University of Stavanger

Faculty of Science and Technology

MASTER'S THESIS

Study program/ Specialization:	
Konstruksjoner og Materialer - Maskinkonstruksjoner	**Spring semester, 2014** Restricted access
Author(s): Tuan Minh Tran	……………………………………… (Writer's signature)

Faculty supervisor: Hirpa G. Lemu

External supervisor(s): Robert Ganski
Ole Gabrielsen

Thesis title:

"Effects of impacts from large supply vessels on jacket structures"

Credits (ECTS): 30 SP

Key words:	Pages: 43
boat impact	**+ enclosure:** 19
non-linear analysis	
finite element method	Stavanger, 16.06.2014
USFOS	

PREFACE

This thesis is submitted in partial fulfilment of the requirements for the degree of Master of Science at the University of Stavanger (UiS), Norway. The work is supported by DNV GL, Stavanger.

Part of the work of this thesis depends on some limitations and relevant assumptions, such that the results are on the conservative side.

I would first like to thank Professor Hirpa G. Lemu, for being my supervisor and his help and support during this thesis.

I would like to thank Mr Robert Ganski, for being my external supervisor and for the guidance during this thesis.

I would also like to thank Mr Ole Gabrielsen for giving me the opportunity to study this topic.

Stavanger, June 2014

Tuan Minh Tran

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SUMMARY
The primary function of a jacket structure is to support the weight of the topside structure by transferring the weight to the foundation. In addition, the jacket structure must also be designed to resist accidental loads, such as boat impact.

This thesis presents the result from high energy ship collision on a jacket structure. There will be four jacket structures that shall be subjected to high energy impact to see the effects it have on the structures.

To date, few researches have been carried out on vessel-to-jacket collisions. This thesis implements the basic design principles of ship collision and several reasonable assumptions. It is expected that the results could provide an overview of how the different potential impact locations and directions will influence the resistance capacity of the jackets. It is also anticipated that this procedure and the assumptions could be a reference for related research in the future.

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NOMENCLATURE

Abbreviations

ALS	Accidental limit States
DP	Dynamic Positioning
FEA	Finite Element Analysis
FEM	Finite Element Method
FLS	Fatigue Limit States
NCS	Norwegian Continental Shelf
NTNU	Norwegian University of Science and Technology
PTIL	Petroleumtilsynet
USL	Ultimate Limit States

1 INTRODUCTION

1.1 Background

Most jacket structures in the Norwegian Continental Shelf (NCS) are designed to resist impacts from supply vessels with a displacement of 5000 tons. This is mentioned in the current version of the NORSOK standard for design of steel structures N-004 (given in Appendix A.3).

Because of higher demands for equipment safety, the supply vessels displacement has increased over the last 5-10 years but the standards have not taken this in to consideration. This is due to the introduction of the Dynamic Positioning (DP) systems which reduced the risk of collision between ships and offshore structures. Like many new technologies, there is still a small possibility that the DP systems “fail” which may result in a catastrophic failure.

In the past 10 years, there have been a total of 26 collisions between incoming vessels and installations in the NCS [Petroleumstilsynet]. These incidents are as a result of poor organization of work and responsibilities, lack of training of personnel and the failure of technical equipment. In other words, the cause of these incidents is not because of a single factor but a number of factors. The people responsible are not just the crew of the vessels but also on the operators and the owners.

No catastrophic failure has yet occurred but many severe accidents have happened. One of them was the “Big Orange XVIII” collision with the platform Ekofisk 2/4-W in the summer of 2009 [Jacobsen, 2009]. The accident caused a lot of material damaged on the vessel and the offshore structures but no personnel were injured. In the investigation report performed by Petroleumtilsynet (Ptil) [Jacobsen, 2009] this accident is categorized as a “major accident” which means a possibility with many serious personal injuries or casualties, or sets the structural integrity in danger. Another incident was in Mars 2004 when the supply vessel “Far symphony” collided into a drilling rig in the “Trollfelt” [Petroleumstilsynet]. No personnel were injured and the material damage was less serious than the “Big Orange XVIII” incident.

1.2 The objective of the project

The objective of the project work reported in this thesis is to study the effects on jacket structures on the NCS from ship collisions where the impact energy is higher than anticipated in the design. This is initiated mainly because of the increasing vessels displacement in the NCS that indicates that the current jacket structures might not resist a potential impact because of the higher energy.

1.3 Scope of work

This Master thesis looks into the currently available knowledge regarding ship collisions with jacket structures to provide better understanding for the thesis. This includes a survey of typical supply vessels operating in the NCS.

A total of 4 jacket models were prepared for the analysis. Studying this effect is done by using USFOS, a leading computer program for nonlinear static and dynamic analysis for space frame structures. The jacket models were simulated by applying representative loading. The results were then reviewed and the consequences of the increased energies where discussed. A simplified hand calculation was also performed for better understanding of the theory.

1.4 Limitations

Since the main aim for this thesis is to study the effects on jacket structures on the NCS from ship collisions, these parameters are not taking into account:

- Ultimate Limit States (ULS)
- Fatigue Limit States (FLS)
- Service Limit States (SLS)
- Design of the jacket structures
- Snow and ice loads
- Typical extreme environmental and accidental actions such as 10−4 wave or wind loading, impact from dropped objects, earth quake, fire and explosion

On the other hand, the reported study is based on case studies of 4 different jacket structures, which all of them will be exposed to ship collision. This is because 4 legged jackets are considered weaker when it comes to structural integrity compared to 6- and 8-legged. In the modelling work of the case studies the installation was assumed to be a “soft” body and the supply vessels were modelled as “rigid” bodies.

1.5 Organisation of the Thesis

This thesis consists of seven chapters.

CHAPTER 2 introduces the scope of the literature collection and related theories behind the procedure of the analyses.

CHAPTER 3 present a survey of typical supply vessels operating on the NCS.

CHAPTER 4 describe the process of converting structural model to USFOS file and the preparation process before the analysis.

CHAPTER 5 present a simplified hand calculation based in order to understand the theory.

CHAPTER 6 discusses the result from the analysis.

CHAPTER 7 present the conclusion and the suggestion for future work.

2 LITERATURE STUDY

2.1 Introduction

To capture the current knowledge regarding ship collisions with jacket structures, a number of standards, books, case studies and technical reports are collected to get a better understanding of the issue.

Since the probability of a supply vessel colliding in a jacket structure is small, research on this field is limited. Case studies like “High-Energy Ship Collision with Jacket Legs” [Amdahl, 2001] and “Ship collision with offshore structure” [Amdahl, 1993] has given a reasonably picture of the effect of ship collision with jacket structures. The result from these studies can be used to compare with this thesis but with caution since both case studies have different criteria and limitations. A case study from the book “Nonlinear Analysis of Offshore Structures” [Skallerud, 2002] will also be used to compare.

2.2 Limit states

A limit state is a state where a structure no longer meets the requirements laid down for its performance or operation [NTS, 2004]. The limit states can be categorized into four different types:

- Ultimate limit states (ULS)
- Serviceability limit states (SLS)
- Fatigue limit states (FLS)
- Accidental limit states (ALS)

These categories are given in both in NORSOK N-001 and ISO19900. When considering the technical and operational safety of the design of the structure, all four categorise should be checked. Since this thesis only focus on ALS, the rest will be neglected.

2.3 Finite Element Method/Analysis

As stated in [Robert, 2001], Finite Element Method (FEM) is a method for numerical solution of field problems that may be a differential equation or an integral expression. This problem requires a distribution of one or more dependent variable such as distribution of stresses and displacement in an offshore structure.

A single finite element can be visualized as a small piece of structure. As illustrated by the finite element model of gear tooth in Figure 1, the finite elements are attached to each other at nodes. An assembly of finite elements are called finite element structures while the arrangements of the elements are called mesh.

FEM can be applied in problems like heat transfer, stress analysis and magnetic field.

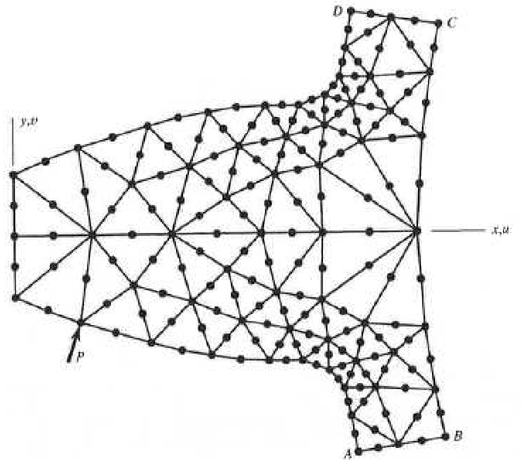


Figure 1 A two dimensional model with elements and nodes [Robert, 2001]

A typical finite element model structure is described by the element displacements at the nodes and the element has in general 6 degree of freedom. Figure 2 shows an example of a three-dimensional beam element. Usually these types of beam elements are more preferable for a linear analysis since a nonlinear analysis requires a more detailed shell finite element modelling of the structure.

For the sake of simplicity, this thesis utilized a three-dimensional beam element.

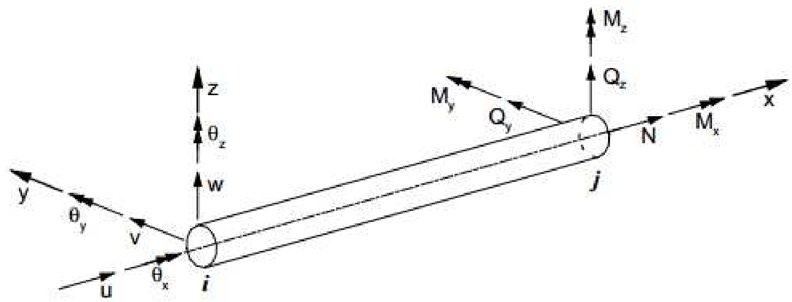


Figure 2 Three dimensional beam element [USFOS, 1999]

2.4.3 Procedure of FEA

Figure 3 presents a basic procedure of solving a problem by FEM. It is not unusually that the steps are repeated more than one cycle. The different steps are briefly presented below [Robert, 2001].

2.4.1 Classification of problems

The first step is to get an understanding of the problem. It is not possible to perform a finite element analysis (FEA) without a proper clarification of the problem at hand. Even though finite element software has the purpose to give the analysis a better capability in decision-making, getting an overall understanding over the problem will decrease the chances of making error during the analysis.

2.4.2 Preparation of mathematical model

After a clarification of the problem is done, the next step is to create a model for the analysis. The model presents the closest to real physical problems. During modelling, the analysis will remove details that are unnecessary and add essential features. The purpose is to make the analysis of the model as accurate as possible, without being too complicated, and give accurate results. Typical simplifications would be ignoring geometric irregularities, regards some loads as concentrated, material may be presented as linear and isotropic and assume some supports are fixed though they are in reality not fully fixed.

2.4.3 Preliminary Analysis

Performing a preliminary analysis before the FEA will give a sense of what kind of result can be expected to get. A typical preliminary analysis can be like a simple analytical calculation, handbook formulas, trusted previous solutions or experiments.

2.4.4 Finite Element Analysis

This step can be split into three minor steps. These steps are briefly mentioned.

Preprocessing: The software receives input data that describes geometry, material properties, load and boundary condition.

Numerical analysis: Matrices are created to show behaviour of elements. These matrices combined into large matrix equation to solve this equation to determine value of field quantities of nodes.

Postprocessing: Solution and quantities from the FEA are being presented in this step. It can be presented as list or graphical display.

2.4.5. Review of results

As mentioned in the preliminary analysis step, the result from the software should be compared with the result from the preliminary analysis. By comparing the two analyses, error and deviation are easier to spot. If such error and deviation are spotted, the analyst should repeat some of the previous steps based on the error.

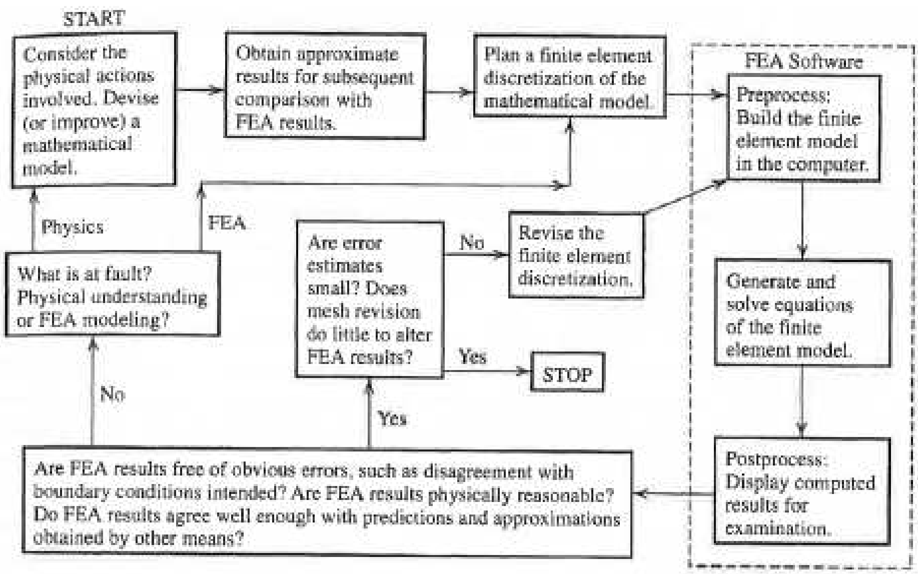


Figure 3 Flow-chart for problem solving by FEA [Robert, 2001]

2.5 Nonlinear Analysis Method

In this thesis work, a nonlinear finite element analysis will be simulated on a structural response to a ship collision. When it comes to nonlinearity behaviour, there are three things that need to be taken into account [Robert, 2001]:

- Material nonlinearity, in which material properties are functions of state of stress or strain. Examples include nonlinear elasticity, plasticity and creep.
- Contact nonlinearity, in which a gap between adjacent parts may open or close, the contact area between parts changes as the contact force changes, or there is sliding contact with frictional forces.
- Geometric nonlinearity, in which deformation is large enough that equilibrium equations must be written with respect to the deformed structural geometry. Also, loads may change direction as they increase, as when pressure inflates a membrane.

2.6 Linear vs. nonlinear analysis

To get a better understanding on nonlinear analysis, this section presents the main differences between linear and non-linear analysis in typical finite element software.

Consider the geometry difference in a linear analysis the geometry remain the same by the applied loads during the equation solving. While in a nonlinear analysis, the geometries are being updated because of the equation systems are being updated and solved repeatedly. Figure 4 illustrates an example of a system where a column supports a beam. As illustrated in the left hand figure, when the load P is applied the beam is subjected to bending only since column carries the axial compression. This is a typical linear analysis phenomenon. In the case of the right side which is a nonlinear analysis phenomenon, deformation increases and the beam become stiffer while the column begins to buckle.

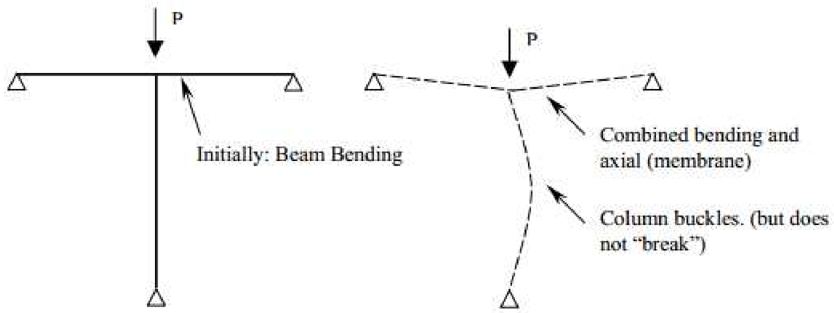


Figure 4 Comparing linear- and nonlinear analysis [USFOS, 1999]

Another difference is illustrated in Figure 5when it comes to material parameters. In a linear analysis, it is applicable until yielding is reached while nonlinear analysis can continue all the way to fracture. In other words, linear analysis is valid in the elastic range while nonlinear analysis is also valid in the plastic range.

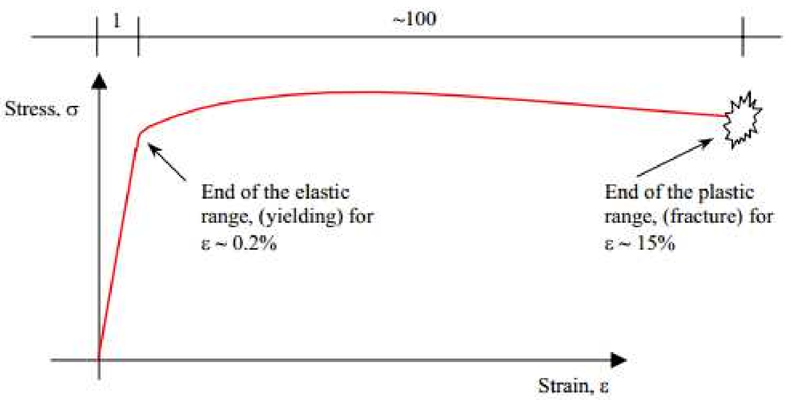


Figure 5 Example of stress-strain curve for steel [USFOS, 1999]

2.7 Description of USFOS

USFOS is a computer program for nonlinear static and dynamic analysis of frame structures and its main aim is for ultimate strength and progressive collapse analysis. It was developed by SINTEF marintek and the Norwegian University of Science and Technology (NTNU) and has been in commercial use since 1985 [USFOS].

The formulation behind USFOS is valid for large displacements, but restricted to moderate strains. It is based on an updated Lagrange formulation called Green strains and defined by:

$$\varepsilon_x = u_{,x} + \frac{1}{2}u_{,x}^2 + \frac{1}{2}v_{,x}^2 \frac{1}{2}w_{,x}^2 \qquad \mathbf{1}$$

where u, v and w are axial displacement and lateral deflection in three dimensions and $u_{,x}$, $v_{,x}$ and $w_{,x}$ are the first derivatives of the displacements u, v and w respectively. For moderate element deflection, the von Karman approximation applies, and ε_x simplifies into:

$$\varepsilon_x = u_{,x} + \frac{1}{2}v_{,x}^2 \frac{1}{2}w_{,x}^2 \qquad \mathbf{2}$$

The stiffness formulation of USFOS is derived from potential energy consideration or the virtual work principle. For an elastic beam element the internal strain energy reads:

$$U = \frac{1}{2}\int_0^1 EA(u_{,x} + \frac{1}{2}u_{,x}^2 + \frac{1}{2}v_{,x}^2 \frac{1}{2}w_{,x}^2)^2 dx + \frac{1}{2}\int_0^1 (EI_z\, v_{,xx}^2 + EI_y\, w_{,xx}^2) dx \qquad \mathbf{3}$$

where EA and EI are the axial and the bending stiffness parameters respectively.

2.7 Jacket structures exposed to Ships collision

2.7.1 Design principles

The methods concerning how to approach ship-to-jacket collision can be found in design codes NORSOK N-004 or DNV RP-C204.

According to N-004 [NTS, 2004], a ship collision may be defined as a kinetic energy, which consists of the mass of the ship which includes the hydrodynamic added mass and the velocity of the ship at the moment of impact. After an impact, some of the kinetic energy will remain as kinetic energy and the rest will be dissipated as strain energy. The strain energy dissipates in the installation and the vessel, and will inflict large plastic strains and structural damage.

Figure 6 shows the distribution of the strain energy dissipation. The different distribution can be described as:

- Strength design
- Ductility design
- Shared-energy design

In this thesis, the ductile design is taken into account. This means that the jacket will dissipate most of the collision energy, in other words the jacket is viewed as a "soft" body while the vessel is viewed as a "rigid" body.

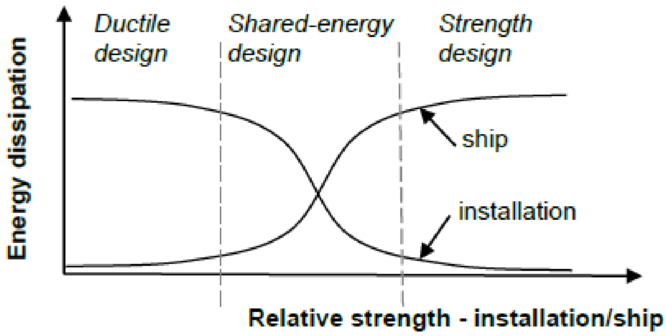


Figure 6 Energy dissipation for different design [NTS, 2004]

The collision energy to be dissipated as strain energy may be taken as [NTS, 2004]:

$$E_s = \frac{1}{2}(m_s + a_s)v_s^2 \qquad \textbf{4}$$

where m_s is the ship mass, a_s is the ship added mass and v_s is the impact speed.

2.7.2 Force-deformation relationships for beams

As stated in NORSOK N-0004, a beams behaviour that is subjected by a collision load is initially determined by bending but if local buckling occurs in the compression side, the bending capacity will decrease. Bending is affected by and interacts with the local denting under load. During beam deformation, the load carrying capacity may increase because of the development of membrane tension forces. This behaviour is governed by the nearby structure capacity to supress the connections at the members ends to inward displacements. This means that the energy dissipation capacity is restricted by tension failure of the member or rupture of the connection as long as the connection doesn't fail. It is also stated that simple plastic methods of analysis are acceptable except in special cases, where these effects must be taken into account [NTS, 2004]:

- elastic flexibility of member/adjacent structure
- local deformation of cross-section
- local buckling
- strength of connections
- strength of adjacent structure
- fracture

Plastic force-deformation relationships including elastic and axial flexibility: Relatively small axial displacements have a significant influence on the development of tensile forces in members undergoing large lateral deformations. An equivalent elastic, axial stiffness may be defined as [NTS, 2004]:

$$\frac{1}{K} = \frac{1}{K_{node}} + \frac{\ell}{2EA} \qquad \textbf{5}$$

where K_{node} is the axial stiffness of node with the considered member removed, ℓ is the length of the beam, E is the modulus section of material and A is a cross section area. For the case that contact point is at mid span, the plastic collapse resistance in bending for the member can be given as [NTS, 2004]:

$$R_0 = \frac{4c_1 M_P}{\ell} \qquad \mathbf{6}$$

where M_P is the plastic moment of cross section, c_1 can be chosen between $c_1 = 1$ (for pinned beam) and $c_1 = 2$ (for clamped beam). The non-dimensional deformation can be given as [NTS, 2004]:

$$\bar{w} = \frac{w}{c_1 w_c} \qquad \mathbf{7}$$

where $w_c = \frac{D}{2}$ is the characteristic deformation for tubular beam, w is is the beam deflection. The non-dimensional spring stiffness can be given as [NTS, 2004]:

$$c = \frac{4c_1 K w_c^2}{f_y A\ell} \qquad \mathbf{8}$$

where f_y is the yield stress of material. For a plastic-deformation relationship for a central collision may be acquire from Figure 7.

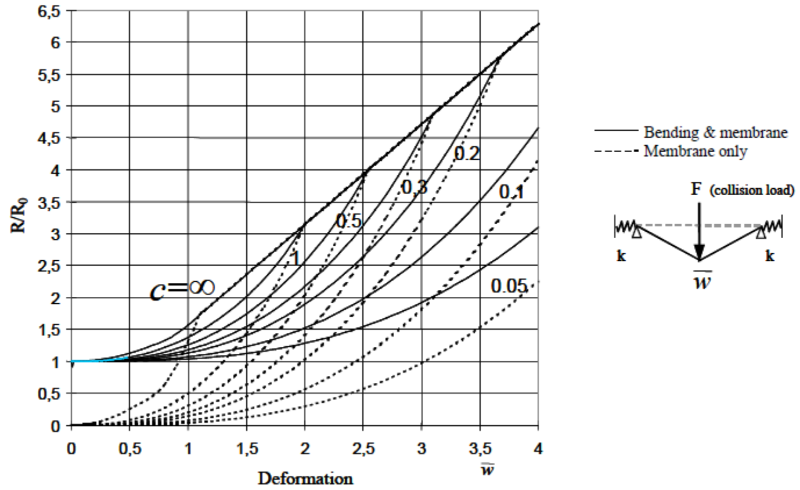


Figure 7 Force-deformation relationship for tubular beam with axial flexibility

Ductility limits: The dissipation of the energy from the impacted member is governed by either local buckling on the compressive side or fracture on the tensile side of the cross-sections undergoing finite rotation [NTS, 2004].

Buckling does not need to be considered for a beam with circular cross-section with axial restraints if the following condition is fulfilled [NTS, 2004]:

$$\beta = \left(\frac{14 c_f f_y}{c_1}\left(\frac{k\ell}{d_c}\right)^2\right)^{\frac{1}{3}} \qquad \mathbf{9}$$

where d_c is the characteristic dimension, $k\ell \leq 0.5\ell$ is the smaller distance from location of collision load to adjacent joint and β can be given as [NTS, 2004]:

$$\beta = \frac{D/t}{235/f_y} \qquad \mathbf{10}$$

where D is the diameter for circular cross-section and t is the thickness. Suppose this condition is not met, buckling may be assumed to take place when the lateral deformation exceeds [NTS, 2004]:

$$\frac{w}{d_c} = \frac{1}{2c_f}(1 - \sqrt{1 - \frac{14c_f f_y}{c_1 \beta^3} * (\frac{k\ell}{d_c})^2}) \quad \mathbf{11}$$

where c_f is the axial flexibility factor and is given as [NTS, 2004]:

$$c_f = (\frac{\sqrt{c}}{1+\sqrt{c}})^2 \quad \mathbf{12}$$

When force deformation relationships for beams are sued rupture may be assumed to occur when the deformation exceeds a value given by [NTS, 2004]:

$$\frac{w}{d_c} = \frac{c_1}{2c_f}(1 - \sqrt{1 - \frac{14c_f f_y}{c_1 \beta^3} * (\frac{k\ell}{d_c})^2}) \quad \mathbf{13}$$

where c_w is the displacement factor and given by [NTS, 2004]:

$$c_w = \frac{1}{c_1}(c_{lp}\left(1 - \frac{1}{3}c_{lp}\right) + 4(1 - \frac{W}{W_P})\frac{\varepsilon_y}{\varepsilon_{cr}})\,(\frac{k\ell}{d_c})^2 \quad \mathbf{14}$$

and c_{lp} is the plastic zone length factor and given by [NTS, 2004]:

$$c_{lp} = \frac{(\frac{\varepsilon_{cr}}{\varepsilon_y} - 1)\frac{W}{W_P}H}{\left(\frac{\varepsilon_{cr}}{\varepsilon_y} - 1\right)\frac{W}{W_P}H + 1} \quad \mathbf{15}$$

where ε_y is the yield strain and ε_{cr} is the critical strain [NTS, 2004].

3 OFFSHORE SUPPLY VESSELS OPERATING IN THE NCS

3.1 Introduction

A supply vessel Figure 8 has the main purpose to supply offshore installation with supplies. The length of these ships can range from 20 to 100 meters.

The typical supplies that the supply vessel brings are drilling mud, pulverized cement, diesel fuel, potable and non-potable water, fuel, water, and chemicals used in the drilling process comprise the bulk of the cargo spaces. Some chemicals will be transported to shore for recycling or disposal. Some supply vessels also have a particular task such as firefighting capabilities for fighting platform fires or oil containment and recovery equipment to assist in the clean-up of an oil spill [Tromsoffshore].

Figure 8 presents the maritime activities in the NCS. Note that the green "spots" are offshore supply vessels.

Figure 8 Reim HRIST [Marine]

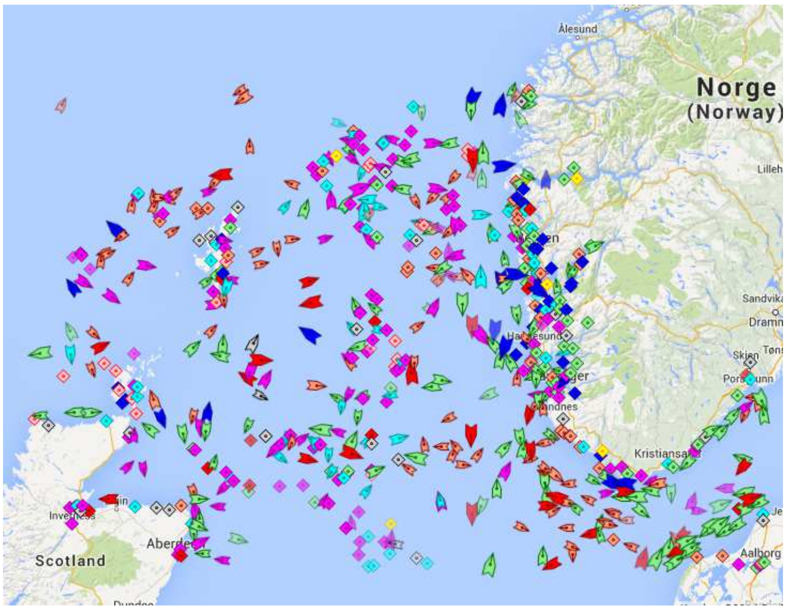


Figure 9 Maritime activities in NCS [Marine]

3.3 Trends of offshore supply vessels

Figure 10 illustrates the year of built-deadweight relationship of offshore supply vessels in the NCS. The list of all supply vessels used in this graph can be found in Appendix B shows that the deadweight of the supply vessels varies from 1400 metric tonnes to 6200 metric tonnes. As depicted in the Figure 10 there is a clear sign of increased displacement in the supply over the past decades.

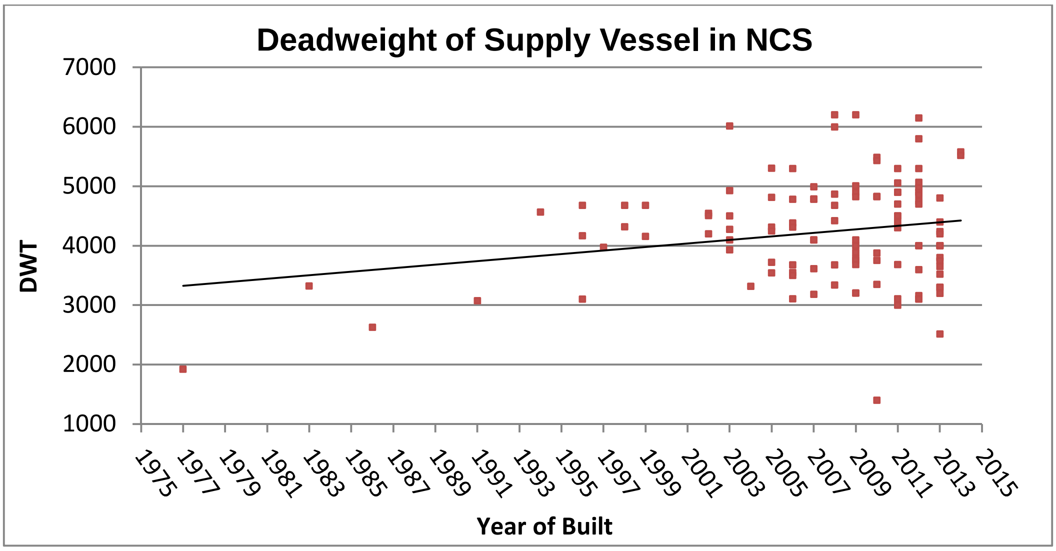


Figure 10 Deadweight of Supply Vessel in NCS and year of built relationship

According to NORSOK N-004, to avoid possible penetration of a cargo tank, the side structure of the unit shall be capable of absorbing the energy of a vessel collision with an annual probability of 10^{-4} or at least a vessel of 5000 tonnes with an impacting speed of 2 m/s [NTS, 2004]. Figure 11 represents the impact energy of supply vessels based on the year of built. The impact energy is calculated by using the formula found in Section 2.7. For bow and stern impact, the added mass is 10% of the ships mass while the broad side impact (sideway) is 40% of the ships mass [NTS, 2007]. For bow and stern impact the value varies from 3 MJ to 14 MJ and for broad side impact the value varies from 4 MJ to 17 MJ. The calculation can be found in Appendix C.

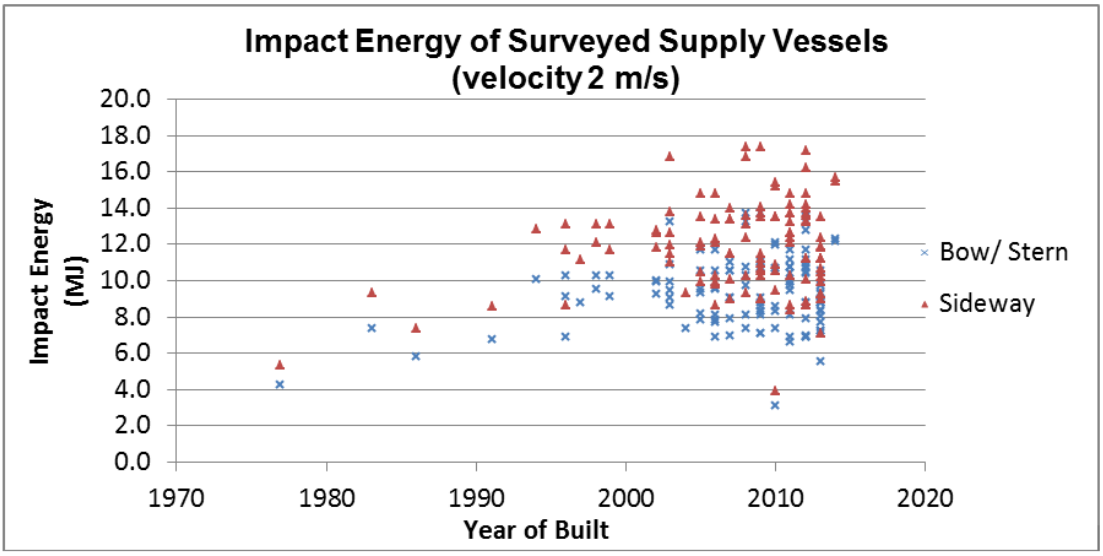


Figure 11 Impact Energy of Supply Vessel in NCS and year of built relationship

4 FE modelling of jackets in USFOS

4.1 Introduction

As mentioned before, this thesis does not take jacket design into the account. The four jacket models that are being used in the analysis are already modelled. This chapter covers the preparation before doing the analysis, like converting the jacket models to USFOS files and then checking that no error has occurred during the converting. The different models have been designated by the letters E, F, G, and H.

4.2 Converting jacket models to USFOS

The existing jacket models were already prepared by GeniE, a software for designing and analysing offshore and maritime structure [DNV, 2011b]. The connection between GeniE and USFOS goes through the SESAM FEM file. As presented in Figure 12, the original model from GeniE becomes "red only" so that an "Intelligent filter" converts the linear model into a model that USFOS can use. The structural information from that model can then be interpreted by USFOS and used directly. Relevant structural information like cross section shape and orientation, element end offset and material properties. Other parameters like load cases, boundary conditions and concentrated mass data are also extracted from the FEM file. Some parameters like hydrodynamic data have to be specified according to USFOS and foundation data can be used through the utility tool "soil" [USFOS, 1999b].

Figure 12 Flow-chart for converting

During the converting phase, the unit defined from GeniE does not change, so that the numbers can be used directly. In this thesis, the basic SI units are employed.

4.3 Extracted structural information

4.3.1 Structural design and material properties

The four jacket models are all 4 legged. A graphical view of the four jackets is also shown in Figure 13. As shown, all four jackets have fully X-braced pattern which gives the jacket a higher horizontal stiffness, ductility and redundancy. The drawback with this design is that it requires a high volume of welding because of the "crowded" pattern [Chakrabarti, 2005].

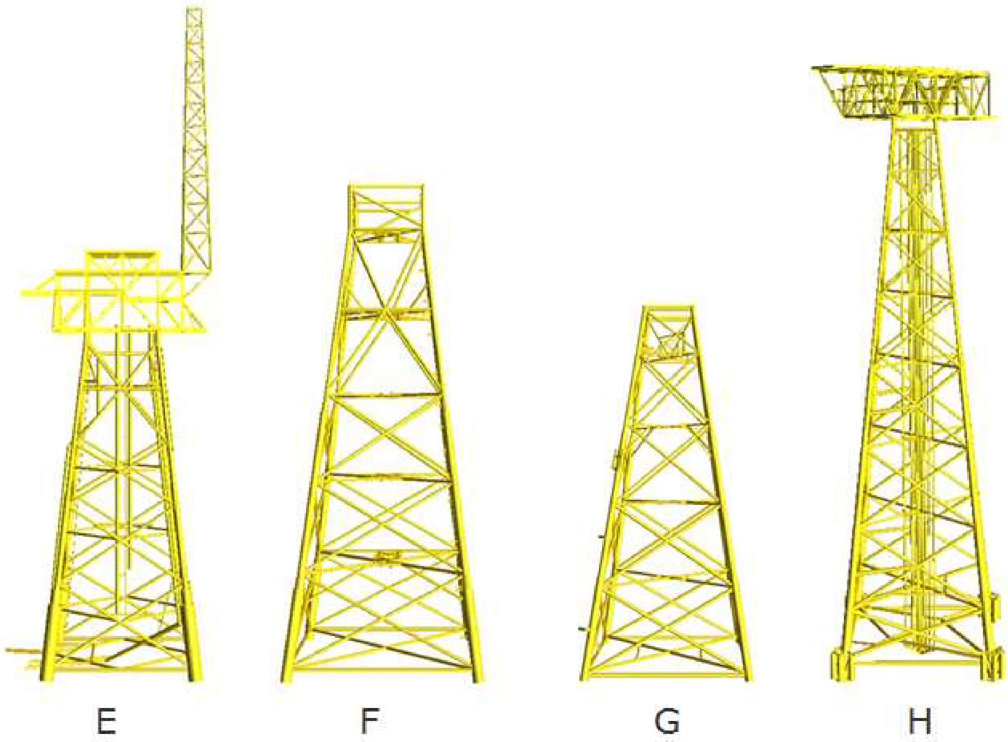


Figure 13 The four jacket models

As mentioned earlier, USFOS also extract the material properties from the FEM file. All the jacket parts considered in this this thesis are made of steel grades with the following physical properties:

- Density: ρ = 7850 kg/m^3
- Young's module: E = 210 GPa
- Poison's ratio: ν = 0,3 and
- Thermal expansion coefficient: $\alpha = 1.2 \cdot 10^{-5}/°\mathrm{C}$

On the other hand, strength properties of the steel grades vary from jacket to jacket, Table 1 presents the yield strength for the different jackets. Because this thesis only focuses on jacket structures, yield strength for topside is not taken into account.

All jacket foundations are set as fixed which give restraint against translation and rotation on each of the pile sleeves

Table 1 Jacket steel material

Jacket	Yield strength (MPa)
E	310 - 355
F	355
G	355 – 460
H	355

4.3.2 Load cases

As mention before, load cases can also be extracted from FEM files. Table 2 gives an overview of the four jackets' load cases that are considered in this thesis. As presented, self-weight of the jacket and topside are considered in all four jackets. The one jacket that are different when it comes to number of load cases are jacket E which also includes buoyancy and the weight of the flame tower.

Table 2 Load table

Jacket	Mass of jacket (Metric Tonne)	Mass of topside (Metric Tonne)	Buoyancy (Mega Newton)
E	3703	4907	11.87
F	4306	6128	N/A
G	2778	6276	N/A
H	10776	6250	N/A

In reality an offshore structure is exposed to more loads than what has been covered in this thesis. For instance, the following load cases are neglected:

- Environmental load (wave, wind, snow...etc.)
- Marine growth

It should be mentioned that jacket F and G doesn't have topside in the model. To compensate for the missing topside model, a node load has been assigned on each of the top of the legs as illustrated in Figure 14.

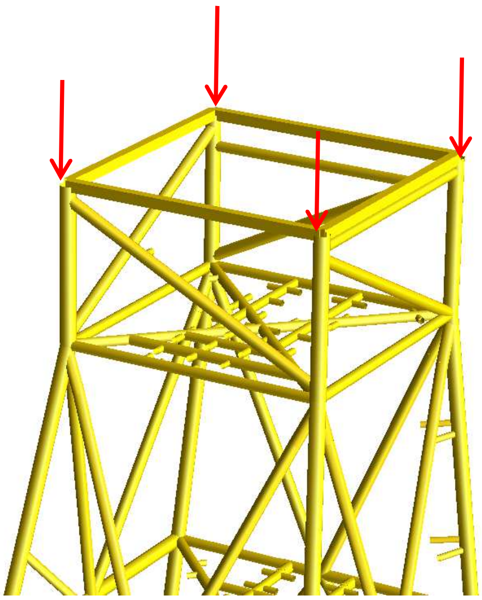


Figure 14 Jacket without topside model

5 Hand Calculation

5.1 Introduction

This chapter present a simplified hand calculation. The methods will be based on the formulas from Chapter 2.7.2. Because of time limitation, this thesis will only perform the hand calculation based on a single impact on a diagonal brace. The calculation was performed in Mathcad and the can be found in Appendix D. Only the important steps are presented.

Because of time limitation, some preparation for the hand calculation has been left out, which mean that the result from the calculation will not be comparable with the result from USFOS.

5.2 Calculation

The chosen jacket for this calculation will be jacket E. The location for the impact shall be on a single diagonal brace mid-span as shown in Figure 15.

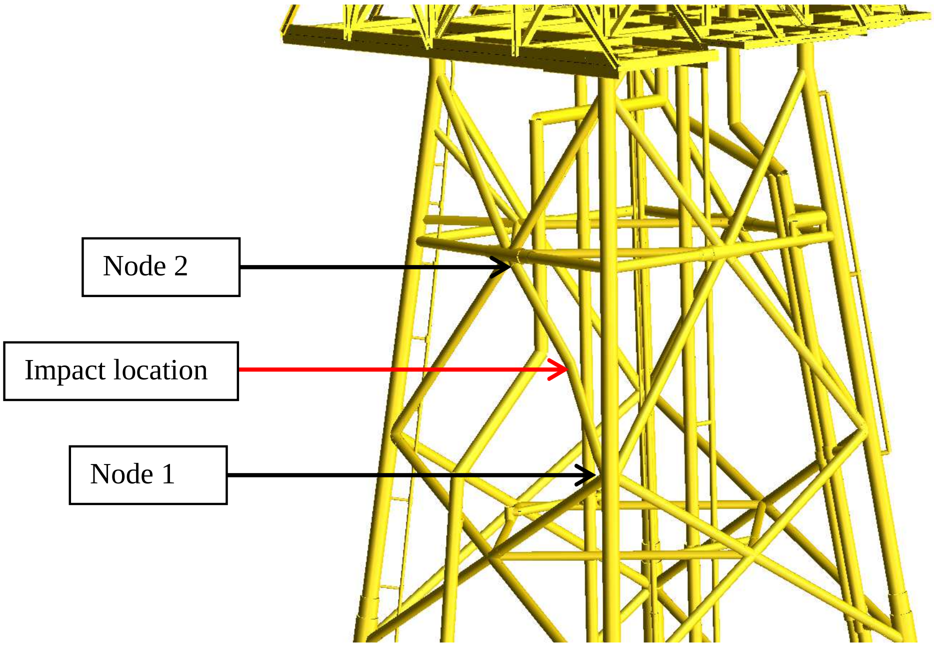


Figure 15 Jacket subjected to ship impact

The brace dimensions are 1300 x 80mm. The stiffness of nodes 1 and 2 against displacement in the brace direction is 736 MN/m and 51 MN/M respectively, when the brace is removed. These values from the stiffness of nodes 1 and 2 are given in NORSOK N-004 [NTS, 2004] from examples which mean the result from this calculation must be treated as conservative.

Then we calculate representative stiffness $k_{node} = 95.39\, MN/m$, so that we can find the effective stiffness $k = 94.55\, MN/m$. Assuming clamped ends $c_1 = 2$ the non-dimensional spring stiffness can be obtained, $c = 0.245$.

Buckling will occur when $\beta > \beta_1$. In this case, $\beta = 24.548$ and $\beta_1 = 17.964$. Which mean buckling occur and the critical deformation (13) is $w = 1.307m$.

The non-dimensional deformation is $\overline{w} = 1.005m$.

From Figure 6, the total resistance is found to be $R = 5.644MN$.

6 BOAT IMPACT

6.1 Introduction

Offshore installations are constantly in need of supply and services from the mainland. Transportation of supply is mainly done by sea using offshore supply vessels. Because of the constant demand for supply from the offshore installation, a certain risk of ship collision on offshore structure is possible. Risk analysis of planned jacket installations has shown that collision with passing vessels, with a kinetic energy in the range of 40-50 MJ, is a potential hazard [Amdahl, 2001]. Even though DP systems have minimized that risk, there is still a small possibility that DP systems fail or a human error may occur.

This chapter present the behaviour of the four jackets when an offshore supply vessel impact on a jacket structure. Both global and local effects shall be evaluated. The impact energy is given as 90MJ, a random number since an integrated algorithm for ship impact analysis is accounting for [Søreide, 1981]:

- Local deformation of the tube wall at the point of impact
- Beam deformation of hit member
- Global deformation of the platform

For single element, joint failure and plastic strain are used as the failure criteria. Local denting is not considered in this thesis.

The value for the critical strain for various steel material grades are given in NORSOK N-004 and are presented in Table 3 [NTS, 2004].

Table 3 Proposed value for critical strain for different steel grads

Steel grade	Critical strain
S 235	0.20
S 355	0.15
S 420	0.12
S 460	0.10

6.2 Impact scenarios

As mentioned in the scope of work, the analysis will focus on local and global effects. This means that every jacket will have single element impact and a multiple impact scenarios. The different impact locations for each model considered in this study are presented in Figure 16 below. The definition of the different load cases considered for the models are also defined in Table 4, including multiple load cases.

Since this thesis only focus on the response behaviour of the jacket, the impact location is chosen without considering zones which have high possibility of being exposed to impact for supply vessels [Skallerud, 2002].

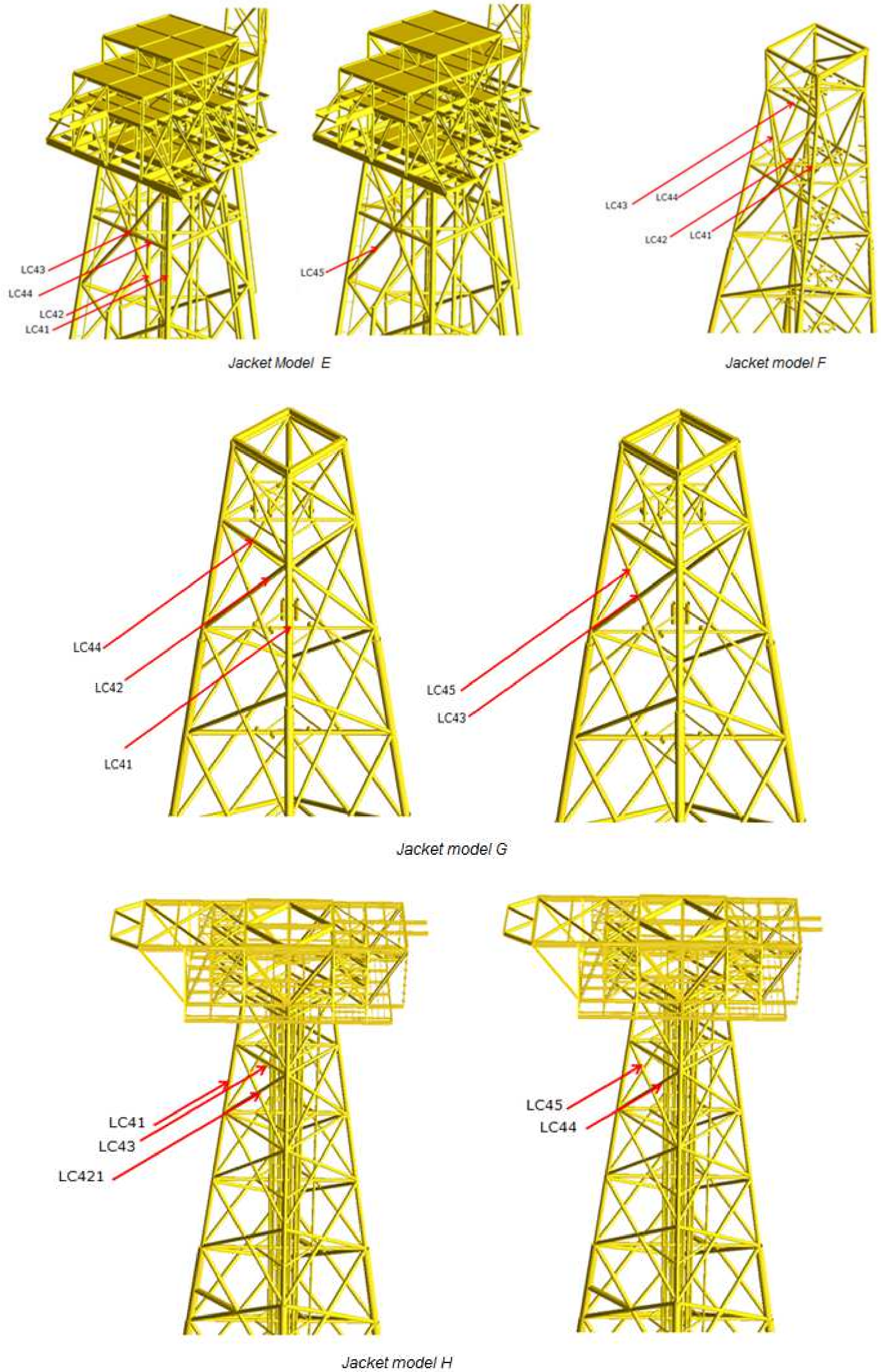


Figure 16 Location of impact points for each jacket model

Table 4 Definition of different load cases for each model

Load case	Description	Applicable for model
41	Single impact on jacket leg	All models
42	Single impact on diagonal brace	E, F and G
421	Single impact on x brace cross	H
43	Single impact on joint	E
	Single impact on horizontal brace	F, G and H
44	Single impact on horizontal brace	E
	Single impact on x brace cross	G
	Single impact on diagonal brace	H
42+45	Multiple impact on two diagonal braces	E and G
42+44	Multiple impact on two diagonal braces	F
44+45	Multiple impact on two diagonal braces	H

6.3 Analysis procedure

The first load applied in the analysis is the self-weight. Right after the first load, the next load which are introduced is the boat impact. USFOS uses an input command called “BIMPACT” for static analysis of collision. This command is used to define ship impact load. When the total impact energy has been dissipated, the impact load will be unloaded into a separate program-defined load case. The impact will be terminated if fracture occurs. In a scenario which requires multiple impacts, USFOS uses the “MULT_IMP” command. This command allows several BIMPACT to be executed in a sequence. In other words, the remaining energy from the first fractured element will be moved to the next specified element [USFOS, 1999b].

6.4 Result of impact scenarios

This section presents the results from the finite element analysis done in USFOS. The energy and the deformation will be plotted in graphs and a 3D graphic display of the impact location will be presented in the moment of failure.

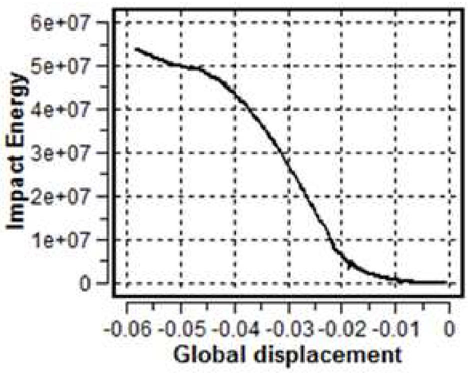
6.4.1 Jacket E

The energy deformation relationship for the five impact scenarios are given in Figure 17. The first load case 41 which is on the leg (mid-span) give the energy 54MJ before fracture occur as displayed in Figure 14. Plastic hinge is formed in both joint that connects the impacted member and there is also some plastification along the leg and the adjacent braces.

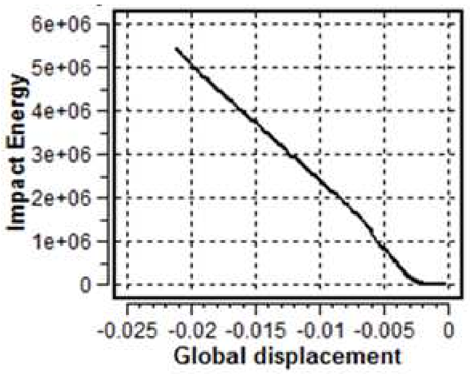
In the other hand, the horizontal and diagonal brace takes much lesser energy. Fracture occurs when the energy is 5 MJ for diagonal brace and 4 MJ for horizontal brace. As displayed in Figure 18 the effect is local since plastic deformation only appears in the impacted member. The global

Impact on the joint is considered as a “strong” point. The highest energy is 36 MJ before fracture occur. The difference in this case compared to the other cases is that fracture occurs in another element than the impacted element. Because of the limitation of USFOS as mention in Section 6.3, the joint can be considered to take more energy than 36 MJ.

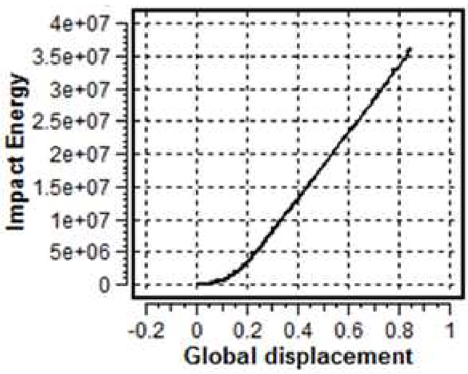
The multiple impact scenarios are a combination of two diagonal braces impacted in sequence. The first sequence is the load case 42, which has been mention previously. When fracture occurred in the first sequence, the remaining energy will be moved to the next impact location which is the second diagonal brace next to the first one. This element fracture at energy level at 6 MJ which give a total energy 11 MJ.



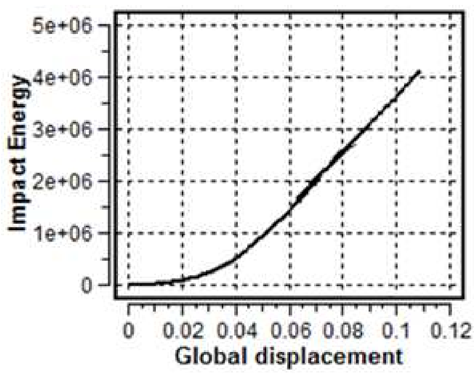
Jacket E – Load Case 41



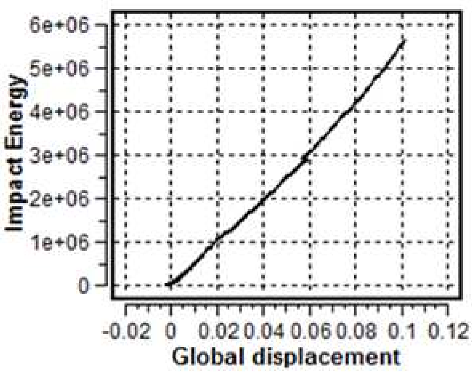
Jacket E – Load Case 42



Jacket E – Load Case 43

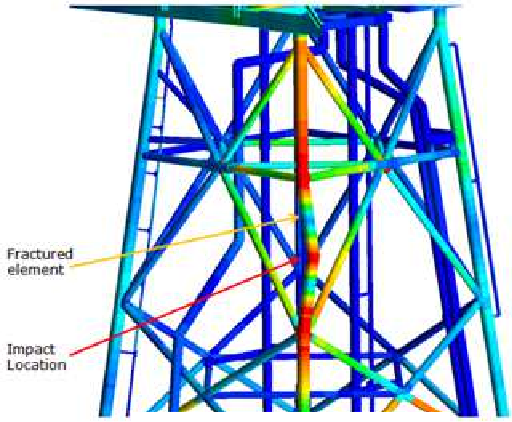


Jacket E – Load Case 44

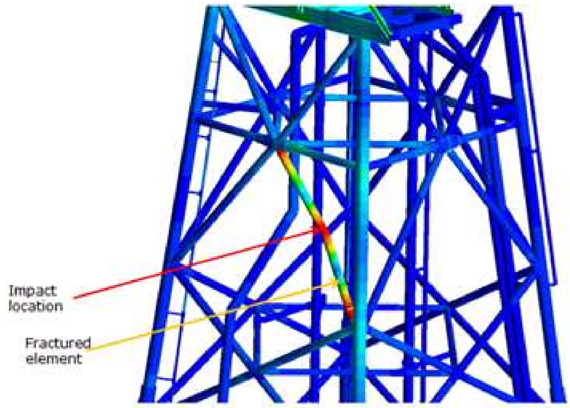


Jacket E – Load Case 45

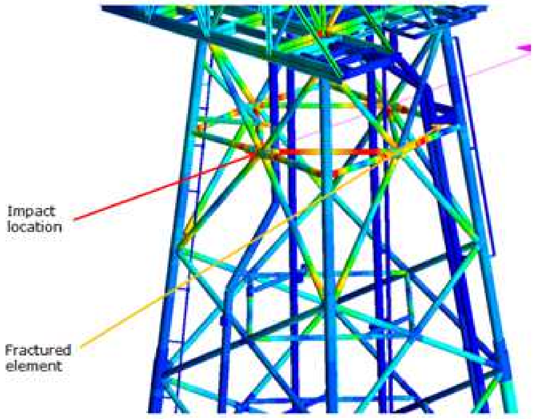
Figure 17 Global displacement vs. impact energy for jacket E



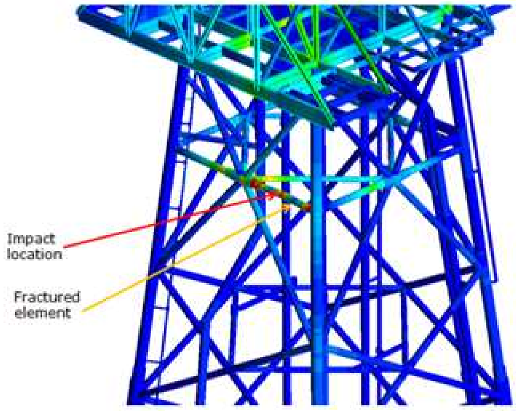
Jacket E – Load Case 41



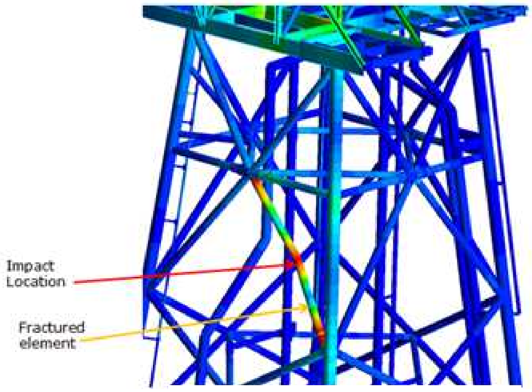
Jacket E – Load Case 42



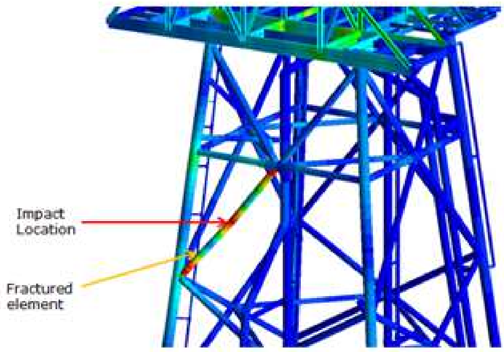
Jacket E – Load Case 43



Jacket E – Load Case 44



Jacket E – Load Case 41+45



Jacket E – Load Case 41+45

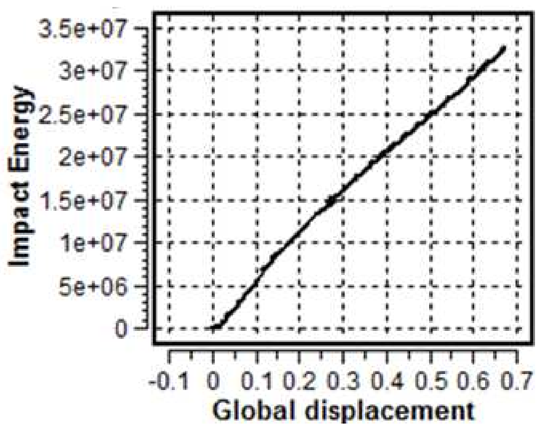
Figure 18 Plastic utilization for jacket E

6.4.2 Jacket F

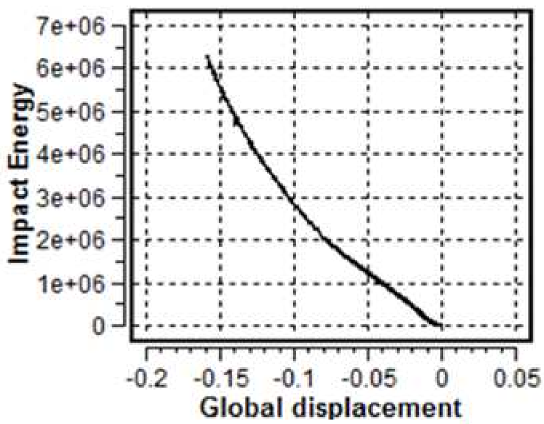
The energy deformation relationship for the four impact scenarios is given in Figure 19. The impact location on the leg (mid-span) gives the energy 33 MJ before fracture occurs as displayed in Figure 20. Some Plastic hinge and plastification can be spotted around the impact location.

Compared to jacket E, the horizontal and diagonal brace can withstand more. Fracture occurs when the energy is 6 MJ for diagonal brace and 26 MJ for horizontal brace. Both cases can be considered as local effects since no other elements except for the impact element experience any plastic hinges or plastification.

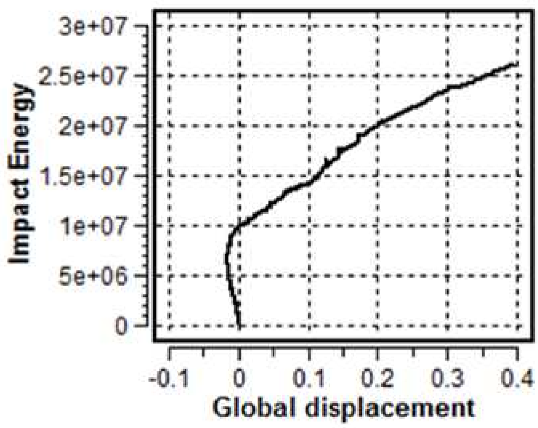
The multiple impact scenarios is similar to jacket E, a combination of two diagonal braces impacted in sequence. First sequence reaches an energy level at 6 MJ where the impacted member fracture and the remaining energy is transfer to the neighbouring brace where it reaches to 6 MJ until fracture occur. This gives a total of 12 MJ.



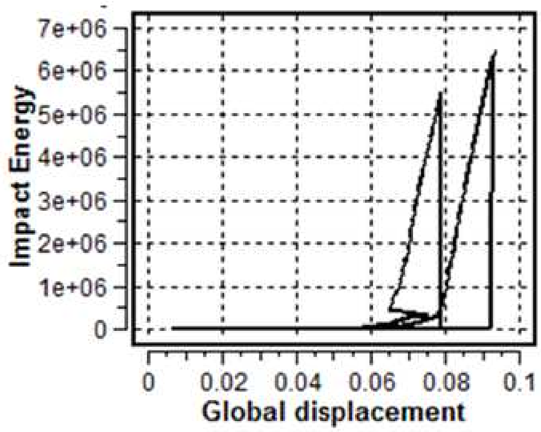
Jacket F – Load Case 41



Jacket F – Load Case 42

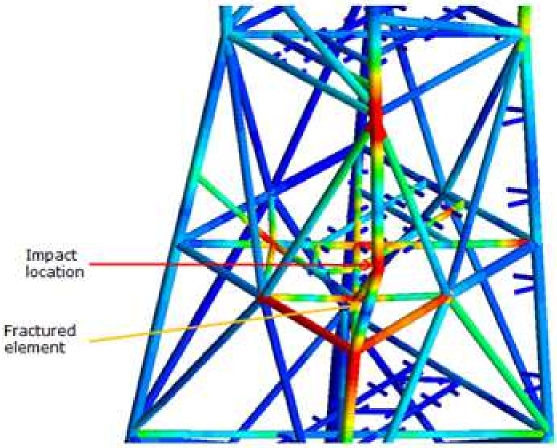


Jacket F – Load Case 43

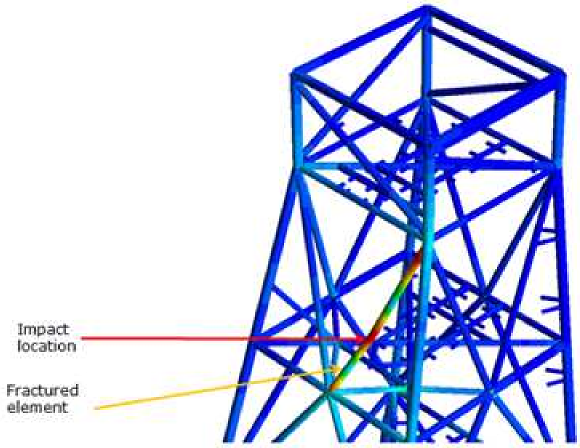


Jacket F – Load Case 42+44

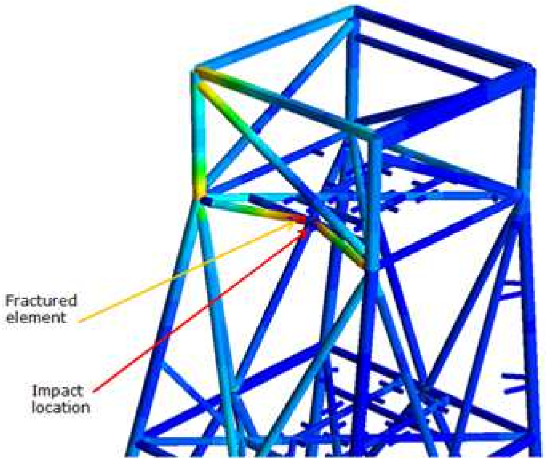
Figure 19 Global displacement vs. impact energy for jacket F



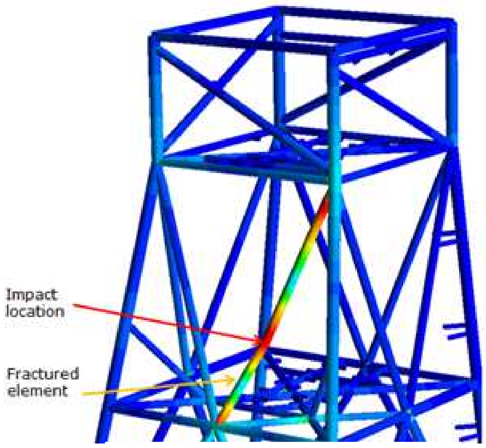
Jacket F – Load Case 41



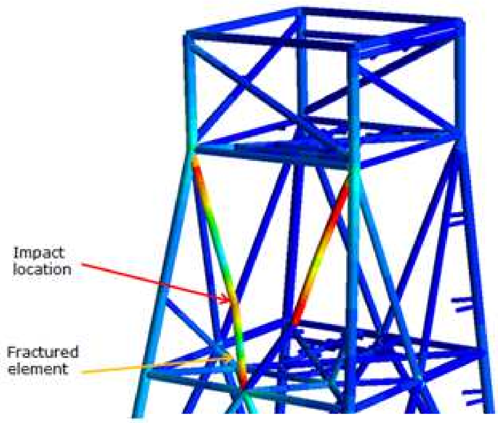
Jacket F – Load Case 42



Jacket F – Load Case 43



Jacket F – Load Case 42+44



Jacket F – Load Case 42+44

Figure 20 Plastic utilization for jacket F

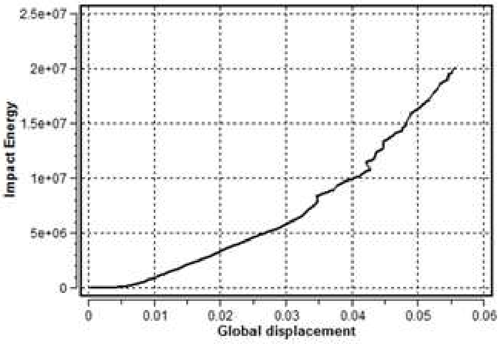
6.4.3 Jacket G

The energy deformation relationship for the five impact scenarios is given in Figure 21. The impact location on the leg (mid-span) fracture at a much lower energy level 21 MJ compared to the jacket E, and F. As the same for the other leg impact, some plastic hinge and plastification can be spotted around the impact location.

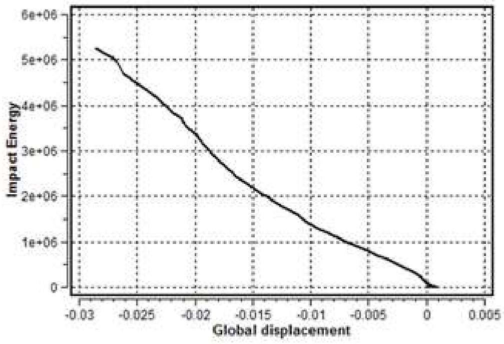
The horizontal and diagonal brace gives a comparable energy level as jacket F. Fracture occurs when the energy level is at 5 MJ for diagonal brace and 16 MJ for horizontal brace. Both cases can be considered as local effects since no other elements except for the impact element experience any plastic hinges or plastification.

A new impact scenario has been introduced in this jacket and that is in the X braced cross displayed in Figure 22. The impact energy reaches to 13 MJ until the joint fails.

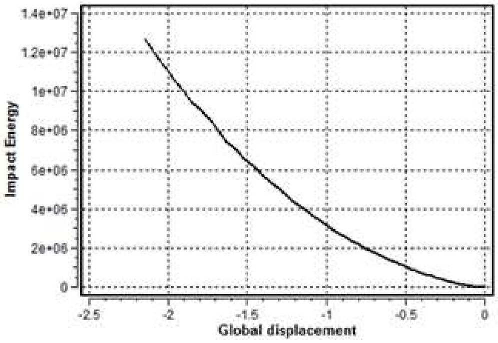
The multiple impact scenarios comprises of two impacts on diagonal braces. First sequence reaches an energy level at 5 MJ where the impacted member fracture and the remaining energy is transfer to the neighbouring brace where it reaches to 9 MJ until fracture occur. This gives a total of 15 MJ.



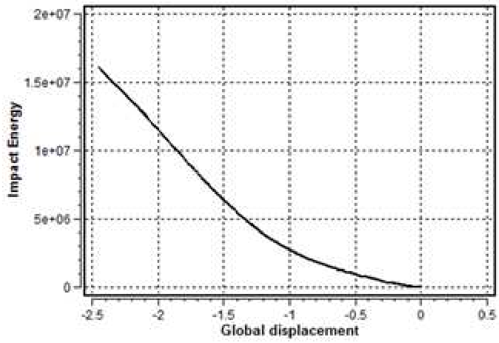
Jacket G – Load Case 41



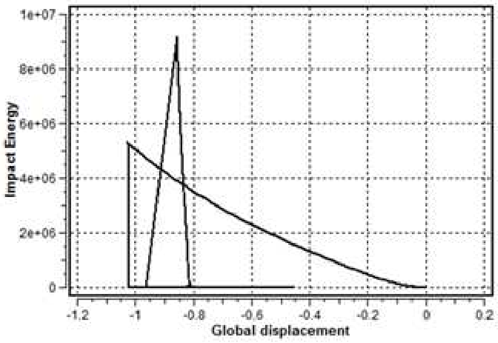
Jacket G – Load Case 42



Jacket G – Load Case 43

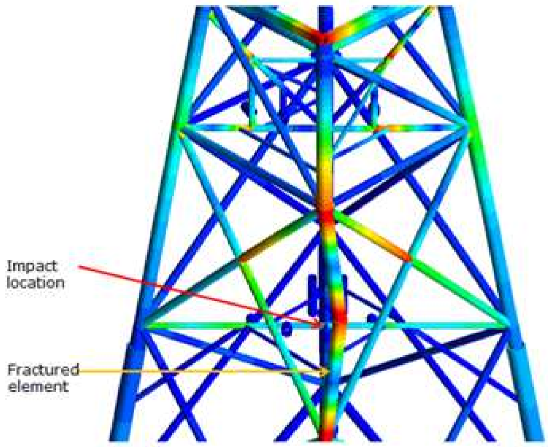


Jacket G – Load Case 44

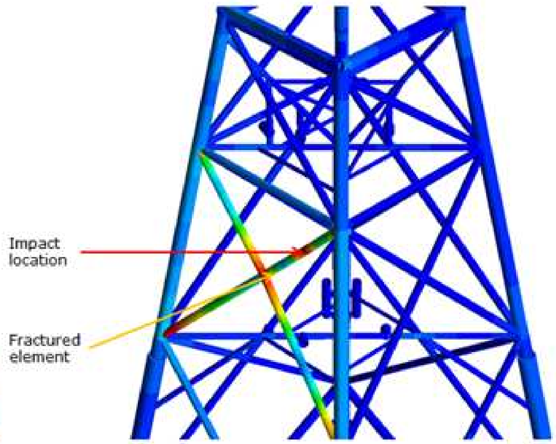


Jacket G – Load Case 41+45

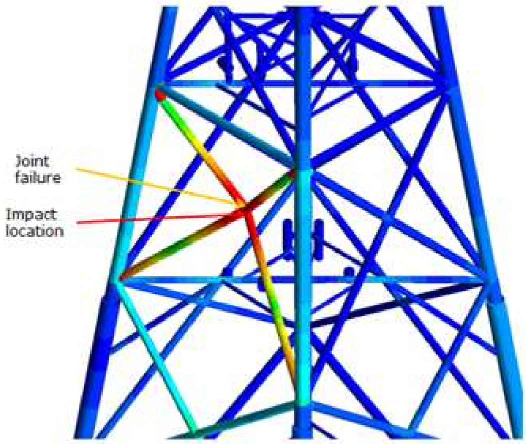
Figure 21 Global displacement vs. impact energy for jacket G



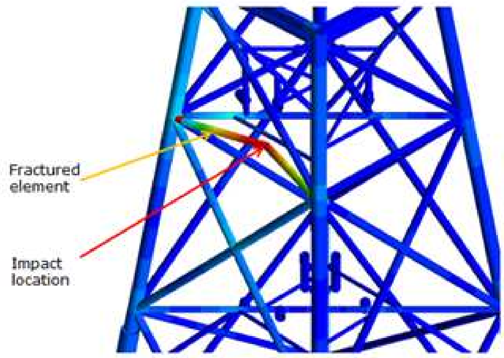
Jacket G – Load Case 41



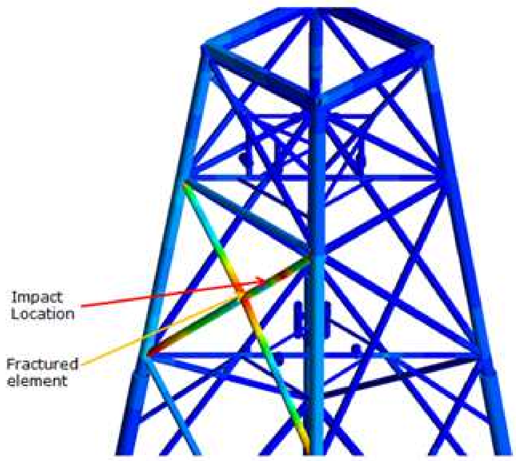
Jacket G – Load Case 42



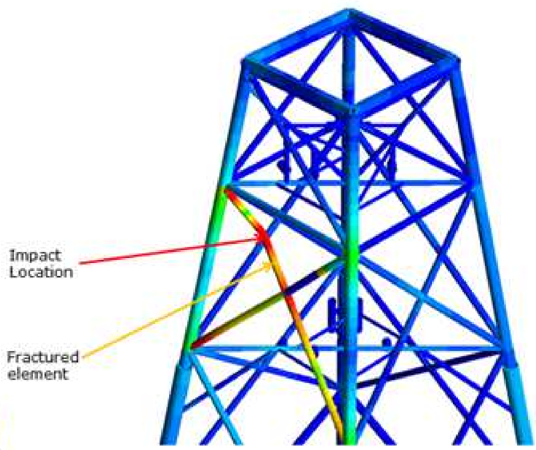
Jacket G – Load Case 43



Jacket G – Load Case 44



Jacket G – Load Case 42+45



Jacket G – Load Case 42+45

Figure 22 Plastic utilization for jacket G

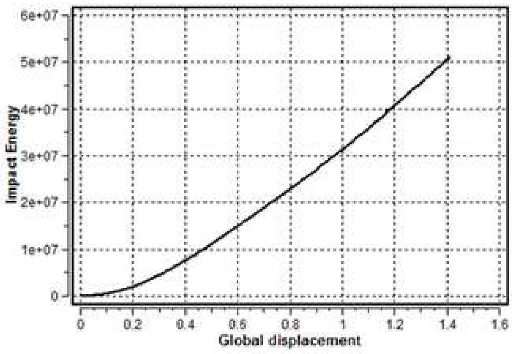
6.4.4 Jacket H

The energy deformation relationship for the five impact scenarios is given in Figure 23. The impact location on the leg (mid-span) reaches an energy level 51 MJ before fracture occur, which is comparable to the jacket E. As the same for the other leg impact, some plastic hinge and plastification can be spotted around the impact location.

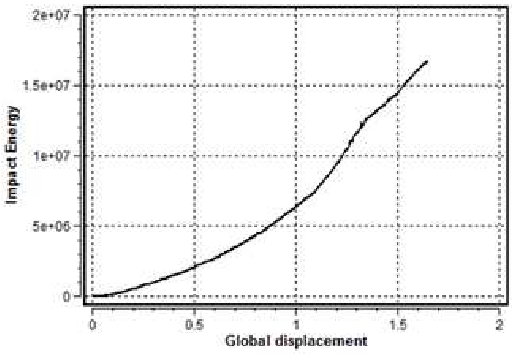
The horizontal and diagonal brace gives a comparable energy level as jacket F. Fracture occurs when the energy level is at 4 MJ for diagonal brace and 8 MJ for horizontal brace. Both cases can be considered as local effects since no other elements except for the impact element experience any plastic hinges or plastification.

The X braced cross scenario displayed in Figure 24, reaches impact energy 13 MJ until the joint fails.

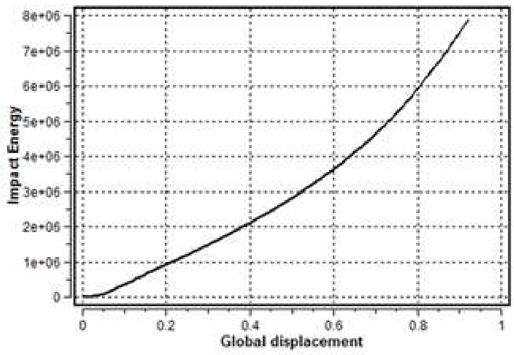
The multiple impact scenarios comprises of two impacts on diagonal braces. First sequence reaches an energy level at 4 MJ where the impacted member fracture and the remaining energy is transfer to the neighbouring brace where it reaches to 3 MJ until fracture occur. This gives a total of 7 MJ.



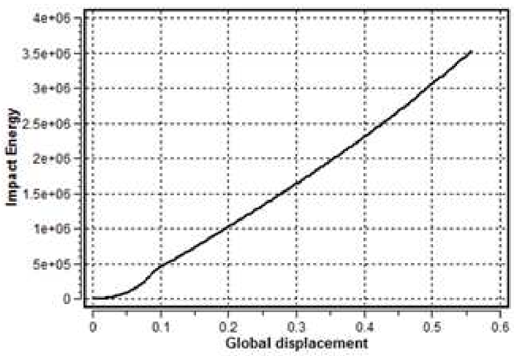
Jacket H – Load Case 41



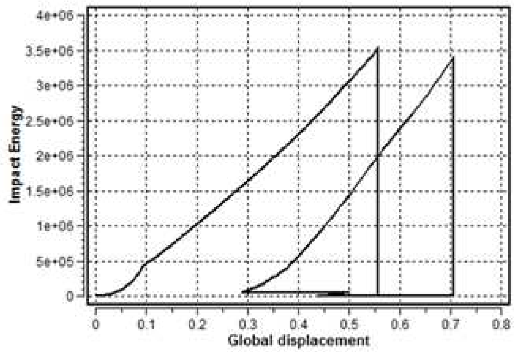
Jacket H – Load Case 421



Jacket H – Load Case 43

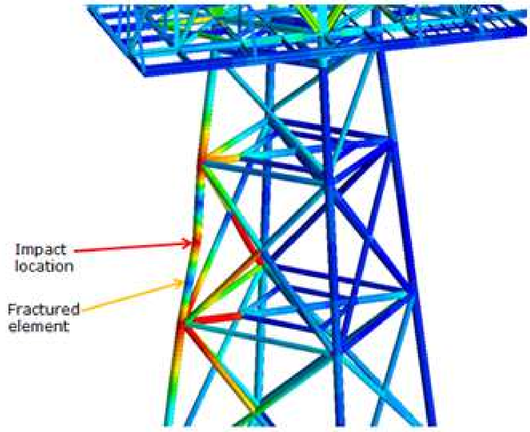


Jacket H – Load Case 44

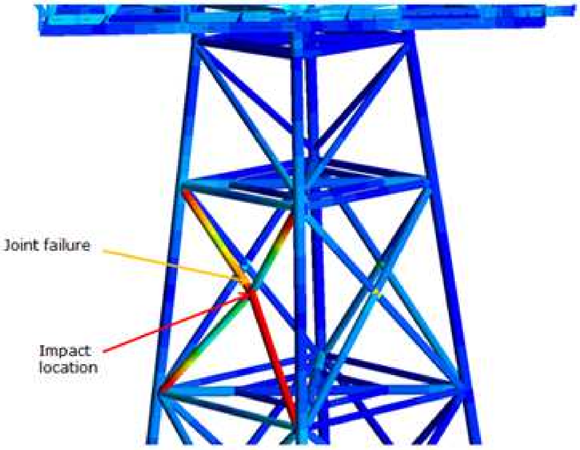


Jacket H – Load Case 44+45

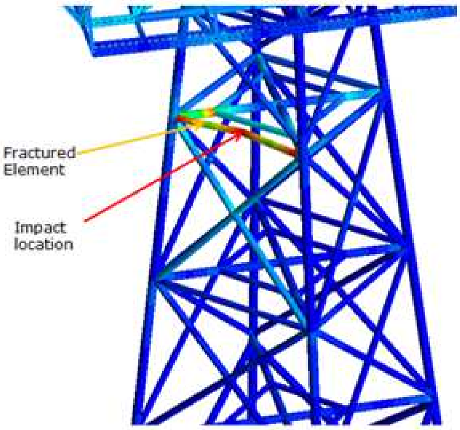
Figure 23 Global displacement vs. impact energy for jacket H



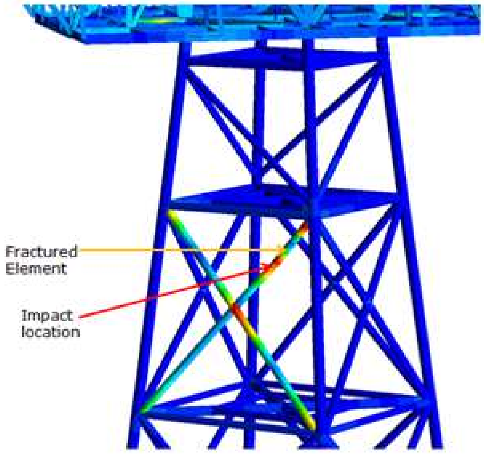
Jacket H – Load Case 41



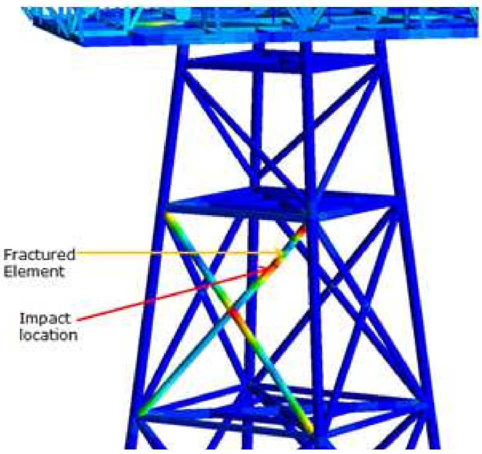
Jacket H – Load Case 421



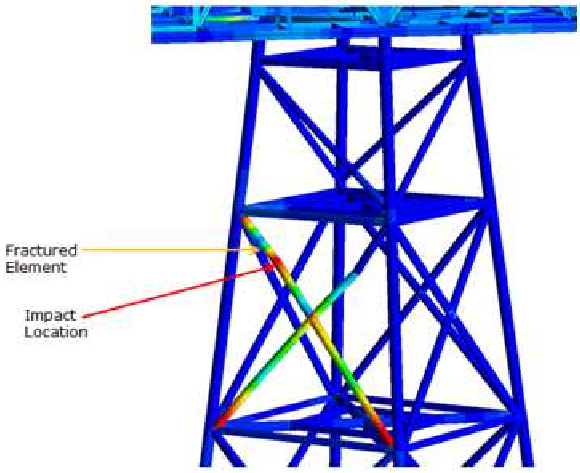
Jacket H – Load Case 43



Jacket H – Load Case 44



Jacket H – Load Case 44+45



Jacket H – Load Case 44+45

Figure 24 Plastic utilization for jacket ..

6.5.5 Summary

To get a better understanding of the jackets capacity when they are subjected to high energy collision, Table 5, Tables 6, Tables 7 and Tables 8 present a summary of the total energy in each scenarios including information about member dimension, yield strength of the material and failure criteria.

All four jacket legs show that it can take much more energy than braces. The total energy for legs before failure ranges from 21–54 MJ. That is much higher than the stated 14 MJ criteria in NORSOK N-004 [NTS, 2004]. But against vessel of 2000-3000 tons displacement traveling with a speed of 5.5–6 m/s, only jacket E and H could withstand the impact energy since the kinetic energy is between 40–50 MJ [Amdahl, 2001]. Comparing with the result from Refs. [Amdahl, 2001], the jacket leg dissipates no more than 10MJ which is lower than jacket G leg. But as mention in Chapter 2, these analyses may have different criteria. For instant when it comes to design principles, Refs. [Amdahl, 2001] uses strength design. This means that the jacket leg is strong enough to resist the collision force with minor deformation. Comparing to this thesis, the jacket dissipate the major part of the energy. Looking at Refs. [Amdahl, 1993] which also include a single leg impact, the jacket dissipate no more than 10 MJ. Again, the conditions for comparing are not the same since the ship and denting also dissipate energy in Refs. [Amdahl, 1993].

In the diagonal braces the energy dissipation varies between 4–6 MJ. This means that none of the diagonal braces can dissipate the ordinary design collision energy of 14 MJ. When it comes to the horizontal braces, the energy ranges from 4-26 MJ so only jacket F and G can withstand ordinary collision energy. Based on the result from Refs. [Skallerud, 2002], the horizontal brace dissipation of energy is no more than 3-5 MJ and the diagonal brace dissipate 9 MJ before being subjected to fracture. These numbers are very comparable except for the horizontal brace from jacket F and G. The energy dissipation from jacket H diagonal brace might also be view as conservative. While a single impact on the X brace cross is between 13-17 MJ.

The total energy absorption from the multiple impact cases ranges from 7–15 MJ. Only jacket G is strong enough to withstand the design collision energy of 14 MJ. The other jacket couldn't withstand the design collision energy which will most likely go under and through the jacket and possibly hit risers and conductors.

Based on the survey which was conducted in Chapter 3.3, only the jacket legs would withstand the highest total energy absorption at 17 MJ considering the vessel travels with a speed of 2m/s.

According to Refs. [Skallerud, 2002], increase in dimension of the impacted element should increase the total energy absorption. When comparing a single impact on horizontal brace between jacket E and F, there is a large deviation between the energy. Jacket E absorb 4 MJ while jacket F absorb 26 MJ. There is a clear sign of an error here, since the dimension between the two elements is close to the same including having the same yield strength. In other word, there might be some factors or errors in the analysis that needs further investigation. The scenarios which are highlighted in red in Tables 5, Tables 6, Tables 7 and Tables 8 needs more investigation and should be considered as conservative.

Table 5 Summary of physical sizes and results for jacket model E

Scenario	Length (m)	OD (mm)	wt (mm)	Fy (MPa)	Energy (MJ)	Failure Criteria
Single impact on jacket leg	12.6	1300	80	355	54 MJ	Fracture
Single impact on diagonal brace	12	850	40	355	5 MJ	Fracture
Single impact on joint	3	1250	65	355	36 MJ	Fracture
Single impact on horizontal brace	9	850	40	355	4 MJ	Fracture
Multiple impact on two diagonal brace	N/A	N/A	N/A	N/A	11 MJ	Fracture

Table 6 Summary of physical sizes and results for jacket model F

Scenario	Length (m)	OD (mm)	wt (mm)	Fy (MPa)	Energy (MJ)	Failure Criteria
Single impact on jacket leg	23	1182	85	355	33 MJ	Fracture
Single impact on diagonal brace	19.7	900	35	355	6 MJ	Fracture
Single impact on horizontal brace	10	800	30	355	26 MJ	Fracture
Multiple impact on two diagonal brace	N/A	N/A	N/A	N/A	12 MJ	Fracture

Table 7 Summary of physical sizes and results for jacket model G

Scenario	Length (m)	OD (mm)	wt (mm)	Fy (MPa)	Energy (MJ)	Failure Criteria
Single impact on jacket leg	21	1300	50	355	21 MJ	Fracture
Single impact on diagonal brace	9	800	40	355	5 MJ	Fracture
Single impact on x brace cross	21	800	40	355	13 MJ	Joint failure
Single impact on horizontal brace	18	700	40	355	16 MJ	Fracture
Multiple impact on two diagonal brace	N/A	N/A	N/A	N/A	15 MJ	Fracture

Table 8 Summary of physical sizes and results for jacket model H

Scenario	Length (m)	OD (mm)	wt (mm)	Fy (MPa)	Energy (MJ)	Failure Criteria
Single impact on jacket leg	22	1205	211	355	51 MJ	Fracture
Single impact on diagonal brace	10	910	30	355	4 MJ	Fracture
Single impact on x brace cross	22	900	30	355	17 MJ	Joint failure
Single impact on horizontal brace	13	900	30	355	9 MJ	Fracture
Multiple impact on two diagonal brace	N/A	N/A	N/A	N/A	7 MJ	Fracture

7 CONCLUSIONS AND RECOMMENDATION

7.1 Conclusions

The aim for this thesis was to study the effect on jacket structures on the NCS from ship collision where the impact energy is higher than anticipated design.

All four jacket legs are capable of withstanding design collision energy of 14 MJ. Disregarding the scenarios which gave unlikely energy level, no braces are capable of withstanding collision energy either sideways or bow/stern collision. The multiple impact scenarios in the other hand, jacket H is the only jacket that could take a sideways collision while jacket E and F can withstand a bow or stern collision of 11 MJ. Some scenarios as mention in the Chapter 6.5.5 needs further investigation.

7.2 Future work

Due to the time limitation, this thesis covers merely a “coarse” analysis. For further work, one could optimize the project in the following way:

- Verification of models against project documents (drawings, design basis, etc.).
- More scenarios on multiple impacts.
- Assessment of 100-year storm condition for damaged jacket.
- Risk evaluation including investigation of operational limits for vessel operations around the jacket.
- Expand the analysis to the other types of jackets.

A study of shared energy design could be carried out by modelling the ship as a finite shell element. In this case both jacket and ship would contribute considerably to the energy dissipation.

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Marine Traffic

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Petroleumstilsynet

http://www.ptil.no/konstruksjonssikkerhet/risiko-for-kollisjoner-med-besoekende-fartoeyer-article7484-826.html

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Tromsoffshore

http://www.tromsoffshore.no/en/fleet/platfrom-supply-vessels-psv

(Link verified June 15th 2014)

USFOS

http://usfos.no/

(Link verified June 15th 2014)

APPENDIX A

Pre-study report for master thesis: “Effects of impacts from large supply vessels on jacket structures”

Author:

Tuan Minh Tran

Abstract

This pre-study report starts with an introduction of the background behind the project. The introduction will also cover the aim of the project, scope of work and limitations.

The rest of the report will cover the different stages during the project life cycle. This is also illustrated with a Work breakdown structure (WBS). A Gantt-chart will present the time schedule.

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1. Introduction

1.1 Background

Most jacket structures are designed to resist impacts from supply vessels with a displacement of 5000 tons In the Norwegian Continental Shelf (NCS). This is mention in the current version of the NORSOK standard for design of steel structures N-004, in Appendix A.3.

Because of higher demands for equipment, the supply vessels displacement has increased over the last 5-10 years but the standards has not taken this in to consideration. This is due to the introduction of the Dynamic Positioning (DP) systems which has reduced the risk of collision between ships and offshore structures. Like many new technologies, there is still a small possibility that the DP systems "fail" which may result in a catastrophic failure.

In the past 10 years there have been a total of 26 collisions between incoming vessels and installations in the NCS. These incidents are a result of poor organization of work and responsibilities, lack of training of personnel and the failure of technical equipment. In other words, the cause of these incidents is not because of a single factor but a number of factors. The people responsible are not just the crew of the vessels but also on the operators and the owners.

No catastrophic failure has yet accord but many severe accidents have happened. One of them was the "Big Orange XVIII" collision with the platform Ekofisk 2/4-W in the summer of 2009. The accident caused a lot of material damaged on the vessel and the offshore structures but no personnel were injured. In the investigation report performed by Petroleumtilsynet (Ptil) categorized the accident as a "major accident" which means a possibility with many serious personal injuries or casualties, or sets the structural integrity in danger. Another incident was in Mars 2004 when the supply vessel "Far symphony" collided into a drilling rig in the "Trollfelt". No personnel were injured and the material damage was less serious than the "Big Orange XVIII" incident.

1.2 The aim of the Master thesis

Because of the increasing vessels displacement in the NCS, the current jacket structures might not resist a potential impact because of the higher energy.

The aim for this Master thesis is to study the effects on jacket structures on the NCS from ship collisions where the impact energy is higher than anticipated in the design.

1.3 Scope of work

- Perform a literature study to capture current knowledge regarding ship collisions with jacket structures, mainly DNV-RP-C204 and similar documents.
- Perform a Survey of typical supply vessels operating on the NCS.
- Collect structural models of jacket structures, categorize them and establish prioritized sequence for coming Finite Element (FE) simulations.
- Perform simplified hand calculations and linear simulations in order to understand the theory plus being able to compare linear with non-linear simulations.
- Prepare non-linear FE models and apply representative loadings. Both local and global effects shall be evaluated.
- Review the results and assess the consequences of increased impact energies on various types of jacket structures.

1.4 Limitations

- Only 6 jacket models are evaluated.
- Jacket structure design
- Topside design
- Foundation design
- Snow and ice loads
- Extreme environmental accidents (earthquake…etc.)

2. Work breakdown structure (WBS) and Gantt chart

A WBS has been created to provide the project scope of work while a Gantt chart has been created to show the project schedule. These can be found in the next section. This section will present short the different tasks that will be performed throughout the project.

2.1 Planning meeting

The planning meeting started in December 2013. The aim for the meeting was to review the MSc thesis proposal, discuss the scope of work and clarify the roles and responsibilities of the participants.

2.2 Literature study

The literature study covers some history of incidents between jacket structures and supply vessels in the past, theory and practices that shall be utilize and presentation of the software that shall be used during the analysis.

2.3 Survey of supply vessels

Collect and present data from a typical supply vessel in the NCS. The data will used to simulate the boat impact.

2.4 Calculation

Hand calculation will mainly be based on linear and non-linear theory. Simplified examples shall be used to demonstrate.

2.5 Categorization of jacket models

Jacket structures models shall be provided. Some converting will be necessary. Prioritize sequence will be based upon the jacket structures age.

2.6 Pre-study report

This report will cover mainly the planning, activities and the purpose of the thesis.

2.7 Analysis

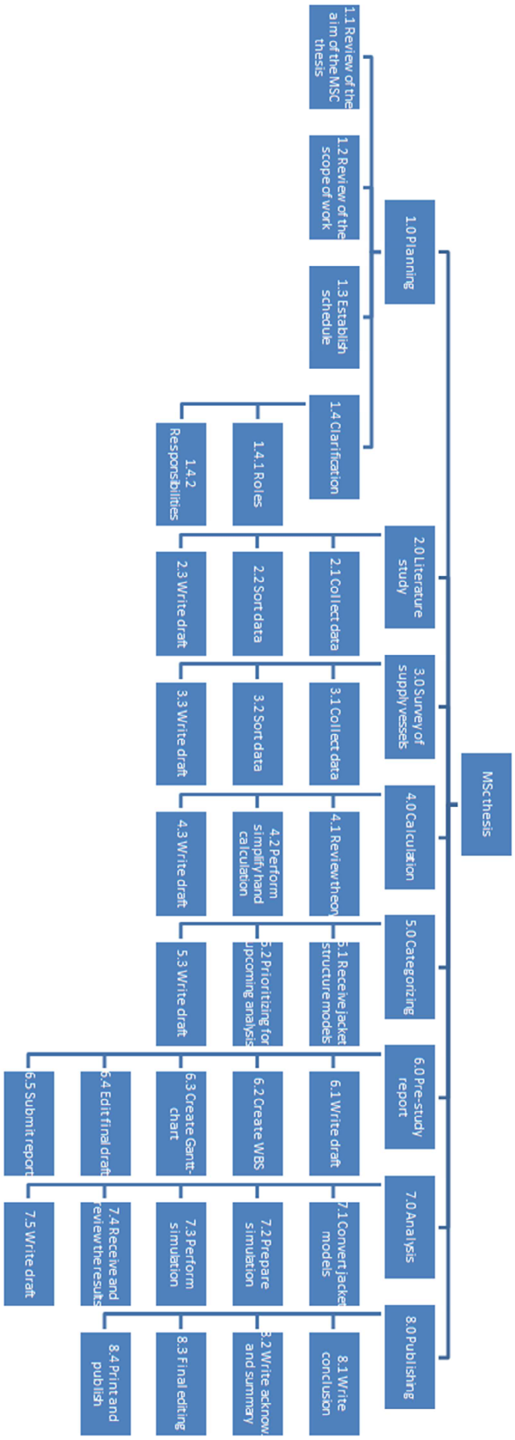
Jacket structure models shall be exposed to increased impact energies by applying representative loadings. The software that shall be used is called USFOS which is specialized in nonlinear static and dynamic analysis of frame structures.

2.8 Publishing

This last phase will be used for final editing of the thesis and publishing.

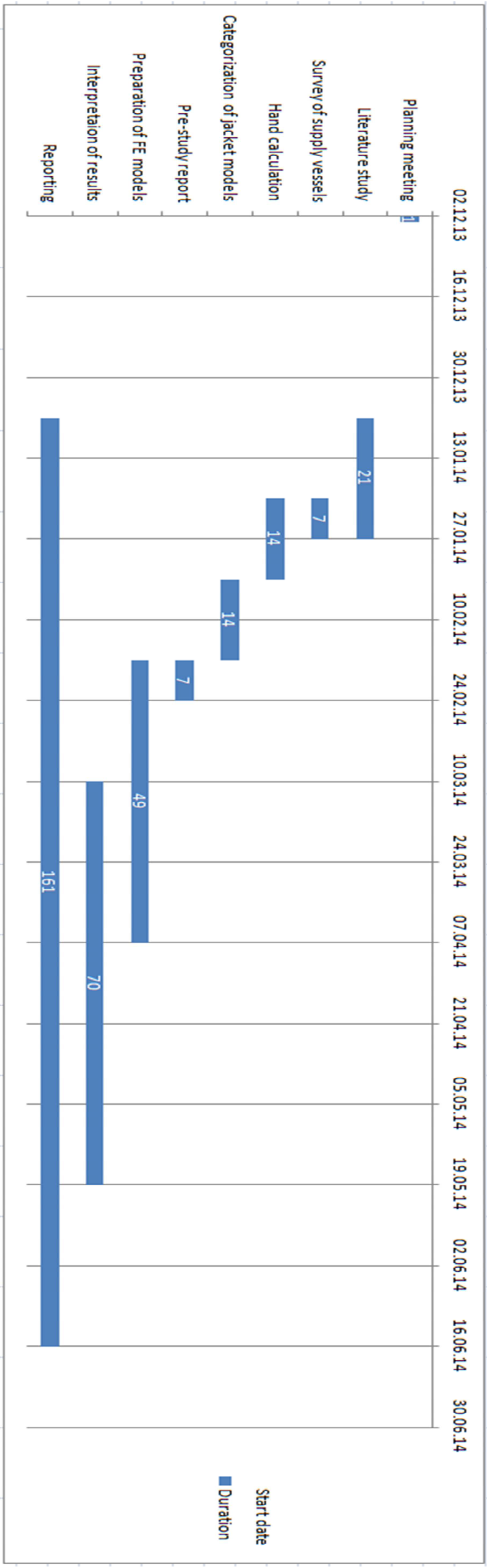
Writing of the report will be included in every phase. Change in the time schedule may occure.

3. Work Breakdown Structures (WBS)



4. Time Schedule

Item	Activities	Start date	Duration	End date
1	Planning meeting	02.12.2013	1	02.12.2013
2	Literature study	06.01.2014	21	26.01.2014
3	Survey of supply vessels	20.01.2014	7	26.01.2014
4	Hand calculation	20.01.2014	14	02.02.2014
5	Categorization of jacket models	03.02.2014	14	16.02.2014
6	Pre-study report	17.02.2014	7	23.02.2014
7	Preparation of FE models	17.02.2014	49	06.04.2014
8	Interpretation of results	10.03.2014	70	18.05.2014
9	Reporting	06.01.2014	161	15.06.2014



Planning meeting
Literature study
Survey of supply vessels
Hand calculation
Categorization of jacket models
Pre-study report
Preparation of FE models
Interpretaion of results
Reporting
02.12.13
16.12.13
30.12.13
13.01.14
27.01.14
10.02.14
24.02.14
10.03.14
24.03.14
07.04.14
21.04.14
05.05.14
19.05.14
02.06.14
16.06.14
30.06.14
1
21
7
14
14
7
49
70
161
Start date
Duration

APPENDIX B

	Vessel Name	L (m)	B (m)	Draft	Year	Deadweight (tons)
1	Blue Power	82	18	5.4	2013	4240
2	Blue Protector	82	18	5.4	2013	4200
3	Bourbon Mistral	89	20	5.6	2006	4779
4	Bourbon Monsoon	88	20	5.5	2007	4779
5	Bourbon Rainbow	88	19	5.6	2013	4400
6	Bourbon Sapphire	91	19	5.4	2008	4678
7	Caledonian Vanguard	93	22	6.2	2005	4312
8	Caledonian Victory	93	22	6.4	2006	4380
9	Caledonian Vigilance	81	18	6.3	2006	5300
10	Caledonian Vision	93	22	6.3	2006	4312
11	E.R Kristiansand	73	16	5.1	2005	3544
12	E.R. Georgina	93	20	6.2	2010	4831
13	Edda Frigg	84	19	4.5	1997	3974
14	Energy Swan	93	19	5.5	2005	5304
15	F.D. Incomparable	75	16	5.3	2012	3161
16	F.D. Indomitable	75	16	4.8	2011	3105
17	Far Serenade	94	21	6	2009	4000
18	Far Solitaire	92	22	5.6	2012	5800
19	Far Spica	81	18	5.3	2013	4000
20	Far Symphony	86	19	6	2003	4929
21	Grampian Sceptre	83	18	4.6	2013	2515
22	Grampian Talisker	82	17	5.2	2009	3890
23	Grampian Talisman	73	17	5	2007	3614
24	Grimshader	80.9	17.5	3.5	1983	3324
25	Havila Aurora	74.87	16.4	6.22	2009	3205
26	Havila Borg	78.6	17.6	7.7	2009	3787
27	Havila Charisma	95	20	5.5	2012	4976
28	Havila Clipper	80.4	17.6	6.5	2011	3683
29	Havila Commander	85	20	6.8	2010	5486
30	Havila Crusader	85	20	6.8	2010	5433
31	Havila Faith	82.85	19	6.31	1998	4679
32	Havila Fanø	80.4	17.6	6.48	2010	3879
33	Havila Favour	82.85	19	6.31	1999	4679
34	Havila Foresight	93.6	19.7	6.3	2007	4785
35	Havila Fortress	82.85	19	6.32	1996	4679
36	Havila Fortune	74.87	16.4	6.22	2009	3205
37	Havila Herøy	80.4	17.6	6.5	2009	3683
38	Havila Princess	73.4	16.6	6.4	2005	3719
39	Highland Duke	75	16	4.9	2012	3105
40	Highland Laird	72	16	4.3	2006	3105
41	Highland Prestige	86	18	5.4	2007	4993

42	Highland Prince	87	19	6	2009	4826
43	Highland Star	81.9	18	3.8	1991	3075
44	Island Challenger	93	20	6	2007	4100
45	Island Champion	93	20	5.8	2007	4100
46	Island Chieftain	94	20	5.6	2009	4100
47	Island Contender	96	20	6.5	2012	4750
48	Island Duchess	85	17	4.8	2013	3750
49	Island Empress	77	16	5	2007	3180
50	Malayiva Seven	82.5	18.8	5.2	1994	4568
51	Malayiva Twenty	72	16	4.5	2004	3316
52	Normand Aurora	86	19	5.5	2005	4813
53	Normand Flipper	80	20	4.4	2003	4276
54	North Mariner	84	18	5.4	2002	4545
55	North Purpose	86	19	5.5	2010	4826
56	North Stream	84	19	5	1998	4320
57	Northern Supporter	67	16	4.6	1996	3100
58	Ocean Scout	77	16	4.8	2013	3200
59	Ocean Viking	70	16	5	1986	2629
60	Olimpic Energy	94	20	5.2	2012	5066
61	Olympic Commander	94	20	6	2012	4857
62	Olympic Electra	80	17	5.2	2011	3000
63	Olympic Princess	84	20	5.6	1999	4159
64	Rem Commander	85	20	6.1	2011	4500
65	Rem Fortress	85	20	5.7	2011	4500
66	Rem Fortune	86	20	5.8	2013	4000
67	Rem Leader	90	24	6.2	2013	4800
68	Rem Mermaid	80	16	5.3	2008	3336
69	Rem mist	89	19	6	2011	4400
70	Rem Ocean	107	22	6.5	2014	5520
71	Rem Server	94	20	5	2011	5300
72	Rem Supporter	94	20	6.2	2012	5300
73	Saeborg	86	18	6	2011	4300
74	Sayan Princess	78	16	5.8	2013	3800
75	SBS Tempest	74	14	5.4	2006	3677
76	Sea Tantalus	82	17	5.6	2013	4000
77	Sea Trout	73	16	5.8	2008	3678
78	Siddis Supplier	73	17	5	2010	3350
79	Skandi Caledonia	84	20	5.3	2003	4100
80	Skandi Feistein	88	19	5.8	2011	4700
81	Skandi Flora	95	20	5	2009	5005
82	Skandi Foula	83	20	5.1	2002	4200
83	Skandi gamma	95	20	6	2011	5054
84	Skandi Kvitsoy	88	19	6	2012	4700
85	Skandi Maroy	82	17	5.2	2012	3594

86		Skandi Marstein	83.7	19.7	5.4	1996	4170
87		Skandi Mongstad	97	22	6	2008	4423
88		Skandi Nova	82	17	5.9	2012	3100
89		Skandi Seven	121	22	7	2008	6000
90		Skandi Sotra	83	20	5	2003	3933
91		Skandi Texel	69	16	4.8	2006	3500
92		Stril Explorer	76.4	16.2	4.6	2010	1400
93		Stril Mermaid	79	18	5.8	2010	3755
94		Stril Myster	90	19	6	2003	4500
95		Stril Orion	93	19	6	2011	4900
96		Stril Polar	93	19	5.5	2012	4900
97		Strill Mariner	79	18	5	2009	3755
98		Strilmoy	86	20	4.2	2005	4248
99		Troms Arcturus	95	21	6.5	2014	5580
100		Troms Artemis	85	20	6.1	2011	4900
101		Troms Castor	85	20	5.6	2009	4900
102		Troms Lyra	82	18	5.5	2013	3650
103		Vestland Mira	86	18	5.5	2012	4000
104		Viking Athene	74	17	4.7	2006	3546
105		Viking Dynamic	90	19	5.4	2002	4505
106		Viking Energy	95	20	6.5	2003	6013
107		Viking Fighter	82	19	5.5	2012	4000
108		Viking Lady	92	21	6.5	2009	6200
109		Viking Prince	90	21	6.3	2012	6150
110		Viking Queen	92	21	6.5	2008	6200
111		Volstad Princess	93	18	5.6	2008	4867
112		Vos Iona	61	14.3	4.3	1977	1921
113		World Diamond	80	16	5.1	2013	3520
114		World Opal	80	16	5	2013	3300
115		World Sapphire	80	16	4.6	2013	3300

APPENDIX C

Impact Energy

Bow/ Stern Impact (MJ)	Broad side (MJ)
9.3	11.9
9.2	11.8
10.5	13.4
10.5	13.4
9.7	12.3
10.3	13.1
9.5	12.1
9.6	12.3
11.7	14.8
9.5	12.1
7.8	9.9
10.6	13.5
8.7	11.1
11.7	14.9
7.0	8.9
6.8	8.7
8.8	11.2
12.8	16.2
8.8	11.2
10.8	13.8
5.5	7.0
8.6	10.9
8.0	10.1
7.3	9.3
7.1	9.0
8.3	10.6
10.9	13.9
8.1	10.3
12.1	15.4
12.0	15.2
10.3	13.1
8.5	10.9
10.3	13.1
10.5	13.4
10.3	13.1
7.1	9.0
8.1	10.3
8.2	10.4
6.8	8.7
6.8	8.7
11.0	14.0

10.6	13.5
6.8	8.6
9.0	11.5
9.0	11.5
9.0	11.5
10.5	13.3
8.3	10.5
7.0	8.9
10.0	12.8
7.3	9.3
10.6	13.5
9.4	12.0
10.0	12.7
10.6	13.5
9.5	12.1
6.8	8.7
7.0	9.0
5.8	7.4
11.1	14.2
10.7	13.6
6.6	8.4
9.1	11.6
9.9	12.6
9.9	12.6
8.8	11.2
10.6	13.4
7.3	9.3
9.7	12.3
12.1	15.5
11.7	14.8
11.7	14.8
9.5	12.0
8.4	10.6
8.1	10.3
8.8	11.2
8.1	10.3
7.4	9.4
9.0	11.5
10.3	13.2
11.0	14.0
9.2	11.8
11.1	14.2
10.3	13.2
7.9	10.1

9.2	11.7
9.7	12.4
6.8	8.7
13.2	16.8
8.7	11.0
7.7	9.8
3.1	3.9
8.3	10.5
9.9	12.6
10.8	13.7
10.8	13.7
8.3	10.5
9.3	11.9
12.3	15.6
10.8	13.7
10.8	13.7
8.0	10.2
8.8	11.2
7.8	9.9
9.9	12.6
13.2	16.8
8.8	11.2
13.6	17.4
13.5	17.2
13.6	17.4
10.7	13.6
4.2	5.4
7.7	9.9
7.3	9.2
7.3	9.2

APPENDIX D

INPUT

Section Profile

$f_y := 355MPa$ Yield stress of material

$D := 1.3m$ Outer diamater of member

$t := 0.080m$ Wall thickness of member

$l_b := 12m$ Member length

$E := 210000MPa$ Modulus section of material

$k_1 := 736\frac{MN}{m}$ Stiffness of adjacent joint 1

$k_2 := 51\frac{MN}{m}$ Stiffness of adjacent joint 2

$c_1 := 2$ c1 = 2 for clamped beam
c1 = 1 for pinned beam

$\varepsilon_{cr} := 0.15$ Critical strain, proposed value by RP C204, table 3-4

$H := 0.0034$ Non-dimensional plastic stiffness, proposed value by RP C204, table 3-4

$\kappa l := \frac{l_b}{2}$ The smaller distance from impact point to adjacent joint.
For central impact taken as half of the length

$d_c := D$ Characteristic dimension of tubular beams, taken as diameter of tubular

Cross Section Properties

$$A_s := 0.25 \cdot \pi \cdot \left[D^2 - (D - 2t)^2\right] = 0.307\,m^2$$

Cross section area

$$W := \left(\frac{\pi}{32 \cdot D}\right)\left[D^4 - (D - 2 \cdot t)^4\right] = 0.088 \cdot m^3$$

Elastic modulus section

$$z := \frac{\left[D^3 - (D - 2t)^3\right]}{6} = 0.119 \cdot m^3$$

Plastic modulus section

$$W_p := z$$

Plastic modulus section

$$y_{bar} := \frac{D}{2} = 0.65\,m \qquad \frac{D}{t} = 16.25$$

Moment arm of cross section

$$w_c := \frac{D}{2} = 0.65\,m$$

Characteristic deformation for tubular beam

$$\varepsilon_y := \frac{f_y}{E} = 0.002$$

Yield strain

$$I := \frac{\pi \cdot \left[D^4 - (D - 2t)^4\right]}{64} = 0.057\,m^4$$

Moment of inertia of tubular cross section

Cross Section Types

Note: The categorization of cross section type is based on DNV-OS-C101 Appendix A

$$\varepsilon_r := \sqrt{\frac{235 \cdot MPa}{f_y}} = 0.814$$

Relative strain

$$SectionType := \begin{cases} \text{"Type I"} & \text{if } \frac{D}{t} \le 50\varepsilon_r^2 \\ \text{"Type II"} & \text{if } 50.\varepsilon_r^2 < \frac{D}{t} \le 70\varepsilon_r^2 \\ \text{"Type III"} & \text{if } 70\varepsilon_r^2 < \frac{D}{t} \le 90\varepsilon_r^2 \\ \text{"Not Specified"} & \text{otherwise} \end{cases}$$

$$SectionType = \text{"Type I"}$$

Representative Stiffness

$$k_{node} := 2 \cdot \left(\frac{1}{k_1} + \frac{1}{k_2}\right)^{(-1)} = 95.39 \cdot \frac{MN}{m}$$

Effective Stiffness

$$k := \left(\frac{l_b}{2 \cdot E \cdot A_s} + \frac{1}{k_{node}}\right)^{(-1)} = 94.55 \cdot \frac{MN}{m}$$

Non Dimensional Spring Stiffness

$$c := \frac{4 \cdot c_1 \cdot k \cdot w_c^2}{f_y \cdot A_s \cdot l_b} = 0.245$$

Collapse Resistance

$$M_p := f_y \cdot z = 42.331\, m \cdot MN$$ Plastic moment of cross section

$$R_0 := 4 \cdot c_1 \cdot \frac{M_p}{l_b} = 28.221 \cdot MN$$

c factors

$$c_f := \left(\frac{\sqrt{c}}{1 + \sqrt{c}} \right)^2 = 0.11$$ Axial flexibility factor

$$c_{lp} := \frac{\left[\left(\frac{\varepsilon_{cr}}{\varepsilon_y} \right) - 1 \right] \cdot \left(\frac{W}{W_p} \right) \cdot H}{\left[\left[\left(\frac{\varepsilon_{cr}}{\varepsilon_y} \right) - 1 \right] \cdot \left(\frac{W}{W_p} \right) \cdot H \right] + 1} = 0.181$$ Plastic zone length factor

$$c_w := \left(\frac{1}{c_1} \right) \cdot \left[c_{lp} \cdot \left(1 - \frac{c_{lp}}{3} \right) + 4 \cdot \left(1 - \frac{W}{W_p} \right) \cdot \left(\frac{\varepsilon_y}{\varepsilon_{cr}} \right) \right] \cdot \left(\frac{\kappa l}{d_c} \right)^2 = 1.934$$ Displacement factor

Ductility Limit: Tensile Fracture in Yield Hinges

Rupture may be assumed to occur when the deformation exceeds the value given by following:

$$w_{crit\_frc} := \left(d_c \cdot \frac{c_1}{2c_f} \right) \cdot \left(\sqrt{1 + \frac{4c_w \cdot c_f \cdot \varepsilon_{cr}}{c_1}} - 1 \right) = 0.371\, m$$ Critical fracture in yield hinge

Ductility Limit: Local Buckling Check

$$\beta := \frac{\frac{D}{t}}{\frac{235}{f_y} \cdot MPa} = 24.548$$ Buckling must be considered since β > β 1 (N-004 Section A.3.10.2)

$$\beta_1 := \left[\left(\frac{14 \cdot c_f \cdot \frac{f_y}{MPa}}{c_1} \right) \cdot \left(\frac{\kappa l}{d_c} \right)^2 \right]^{\frac{1}{3}} = 17.964$$

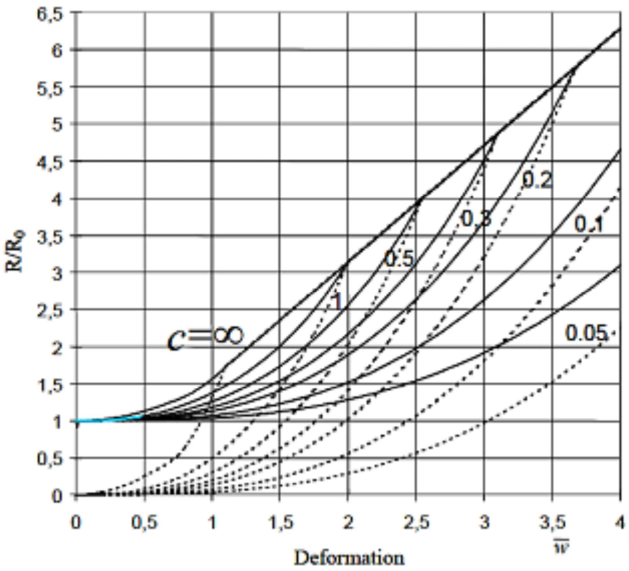
$$w_{crit\_buck} := \left(\frac{d_c}{2 \cdot c_f} \right) \cdot \left[1 - \sqrt{1 - \left(\frac{14 \cdot c_f \cdot \frac{f_y}{MPa}}{c_1 \cdot \beta^3} \right) \cdot \left(\frac{\kappa l}{d_c} \right)^2} \right] = 1.307\, m$$

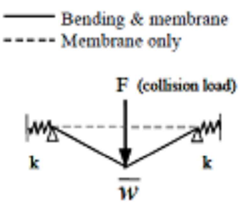
Impact Energy

For fracture-limited dissipation

$$w_{bar} := \frac{w_{crit\_buck}}{c_1 \cdot w_c} = 1.005$$

Non-dimensional deformation





From the graphic above:

where y equals R/R0

$y := 0.2$

$R := y \cdot R_0 = 5.644 \cdot MN$