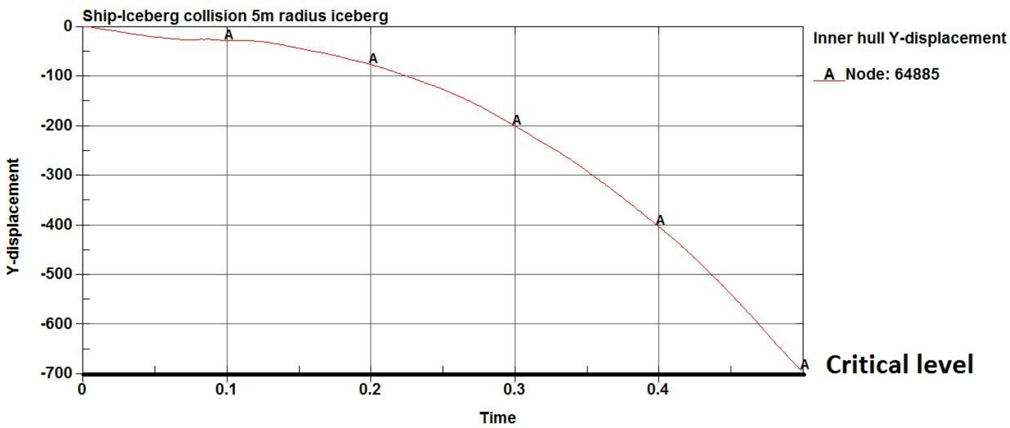

Figure 33: 5m iceberg, Displacement of centre of inner hull

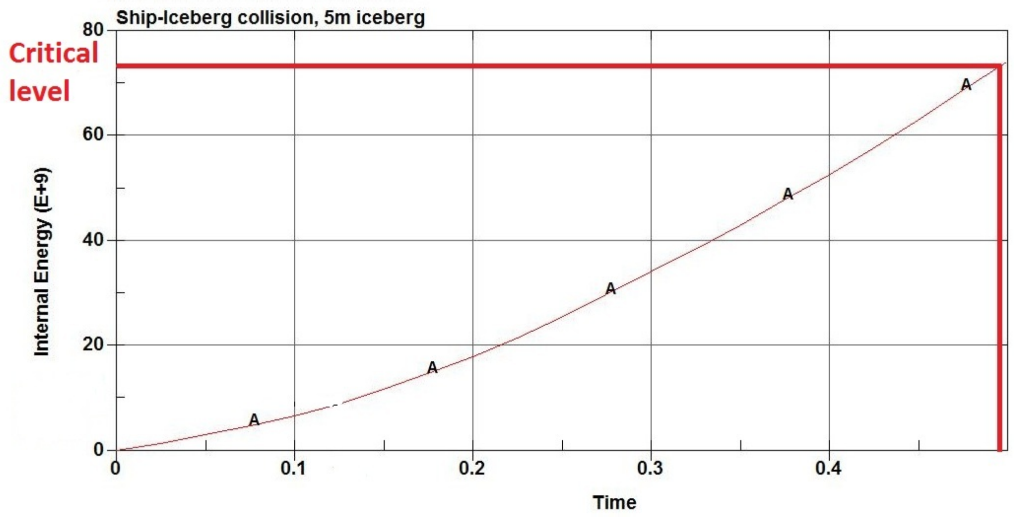

Figure 34: 5m iceberg, Total internal energy in ship and ice

7.4 10m radius iceberg

The final analysis is done with a spherical 10 meter iceberg, which is scaled up from the 5 meter model. This means that the element edges are twice as large and thus the accuracy is reduced. As expected the level and extent of deformations are larger. The strain energy level for critical inner hull deflection seem to correspond well to the 5 meter iceberg. The critical deflection is reached after 0.42s, and the critical total energy level is just over 70MJ.

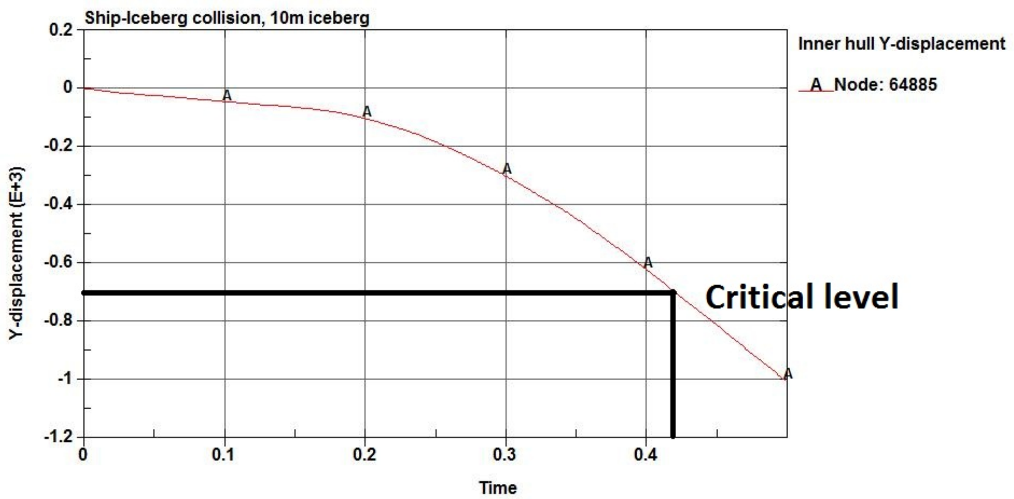

Figure 35: 10m iceberg, Displacement of centre of inner hull

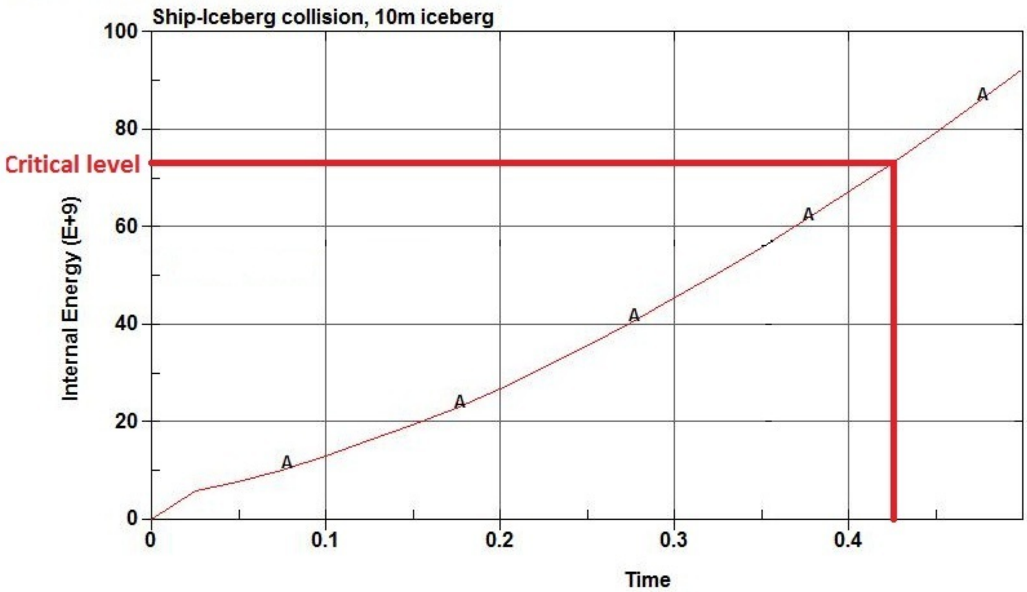

Figure 36: 10m iceberg, Total internal energy in ship and ice

7.5 Steel material with fracture criterion

The simulation results where a fracture criterion is applied to the steel material model, prove to be close to the previous results. For the 5m case, the inner hull deflection limit is reached a few hundredths of a second earlier with the steel fracture criterion. The total strain energy is also slightly lower. This is as expected, since the fracture criterion removes all elements that are beyond their load carrying limit. There is no fracture in the outer skin, while at the ends of the stringers the most extensive deletion of elements takes place, see Figure 37. In the earlier analyses were no fracture criterion is applied, the elements still provide some resistance even when they reach their ultimate strength. This indicates that the fracture material model is more realistic, particularly for the large deformations at the end of the simulation.

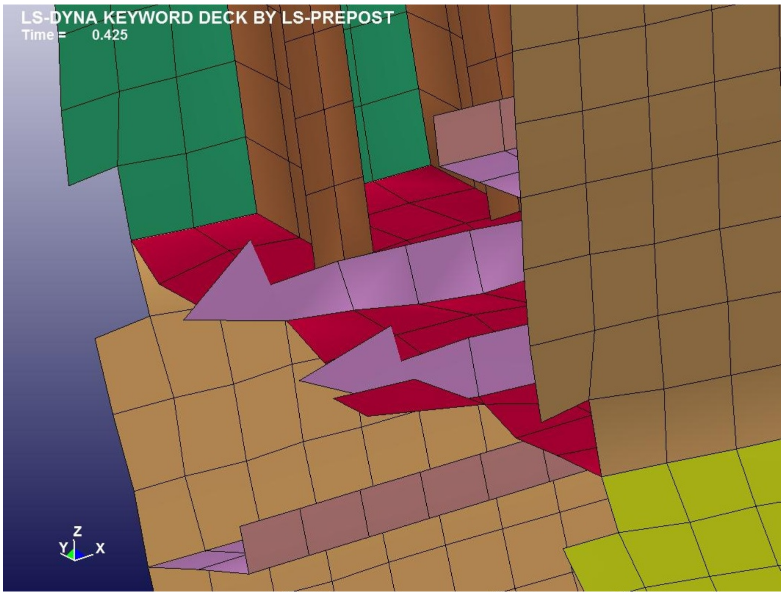

Figure 37: Erosion of elements at the boundaries of the stringers

7.6 Accelerations

The accelerations of the inner hull have been checked on the locations shown in Figure 38. The collision scenario is the 5 meter radius iceberg.

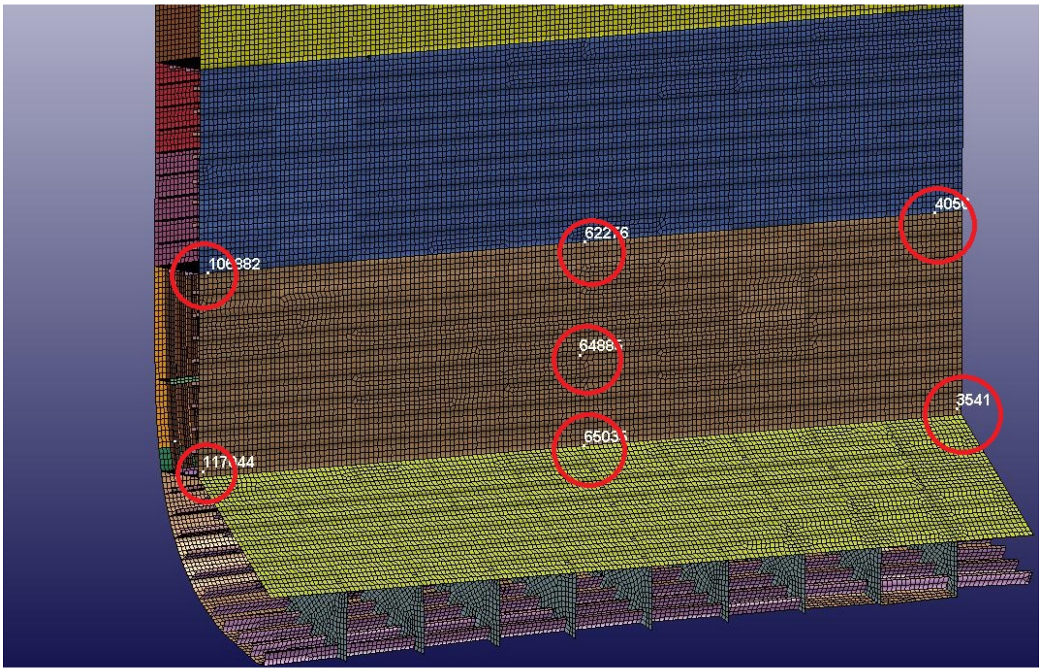

Figure 38: Locations for acceleration check

The worst location is the one in the centre, which is also directly inside of the centre of the collision zone. The concern about accelerations causing problems far from the collision point seems to be exaggerated. Yet for the cubic iceberg used in [8], collision shock waves will surely cause larger levels of accelerations than found here. For all cases the accelerations of the inner hull does have some high peaks. Some are even in excess of $2000m/s^2$, but none seem to last more than a few milliseconds. This means that the displacements caused by the accelerations should be small. It is the quick changes in displacements caused by the accelerations and not the accelerations themselves that are the danger. None of the corresponding displacement curves have any sudden jumps, hence the acceleration peaks are too short-lasting to be a serious problem.

In the case of the 10 meter radius iceberg, the accelerations are checked only in the central position. The curve looks very similar to the smaller iceberg, with some high peaks over 2000m/s, but the duration is very short.

This indicates that the accelerations are not of major importance.

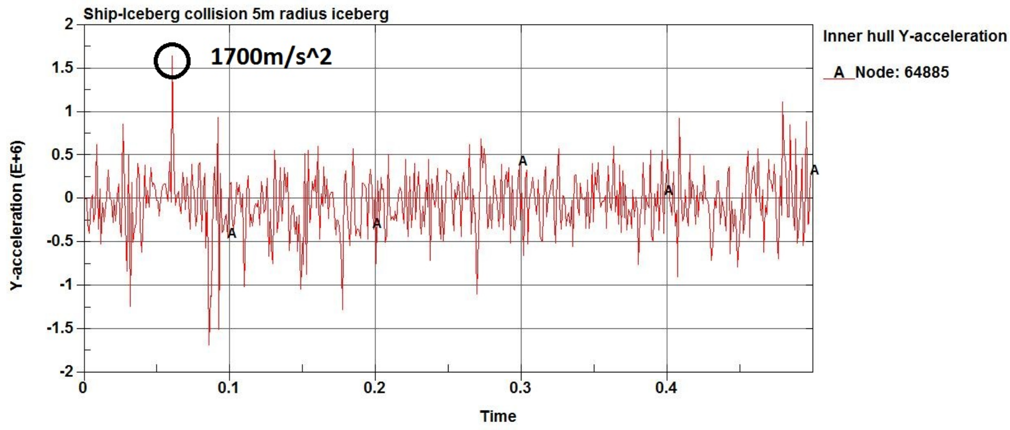

Figure 39: Acceleration of the inner hull, central position

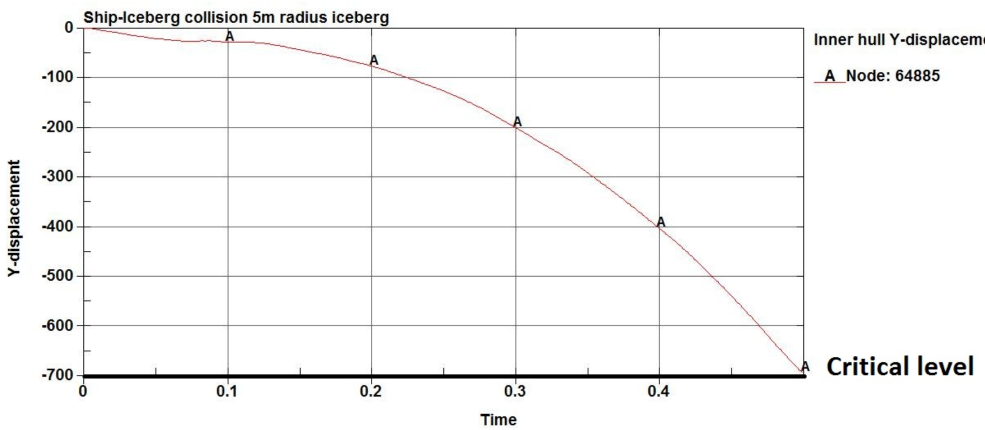

Figure 40: Displacement of the inner hull, central position

7.7 Local distribution of strain energy

As expected most of the internal energy is dissipated in parts of the structure closest to the impact zone, but as the deformations increase the strain energy is being dissipated in larger areas of the structure. The plot in Figure 41 shows the results of the 5 meter radius iceberg collision, where the fracture criterion in the steel material model is not applied. The curve is assumed to represent the strain energy distribution of the other collision scenarios as well. The curves are somewhat misleading when it comes to deformations, since they shows the absolute value of strain energy while the parts vary in size.

The greatest contributors are the outer plating and the stringers. After about 0.2s the outer plates are by far the largest contributor, but towards the end of the simulation the stringers have a sharp increase in internal energy. This is probably a result of the deformation zone increasing in size and the forces are transferred to the stringers. The vertical ice stiffeners and the so-called ice stringer in the middle of the ice belt are large contributors as well. The strain energy dissipated in the erosion of ice elements can also be seen. The total strain energy dissipated in the iceberg is thus a combination of the current internal energy and the eroded internal energy. The total iceberg strain energy peaks at around 6MJ at the end of the simulation. This is not in perfect correspondence with the force-deformation curve. A quick estimate of the area under the curves in Figure 32 show that the area under the ship curve is no more than 6-7 times that of the ice. In comparison, the ratio between the total strain energy and the energy being dissipated in the ice is 72/6=12. The method using the force-deformation curves is a very crude and simplified method, and the energy values taken directly from LS-DYNA are surely the most accurate.

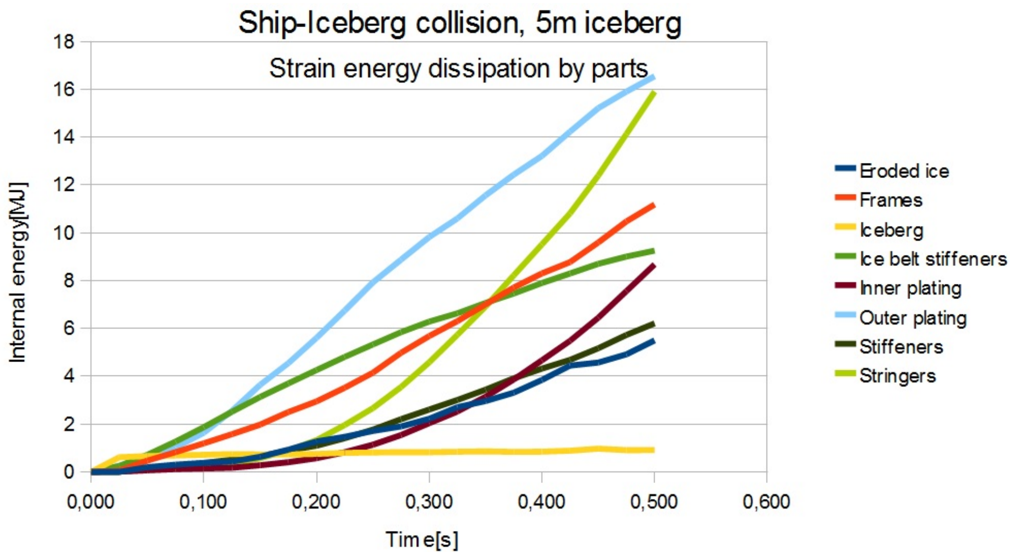

Figure 41: Internal energy by parts

8 3D external mechanics analyses

The waterline angle, α of the ship at the foremost frame in tank 1 is found to be approximately 25 deg. The waterline is halfway between stringers 2 and 3, and the angle is determined by comparing the angles at the frame of impact. When it comes to frame angle, β, it can be found directly from the drawing of frame 123 and found to be 20 deg.

The point of impact is defined from the centre of gravity of the ship. This was not given, so the centre of gravity is assumed to be located in the middle of tank 3, and just below the ice belt in the vertical direction. The same assumptions was made in [14].

The collision point is in the forward part of tank 1, i.e just to the rear of the collision bulkhead. As seen in Figure 42 the coordinates in the local coordinate system for the ship are: $[x = 110m, y = 20m, z = 4m]$.

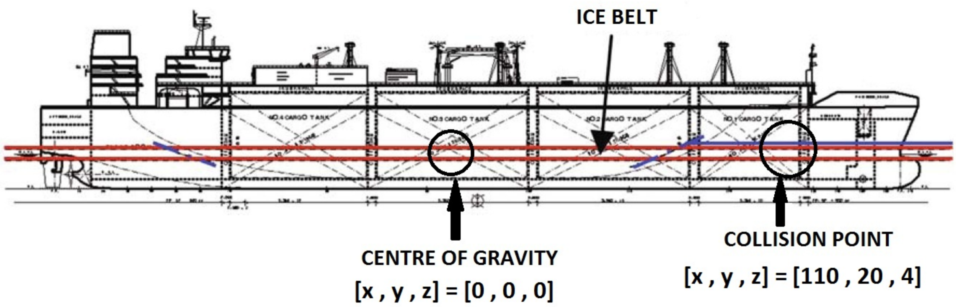

Figure 42: Profile view of 4-tank membrane LNGC

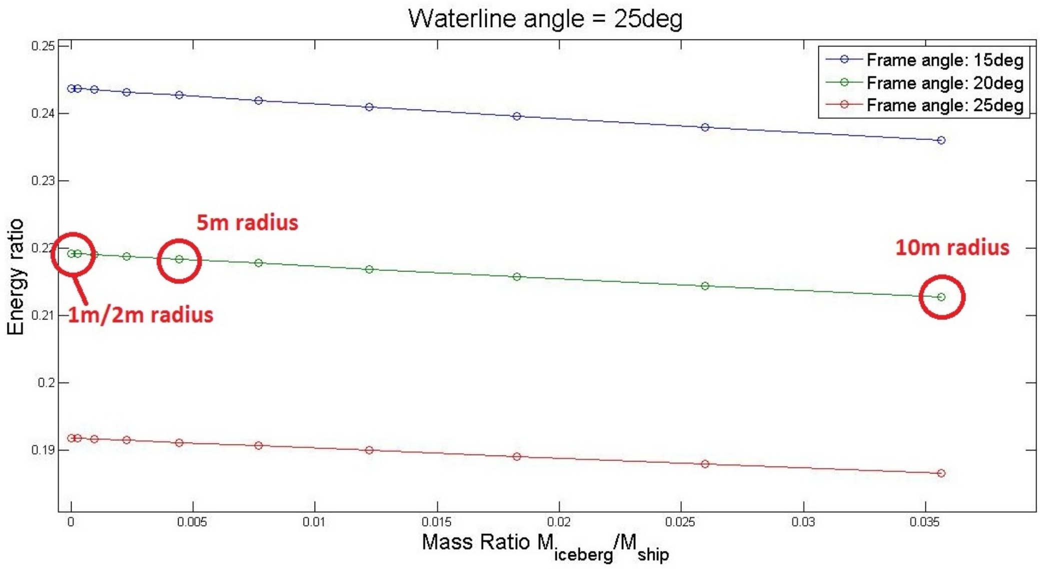

Figure 43: Strain energy ratio from MATLAB

Figure 43 shows the strain energy ratio between the actual collision energy and the energy from a head-on plastic impact scenario, E_0. The three different curves are for different frame angles, 15, 20 and 25 degrees. The middle curve of 20 degrees is closest to the real life geometry.

Iceberg radius[m]	Iceberg mass[tons]	$E_0[MJ]$	Energy fraction	Required total strain energy dissipation[MJ]
1	3.8	0.283	0.22	0.062
2	30.2	2.26	0.22	0.497
5	471	35.2	0.22	7.74
10	3770	273	0.21	57.33

Table 4: Strain energy values from Liu's external mechanics model

When comparing the values in Table 4 to the LS-Dyna energy levels required to reach the inner hull failure criterion, the energy values for the three smallest icebergs are very low. The largest 10 meter iceberg is on the other hand closer to the critical strain energy level of approximately 70MJ. This level is found in LS-DYNA for both 5 meter and 10 meter radius icebergs.

In order to see which conditions it will take to reach all the way up to

70MJ, some parameters must be changed. When it comes to the velocity of the ship, this is already set at cruising speed. Thus the size of the iceberg is the probably best way to increase the impact energy. It should be said that icebergs which are significantly larger than 5 meter radius will probably be detected prior to impact. On the other hand there are so many shapes and sizes of iceberg that some might have a low sail height and still have a dangerously large mass. In any case the critical spherical iceberg size seems to have the following values.

Iceberg radius[m]	Iceberg mass[tons]	$E_0[MJ]$	Energy fraction	Required total strain energy dissipation[MJ]
11	5018	359	0.21	75.39

Table 5: Critical iceberg size

Iceberg radii beyond 11 meters have a dramatic effect on the collision energy. This is expected since the kinetic energy ($E_k = \frac{1}{2}mv^2$) increases with mass, and the mass of a sphere($m = \rho\frac{4}{3}\pi r^3$) is proportional to r^3.

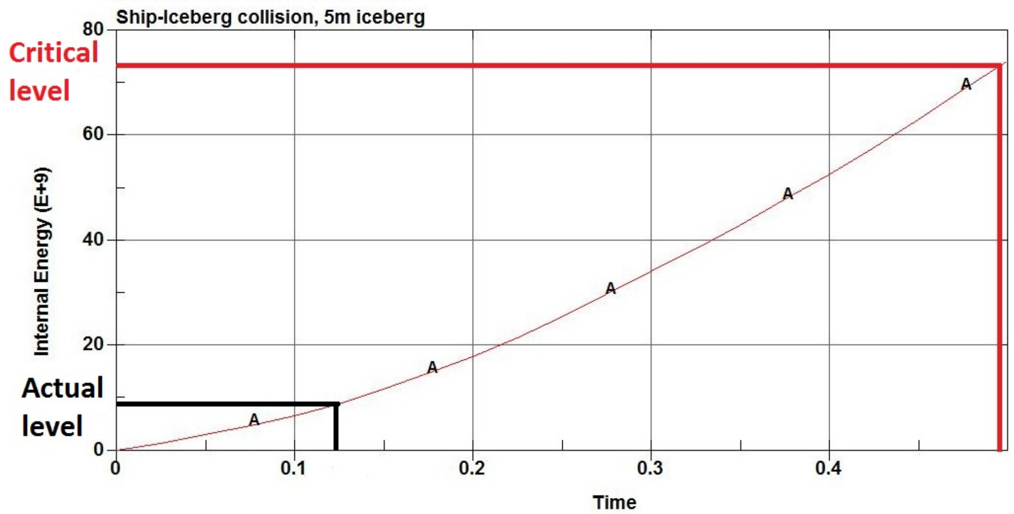

Figure 44: Internal energy plot from LS-DYNA, compared with levels found from 3D external mechanics

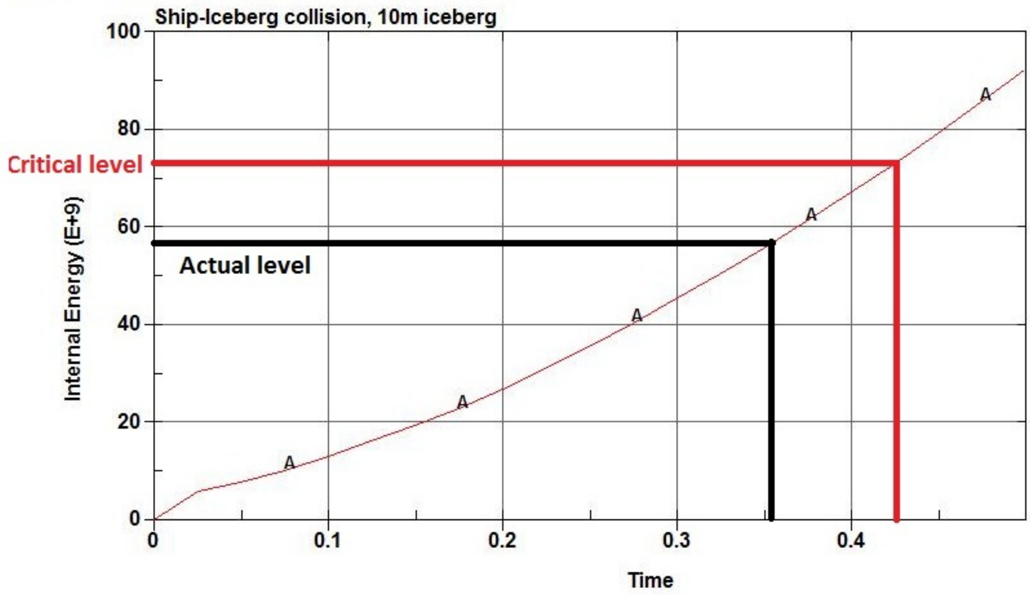

Figure 45: Internal energy plot from LS-DYNA, compared with levels found from 3D external mechanics

9 Conclusion

The real world collision point is assumed to be the worst case scenario. Any further forward and the collision bulkhead will probably absorb most of the impact. Further to the rear and the iceberg is deflected off more easily. In the FEM analyses the collision point is moved towards the centre of the tank. This is to not be influenced by the boundary conditions. The geometry of the tank in the central position is assumed to represent the forward part, since the frames are quite similar along the tank. The failure criterion for the cargo containment system is based on deflection in the middle of the tank. Therefore, it is not completely accurate to use this criterion to represent the actual collision.

The iceberg model seems to behave well, and the erosion of elements using Zhenhui Liu's material model appears to simulate crushing in an acceptable manner. There is some back erosion in the iceberg on the interface between rigid material and ice material. This can probably be avoided by increasing the size of the ice material, and thus reducing the influence from boundary conditions. On the other hand, the computational time is largely influenced by the number of solid ice elements. As long as the crushing of ice in the iceberg/ship interface is much more extensive than the back erosion, this back erosion is not of major importance.

Compared to the rigid iceberg model, the iceberg with ice material shows a drastic reduction in deformations of the ship structure. The undeformable 2m radius rigid iceberg reaches the inner hull failure criterion after around 0.45s. The real iceberg only pushes the inner hull in about a third of the way to the failure criterion in 0.5s. This indicates that the rigid iceberg is not suited for analyses with these small icebergs. For larger icebergs on the other hand, the difference is expected to be smaller. The collision force is spread out on a larger area and the crushing of the ice is less extensive. This can be one of the topics for further investigation, since a rigid iceberg reduces the computational time drastically.

When it comes to the actual energy dissipation and behaviour of the ship structure during the collisions, the energy dissipation is below the critical levels for all icebergs. The energy levels are found by means of Liu and Amdahl's 3D external mechanics model. The conclusion appears to be that for all icebergs tested during the work on this thesis, the actual energy levels are far below the critical level. The critical level refers to the strain energy needed to reach the inner hull failure criterion of 700mm deflection in the middle of the tank. However, for the biggest 10 meter iceberg, the margin is not very large. Still, the conditions in this collision scenario are very conservative. It is expected that such large icebergs will be detected well before impact, and the velocities involved should be lower than the cruising speed. A natural topic of discussion is whether the 2 meter sailing height criterion is totally realistic. But with no real insight into the accuracy of radars or other warning systems, it can only be assumed that this limit is absolute. With this in mind the 10 meter iceberg scenario might not be as close to the limit of failure after all. The critical iceberg size found is 11 meter radius.

This steel material model with RTCL fracture criterion implemented is more conservative when it comes to deformations. The energy level required to reach the 700mm deflection criterion is slightly lower than the regular power-law steel. The boundary conditions are also more realistic with the RTCL-steel, since elements are removed when they reach their ultimate capacity.

The accelerations are not even close to the 2000g that some studies have found. Some peaks in the acceleration plots for the inner hull are more than 2000m/s2, or around 200g, but the corresponding displacement curves have no sudden changes. The highest peaks in the accelerations seem to be more of numerical nature than a real life problem.

The boundary conditions appear to be a good approximation as long as the deformations not are very large. The steel material with fracture criterion seems to be a better model for large deformations.

10 Further work

There are several areas where simplifications have been made which could be investigated further.

BWH stability criterion for steel

The BWH (Hill-Bressan-Williams)-criterion is an alternative to the RTCL-criterion, and is a simplified way of determining the onset of local necking. It is a combination of a local necking analysis and a shear stress criterion. The criterion searches for local instabilities. Therefore it applies only to membrane stresses and strains, and the effect of bending is not included.

Initial velocity

A somewhat more realistic approach to Finite Element collision analyses will be to give the iceberg and/or ship model initial velocity instead of prescribed displacement. This means that the iceberg is not pushed relentlessly into the ship side, and the iceberg is free to move after the impact. The mass, inertia of the iceberg and friction between ship and ice now become important parameters.

Boundary conditions, sensitivity analysis

The boundary conditions is probably the most difficult to get right when a FE-model is cut out of a larger structure. Therefore more analyses should be done on different boundary conditions, such as fixing the frames to where the double bottom would be.

Ship model size

The most realistic results will be obtained if the model itself is increased in size. The influence from the global stiffness of the ship is more realistic, and the boundary conditions are not as important for the results. What speaks against this is the modelling time, since modelling for FEM-purposes is very time consuming. The question is then whether the small improvement in the results will make up for the time used in the modelling process. An increase in model size will also affect the computational time.

Iceberg shape

The iceberg shape used in this thesis is a highly simplified shape. No icebergs in nature come with a perfect spherical shape. Though the curvature may be realistic, further analysis should investigate the effect of changing the geometry.

Iceberg material parameters

Since ice is a very unpredictable material, different material properties should be used in order cover many different types of ice. This could either be done by changing the properties for the ice, or changing the strength of the steel material on the ship. The latter is what Zhenhui Liu has done.

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[3] H. S. Alsos, J. Amdahl, and O. S. Hopperstad, "On the resistance to penetration of stiffened plates, Part II numerical analysis," *International Journal of impact engineering*, 2008.

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[8] Lee, Lee, and Baek, "Structural safety assessment in membrane-type ccs in lngc under iceberg collisions,"

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[12] M. Storheim, "Analysis of structural damage of tankers subjected to collision," Master's thesis, NTNU, 2008.

[13] DNV, 2011. Rules for classification of ships Pt.5 Ch.1.

[14] S. A. Myhre, "Analysis of accidental iceberg impacts with membrane tank LNG carriers," Master's thesis, NTNU, 2010.

A Appendix

A.1 Acceleration plots

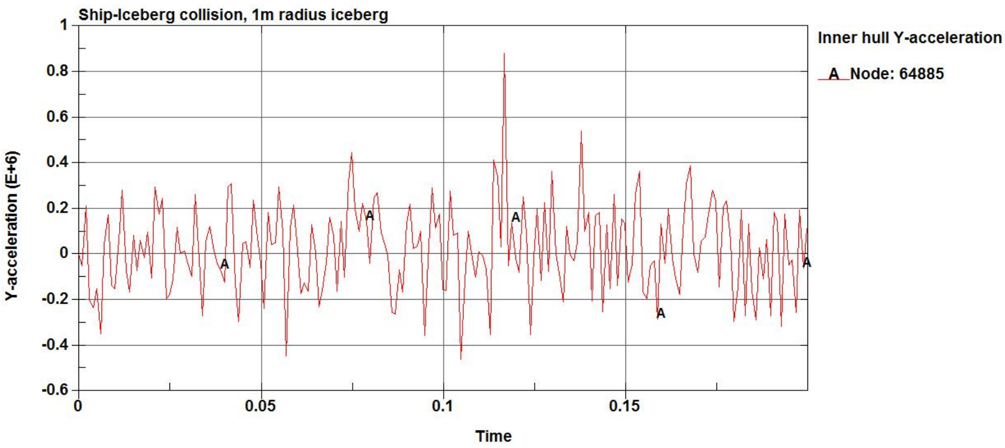

Figure 46: Y-acceleration 1m iceberg

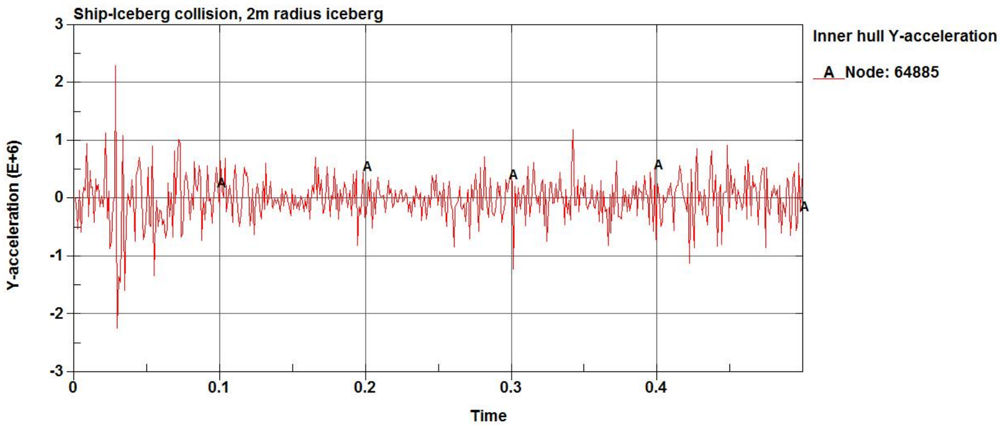

Figure 47: Y-acceleration 2m iceberg

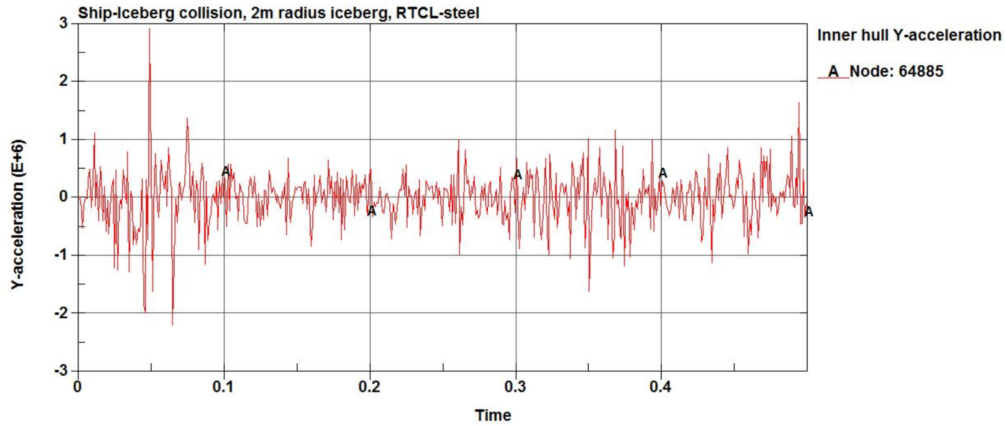

Figure 48: Y-acceleration 2m iceberg, RTCL-steel

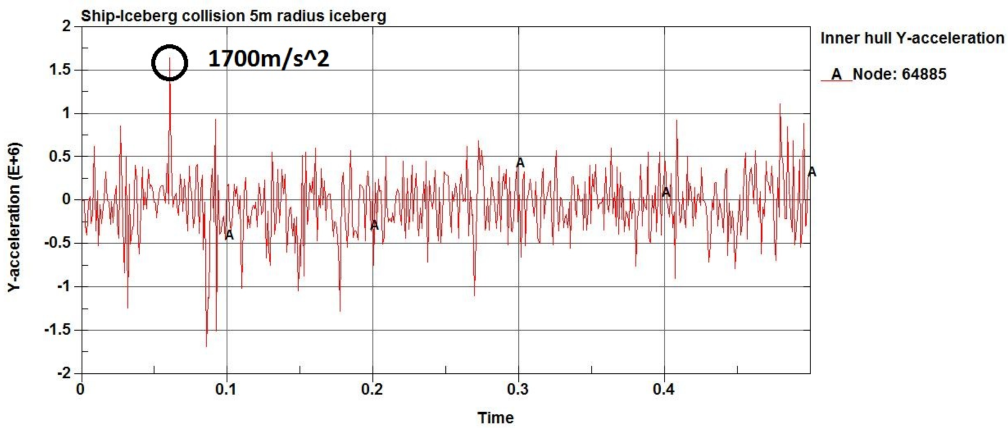

Figure 49: Y-acceleration 5m iceberg

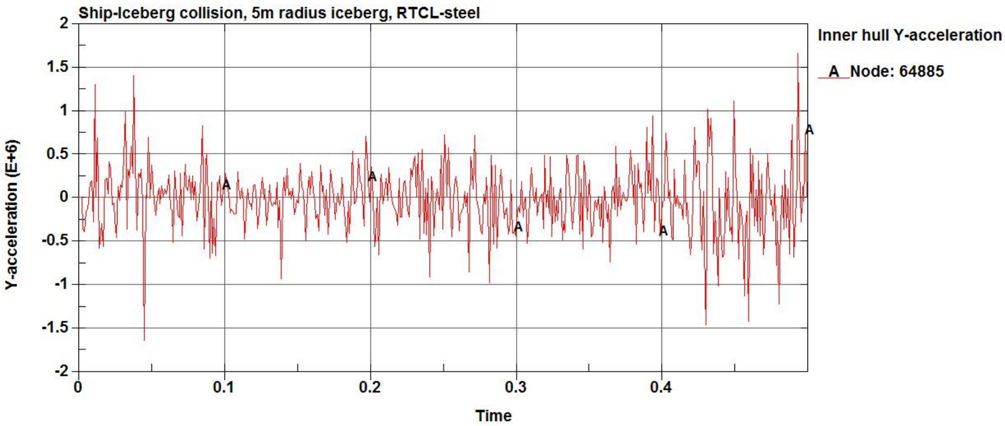

Figure 50: Y-acceleration 5m iceberg, RTCL-steel

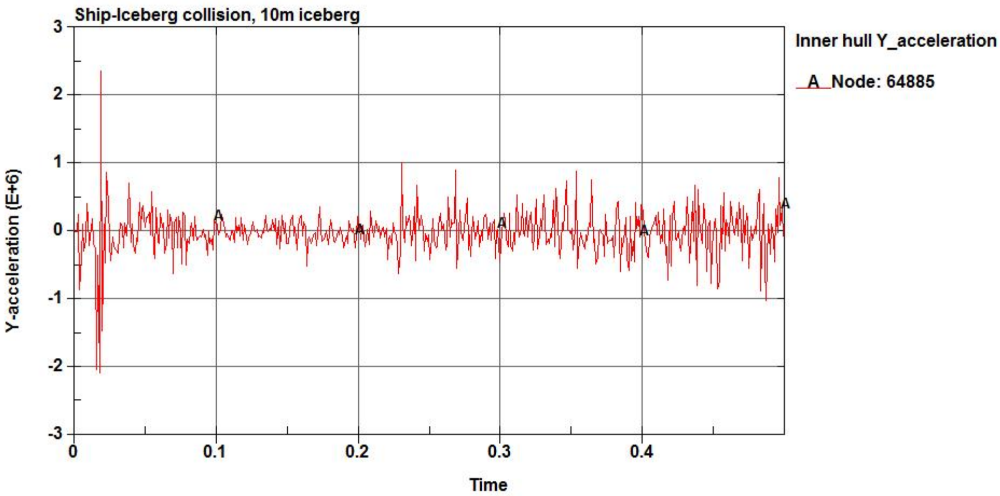

Figure 51: Y-acceleration 10m iceberg

A.2 Displacement plots

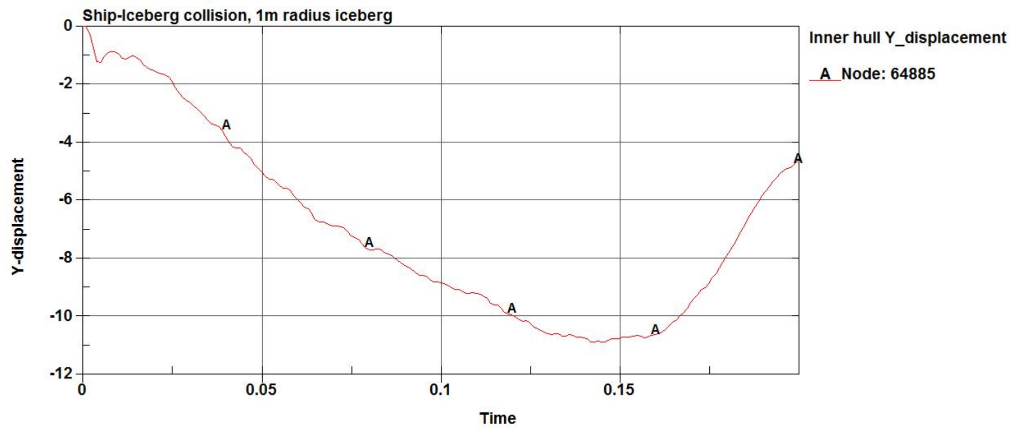

Figure 52: Y-displacement 1m iceberg

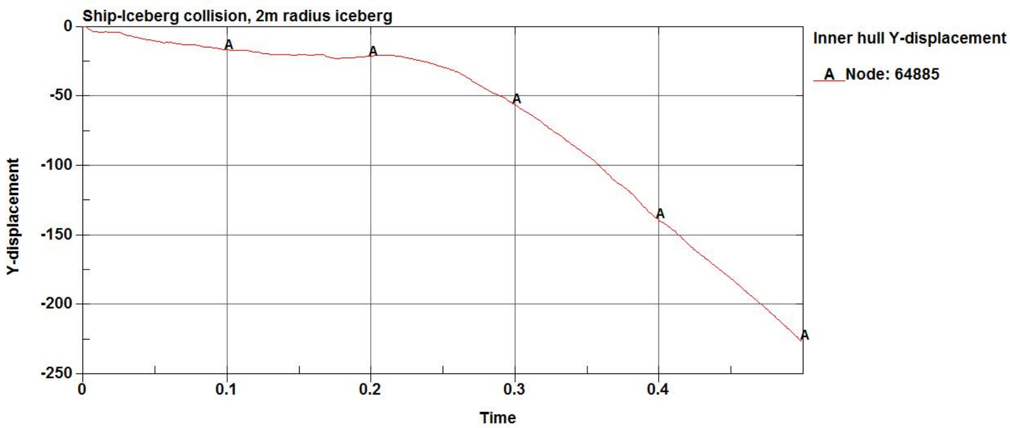

Figure 53: Y-displacement 2m iceberg

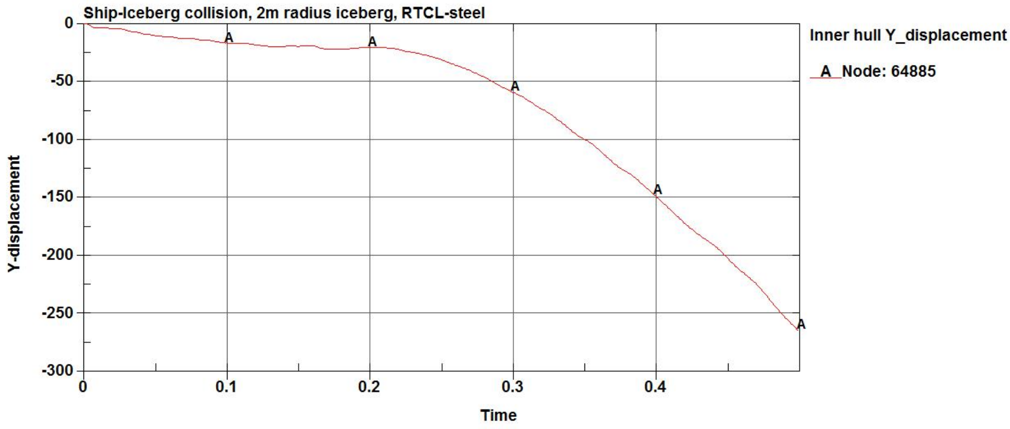

Figure 54: Y-displacement 2m iceberg, RTCL-steel

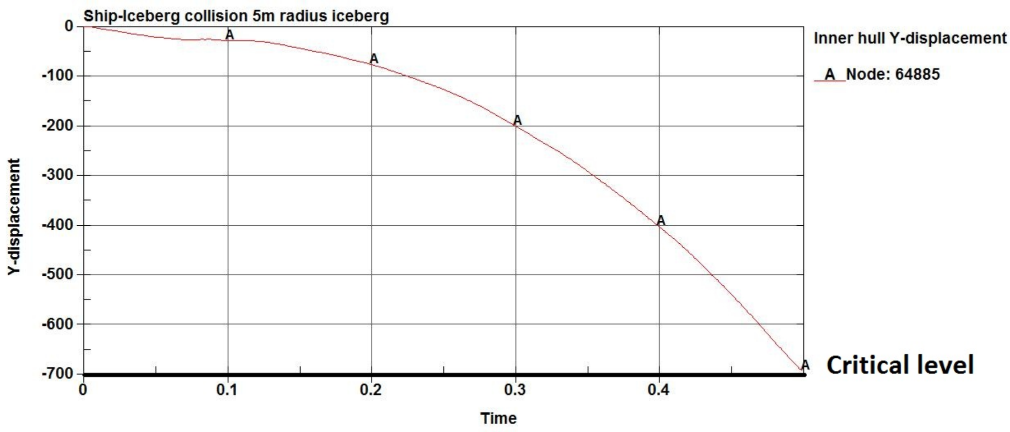

Figure 55: Y-displacement 5m iceberg

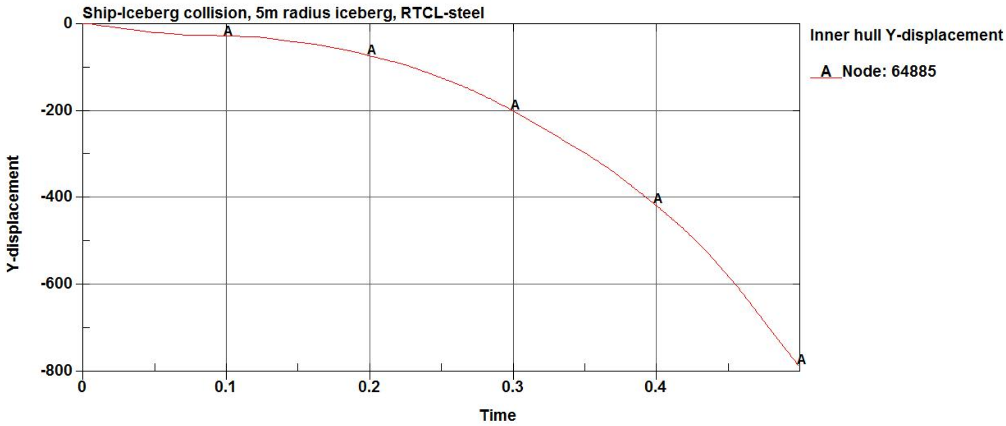

Figure 56: Y-displacement 5m iceberg, RTCL-steel

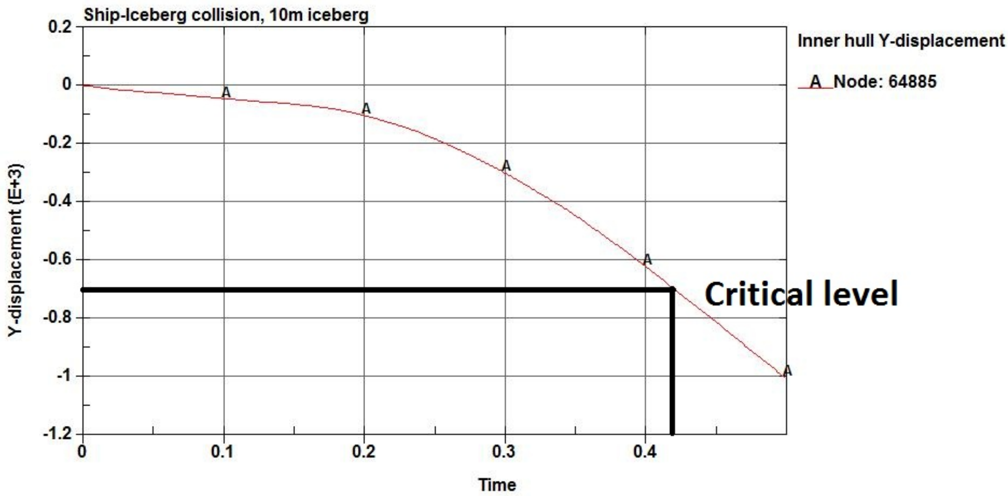

Figure 57: Y-displacement 10m iceberg

A.3 Internal energy plots

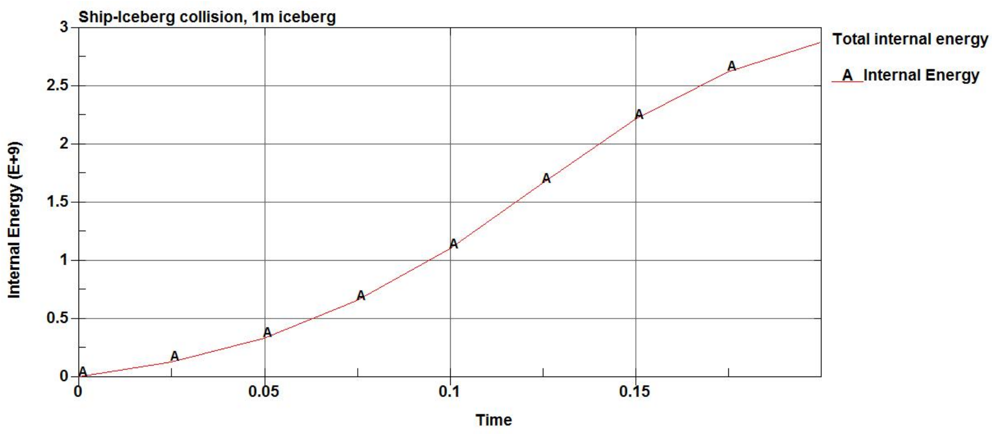

Figure 58: Internal energy 1m iceberg

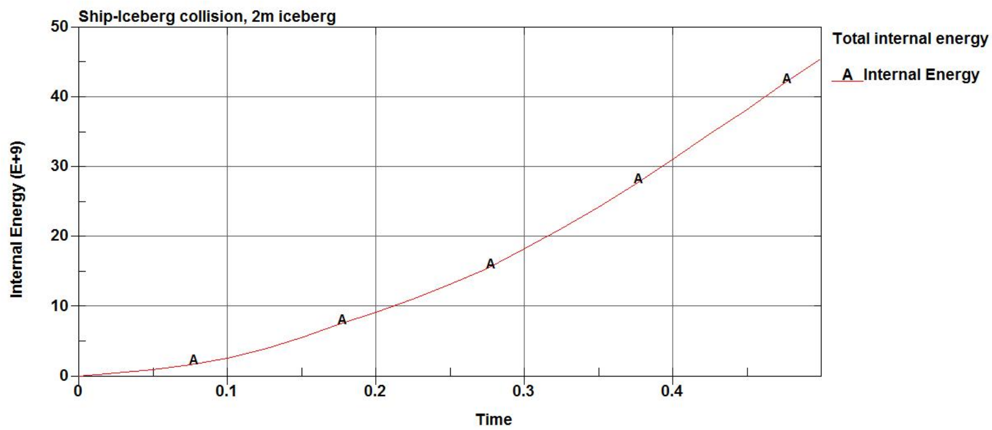

Figure 59: Internal energy 2m iceberg

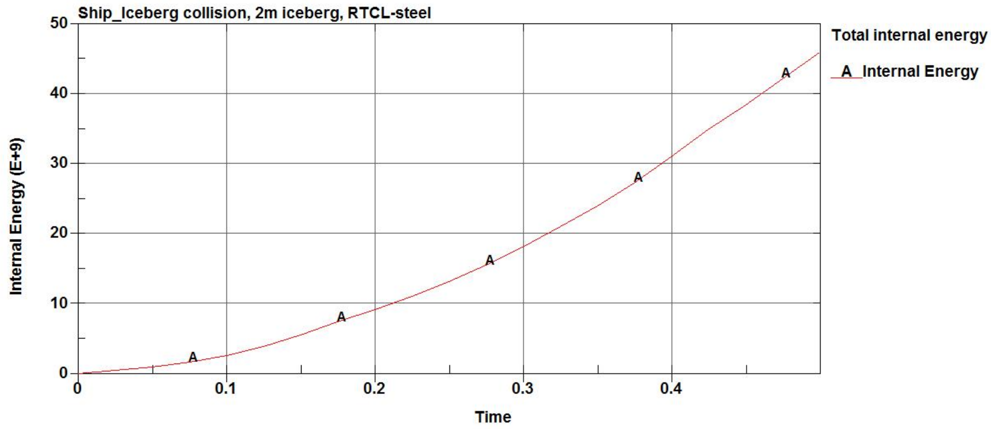

Figure 60: Internal energy 2m iceberg, RTCL-steel

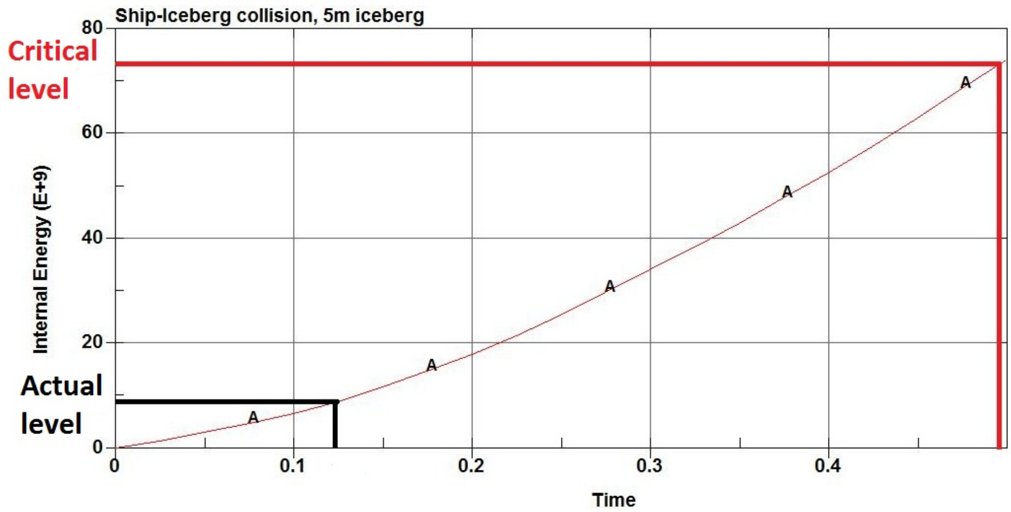

Figure 61: Internal energy 5m iceberg

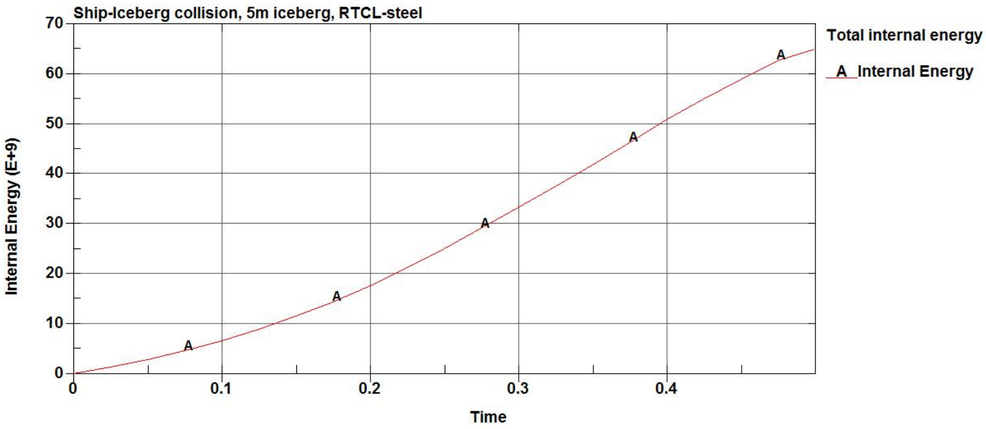

Figure 62: Internal energy 5m iceberg, RTCL-steel

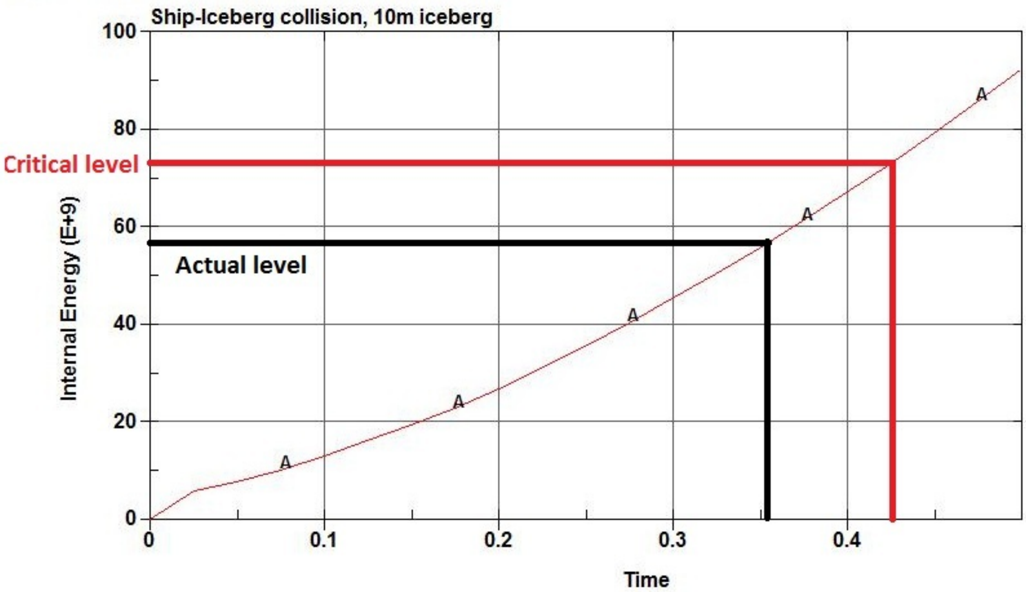

Figure 63: Internal energy 10m iceberg

A.4 Enclosed DVD

The disc enclosed at the back of the report contains the following:

- The report
  - MasterBerling.pdf
- Movie files of the simulations
  - 1m_iceberg.avi
  - 2m_iceberg_1.avi
  - 2m_iceberg_2.avi
  - 5m_iceberg_1.avi
  - 5m_iceberg_2.avi
  - 10m_iceberg_1.avi
  - 10m_iceberg_2.avi
- LS-DYNA key files
  - 1m_iceberg.key
  - 2m_iceberg.key
  - 2m_iceberg_fracture.key
  - 5m_iceberg.key
  - 5m_iceberg_fracture.key
  - 10m_iceberg.key
- MATLAB file, external mechanics
  - 3D_external_mechanics.m