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Investigating the elastic and plastic behaviour of a steel caisson subjected to ship impact

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Abstract

A four-fold increase in global sea traffic over the past twenty years has led to an increased risk of ship impact on dock infrastructure. Ship impact is a highly non-linear process and it is important that the elastic and plastic behaviour of caissons subjected to ship impact is well understood so that they can be sufficiently designed against it. This project used LUSAS finite element analysis software to perform a time history analysis of a proposed steel caisson subjected to five different ship impact loads and from the results compares the behaviour of each impact case to gain a better understanding of impact behaviour. The main conclusion is that increased ship mass means an increase in kinetic energy which leads to increased plastic behaviour within the caisson. The maximum plastic strain within the caisson was 0.15% and the permanent deformation was 2.70mm. Another conclusion was that the proposed steel caisson is sufficiently designed to safely withstand ship impact from a 1500 tonne vessel traveling at 0.5m/s with only local plastic deformation occurring.

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1. Introduction

1.1 Introduction

The redevelopment of an existing Edwardian dry dock located in Devonport Dockyard, Plymouth and constructed circa 1908 has led to the design, construction and installation of a new steel caisson. The caisson's purpose is to retain a large amount of water from flooding into the dry dock whilst a vessel is on the dock bottom undergoing work. The water retained by this caisson equates to a high amount of potential energy. If an impact scenario were to occur which led to a sudden failure in the caisson the effects of the retained water flooding into the dock would be catastrophic and would undoubtedly lead to the death of all workers in the dock bottom. This is a current issue because ports are getting busier which has led to an increased risk of ship collision with dock infrastructure. It is important that the elastic and plastic behaviour of caissons subjected to ship impact is well understood so that they can be sufficiently designed against it.

The protection of structures against accidental actions has come a long way in the past 20 years which is most likely due to the increased threat of malicious attacks from terrorism (Jones, 2011). The current procedure for the design of unprotected dock structures in the UK is governed by Eurocodes, where the collision force is determined as an equivalent static load which is related to both the approach velocity and dead weight tonnage (DWT) of the ship. Unfortunately, ship impact is a highly non-linear process meaning the calculation of these impact forces and subsequently the overall structures response isn't exact. This non-linearity is primarily due to the considerable number of different factors involved within ship impact. This uncertainty related to ship collision makes it a major risk to marine structures and the safety of any employee working in the dock bottom (Pedersen, 1993). It is important to be able to say with a high degree of certainty that the structure can withstand any 'likely' impact. If this is not possible, then it is important that if the structure did fail, it would be a ductile failure and not a catastrophic failure. Hence ensuring that dock workers have sufficient time to escape and are operating in the safest working environment possible.

This project uses LUSAS which is a finite element analysis software used extensively throughout the industry to perform a time history analysis of a steel caisson subjected to ship impact. The numerical model is based upon the design of the proposed caisson within Devonport Royal Dockyard, Plymouth. The caisson models were tested parametrically under 5 varying impact loads to identify any patterns in the elastic and plastic behaviour of the steel caisson. The LUSAS output values were validated by comparing the force acting upon the caisson due to the impact calculated by LUSAS and the theoretical force calculated via the kinetic energy equation. An in-depth explanation and summary of this validation can be seen in Section 4.1.

Section 1.3 will outline the structure and function of the steel caisson to be built within Devonport Dockyard that forms the basis of this project. It will also give details on the redevelopment project currently in progress and the main drivers for the design of the new caisson. Chapter 2 gives an overview of the current literature surrounding ship impact on dock infrastructure. Chapter 3 describes the methodology used to create the numerical model in LUSAS, it also shows how the analysis was run. Chapter 4 presents the results and shows how the model was validated to confirm its reliability. Chapter 5 discusses the results presented in the results section. Chapter 6 presents the conclusions of the analysis and discussion. Section 6.2 makes recommendations for any future research in the field.

1.2 Aims and objectives

This project aims to investigate the elastic and plastic behaviour of a steel caisson when it is subject to ship impact and confirm that the caisson has been sufficiently designed to withstand any likely impact. In order to achieve the aim, the project has four main objectives:

- To create and run a numerical model of the caisson using finite element analysis and parametrically test the model under 5 different impact loads.
- Validation of the model results to ensure that the values are correct, accurate and reliable using kinetic energy and work done equations.
- Analyse and compare the results from all impact cases to understand the elastic and plastic behaviour of the steel caisson subjected to ship impact.
- Determine whether caisson has been sufficiently designed to withstand impact from 500 – 1500 tonne vessels travelling at a speed of 0.5 m/s.

1.3 Case study (Devonport Royal Dockyard)

Devonport Royal Dockyard also known as HMNB Devonport is the largest naval base in Western Europe. It is a historic site that has supported the Royal Navy since 1691. The site covers over 650 acres and has 11 dry docks, 4 miles of waterfront, 25 berths and 5 basins (Royal Navy, 2016). The redevelopment of one of these dry docks has led to the design, construction and installation of a new steel caisson. A caisson is a large water retaining structure usually of both steel and concrete construction. They can be filled with water to create a water tight seal at the dock entrance allowing water to be pumped out of the dry dock and work to commence.

Unfortunately, due to the sensitive nature of Devonport Dockyard being a nuclear site, the exact details, name and location of the dock cannot be disclosed. The aged dry dock constructed circa 1907 has not been used in over 25 years and is currently undergoing an extensive redevelopment to bring it back into service. The new caisson is an essential part of the infrastructure required to perform any refit and maintenance work within the dock and it is crucial that it is sufficiently designed against ship impact. The numerical model that will form the basis of this investigation has been modelled from the exact design of the proposed caisson.

Figure 1: Redevelopment project and existing steel caisson

The proposed caisson will be 29.4m high, 17m wide and 6.2m deep and will slide into the dockside when vessel movement is required. It will be built by a Dutch company called Ravenstein who specialise in large steel constructions and will be constructed out of a mix of S355 grade steel plates and both vertical and horizontal stiffeners.

The caisson has been designed to withstand all the following actions:

- Self-weight of steel caisson.
- Hydrostatic loads assumed as the full caisson height on one side only as this would be the worst-case scenario.
- Ambient temperature changes (especially important in fully steel structures).
- Localised Drop loads of 10 tonnes from 30 metres.
- Vessel impacts of 0.25 knots for a DWT of 20000 tonnes.
- Seismic loads including hydrodynamic water and inertia of actions.
- Wind loading on the caisson for a wind speed of 42 m/s.

Table 1: Thickness of caisson plates and equivalent steel sub-grades

Thickness (mm)	Grade of Steel
$0 < t \leq 50$	S355 J2
$50 < t \leq 66$	S355 K2
$66 < t \leq 150$	S355 K

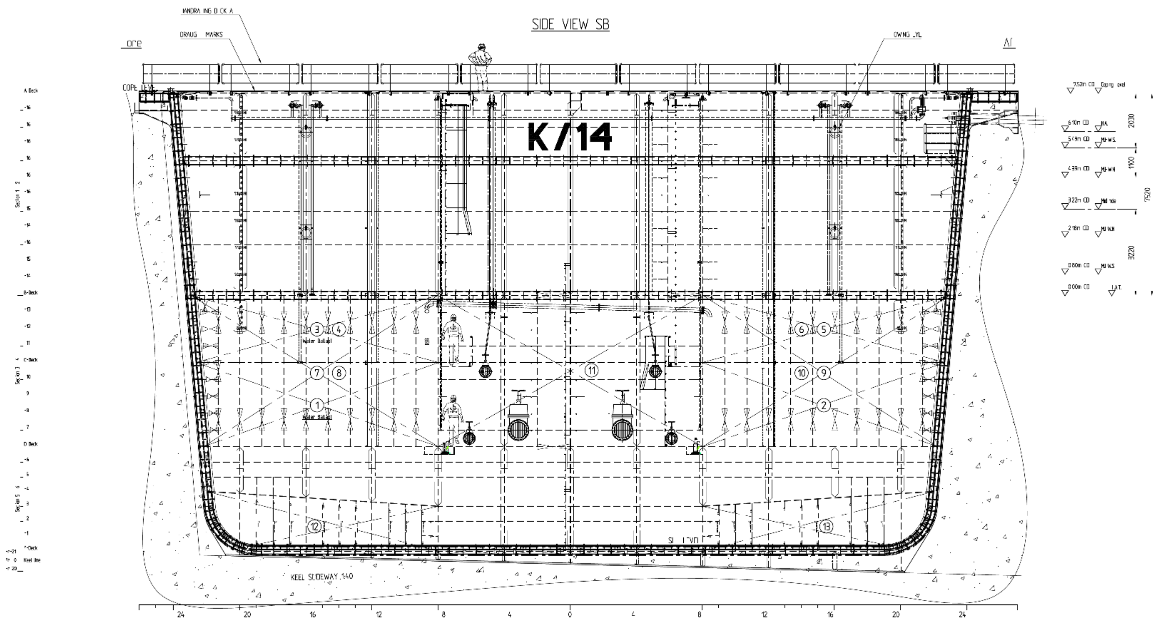

Figure 2: General arrangement drawing of proposed steel caisson

2. Overview of ship impact on dock infrastructure

2.1 Introduction

Investigating the behaviour of dock infrastructure such as steel caissons subjected to ship impact is an important topic that has little literature currently surrounding it. Past events such as the Dubai Dry Dock disaster which will be outlined in the next section and the increase in both global sea traffic and ship size (Calle and Alves, 2011) clearly show that this is a dangerous and current issue that is only going to get worse. Through continued research, dock infrastructure can be designed more efficiently to keep workers and assets safe in the future.

This literature review will firstly explore the importance of understanding ship impact on marine structures and highlight a relevant case study. It will then identify several types of loading and how they affect structures differently. The next section will focus on the stress, strain and Young's modulus properties of structural steel and outline mechanics behind ship impact. Next it will outline current design procedures in the UK and EU in regard to ship impact design. It will then explore and critically evaluate the variety of different approaches used to analytically predict ship impact forces. Finally, it will explore relevant papers surrounding ship impact and how it can be modelled using finite element analysis. This review will primarily study dock infrastructure subject to impact such as dock walls and caissons.

2.2 The importance of understanding ship impact on dock infrastructure

An analysis on global ship density performed by (Tournadre, 2014) showed that between 1992 and 2012 there was a fourfold increase in international shipping traffic. To meet this demand there are currently around 50,000 merchant ships trading internationally (International Chamber of Shipping, 2015). In addition to just merchant traffic, passenger vessels also make up a substantial proportion of global sea traffic. A statistical release from the Department of Transport, showed that there were 23.2 million sea passengers on all international routes to and from the UK in 2014 (Great Britain. Department for Transport, 2016). In addition to the increase in sea traffic, container ships are also getting bigger. Early container ships in 1956 were around 137m in length, this number has almost quadrupled in the last 60 years with the latest line of Maersk's "Triple E" ships being 400m long (Kremer, 2013). Each of these vessels be it freight or passenger come into ports all around the world, and increased traffic and ship size is increasing not only the likelihood but also the severity of collisions on dock infrastructure.

It is important to understand the behaviour of marine structures such as caissons subject to ship impact to be able to say with a high degree of certainty that the structure can withstand any 'likely' impact. If this is not possible, then it is important that if the structure did fail, it would be a ductile failure and not a catastrophic failure. Hence ensuring that dock workers have sufficient time to escape and are operating in the safest working environment possible. This is required by employers under regulations set out by the Health and Safety Executive:

"Employers must do whatever is reasonably practicable to ensure that workers and others are protected from anything that may cause harm, effectively controlling any risks to injury or health that could arise in the workplace" (Health and Safety Executive, 2016).

To ensure that appropriate safety measures have been taken, employers are required to create risk assessments, this uses a likelihood vs severity approach. Obviously, a caisson collapse would be a high severity scenario, subsequently meaning appropriate action should be taken to ensure that the likelihood of a collapse is as low as “reasonably practicable”.

2.2.1 Dubai dry dock disaster

On the 28th March 2002, two sections of a dry dock gate collapsed allowing a 12-metre high tidal surge to crash in, fully submerging a number of large vessels undergoing repair and killing at least 22 workers (BBC, 2002). The incident occurred within Dock 2 at Dubai Drydocks in the United Arab Emirates which is the largest dry docking facility in the Persian Gulf. The yard has repaired over 7,500 vessels since its opening in 1983 (Dubai Drydocks, 2012). It was reported that the 100m long, 12m high gate had been struck by a barge several days before the disaster. This caused severe damage to four of the 28 props which support the gate (Lovelace, 2002). The function of the props is to transfer the hydrostatic load acting on the dock gate directly into the dock bottom, reducing the required thickness of the gate.

Haigh (2002) commented: “It is likely that workers failed to provide enough temporary support for the gate, while carrying out the repairs”. This quote highlights the importance of understanding the behaviour of marine structures subject to ship impact so that disasters like this can be avoided. If a failure like this can occur under a tidal surge it is just as likely to occur under ship impact, the effects of which would be catastrophic. This is an important issue not only for the health and safety of workers in the dock bottom, but also to minimise the financial costs of damage to high value vessels.

2.3 Different types of loading

To ensure safe and economic design, it is very important to estimate the loads that the structure will be subjected to over its lifetime (Durka et al, 2010). It is also just as important to know how the load will be applied, because the behaviour of the structure varies under different load types (Roark et al, 2012). The behaviour of buildings under varying load applications has become more of a concern for designers in the last 30 years due to an increase in slender elements and lighter cladding (Kappos, 2002). Incidents such as the partial collapse of Ronan Point in 1968 have highlighted the importance of understanding different loading types. A gas explosion blew out two load bearing walls which led to the collapse of the entire corner. This incident led to an amendment in the building regulations related to structural robustness and disproportionate collapse (Durka et al, 2010). This meant increasing the industry’s understanding of different load types and ensuring structures were robust enough to withstand any likely impact loads.

The different types of loading can be separated into four main categories:

(i) Static

Static loads are applied gradually and exert a constant force making them time-independent. In structural analysis, they’re usually referred to as ‘dead loads’. Dead loads are the self-weights of structural elements and permanent items that remain in place within the structure (Paik and Thayamballi, 2003). Because the loads are applied gradually, there is assumed to be no acceleration, therefore meaning inertial effects are considered negligible in static load cases. Compared to the loads that will be examined next, static loads are considered easy to predict and design for.

One of the most common behaviours under static loading conditions is known as creep. The application of stress onto a material leads to a proportionate amount of strain within the material (more details in section 2.4). This strain increases over time despite the applied stress remaining the same (Domone, 2010). This increase in strain over time is defined as “creeping”.

(ii) Dynamic

Dynamic loads exert a varying amount of force and are highly time-dependent. They cause dynamic effects in the structure such as vibrations (Durka et al, 2010). Examples of this type of loading include: wind, earthquakes and machinery. The response of the structure is dependent on the magnitude and acceleration of the dynamic load and the stiffness and mass of the structure. Due to the acceleration of the dynamic load, inertial forces are present (see impact section for details).

(Paik and Thayamballi, 2003) identified that due to the varying nature of dynamic loads they are generally hard to accurately predict. For this reason, codes of practice use various live load occupancy classifications based on experience and proven practice

(iii) Cyclic

Cyclic loading is the application and removal of a load repeatedly over any period. The repeated cyclic loading can often cause materials to fail well below the yield stress, making it potentially a very dangerous failure. This phenomenon is referred to as fatigue (Roark et al, 2012). Common examples of cyclic loading include vehicles over a bridge or the breaking of waves on off-shore structures.

(iv) Impact

Impact loads are defined as a rapid application of force over a short time usually within the region of a few micro-seconds (Domone, 2010). Impact is the transformation of kinetic energy from the impacting body into elastic and plastic strain energy and subsequently elastic and plastic deformation within the impacted structure. A proportion of the kinetic energy is lost in the form of heat, sound, noise and vibration energy. It is important to understand that the impact force during the collision is not constant and decreases over the time of impact due to the crushing and deceleration of the impacting body. Much work has been done surrounding the approximation of these impact forces which can be found in section 2.6. Kinetic energy depends on the mass and velocity of the impacting body and is denoted by the equation:

$$E = 0.5MV^2$$

There are two common types of impact that are considered in modern structural design. The first is *soft impact*, in this case there is no rebound of the projectile because the two masses stick together to become one body after impact. The impacting body is often very deformed (crumpled) in comparison to the impacted structure, which experiences little deformation or strain (Kœchlin and Potapov, 2009). A good example of a soft impact would be the collision between an aircraft and a reinforced concrete structure (i.e. an impactor of large mass and relatively low velocity). The second is *hard impact*, in this case the projectile is very rigid in comparison to the impacted structure and the projectile often sustains little damage upon impact (Kœchlin and Potapov, 2009). An example of a hard impact would be a bullet impacting a steel plate (i.e. an impactor of low mass and relatively high velocity).

The response due to impact can also be separated into two types: local and global effects. The local damage is likely to be penetration (hard impact) or perforation (soft impact) from the impacting body, cratering or depression on the impacted structure and scabbing or bulging on the opposite side from the impact (Watson, 2002). The global effect is likely to be the dynamic response of the structure to the loading such as oscillation and vibration until it returns to its steady state.

The main factor that separates impact and dynamic loading with static loading is the effects of inertia which is defined as the resistance of an object to its change of motion. Inertia comes from Newton's first law which states that "every object will remain at rest or in uniform motion in a straight line unless compelled to change its state by the action of an external force". The resistance of the impacting body due to its change of velocity upon impact consequently means that more force is applied by the impacting body. This means that an element of a structure that would be fine under a static could in fact fail under an impact load of the same magnitude. This is obviously dangerous and highlights the importance of understanding the different methods of loading and how they behave differently. Another danger with impact loading is the increase in brittle behaviour, leading to a brittle failure in a usually ductile material (such as steel). This behaviour is most common in high velocity impacts (Domone, 2010).

2.4 Stress, strain and Young's Modulus

Within a structure, most materials operate within their elastic region. This is purposefully done by design engineers to ensure that the structure is as safe as possible, but due to factors outside of their control (accidental actions such as impact) this isn't always the case (Domone, 2010). The application of any load onto a structure generates stress and strain within it. In order to investigate the elastic and plastic behaviour of a structure under impact it is very important to first understand the relationship between stress and strain (known as Young's Modulus) and how these relate to elastic and plastic behaviour.

Stress is described as a distributed force on the external and internal surfaces of a body (Roark et al, 2012) and is defined in terms of force per unit area. When an external force (load) is applied to a body, in order for the structure to remain in equilibrium, there must be an equal internal reaction within the body. The particles within the material exert forces upon each other due to cohesion to maintain this equilibrium (this known as stress). Stresses can be separated into three types: compressive (shortening), tensile (stretching) and shear (sliding). The stress due to loading on an element is dependent on its geometrical size and material properties (Domone, 2010). This makes sense as the larger the area of application is, the more distributed the stress will be.

The effects of element size can be removed by converting the load to stress:

$$\sigma = \frac{P}{A}$$

Strain is defined as the ratio of total deformation to original length. It is important to note that because it is a ratio, it is dimensionless. Strain is often expressed in terms of a percentage. Strain can be calculated using the equation:

$$\varepsilon = \frac{\Delta L}{L}$$

In 1660, British physicist Robert Hooke stated that the force applied to a spring was proportional to the extension of the spring, this is known as *Hooke's Law*. This law can be generalised to prove that strain within a material is proportionate to the stress applied upon it (Roark et al, 2012). This relationship is governed by Hooke's Law and only applies within the elastic region of the stress-strain curve:

$$F = K\,.\,x$$

where:

$F\ =\ Force\ (Stress)$

$K\ =\ Stiffness\ (Young's\ Modulus)$

$x\ =\ Deformation\ (Strain)$

Can be rearranged to give:

$$E = \frac{Stress}{Strain} = \frac{\sigma}{\varepsilon}$$

This relationship is most commonly referred to as Young's Modulus but it is also known as the modulus of elasticity or elastic modulus (Morrow and Kokernak, 2011).

It is commonly described as the stiffness of a material and has the same units as stress due to strain being dimensionless. Steel is a linearly elastic material, meaning the stress-strain curve is linear within the elastic region (Figari, 2014); this makes it an ideal material for construction. Examples of non-linear elastic materials include rubber and soil. The point at which Hooke's law stops governing the linear stress-strain relationship is known as the yield point. The stress-strain behaviour for most materials can be split into two distinct regions, the 'elastic region' which is before yield and the 'plastic region' which is after yield (Domone, 2010). The behaviour of these regions will be explained in the next section.

2.4.1 Elastic behaviour

Within the elastic region any strain that occurs due to the applied stress is fully recoverable, meaning once the load is removed it return to its original shape and dimensions; this is commonly known as elastic deformation. This is a desirable behaviour and all elements within a structure are designed to operate within this limit.

2.4.2 Plastic behaviour

Within the plastic region any strain that occurs due to the applied stress is non-recoverable and will result in a permanent deformation, meaning the material will not return to its initial shape or dimensions. This is commonly known as plastic deformation. Strain hardening is the increase in strength of a material due to plastic deformation, it occurs when stress is applied past the yield point. This increase in strength continues all the way up until the ultimate limit state. After which time a neck begins to occur. This is a reduction in cross sectional area due to strain after which fracture usually occurs.

Plastic behaviour can be separated into two different types, which is dependent on the material properties. Brittle materials tend to have a low Young's modulus value and are not used as structural elements in construction. This is because there is no plastic deformation before failure meaning they often fail catastrophically (i.e. no warning). Glass is a good example of a brittle material.

Ductile materials require a higher amount of strain energy before failure. Within the plastic region ductile materials give warnings of distress before any failure occurs (Domone, 2010). Steel is a good example of a ductile material.

2.5 Current design procedure for ship impact in the UK and EU

The current procedure for the design of unprotected dock structures in the UK is governed by (BS EN 1991-1-7 Annex C). The collision force is determined as an equivalent static load which is related to both the approach velocity and dead weight tonnage (DWT) of the ship. Eurocodes have adopted the ship collision analysis developed by Pedersen in 1993. Pedersen's equations for the approximation of the ship impact force also includes the added hydrodynamic mass of water. The hydrodynamic mass is the water entrained around the hull that carries on moving once the ship has stopped, exerting an extra force against the dock structure (Thoresen, 2014).

When designing structures against impact there are two approaches. The design impact force can be calculated as the *maximum* force or an *average* force. This is significant because the maximum force is approximately twice the average force (Alves and Calle, 2011). The Eurocodes calculate the impact force as the maximum equivalent static load. However, Woelke et al. (2012) believes this to be a large over prediction of the impact force. The approximate impact force is calculated by the equations:

$$F_{bow} = F_0 * \bar{L}\left[\bar{E}_{imp} + (5.0 - \bar{L})\bar{L}^{1.6}\right]^{0.5} \qquad for\ \bar{E}_{imp} \geq \bar{L}^{2.6}$$

$$P_{bow} = 2.24 * F_0\left[\bar{E}_{imp} * \bar{L}\right]^{0.5} \qquad for\ \bar{E}_{imp} < \bar{L}^{2.6}$$

where:

$$\bar{L} = L_{pp}\ /275\ m$$

$$\bar{E}_{imp} = E_{imp}\ /\ 1425\ MNm$$

$$E_{imp} = \frac{1}{2} m * v^2$$

and:

F_{bow} $is\ the\ maximum\ bow\ collision\ force\ in\ (MN)$;

F_0 $is\ the\ reference\ collision\ force\ =\ 210\ MN$;

E_{imp} $is\ the\ energy\ to\ be\ absorbed\ by\ plastic\ deformations$;

L_{pp} $is\ the\ length\ of\ vessel\ in\ (m)$;

m $is\ the\ mass\ plus\ added\ mass\ with\ respect\ to\ longitudinal\ motion\ in\ [106\ kg]$;

v $is\ the\ sailing\ speed\ (impact\ velocity)\ of\ the\ ship.$

NOTE: The impact force applies to vessels between 500 DWT - 300,000 DWT.

This formula developed by Pedersen accounts for the effects of strain rate, impact velocity, vessel loading condition and vessel size but does not consider eccentric impact or width of impacted structure (Pedersen, 1993).

2.6 Analytical methods for predicting ship impact forces

Ship impact is a highly non-linear process meaning the calculation of the impact force isn't exact. This non-linearity is primarily due to the large variety of different factors involved with ship impact. Obviously, the inclusion of all factors is unrealistic but an awareness of them is crucial when investigating impact behaviour. Some of the factors that influence ship impact forces identified by Woelke et al. (2012) are listed below:

- Angle of impact (head on collision being the worst case).
- Bulbous bow presence and shape (this governs the point of impact and whether the forces will be distributed between the nose and bulbous bow)
- Local stiffening pattern of the vessels bow (bulkheads and deck).
- Shape of the impacted structure also influence force distribution.
- Rigidity of the impacted structure (stiffer structures will cause more damage than an elastic structure)

There are a number of different analytical methods for calculating the impact force of a ship upon collision that have been developed by a number of different people over the last sixty years. The vast majority of analytical methods developed in the past focus on ship to ship collision scenarios. The following section will review some of the more prominent ship impact theories.

2.6.1 Minorsky's method

In 1959, Minorsky made the first attempt in the formulation of ship impact analysis and predicted the structural resistance required in a nuclear-powered ship hull to safely absorb the kinetic energy dissipated in a collision. He developed a linear relationship between the volume of steel deformed and energy dissipated in deformation (Pedersen, 1993). His analytical method allowed him to calculate the maximum depth of penetration from the impacting ship provided the stiffness pattern of the ship was known. The depth of penetration was an important factor in his case for predicting the integrity of the nuclear reactor space within the vessel during impact.

Minorsky's method was an empirical formula derived from data of 26 real ship to ship collision scenarios between ships of similar sizes (Alves and Calle, 2011). This means Minorsky's method is ship specific which questions its accuracy when applied to ships of varying sizes. Minorsky's formula were considered invalid at low energy impacts (below 100 MJ) due to the poor correlation between absorbed energy and resistance factor (Woelke et al, 2012). This leads to the assumption that Minorsky's method is inaccurate at low energy scenarios.

The limitations of Minorsky's work is that because it's based on total absorbed kinetic energy and measured penetration depths, it cannot be used to find accurate impact forces and only average forces over the impact (Pedersen,1993). It is clear that the application of Minorsky's analytical method would be unsuitable for this project's aims. This is primarily due to its inaccuracy for low energy impacts, that it has been fully derived from ship to ship collision scenarios and that it cannot be used to accurately predict maximum ship impact forces.

2.6.2 AASHTO specification method

In 1980, a vessel collided with the unprotected pier of the Sunshine Skyway Bridge on the west coast of Florida, USA. Approximately 400m of the bridge deck fell into the water killing 38 people (Svensson, 2009). This tragic incident identified that the USA was in urgent requirement of a design code for bridge structures against ship impact.

In 1991 the American Association of State Highway and Transportation Officials (AASHTO) published the Guide Specification and Commentary for Vessel Collision Design of Highway Bridges. This remains the current design procedure in the USA and uses a far simpler equation for the approximation of impact forces:

$$F_{bow} = 0.12V_0\sqrt{DWT}$$

where:

$F_{bow} = Average\ bow\ collision\ force\ (MN)$;

$V_0 = Initial\ ship\ velocity\ (m/s)$;

$$DWT = Deadweight\ of\ the\ vessel\ in\ metric\ tonnes.$$

The AASHTO equation gives lower impact forces than the Eurocodes equation. This is because the use of the maximum impact force is not recommended by AASHTO, since it believes that the duration of the maximum force is too small to cause any significant damage to the structure (Svensson, 2009).

2.7 Application of Finite Element Analysis for modelling ship impact

Finite Element Analysis (FEA) is a numerical method used to approximately predict how a structure will respond to different forces (and other variables such as heat and vibration). FEA can be carried out by a number of different software programs and is used across all engineering disciplines. It is mainly used to identify weak points in conceptual structures. It works by dividing the problem into a large amount of smaller finite elements, this is known as the 'mesh'. Each of these elements within the mesh have their own equations and all the combined equations are solved to examine the overall behaviour/response.

2.7.1 Ship impact on rigid dock wall

Woelke et al. (2012) conducted a ship impact study using finite element analysis considering a collision between a typical container vessel and a dock wall. The main objectives of this study were to compare the predictions of the collision forces imparted by the vessel on the wall between analytical methods and detailed finite element analysis. Although the study considered many different analytical methods for ship impact only two were selected for the comparison, one developed by Pedersen and another method from the AASHTO guide. The paper does a good job in showing that the AASHTO method predicted the impact force most accurately, and that Pederson's method was overly conservative. It is important to consider that neither analytical method considers bow geometry (Woelke et al, 2012).

The finite element analysis conducted in the paper models a typical 247m long container vessel of 80,000 tonnes (DWT) travelling at 10 knots impacting a rigid wall. Woelke modelled the wall as a rigid structure as his aims were focussed on the response of the ship and not the dock wall. This once again highlights that most literature in this field focuses on the ship and not on the impacted structure. This has influenced the decision to model the structure and assume the ship is rigid, allowing analysis of the impacted structure instead, filling the knowledge gap that currently exists in the field.

Woelke concludes that all ship collision scenarios should be considered on a case by case basis and that using a design and analysis procedure that fits all scenarios would be very hard if even possible. (Roark et al 2012) makes the point that the assumption that the impacting body will be fully rigid is a false assumption that makes the findings less accurate.

2.7.2 Ship impact on offshore platform legs

Offshore oil platforms are usually constructed on either steel or concrete legs and are anchored into the seabed. They are primarily designed to withstand the static dead load of the structure and the cyclic wave loading. The legs are also susceptible to accidental loading such as ship impact. Jones and Fraser (2008) investigated the effects of a low speed impact on the integrity of a reinforced concrete platform leg using finite element analysis. The model was based on a reinforced concrete leg that was impacted by a supply vessel in the North Sea. The purpose of the investigation was to see if the finite element model accurately predicted the local damage that occurred in reality.

The whole structure was modelled to ensure that the global response was a true representation of the structure. This included the compressive axial load that would have been acting down through the reinforced concrete leg from the oil platform above. The oil platform was modelled using beam elements and the concrete leg where the impact occurred was made up of solid elements. The ship model consisted of a 1m radius cylinder attached to a non-linear spring that resembled the force-displacement behaviour of a typical hull that deforms plastically. Attached to the other side of the spring was a mass element of 1870t (Jones and Fraser, 2008). The 1870t mass also included the hydrodynamic mass of the water that would have been entrained around the ship's hull upon impact. The impacting body was assigned a velocity of 0.3 m/s giving an impact energy of 84 kJ.

The paper concludes that the numerical finite element model accurately predicts the damage on the concrete structure from the impact. This confirms finite element analysis as an invaluable tool in the prediction of impact forces in real world scenarios. Reinforced concrete is a composite material; composites are often considered harder to model than solid sections such as steel. This shows that the application of FEA on a steel structure would be very appropriate providing an accurate prediction of the elastic and plastic behaviour present. This paper has influenced the approach of the project by encouraging the use of a case study.

2.8 Conclusion

Research shows that ships are getting bigger and that global sea traffic is increasing. This means that ports are getting busier inevitably increasing the risk of ship impact on dock infrastructure. It is important that the response of structures should be checked and understood to keep people and assets safe. This is required by law through standards set out by the HSE which states that employers should do whatever is 'reasonably practicable' to ensure workers are protected from anything that could cause them harm (i.e. water crashing into the dock bottom following the catastrophic failure of a caisson due to ship impact).

Research in the field of ship impact primarily focuses on the interactions between two ships colliding and not on the effects of ship impact on fixed structures where arguably a collision is most likely to take place. The majority of research related to ship impact on fixed structures seems to focus on the damage to the vessel and not on the structures response. This literature review concludes that there is a definite knowledge gap associated with the effects of ship impact on dock infrastructure that will be addressed through this project.

The effects of inertia mean that impact loading is completely different to static loading and should be treated as such. Failure to do so could lead to potentially catastrophic failures within industry. Structures are getting slenderer but need to remain just as robust against accidental loads (dynamic or impact). Ship impact is largely dependent on the mass and velocity of the ship however there are also a number of other factors that influence the impact force, although the inclusion of all the factors would be unrealistic it is important to have an awareness of them.

Finite element modelling is proven as an accurate and efficient means of gathering ship impact data. It is not limited by budget as is the case with physical modelling. It is also very versatile in the sense that material properties of the model can be changed with the click of a button. The modelling of a steel caisson subjected to ship impact should provide very accurate data, due to the isotropic nature of steel. Allowing the investigation of the elastic and plastic behaviour with relative ease.

3. Model methodology

3.1 LUSAS model setup

The objective of this project is to investigate the elastic and plastic behaviour of a steel caisson subjected to ship impact. As part of this it was identified that finite element was the most appropriate form of analysis. In order to perform finite element analysis the steel caisson had to be modelled in LUSAS (Figure). The following section will explain how the model was built and all of the different factors included in the model.

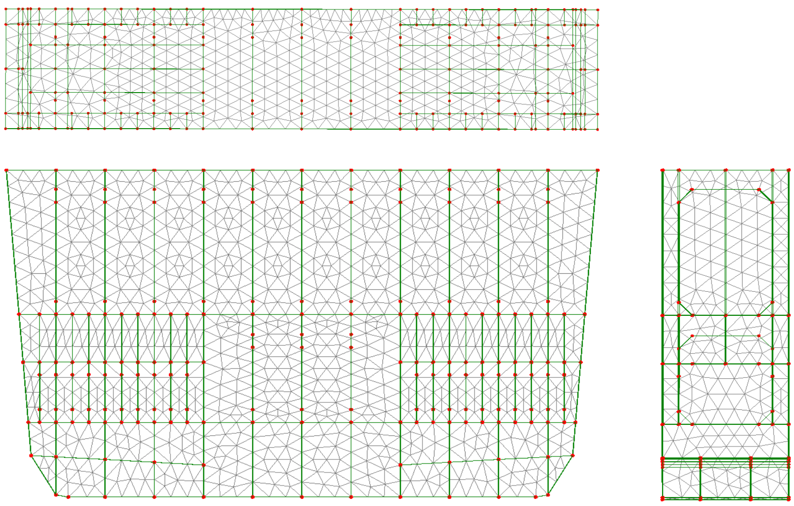

Figure 3: Model of caisson showing mesh and nodes

3.1.1 Geometry creation

The basic geometry of the structure was produced using nodes from the geometry menu. As the geometry of the steel caisson is known, the location of the nodes could be added using the coordinate system built into LUSAS. The lines between the nodes could then also be added using the line function in the geometry menu. This was repeated for all of the internal elements within the caisson such as the vertical stiffeners, horizontal stiffeners and ballast tanks.

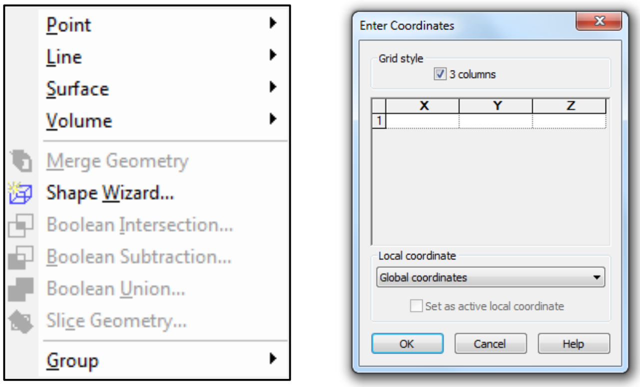

Figure 4: LUSAS Geometry and coordinates menu

3.1.2 Applied mesh

The meshes were selected as thick shell elements with triangular shapes from the surface mesh menu. The irregular mesh option was ticked allowing the appropriate mesh size to be set. In this model, the mesh has been refined in areas of interest (i.e. the impact zone). This basically means insignificant areas such as the edges were assigned a larger mesh size (0.80) and the impact zone was assigned a smaller mesh (0.40). This was done to reduce the already lengthy run time of the model.

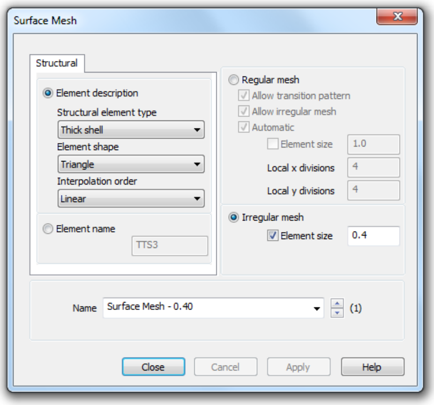

Figure 5: LUSAS Surface mesh menu

3.1.3 Material definition

Elastic properties – The elastic properties of the steel were entered within the elastic tab, using a Young's modulus of 210 x 10^6 kN/m^2 and a Poisson's ratio of 0.3. The dynamic properties option was ticked because impact loading is a dynamic action. This introduces a Rayleigh damping constant which ensures that the model dampens after impact, if this wasn't included there would be no energy loss and the model would infinitely oscillate.

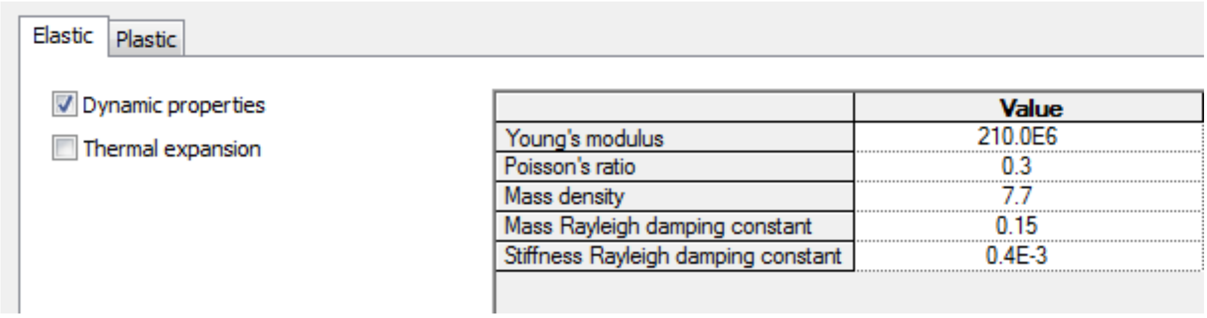

Figure 6: Elastic properties of steel

<u>Plastic properties</u> – The plastic properties of the steel were entered within the plastic tab by selecting the Von Mises stress potential option and a yield strength of 275 N/mm^2. Although the structure is to be constructed using S355 structural steel, S275 was used to ensure that the impact caused sufficient yielding allowing any patterns in the stress and strain distribution to be identified. The hardening gradient option was ticked, the strain gradient (slope) was selected as 0.01 and the plastic strain was selected as 1.0 or 100%.

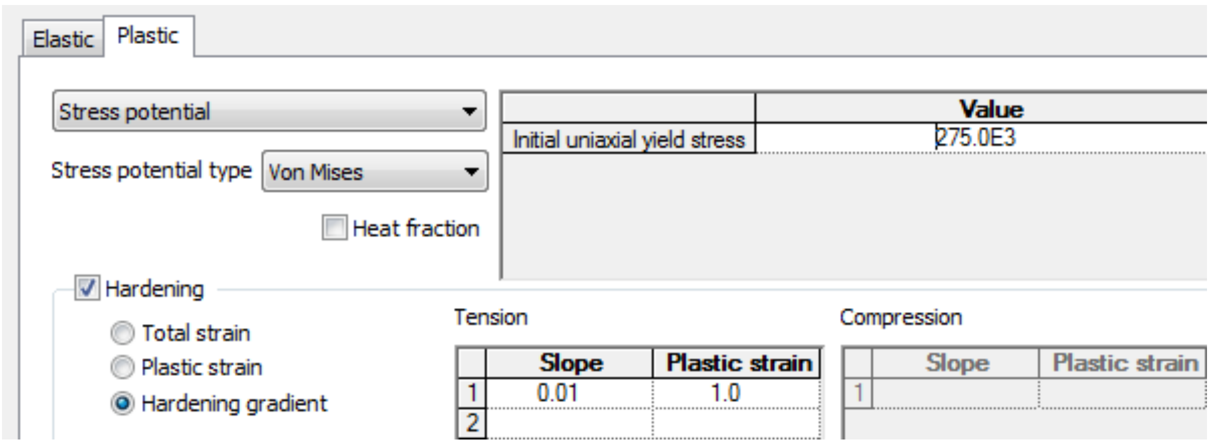

Figure 7: Plastic properties of steel

3.1.4 Supports definition

Simple supports were applied using the structural supports menu seen below in Figure 8. The model has vertical supports along the two bottom edges of the caisson which have been restrained in the z axis, and lateral supports along the two sides which have been restrained in the y axis. The purpose of simple supports is to ensure the impact causes sufficient rotation to identify any patterns. Due to the impact component of the model a friction side line has been included.

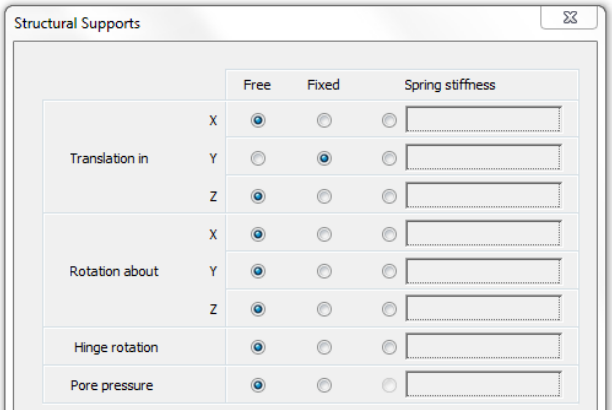

Figure 8: LUSAS Structural support menu

3.1.5 Loading definition

As this project is only interested in the behaviour of the caisson under impact, the vessel used in the LUSAS analysis was treated as rigid. The density of the vessel was changed within LUSAS for each analysis to change the different ship masses. The collisions were assumed to be head-on as this is always considered to be the worst-case scenario. The velocity of the vessel was entered as a negative 0.5 in the y-axis in the dynamic loading interface. Hydrostatic and self-weight were excluded from the analysis as the project is focussed on the behaviour of the caisson under impact.

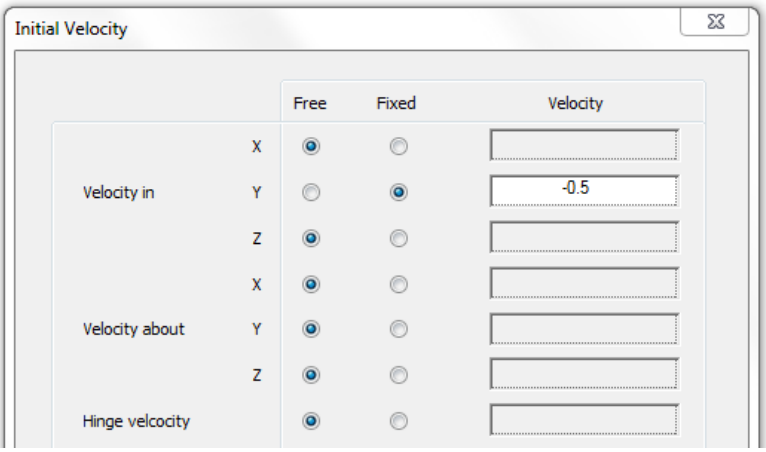

Figure 9: Dynamic loading interface

3.1.6 Non-linear controls

Impact loading is a highly non-linear process due to its dynamic nature. This means that the analysis needed to include several controls to deal with the non-linear behaviour. There are three main sources of non-linear behaviour dealt with in the model:

- Geometric NL using Total Lagrangian – Used due to large strain and is the only available GNL for large strain with shell elements in LUSAS.

- Material NL using Von Mises Stress Potential – Used a low strain hardening gradient to include non-linear plastic behaviour of material.
- Boundary NL using Friction Slideline – This is caused by the Impact component, where a sudden change in stiffness can occur due to bodies coming into or out of contact with each other.

3.2 LUSAS model analysis

Finite element analysis was chosen for this method due to the fact that it is fairly inexpensive in comparison to real models and there are a wide range of factors that can be built into the model to make it as realistic as possible. Unfortunately, the main limitation is that the more detailed the analysis is made, the longer it takes to run and requires a PC with a large amount of processing power.

Once the finite element model had been setup it was possible to run the analysis. The limitations outlined above was avoided through the use of a specialist computer owned by AECOM with large processing capabilities. The impact cases were coded to batch run continuously and the analysis took approximately 24 hours in total. The results files for each analysis are approximately 3.5GB each.

Table 2: Summary of ship masses tested and associated parameters.

Impact Case	Ship Mass (T)	Mass (kg)	Impact Velocity (m/s)	Kinetic Energy (J)
1	500	500,000	0.5	62500
2	750	750,000	0.5	93750
3	1000	1,000,000	0.5	125000
4	1250	1,250,000	0.5	156250
5	1500	1,500,000	0.5	187500

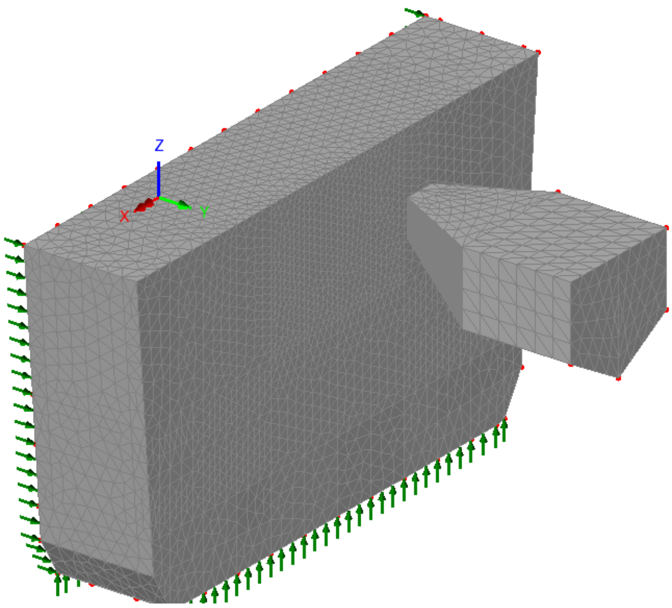

Figure 10: LUSAS 3D Model of caisson and impacting vessel

4. Results and validation

This chapter will present the results of the LUSAS analysis and validate the results. Validation is a critical part of using any numerical analysis to ensure that it is both accurate and reliable.

4.1 Validation of LUSAS model

The following calculation is a sample validation for the 1500T impact scenario. The below graph is a plot of Force vs Time acting upon the caisson due to the impact.

Maximum force from LUSAS @ Node 6652 from Figure 11 was found to be:

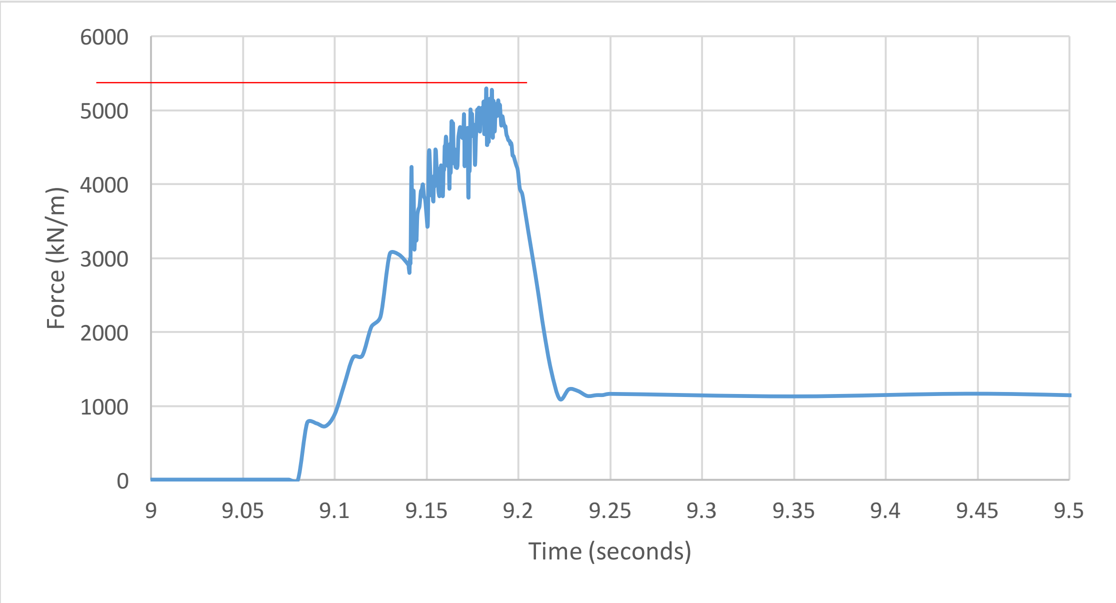

Figure 11: LUSAS Force time history output

$F = 5.291\ x\ 10^3\ kN/m \qquad when\ T = 9.18258\ Seconds.$

Because the force is kN/m it must be multiplied by the width of the impacting vessel:

$F = 5.291\ x\ 10^3\ x\ 1.8 \quad \therefore \quad \boldsymbol{F = 9.524\ x\ 10^3\ kN}$

In order to validate the LUSAS force a theoretical force must be calculated using:

$$Kinetic\ Energy = 0.5mv^2\ (\boldsymbol{Eq.1})$$

$$Work\ done = Force \times Displacement\ (\boldsymbol{Eq.2})$$

Next the kinetic energy is equated to work done:

$$Kinetic\ Energy\ (\boldsymbol{Eq.1}) = Work\ done\ (\boldsymbol{Eq.2})$$

Which can be rewritten as:

$$0.5mv^2 = F \times \delta$$

Which can be rearranged to give:

$$F = \frac{0.5mv^2}{\delta}\ (\boldsymbol{Eq.3})$$

Where:

$F = Force\ (kN)$

$m = Mass\ of\ impacting\ vessel = 1{,}500{,}000\ kg$

$v = Velocity\ of\ impacting\ vessel$ = 0.5 m/s

$\delta = Displacement\ due\ to\ impact$ = 19.54 mm or 0.0195406 m

The displacement was extracted from LUSAS at Node 6652 for consistency with the force data and was found to be 19.54 mm. All of the values were then substituted into Equation 3 which can be seen below:

$$Force = \frac{0.5 \times 1{,}500{,}000 \times 0.5^2}{0.0195406} = \frac{187{,}500}{0.0195406} \therefore \boldsymbol{F = 9.595 \times 10^3\ kN}$$

LUSAS Force = $9.524\ x\ 10^3\ kN$ **Theoretical Force** = $9.595 \times 10^3\ kN$

The percentage difference between the force found in LUSAS and the theoretical force found is 0.75%. This shows there is a strong correlation and that the LUSAS analysis is valid. Table 3 below shows validation for the other scenarios.

The full calculation and values can be seen in the attached excel spreadsheet (CD).

Table 3: Summarised validation calculations for all impact cases

Mass (T)	Energy (J)	Displacement (m)	Theoretical Force (kN)	LUSAS Force (kN)	Percentage Difference (%)
1500	187500	0.0195406	9595.406	9523.872	-0.75
1250	156250	0.0177257	8814.885	9749.394	9.59
1000	125000	0.0156409	7991.867	8848.260	9.68
750	93750	0.0134935	6947.790	8462.448	17.90
500	62500	0.0109239	5721.400	7635.330	25.07

4.2 Results for 1500T impact scenario

This section will present the results from the 1500 tonne impact load case in detail. This case has been identified as the worst-case impact scenario as it is the largest ship mass and will show the maximum elastic and plastic behaviour of the caisson.

4.2.1 Deformed mesh

Figure 12 shows the deformed mesh of the caisson subjected to a 1500 tonne impact load. The contouring tool built into LUSAS was also utilised as a visual aid in the identification of any patterns in the deflection across the caisson

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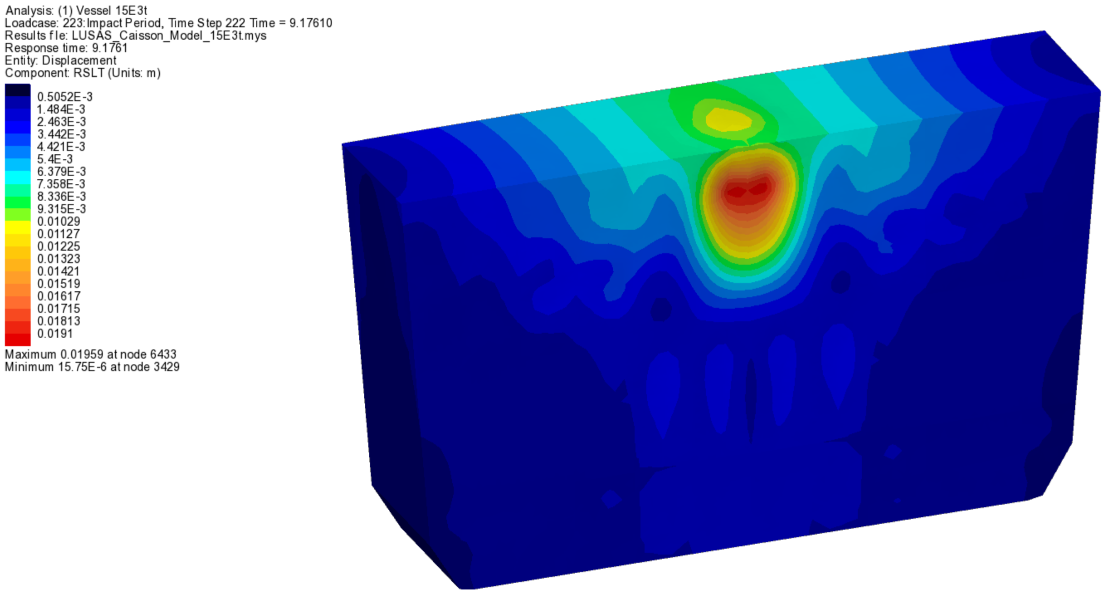

Figure 12: Deformed mesh of steel caisson subjected to 1500 tonne impact

The maximum deflection is 19.59mm and occurs at approximately 1.5m below the top middle of the caisson. The LUSAS contours shows that the impact creates a large depression (also known as cratering) on the impacted surface and a smaller bulging (also known as scabbing) effect along the top and back of the caisson. It is also clear that the deflection is largest along the upper unsupported edge and there is little to no deflection along the bottom and sides of the caisson.

4.2.2 Stress vs Strain

Figure 13 shows the stress and strain curve throughout the whole impact scenario. The values have been taken from Node 6812 which is within the central stiffener that provides lateral support against the impact. The stress and strain curve conveys the elastic and plastic behaviour of the caisson subjected to ship impact and will be discussed in more detail within the discussion section.

The stress vs strain graph starts with an initial linear section where the stress and strain have a positive linear relationship. Once the stress reaches 200 N/mm^2 the gradient of the line begins to plateau becoming a non-linear relationship (i.e. the strain increases more than the stress per run). The graph continues at a reduced gradient with fairly volatile oscillations occurring. At a stress of approximately 270 N/mm^2 the stress and strain both decrease, causing the line to run parallel to the initial linear section. The linear line then stops reducing at a strain rate of 0.15%.

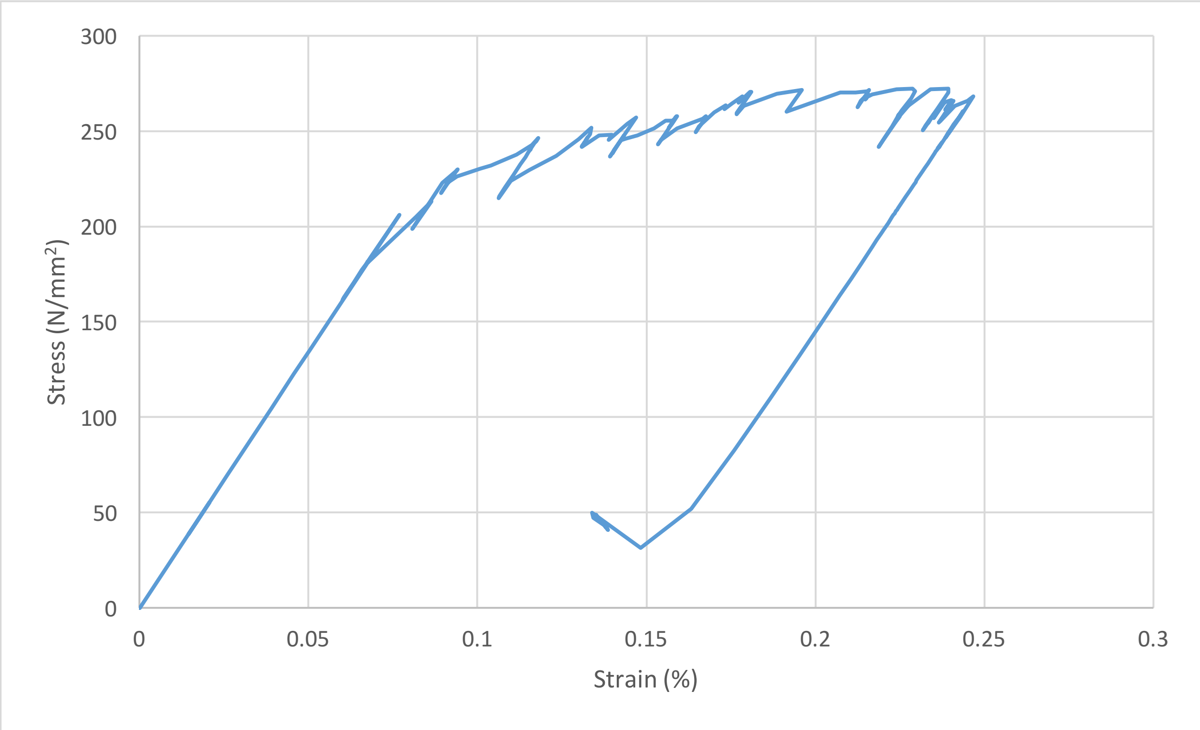

Figure 13: Stress vs strain at Node 6812 for 1500T impact

Figure 14 shows the plastic strain throughout the whole impact scenario along the central stiffener of the caisson. Just after impact occurs at t = 9.14186 seconds the plastic strain jumps from 0% to 0.15%. The plastic strain then remains at 0.15%.

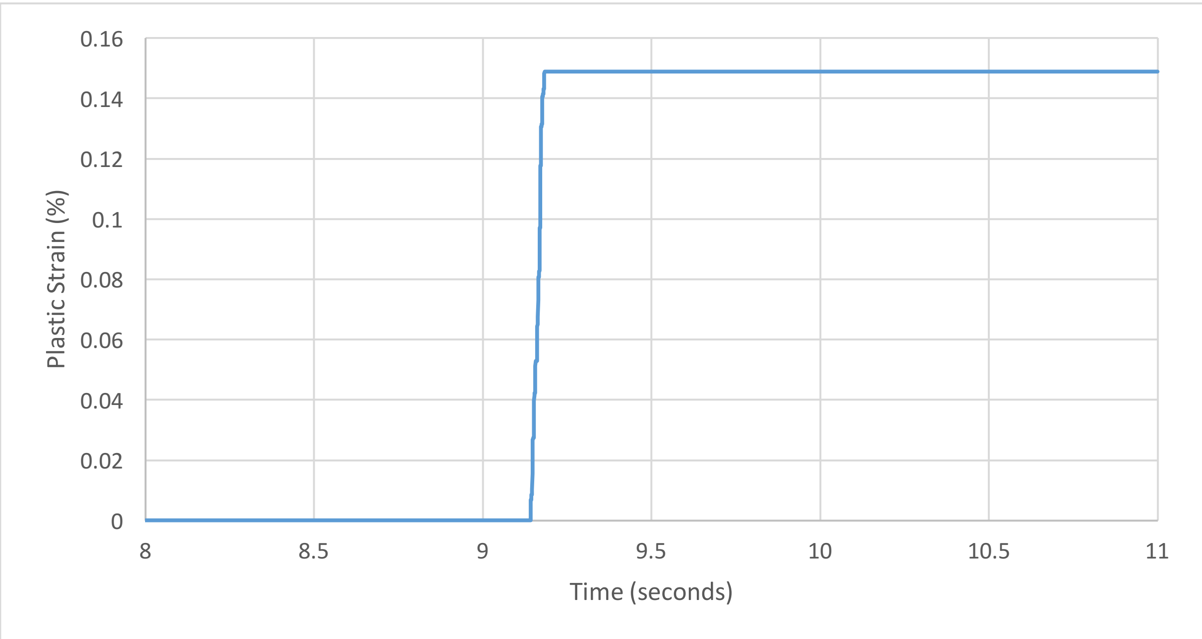

Figure 14: Plastic strain at Node 6812 for 1500T impact

Figure 15 shows that when impact occurs at t = 9.085 seconds the displacement jumps from 0 mm to 17.87 mm and then back down to 2.49 mm over a time period of 0.09155 seconds. It then stabilising at approximately 2.70 mm.

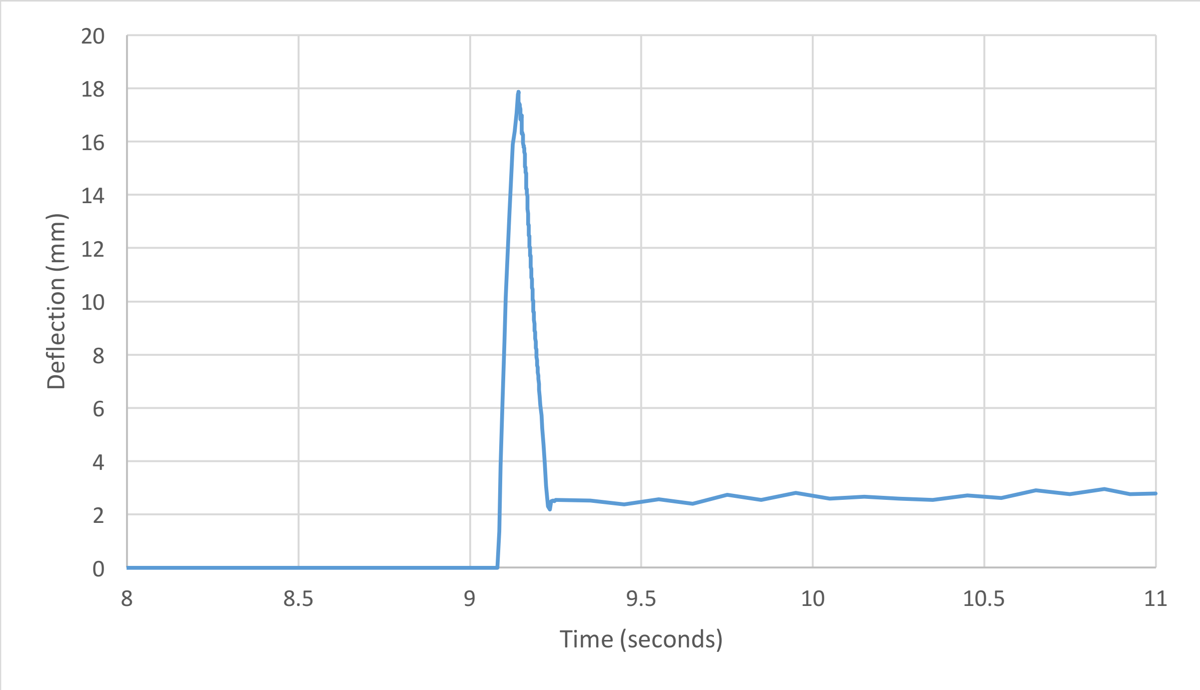

Figure 15: Displacement at Node 6812 for 1500T impact

4.2.3 Stress distribution

Figure 16 shows the distribution of stress across the caisson due to the 1500 tonne ship impact scenario. Although the ship makes direct contact with the steel plate of the caisson, the maximum stress along the steel plate is only approximately 102 N/mm^2.

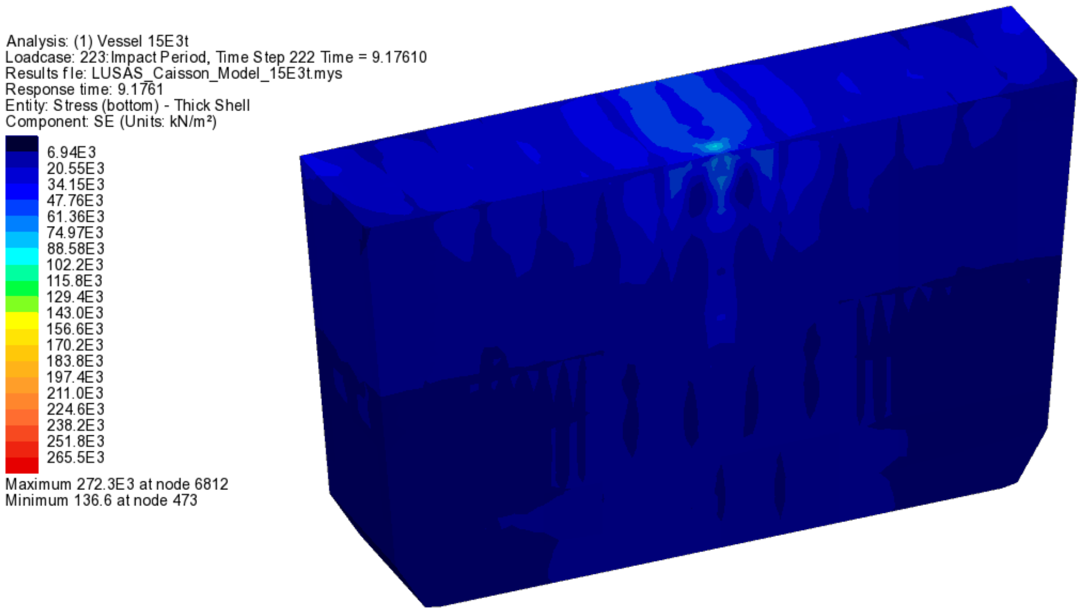

Figure 16: Stress distribution along impacted surface due to 1500T impact

Figure 17 shows that most of the stress acts upon central stiffener as it provides the lateral stability against the impacting vessel. This is mostly concentrated along the inside edge of the horizontal in plane stiffener. The maximum stress along the central internal stiffener is 272 N/mm^2 and has been labelled below.

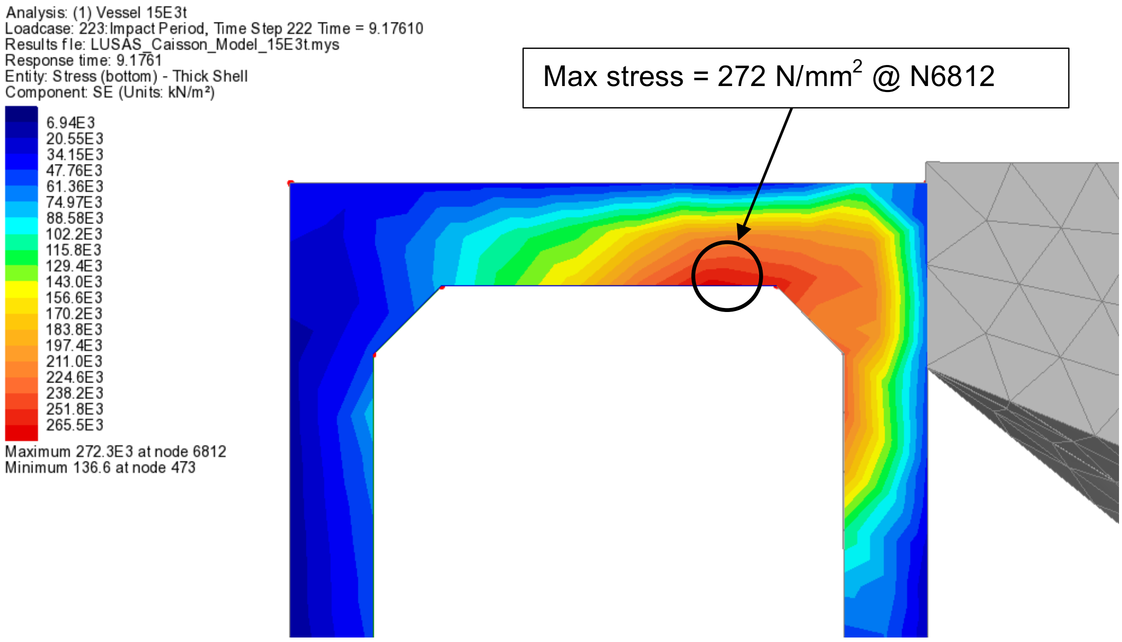

Figure 17: Stress distribution across internal stiffener due to 1500T impact

4.2.4 Strain distribution

Figure 18 shows the distribution of strain across the central stiffener of the caisson due to the 1500T tonne ship impact scenario. The strain is concentrated on the inside edge corner between both the horizontal and vertical stiffener. The maximum strain within the caisson due to impact is 0.247% @ N6812 as seen below.

Figure 19 shows the distribution of plastic strain across the central stiffener of the caisson subjected to the 1500T tonne ship impact scenario. The plastic strain is concentrated around the corners of the vertical and horizontal stiffeners. The maximum plastic strain within the caisson is 0.145% @ N6812.

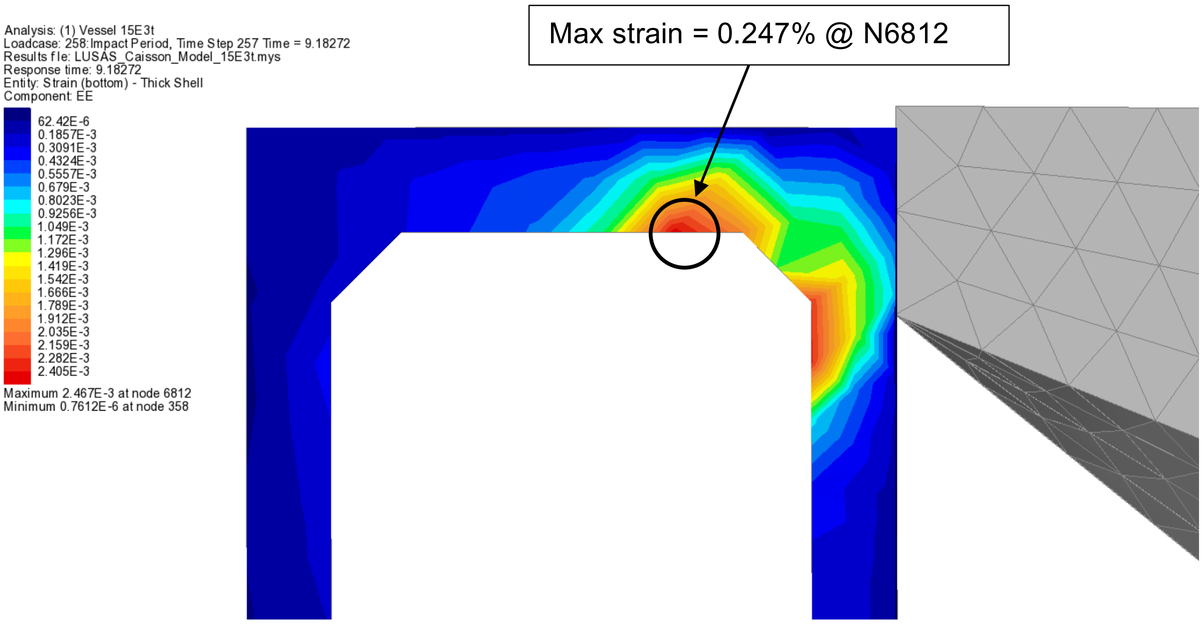

Figure 18: Strain distribution across internal stiffener due to 1500T impact

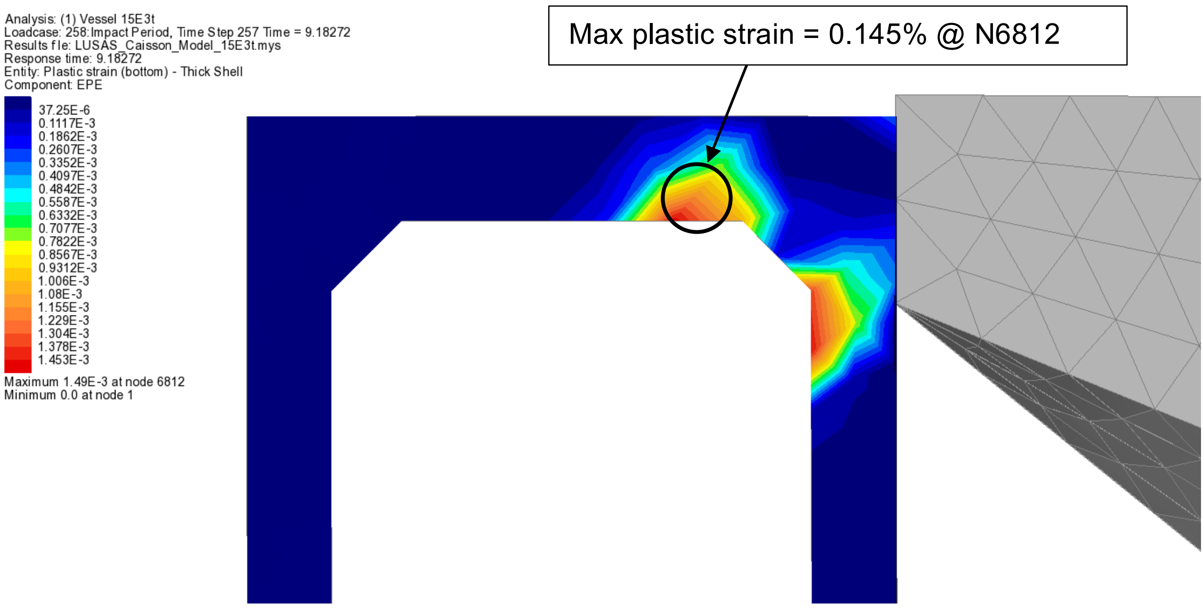

Figure 19: Plastic strain distribution across internal stiffener due to 1500T impact

4.3 Results for 500T impact scenario

This section will present the results from the 500 tonne impact load case in detail. This case has been identified as the best-case impact scenario as it is the smallest ship mass and will show the minimum elastic and plastic behaviour of the caisson.

4.3.1 Deformed mesh

Shown below in Figure 20 is the deformed mesh of the caisson subjected to a 500 tonne impact load. The contouring tool built into LUSAS was utilised as a visual aid in the identification of any patterns in the deflection across the caisson.

The maximum deflection is 11.2mm and occurs at approximately 1.5m below the top

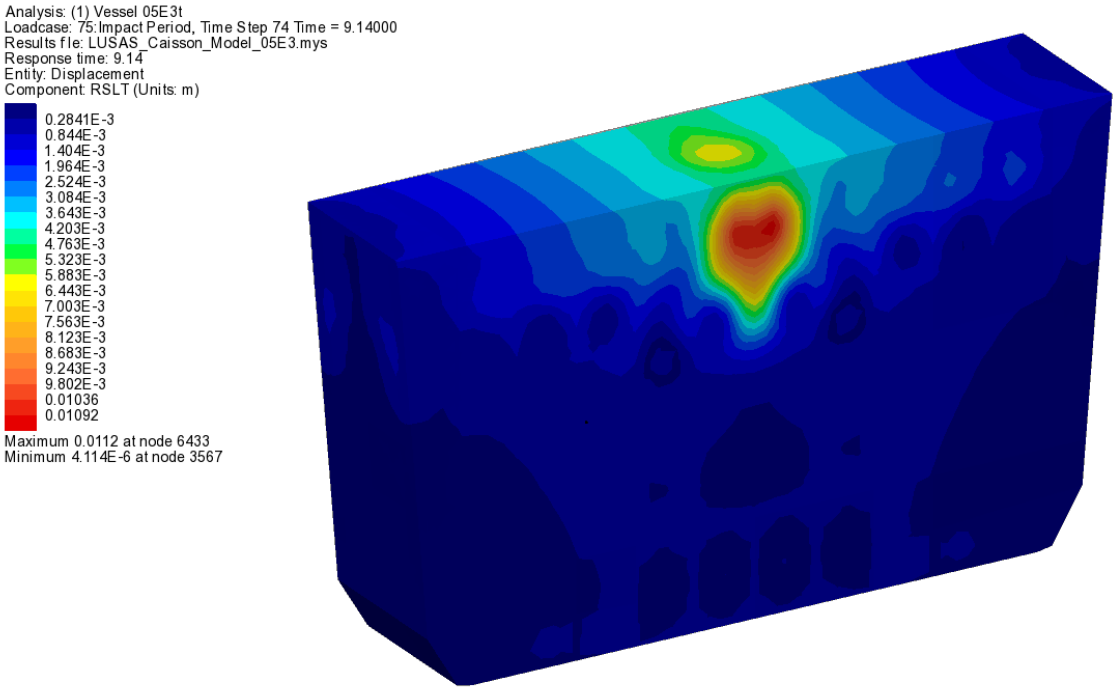

Figure 20: Deformed mesh of steel caisson subjected to 500 tonne impact

middle of the caisson. The LUSAS contours shows that the impact creates a depression on the impacted surface and a smaller bulging effect along the top and back of the caisson. It is also clear that the deflection is largest along the upper unsupported edge and there is little to no deflection along the bottom and sides of the caisson.

4.3.2 Stress vs strain

Figure 21 shows the stress and strain curve throughout the whole impact scenario. The values have been taken from Node 6812 which is within the central stiffener that provides lateral support against the impact. The stress and strain curve conveys the elastic and plastic behaviour of the caisson subjected to ship impact and will be discussed in more detail within the discussion section. The stress vs strain graph starts with an initial linear section where the stress and strain have a positive linear relationship. Once the stress reaches 200 N/mm^2 the gradient of the line flattens for a very brief period.

At a stress of approximately 215 N/mm^2 the stress and strain both decrease, causing the line to run parallel to the initial linear section. The linear line then stops reducing at a strain rate of 0.015%.

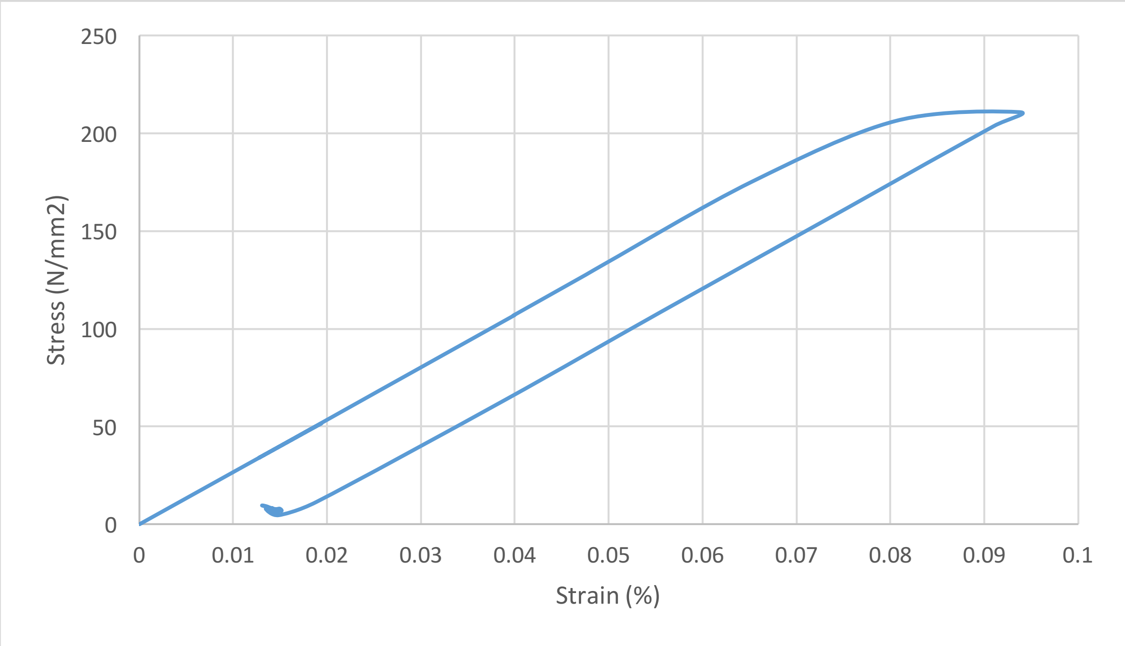

Figure 21: Stress vs strain at Node 6812 for 500T impact

Figure 22 shows the plastic strain throughout the whole impact scenario along the central stiffener of the caisson. At t = 9.135 seconds the plastic strain jumps from 0% to 0.0157%. The plastic strain then remains at 0.0157% for the rest of the analysis.

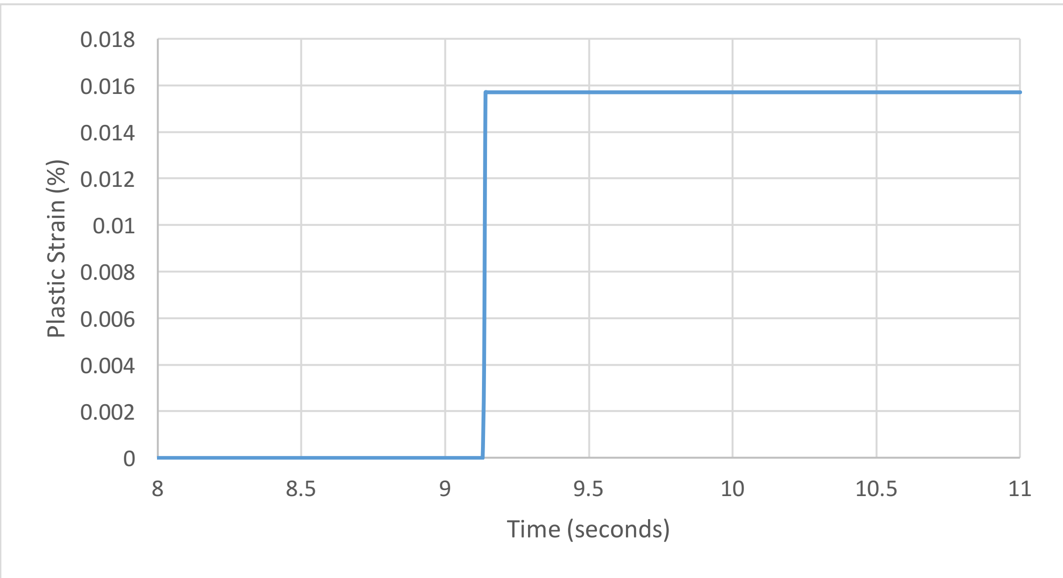

Figure 22: Plastic strain at Node 6812 for 500T impact

Figure 23 shows that at t = 9.085 seconds the displacement jumps from 0mm to 9.76mm and then back down to 0.3mm over a period of around t = 0.09 seconds. It then oscillates around 0.5mm for the remainder of the analysis.

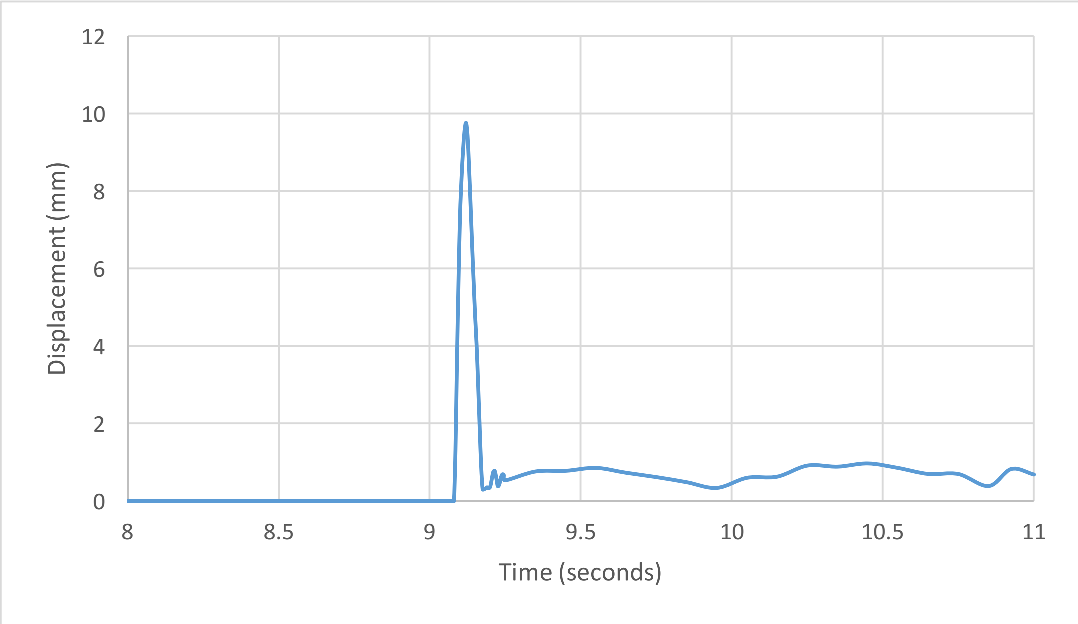

Figure 23: Displacement at Node 6812 for 500T impact

4.3.3 Stress distribution

Figure 24 shows the distribution of stress across the caisson due to the 500 tonne ship impact scenario. Although the ship makes direct contact with the steel plate of the caisson, the maximum stress along the steel plate is only approximately 79N/mm^2.

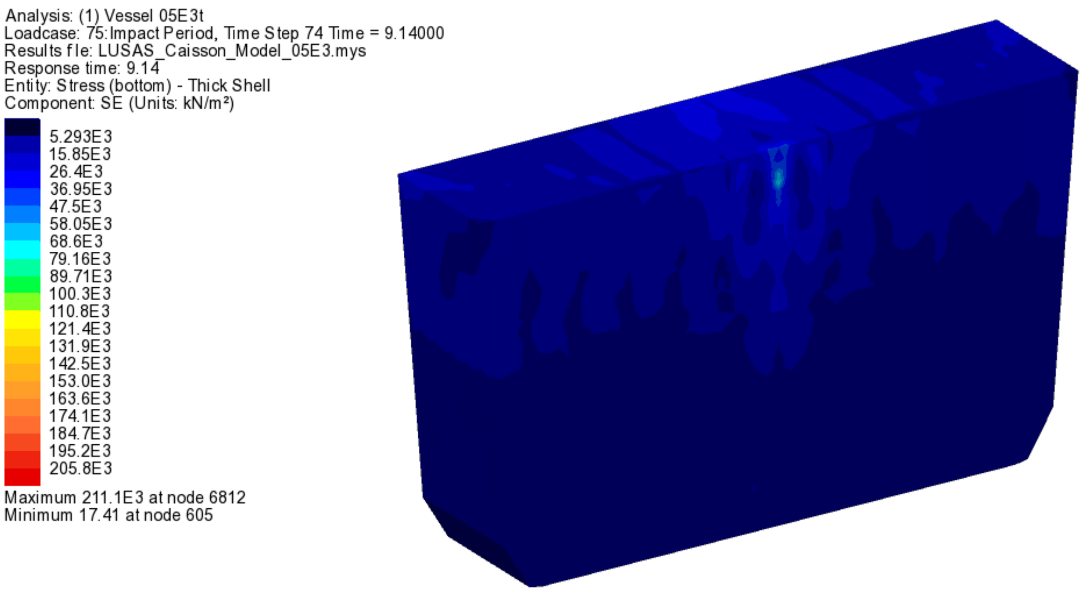

Figure 24: Stress distribution along impacted surface due to 500T impact

Figure 25 shows that most of the stress acts upon central stiffener as it provides the lateral stability against the impacting vessel. This is mostly concentrated along the inside edge of the horizontal in plane stiffener. The maximum stress along the central internal stiffener is 211N/mm^2 and has been labelled below.

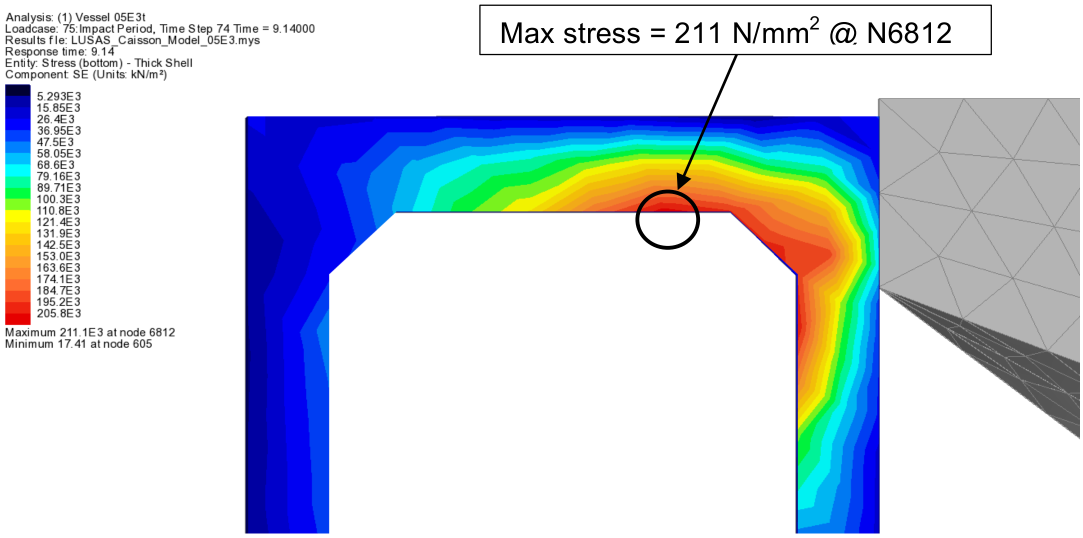

Figure 25: Stress distribution across internal stiffener due to 500T impact

4.3.4 Strain distribution

Figure 26 shows the distribution of strain across the central stiffener of the caisson due to the 500T tonne ship impact scenario. The strain is concentrated on the inside edge corner between both the horizontal and vertical stiffener. The maximum strain within the caisson due to impact is 0.11% @ N6548 as seen below.

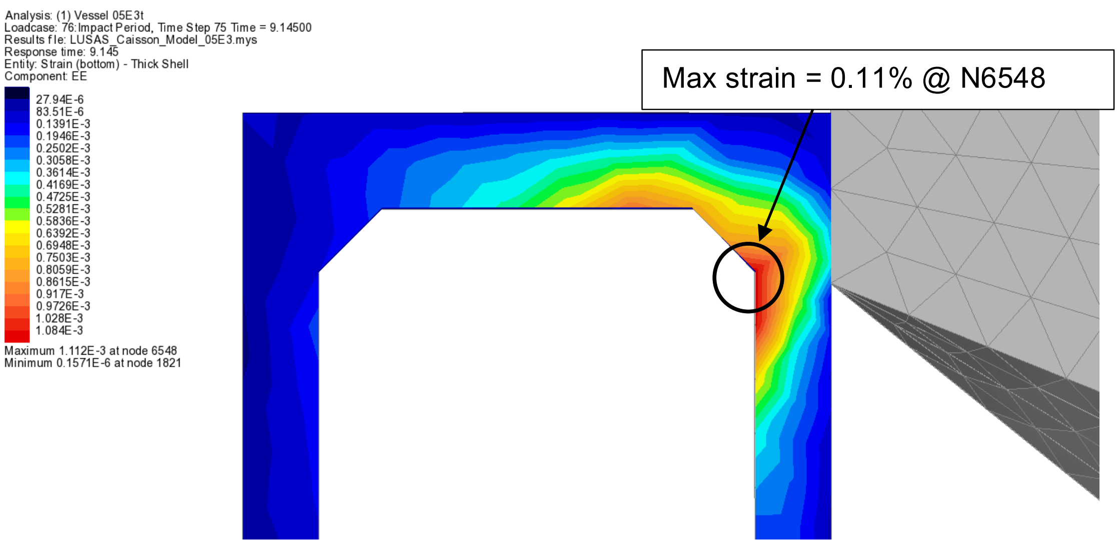

Figure 26: Strain distribution across internal stiffener due to 500T impact

Figure 27 shows the distribution of plastic strain across the central stiffener of the caisson subjected to the 500T tonne ship impact scenario. The plastic strain is concentrated around the corners of the vertical and horizontal stiffeners. The maximum plastic strain within the caisson is 0.032% @ N6323.

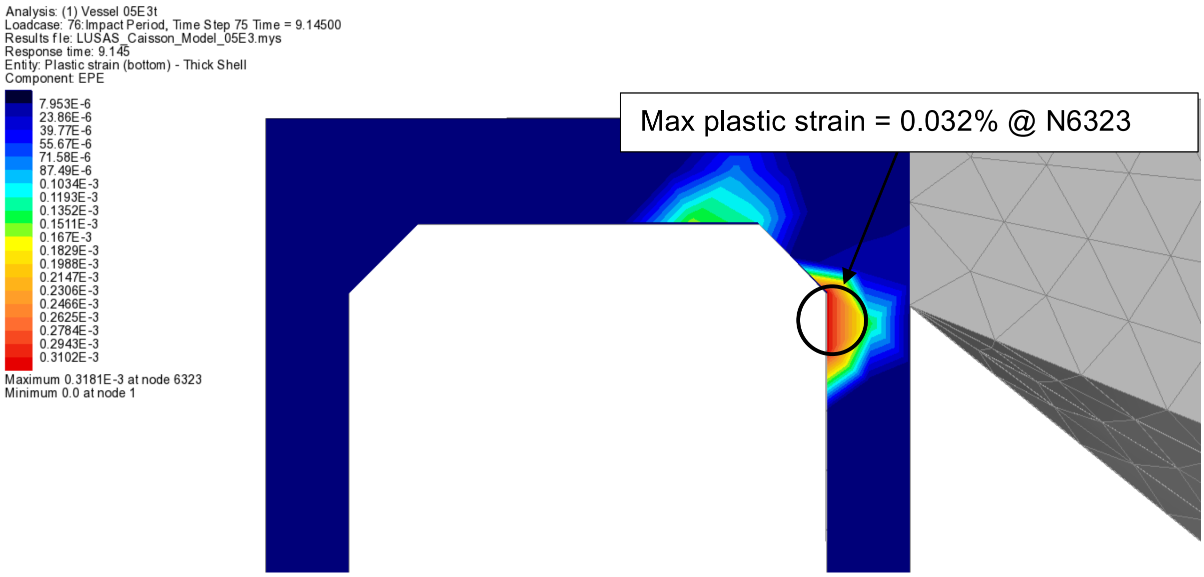

Figure 27: Plastic strain distribution across internal stiffener due to 500T impact

4.4 Comparison of all results

Figure 28 shows the stress vs strain across the whole impact period for all ship masses. The values have been taken from Node 6812 which is within the central stiffener that provides lateral support against the impact. The stress and strain curve conveys the elastic and plastic behaviour of the caisson subjected to ship impact and will be discussed in more detail within the discussion section.

The linear section of the stress vs strain graphs is exactly the same for all impact scenarios tested in this project. At a strain of approximately 0.07% they all begin to plateau and carry on increasing at a reduced gradient with slight variations. They all then begin to return back to a linear relationship again in order starting with the smallest mass and finishing with the largest mass. It can be seen that the larger the mass, the larger the strain remains at the end of each stress vs strain plot. The reasons for this behaviour will be covered in the discussion section.

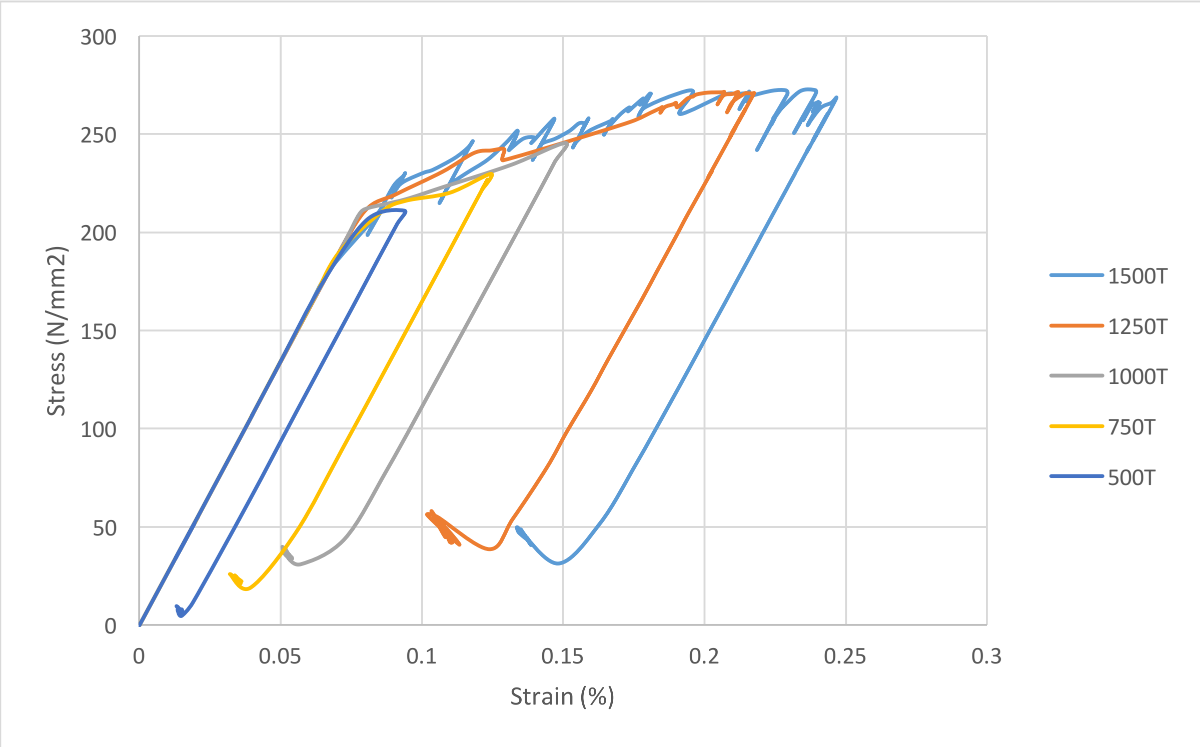

Figure 28: Stress vs strain for all impact cases

5. Discussion

5.1 Model validation

In order to discuss the results of this project it is first crucial to outline the validity of the numerical model to ensure that all results presented are both accurate and reliable. Since ship impact is a highly non-linear process, the equations used for the approximation of impact forces are often very inaccurate. Therefore, it was decided that the most appropriate and efficient form of validation was through the use of the kinetic energy and work done equations to calculate a theoretical force which could then be compared to the force extracted from the LUSAS analysis.

As seen in Section 4.1 the theoretical impact force was calculated using Equation 3 which was derived by equating the kinetic energy of the impacting vessel with the work down in the deformed steel caisson due to the impact. This theoretical force was then compared to the LUSAS force extracted from the numerical model. The difference between the theoretical and LUSAS force for the 1500 tonne impact scenario was found to be 0.75%. The lowest impact scenario which is 500 tonnes has a difference of 25.07% between the theoretical and LUSAS force. Interestingly, this shows that the validation appears to be less accurate at lower energies, which can be seen in Table 3. This inaccuracy at lower energies can be explained by a study conducted by Minorsky (1959) who stated that his analytical method was also inaccurate for low energy impacts due to the poor correlation between kinetic energy and volume of steel deformed. The percentage difference shown in Table 3 between the theoretical force and LUSAS force show that there is sufficient evidence to conclude that the numerical model is valid.

5.2 Model assumptions

It is next important to discuss and justify some of the main assumptions that were made within the analysis. The first being the exclusion of both the self-weight of the steel caisson and the hydrostatic load acting upon the caisson due to the water in the basin. This decision was made because this project is focussing on the behaviour of the caisson subjected to ship impact and the inclusion of any other loading could have potentially hidden the true behaviour due to impact. It would also have made the validation of the numerical model far more complex and potentially not possible. Another consideration that was included in the model was the use of a Rayleigh damping coefficient to ensure that the model correctly dampened post-impact. This ensured that the kinetic energy from the impacting vessel was correctly dissipated as accurately as possible, if this wasn't included then the model would have continued to oscillate infinitely and would not have been an accurate representation of the behaviour exhibited by the caisson post-impact. The final assumption that was made within this project is that the effects of the hydrodynamic mass was negligible. The hydrodynamic mass is the water entrained around the hull that carries on moving once the ship has stopped due to the impact, exerting an extra force against the dock structure (Thoresen, 2014). Due to this projects focussing on relatively small vessels ranging between 500T and 1500T traveling at a relatively slow speed of 0.5 m/s upon impact, this extra hydrodynamic force was considered insufficient to cause failure.

5.3 Elastic and plastic behaviour

This projects main aim was to investigate the elastic and plastic behaviour of the steel caisson subjected to impact. The most logical way that this can be done is through close examination of the stress vs strain plots for the 1500 tonne impact scenario.

Figure 13 shows the stress vs strain plot for the 1500T impact case at Node 6812. This node was selected as it is located within the central stiffener of the caisson and experiences the highest stress and hence plastic strain during the impact period. The initial linear portion of the stress vs strain plot shows the elastic behaviour of the material response due to impact and is known as the elastic region. Within this region any deformation would be fully recoverable upon the unloading of the caisson. This linear relationship is governed by Hooke's law and can be seen in more detail within the literature of this project (Section 2.4). Once the stress reaches approximately 200N/mm^2 the gradient of the line begins to plateau which shows that the steel is yielding and marks the start of the plastic region of the curve where plastic behaviour begins to occur.

The line continues at a reduced gradient which is known as strain hardening (the plastic deformation leads to an increase in strength of the steel). The rapid unloading and reloading within the non-linear material response could possibly be attributed to oscillations occurring within the impact event. These oscillations are pronounced due to the node being an in plane vertical stiffener, noticeably it only occurs for the 1500 and 1250 tonne cases which is most probably due to the higher energy levels. Once the stress reaches approximately 270N/mm^2 the plot begins to decrease in a linear fashion parallel to the initial linear line. This is the unloading of the caisson because all of the kinetic energy due to impact has been dispersed by the steel caisson in the form of heat, sound, plastic strain, vibration and oscillation. It is important to note that this peak does not represent the ultimate limit state and is purely the unloading of the caisson. This decreasing linear line shows the elastic strain within the steel being undone due to it being recoverable. The line stops decreasing at a strain of 0.15% and never returns to zero as the plastic strain has caused plastic deformation which is an irreversible process unlike elastic deformation. This plastic strain can be seen in Figure 14 across the whole impact period and increases to 0.15% at the point of impact and remains there for the rest of the analysis. This corresponds to the 0.15% value in the stress vs strain plot in Figure 13.

Figure 28 shows a comparison of the stress vs strain plots for all five of the ship masses tested in this project. It shows that they all follow the same pattern during the initial linear elastic phase of the impact. The plot then shows each scenario beginning to unload, starting with the smallest mass of 500 tonnes and finishing with the largest mass of 1500 tonnes. This is most probably attributed to the fact that during the 500 tonne impact, there is less kinetic energy to be dissipated into the caisson structure so it begins to unload first. This also means that less energy is dissipated in the form of plastic strain which means there is less permanent plastic deformation occurring. Looking at the other ship masses shows that the larger the impacting ship mass is, the longer the unloading takes because of the increased kinetic energy. This increased kinetic energy leads to an increased amount of plastic strain required to dissipate the energy which means an increased amount of plastic deformation occurring within the steel caisson.

5.4 Total displacement

Next the total displacement plots across the whole impact period for the 1500 and 500 tonne ship masses will be compared. The maximum displacement of the caisson due to the 1500T impact was found to be 19.59mm from Figure 12. After the maximum deflection occurs the unloading begins and the deflection reduces to 2.49 mm and then stabilises at around 2.70mm but does not return to 0mm due to the plastic deformation. This shows that a permanent deformation of approximately 2.70mm has occurred within the caisson due to the 1500 tonne ship impact. Next the maximum total displacement for the 500 tonne impact was found to be 9.79mm from Figure 23. After the maximum displacement the unloading begins causing the deflection drop to 0.3mm before stabilising at around 0.5mm but never returning to 0mm. Comparing both of these values show that the total plastic deformation was 0.5mm for the 500T case compared to 2.70mm for the 1500T case. This clearly shows that plastic behaviour is present and that as the impacting ship mass increases so does the plastic deformation and hence damage to the steel caisson.

5.5 Stress and strain distribution

The stress distributions across the caisson were similar for all ship masses analysed in this project and only varied by the magnitude of the stresses. The stress distribution acting upon the impacted surface due to the 500 and 1500 tonne impact can be seen in Figure 25 and Figure 17. The maximum stress acting upon the impacted surface for the 500 and 1500 tonne impact where 79N/mm^2 and 102N/mm^2 respectively. The next step is to consider the stress distribution along the internal central stiffener due to the 500 and 1500 tonne (Figure 26 and Figure 18). The maximum stress acting upon the central stiffener for the 500 and 1500 tonne impact where 211N/mm^2 and 272N/mm^2 respectively. Comparing the stress acting upon the impacted surface to the central stiffener for both cases shows the stress acting upon the central stiffener far exceeds the stress acting upon the impacting surface even though the steel plated face of the caisson comes into direct contact with the impacting vessel. This can be explained by the simple engineering theory that load goes to stiffness. The steel plate has very low relative stiffness in comparison the central vertical stiffener and hence the stress is concentrated within the central stiffener and not the steel plate. The steel has been tapered to deal with this build of stress within all of the stiffeners.

Next the distribution of the plastic strain within the caisson will be examined and discussed. The plastic strain for the 500 tonne impact can be seen in Figure 28 and shows a concentration of plastic strain (0.032%) around the right hand corner of the vertical stiffener and a slight build-up of plastic strain (0.014%) on the corner of the horizontal stiffener. The plastic strain for 1500 tonne impact can be seen in Figure 20 and shows a large concentration of plastic strain around both corners of the horizontal and vertical stiffeners (0.145% and 0.142% respectively). There is also slight plastic strain on the impacted surface of 0.034%. Comparing the plastic strains between both impacts cases shows that increased ship mass leads to an increased amount of plastic strain within the steel caisson due to impact.

5.6 Overall response of caisson

In order to discuss the overall response of the caisson and whether the caisson has been sufficiently designed to withstand any likely impact it is first essential to determine the definition of damage. Damage will be classified as any 'plastic deformation' in the steel. All references within this section refer to the 1500 tonne impact case as it is the maximum mass tested.

The local response of the steel caisson subjected to impact can be determined by inspection of both the deformed mesh and plastic strain diagrams in Figure 13 and Figure 20 respectively. Local damage can be seen on the top edge of the caisson at the point of impact where a depression has occurred. There is also minor damage on the top and back of the caisson where bulging has occurring. Most of the local damage was concentrated within the central stiffener where the stress reached a maximum of 272N/mm^2. Comparing this stress to the yield strength of the steel which is 275N/mm^2 shows that the central stiffener is very close to a possible fracture.

The global response of the steel caisson is apparent through careful inspection of the stress over time (Appendix X). It is apparent that the whole caisson oscillates as the kinetic energy from the impact is dissipated. The caisson continues to dampen until all the kinetic energy has been dissipated and the caisson returns to a steady state. None of the stress/strain generated by the global response is sufficient to cause any permanent damage to the caisson with only slight elasticity occurring.

Through the investigation of the elastic and plastic behaviour of the caisson in this project it is clear the caisson would be able to withstand impact from a 1500T vessel travelling at 0.5m/s with only minor permanent damage occurring. It is very important to remember that this project has excluded the hydrostatic load and self-weight due to its focus on the behaviour due to impact. Further research would need to be conducted and all loadings would need to be added to confirm the caisson has been sufficiently designed to withstand. This research would also need to consider instability type failures such as buckling and lateral torsional buckling checks.

5.7 Limitations of analysis

Due to the projects focus on the behaviour of the caisson subjected to ship impact the hydrostatic load and self-weight were excluded from the analysis due to the fear that it would hide the behaviour due to impact and make the validation too complex.

LUSAS does not have any way of implementing the hydrodynamic mass load caused by the water entrained around the hull of the vessel that continues to move after the vessel has impacted exerting an extra force upon the caisson.

The literature research carried out for this project uncovered several factors that influence ship impact forces and including all of them would have increased the runtime of the model which would require a computer with more processing power.

6. Conclusion

The comparison of the LUSAS forces to the calculated theoretical forces show that the model is valid and the results are both reliable and accurate. The validation performed via the energy equation and work done equations are less accurate for low energy which was expected due to the poor correlation between kinetic energy and volume of steel deformed found by (Minorsky, 1959).

The results show that local damage occurred in the form of a large depression across the front face and bulging across the back and top faces of the caisson. The maximum deflection during the impact was 19.59 mm and occurred within the central stiffener. As the impact energy was dissipated the caisson was unloaded and 16.89mm of the total deflection was recovered due to it being elastic deformation, this resulted in a permanent deformation of 2.70mm within the caisson due to ship impact. Oscillation of the whole caisson structure was also apparent post-impact until the kinetic energy was fully dissipated.

Comparison of the stress vs strain plots for all impacts cases shows that increased ship mass leads to an increased amount of both elastic and plastic behaviour. The increased ship mass means an increased amount of kinetic energy which needs to be dissipated by plastic strain in the steel. This increase in plastic strain subsequently causes increased plastic behaviour within the steel caisson. The maximum plastic strain was 0.15% within the central internal stiffener due to the 1500 tonne case.

The maximum stress within the caisson due to the 1500 tonne impact was 272N/mm^2 meaning the ultimate limit state of the steel is never reached and the caisson has the potential to absorb more energy before catastrophic failure occurs. This leads to the conclusion that the proposed steel caisson for installation within Devonport Royal Dockyard is sufficiently designed to safely withstand 500T – 1500T impact loads travelling at 0.5 m/s with only local plastic deformation occurring. It would be unsafe to comment on increased ship masses or increased impact speeds not tested in this project and further research would need to be conducted.

6.1 Recommendations

It would be interesting and valuable to see how the elastic and plastic behaviour changes due to further increased ship masses and impact velocities, and determining the behaviour at the point of failure. Further research would also need to include instability failure checks such as buckling and lateral torsional buckling checks to safely confirm that the caisson has been sufficiently designed.

Inclusion of the self-weight and hydrostatic loading upon the caisson would be needed to give as realistic representation as possible.

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