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Finite element method as a tool for preliminary steel weight estimation

Master's thesis in Marine Technology

Supervisor: Prof. Bjørn Egil Asbjørnslett / Ali Ebrahimi

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Department of Marine Technology



Preface and acknowledgments

This master thesis is the result of the work and research completed in the 10th semester at the Norwegian University of Science and Technology - NTNU, Department of Marine Technology.

The thesis is a continuation of the project thesis, written fall 2018. The subject: Determining a parametric relation for the steel weight for cruise vessels. The problem addressed in the thesis was introduced by Ulstein through my co-supervisor Ali Ebrahimi.

I would like to thank Ali Ebrahimi for allowing me to investigate this topic and for the guidance he has provided. I would also like to thank my supervisor, Bjørn Egil Asbjørnslett for his guidance. Lastly, the contributors to ANSYS Student community deserves a mention for helping with the technical difficulties encountered in ANSYS.

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Date

Summary

The design process for a cruise vessel starts with a concept phase. At this stage a range of different decisions are made, included the main dimensions and an estimated total cost. Steel weight is one of the larger contributors to that estimated cost. Hence utilizing the global dimensions to obtain as much volume as possible, while minimizing the steel usage is key. Being able to predict the steel weight based on a limited set of information is also important to keep the project within budget.

This thesis discusses the use of finite element method (FEM) as a tool, to minimize steel weight, and develop a parametric steel weight relationship for the main dimensions of a cruise vessels.

The approach towards the solution can be seen as a stepwise process. The first aspect of the problem was to look into ways of minimizing the steel weight. This was done first in order for the results to be utilized in the second stage of the project. The second stage was the development of the FEM models needed to create the parametric equations for the global parameters and their relation in regards to the steel weight.

Trying to minimize the steel weight was tried in three different ways, all attacking the problem in a new way. A particle swarm optimization in combination with FEM, a topology optimization of the hull and a parameter correlation test for different stiffener profiles were tested. The result varied but yielded some results that proved useful for minimizing steel weight. In short; a higher number of stiffeners, minimize plate thickness, reduced flange width, and L-profile stiffeners were found to have a positive effect on the steel weight.

The weight minimizing knowledge was applied to the midships hull models, created using CAD and tested with FEM to produce the parametric relation for the main dimensions. The results found that increasing the volume of the vessel by increasing the length was 2,54 times worse in terms of added steel weight, compared to increasing the volume by increasing the breadth and 3,94 times worse compared to increasing the depth.

The method and findings in this thesis did not provide a full steel weight estimate for an arbitrary cruise vessel, a more comprehensive and detailed model was needed for that. The experiments did produce weight reducing measures and a relationship between the main parameters in terms of steel weight. The results were applicable to an arbitrary hull and can hopefully serve as decision support for ship designer developing new concepts.

Sammendrag

Design prosessen for et cruiseskip starter med en konseptfase. På dette stadiet bestemmes en mengde forskjellige avgjørelser, inkludert hoveddimensjoner og et estimert budsjett. Stålvect er en av de største utgiftspostene til den totale kostnaden. Av den grunn er det viktig å utnytte hoveddimensjonene slik at minst mulig stålvect går med til å produsere det ønskede volumet. Å kunne estimere stålvecten basert på begrenset informasjon er også viktig for å forsikre seg om at budsjettet blir holdt.

Denne avhandlingen diskuterer bruken av elementmetoden som et verktøy for å minimere stålvect og utvikle et parametrisk stålvectforhold for hoveddimensjonene til et cruiseskip.

Fremgangsmåten til løsningen kan ses som en stegvis prosess. Det første aspektet av problemet var å se på måter å minimere stålvecten. Dette ble gjort først for at resultatene skulle kunne brukes i andre fase av prosjektet. Den andre fasen var utviklingen av FEM-modellene, som brukes til å danne parametriske ligninger for de globale parameterne og vurdere deres innvirkning på stålvecten.

Tre forskjellige metoder for å minimere stålvecten ble forsøkt, hver metode angriper problemet på en egen måte. En particle swarm optimalisering (PSO) i kombinasjon med FEM, en topologi optimalisering av skroget og en parameter korrelasjonstest for forskjellige stiverprofiler ble testet. Resultatene varierte, men ga noen resultater som viste seg nyttig for å minimere stålvect. Kort oppsummert; Flere stivere, minimere platetykkelsen, redusere flensbredde og L-profil for stivere ble funnet å ha en positiv innvirkning på stålvecten.

Kunnskapen fra vekt minimerings forsøkene, ble brukt på midtskips-skrogmodellene som var produsert ved hjelp av CAD og testet med FEM for å danne parametrisk relasjoner for hoveddimensjonene. Resultatene viste at å øke volumet av fartøyet ved å øke lengden var 2,54 ganger verre hva gjelder stålvect, enn tilsvarende volumøkning gjort med økning av bredden. Den samme volum økningen gjort med økning av dypgang hadde vært 3,94 ganger bedre enn ved økning av lengden.

Metoden og funnene i denne oppgaven ga ikke et fullstendig stålvectestimat for et vilkårlig cruiseskip. En mer omfattende og detaljert modell var nødvendig for det. Forsøkene produserte vektreduserende tiltak og et forhold mellom hovedparameterne i form av stålvect. Resultatene var gjeldende for et vilkårlig skrog og kan forhåpentligvis fungere som beslutningsstøtte for skipsdesignere som utvikler nye konsepter.

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Nomenclature

Acronyms

FEM	Finite element method
DOE	Design of experiments
CAD	Computer aided design
CASD	Computer aided ship design
CAE	Computer aided engineering
FEA	Finite element analysis
CFD	Computer fluid dynamics
PSE	Parametric section-based estimation
BFGS	Broyden Fletcher Goldfarb Shanno algorithm
PSO	Particle swarm optimization
COG	Center of gravity

List of symbols

W_{st}	Steel weight
C_b	Block coefficient
C_m	Midship section coefficient
L_{pp}	Length between perpendiculars
L	Length
B	Breadth
D	Depth
T	Draft
SP	Swarm particles
G	Number of generations
C	Computational time in ANSYS
g	Gravity

E	Youngs modulus
KM	Distance from keel to metacentre
KG	Distance from Keel to center of gravity
M_S	Midships moment
t	Thickness
p	Pressure
s	stiffener spacing
σ	Stress

Chapter 1

Introduction

1.1 Background

The design process for a cruise ship starts with a concept phase. At this stage decisions regarding the field of operation (arctic, tropical, etc.), approximate main dimensions, cost and earnings are established. In order to estimate the cost for a cruise ship to be built, a steel weight estimate must be completed, as it is a large part of the total cost. The goal of this thesis is to use finite element method analysis to establish and develop a parameter correlation test in regards to steel weight for cruise vessels. In other words, it can be seen as a design of experiments (DOE).

Earlier the design process used to be an iterative process where the design firm and shipyard sequentially developed the final product. This is a time-consuming process but ensures that changes and decisions happen sequentially, minimizing the risk encountering unforeseen difficulties.

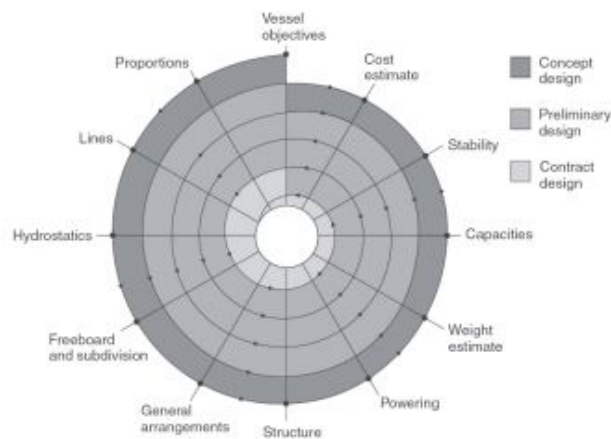


Figure 1.1: One example of a design spiral found from (Rigo and Caprace, 2011)

1.2 New technology

Now a more concurrent design process is used, where several aspects are established simultaneously (Rigo and Caprace, 2011). This has been made possible through computer-aided ship design (CASD) platforms where all parties involved can work simultaneously and relative seamlessly with the product. This makes for a faster timeline until delivery, but as shipbuilding is an ad hoc process, it also includes difficulties (Jean-David Caprace Frederic Bair, 2010).

Whether the design process is sequential or concurrent, the fact that there will occur changes along the design process is inevitable. And changes can be costly, especially if they occur towards the end of the process (Chang et al., 2011). The amount of information about the vessel that is to be designed and manufactured is key to achieve a smooth and fast manufacturing process. The earlier this information is obtained, the easier it is to avoid costly changes and staying within budget.

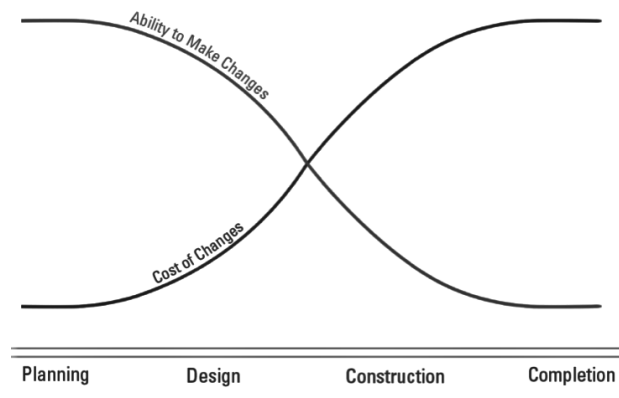


Figure 1.2: Cost of changes as a function of timeline (Hooker, 2015)

The hierarchy/timeline of the design process is one aspect that has changed, but the way of designing the ship structure itself has also changed, compared to earlier. Scantling was obtained from simplified formulas given by classification societies. This made for a short approval process for the vessel, which is good, but there are drawbacks to this method. Using simplified equations that are set to be valid for a variety of designs, demand a certain level built-in margins (F. Hughes et al., 1980). These margins and this simplified way of determining vessel parameters such as plate thickness, etc. will make the final design far from optimum.

Now that new technology and a lot more computational power is available, the design process is much more advanced. Computer-aided ship design (CASD) and computer-aided engineering (CAE) have set the industry standard for what to expect when developing new vessel designs. CAE is the collective name for computer analysis such as finite element analysis (FEA), computational fluid dynamics (CFD), etc. (B. O. Saracoglu, 2006). These programs require skilled engineers in order to develop and validate the results. The process is also based on optimizing the vessel structure to make sure the optimum design is chosen, and this is time-consuming.

In the early stages of the design process, the concept phase, a full computer aided design analysis is way to expensive and time consuming. Hence a best possible approximation is what we want and a guidance set of where the need and use for added steel weight contribute

the most to the design and strength of the hull.

1.3 Objective

The design of a cruise vessel can be arranged in a multitude of different ways. It is possible to achieve the same amount of displacement in numerous ways, but not all designs are equally good. Stability, drag forces, space utilization, etc. are all affected by the design. This paper will be looking into how the steel weight is affected for different designs and if there is a superior design in terms of minimizing the steel weight. In order to develop a best possible approximation of the steel weight and for it to be as low as possible, the correlation between hull-design, strength, and weight is important to be aware of. The use of FEM analysis is the basis for this information and how these relations can be obtained. The goal of this thesis is to find the relation between hull-design, strength and steel weight and from this develop a parametric relation of the parameters contributing to the steel weight.

The objectives of this thesis can be summarized by the following questions:

- Can FEM and an optimization algorithm be used to provide a superior design in terms of weight and strength?
- Is topology optimization able to provide new and lighter hull designs?
- How to best strengthen a midship hull section?
- Using the findings for the questions listed above to answer. Can FEM be used to develop a quantifiable relationship between the main parameters, in terms of steel weight?

Chapter 2

Existing methods to estimate steel weight

The estimated steel weight for a vessel can be calculated in a number of different ways. The methods vary in terms of complexity and how good they actually represent the vessel in question. All the methods can be categorized under one of four classes. Each class is based on its individual approach and vary somewhat in how much knowledge is needed for it to be utilized.

The first class is based on the ship characteristics, meaning the steel weight is calculated as functions of the main dimensions (Schneekluth and Bertram, 1998). The next class relies on information and data from already existing vessels. This method can be hard to implement if the new-building is hard to relate to existing vessels, hence a solution is to combine this method with the first class mentioned (Watson, 2002). This approach is widely used and often referred to as parametric estimation and can in many cases produce good preliminary weight estimates. The third class is related to the surfaces of the hull, but an approximately correct general arrangement drawing is required for the bulkheads weight contribution to be included correctly. The last class/method is based on using a midship section and uses the weight of the scantlings from this section to project the full steel weight. This approach is commonly used by classification societies. Some examples of the different methods will be provided to showcase their structure.

Weight estimates for bulk carriers and cargo ships are relatively easy to determine, due to their simple form and hull specifications. Hence an estimate at concept stage will usually be within 1-3% of the actual weight (Schneekluth and Bertram, 1998). This can be done using coefficient methods, on the form:

$$W_{st} = L^a \cdot B^b \cdot D^c \cdot C_B^d \cdot e \quad (2.1)$$

Here a-e are constants. There are also examples of some more specified formulas explicitly for container ships, dry cargo vessels or tankers. They have in common that they are all based on relatively few inputs such as length, width and depth (Schneekluth and Bertram, 1998).

For a passenger vessel such as a cruise ships, more information is needed due to the complexity of the structure and varying structures. A common way of determining the steel weight and lightship weight for these new vessels is to look to similar ships, preferably parent ships as they provide the most correct representation of the new vessel.

If the new building is not a reproduction and cant be based solely on a parent ship. Then statistics is another good way of estimating the steel weight. Using data from similar ships that have already been built and then use this data to determine weight factors similar to the system based ship design approach. These weight factors can be related to ship functions, areas or for specific decks i.e. The total steel weight can then be found by determining what functions, areas or number of decks i.e the new vessel is to have and then obtain the steel weight by multiplying corresponding factors and attributes (Anondsen, 2018). This method is heavily based on having a wide pool of information to make sure the weight factors are correctly developed. (Lin, 2016) presents one version of this method called parametric section-based estimation (PSE), where the goal is to produce a more precise steel weight estimation based on the limited information available from the preliminary design.

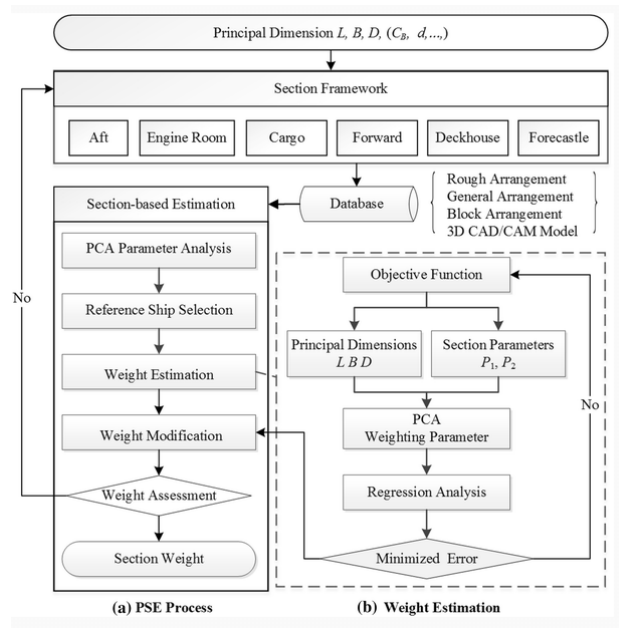


Figure 2.1: PSE approach (Lin, 2016)

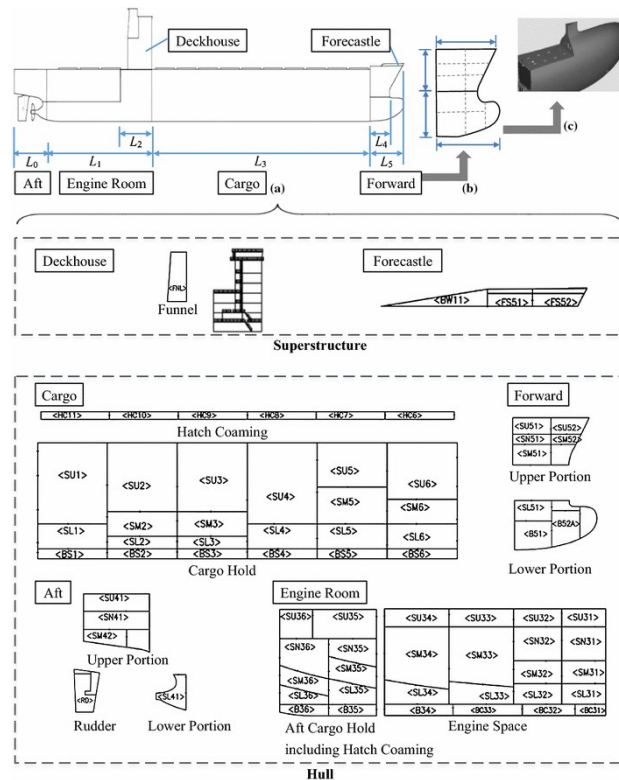


Figure 2.2: PSE for cargo vessel (Lin, 2016)

The model was developed for cargo vessels but can be used for all types of vessels as long as you have access to information from other similar already built vessels. The information will have to be quite specific as the method is based on dividing the hull into subgroups that are all to have their own weight factor. This information is often hard to obtain as the shipyards are very restrictive in handing out information to the public and competitors. A result of this, is that the amount of information about steel weight for different ship types are scares compared to lightship or deadweight information. Hence the correlation between steel weight and lightweight can be used to estimate the steel weight.

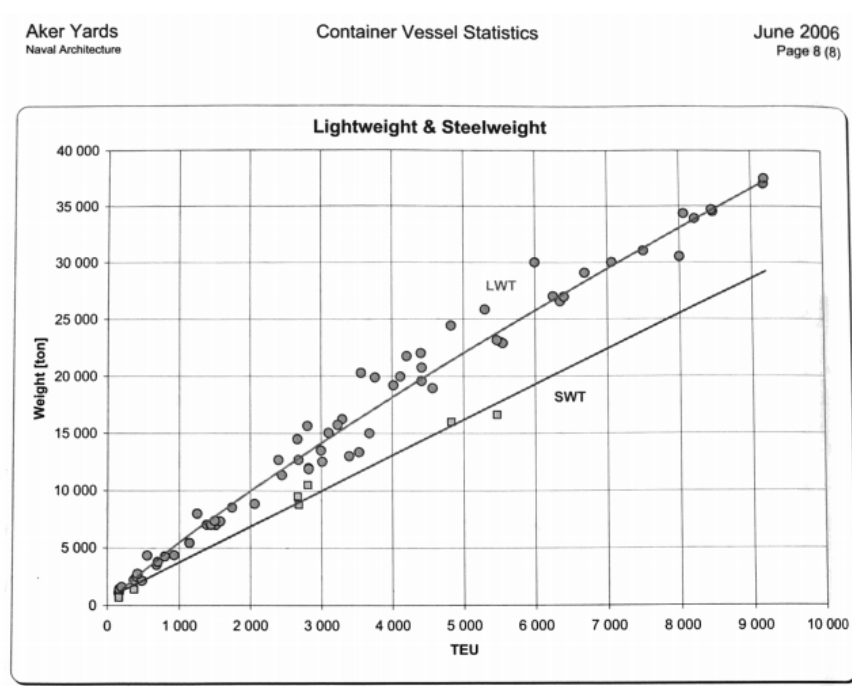


Figure 2.3: Lightweight and steel weight (Levander, 2006)

There are also other generalized methods of determining steel weight that are set to work for a broad specter of vessels. Strohbuschs approach from 1928 is one way, which to this day is still relevant for steel weight calculations, especially for unconventional ships (Papanikolaou, 2014). For the approach to be applicable, one has to assume that some information about the ship is available. More specified, it uses the structural plans of characteristic sections and based on this information, the steel weight for the vessel is calculated.

$$W_{st} = \int_L \frac{dW_{st}}{dx} = \int_L W_{st}(x) dx \approx \sum_N W_{st}(x_i) \delta x_i \quad (2.2)$$

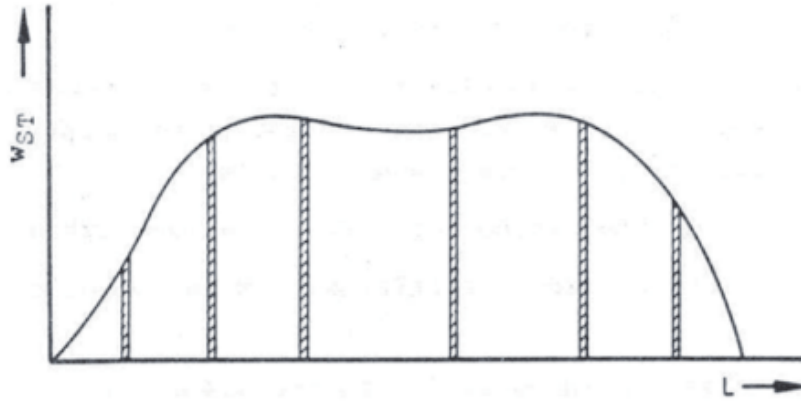


Figure 2.4: Strohbusch steel weight graph (Papanikolaou, 2014)

Another method that is also applicable for passenger/cruise vessels is Schneekluth's Method which builds on the Strohbusch method. The method was originally used for dry-cargo ships, but was later found to give quite good estimations for a broad range of vessels. The drawback with this method is that it only addresses the weight of the hull and do not account for the superstructure. Hence the approach needs to be combined with another method that evaluates the superstructure. This could for instance be done using Müller-Kösters method or other equivalent methods that account for the super structure. Both hull and superstructure must be accounted for to compute the total steel weight for the whole ship.

Schneekluth's approach for calculating the steel weight is given as a function of the total (estimated) volume and is based on a set of assumptions. The following parameters will have to be known in order to complete the steel weight calculations (Papanikolaou, 2014).

L	Length between perpendiculars [m]
B	Breadth [m]
T	Design draft [m]
D	Side depth of the uppermost continuous deck [m]
C_B	Block coefficient at design waterline (draft)
C_{BD}	Block coefficient at height D
C_M	Midship section coefficient
S_F	Sheer height at FP [m]
S_A	Sheer height at [m]
b	Camber height at the midship section [m]
n	Number of decks
∇_U	Volume below the uppermost continuous deck [m ³]

Table 2.1: Parameters needed for Schneekluth's approach

$$\begin{aligned}
W_{st} = \nabla_u C'_{st} \cdot [1 + 0.033(\frac{L}{D} - 12)] \\
\cdot [1 + 0.06(n - \frac{D}{D_0})] \cdot [1 + 0.05(1.85 - \frac{B}{D})] \\
\cdot [1 + 0.2(\frac{T}{D} - 0.85)] \cdot [0.92 + (1 - C_{BD})^2] \\
\cdot [1 + 0.75C_{BD}(C_M - 0.98)]
\end{aligned} \tag{2.3}$$

The C'_{ST} is given as a function or a coefficient, depending on the ship type. For a passenger vessel $C'_{ST} = 0.113-0.121$ and is valid for vessels between 80 and 150 meters.

Yet another method is Harvald and Jensen's method which was developed in 1992 and is based on cargo ships built in Denmark from 1980 to 1990. Their method was found to predict the steel weight within 10% of the actual weight (Schneekluth and Bertram, 1998). Their formula is not dependant on many parameters or inputs, and can be computed based solely on the length, breadth, depth, displacement and a constant depending on the ship type.

$$\begin{aligned}
W_{st} &= (L \cdot B \cdot D_A) \cdot C_s \\
C_s &= C_{so} + 0.064e^{-(0.5u+0.1u^{2.45})} \\
u &= \log_{10}\left(\frac{\Delta}{100t}\right)
\end{aligned} \tag{2.4}$$

$C_{so} \left[\frac{t}{m^3}\right]$ is given for the ship type of interest. $C_{so} = 0.058$ for passenger vessels, compared to i.e

$C_{so} = 0.070$ for bulk carriers.

2.1 Commentary to existing methods

All methods that have been presented in this chapter are in various degree based on knowledge/information from already built ships. When a new vessel is based on information from previously built ships, that decreases the risk of choosing an infeasible design in terms of steel weight. The drawback is that the design might be based on vessels that have been designed using built in margins that cause an unnecessary increase in the steel weight. This problem could be diminished if the design was based on a FEM approach, where the weight can be minimized. This process starts by developing a relation between how design-parameters affect the strength and steel weight. Another point worth mentioning is that the majority of the methods listed require quite a lot of information about the vessel that is to be built. That information can be hard to obtain and require the whole design process to be quite far ahead for the estimates to yield any results. That means that when there is enough information to compute a steel weight estimate, the design is already set, and can hardly be changed. This introduces a new problem with the approaches, the methods does not provide any clear information about the parametric relation between the impact of the different parameters. Hence there is no clear guidance for how to improve the design in terms of steel weight.

Chapter 3

Problem

The problem that will be investigated in this thesis is related to the design of cruise vessels, and more specifically the steel weight of cruise vessels. As explained in section 1.1, the process of designing and developing new cruise concepts has changed as new technology has been made available. One crucial problem that is the same now as it was earlier is trying to estimate the steel weight based on the minimum of information provided at a concept stage. The steel weight is again as mentioned earlier a large contributor to the total cost of a cruise vessel. Hence minimizing the steel weight and being able to predict the steel weight, is crucial in regards to keeping the cost low and within budget. The goal of this thesis is to try to help provide a solution to this problem through the use of FEM for developing a quantified relation between the main parameters (L-B-D).

3.1 Approach

A short introduction to the thought process and way of approaching the problem is stated below. The comprehensive discussion and findings will be addressed later in their respective chapters. The theory worth mentioning is also presented throughout the chapters.

The approach towards a solution can be seen as a stepwise process. The first aspect of the

problem was to look into ways of minimizing the steel weight. This was done first in order for the results to be used in the second stage of the project. Developing the FEM models needed to create the parametric equations for the global parameters and their relation in regards to the steel weight.

Trying to minimize the steel weight was tried in three different ways, all attacking the problem in a new way. The first approach to minimizing the steel weight was by using a particle swarm optimization (PSO) algorithm to optimize a deck structure in terms of steel weight and strength. The second approach was by looking to develop completely new and better midship designs in regards to steel weight, by utilizing topology optimization. The last weight minimizing experiment was to do parameter correlation tests for stiffeners and comparing them against each other.

The information obtained from the minimizing projects were then to be utilized as help and guidance in the construction of the models used to create parametric steel weight relations for the main parameters. The FEA models constructed for the tests were also heavily influenced by rules and requirements given by DNV GL.

Chapter 4

PSO approach

The first step towards developing a decision support/guidance tool was through analyzing and optimizing the design of a deck. The goal for this experiment was to develop a superior design in regards to steel weight and strength. FEM and a best suited optimization algorithm should provide the means necessary to calculate and develop a parametric relation that could be used to calculate steel weight for a given deck with various dimensions. This information from the deck structure was then thought to be used as a basis and template for a complete midship section.

4.1 Optimization problem

The optimization part of the problem was essentially minimizing the total weight of the structure, while still being within the feasible domain denoted by the constraints related to stress, strain and deflection. The weight function for a cruise vessel hull is non linear and the complexity of the function increases as the complexity of the structure increases. This optimization of steel weight was the first attempt at developing a superior relation between the design parameters.

4.2 Different optimization algorithms

When optimizing/minimizing the steel weight for a cruise ship, reducing the amount of steel in the structure, while still meeting the requirements in terms of strength was what I aimed for. There are in general two different strategies to find a optimum solution. There are deterministic ways and there are stochastic approaches.

4.2.1 Deterministic

The deterministic way of deriving a optimum solution, starts in a given solution (feasible or not) and identifies what direction to move based on the best way of propagation from that given starting point. Examples of deterministic methods are, BFGS, simplex, steepest decent etc. All methods are great at finding local optimum, and also global optimum especially if the domain is convex. If the deterministic methods don't find the exact global optimum, they are guaranteed to obtain a local optimum that will differ no more than a given ϵ from the global optimum (Leo Liberti, 2005). The amount of iterations and time needed for a deterministic solver to reach the global optimum is highly dependent on the form of the problem to be optimized, especially asymptotic functions have proven difficult. Zabinsky found that there were limitations to several deterministic approaches in regards loose bounds and solving practical problems with more than 10 variables (Zabinsky, 1998).

4.2.2 Stochastic

A stochastic approach is not based on starting in a specified point as the deterministic approach, rather it relies on randomly testing solutions in the feasible domain, hoping to find the global optimum. There are two categories of stochastic methods, the Monte Carlo- and the Las Vegas- methods. Las Vegas algorithms guarantee that global optimum is found, but "gambles" at the expense of resources/time used to reach the global optimum. The Monte Carlo approach on the other hand is defined as an algorithm that may or may not reach/find

the global optimum, but the probability of an incorrect answer is bounded (R. Tempo, 2007). Dixon and Jha found in a study that a stochastic MLSL method compared to a deterministic interval method would be at least 8 times faster (dependent on complexity of the problem) (L. Dixon and Szego, 1975).

A heuristic approach is essentially a "smart" version of a stochastic approach. These methods "scan" the feasible domain and use this knowledge when choosing what points to be tested, rather than choosing points completely at random. Examples of heuristic search algorithms are; Particle swarm optimization (PSO), Tabu search, Genetic algorithms etc.

4.2.3 Choice of algorithm

There was a set of characteristics that were desirable when we looked to find the optimum steel weight. The algorithm had to be precise enough to make sure the basis for the parametric equation was as correct as possible. Computational time would have to be low in order to evaluate all the different design parameter combinations fast and do so without getting "caught" in any local minimums. The algorithm also had to be able to handle non-linear and complex functions. All design combinations had to be analyzed with FEM to make sure the solutions were feasible in regards the regulations given by the classification society. Hence a minimum amount of steps used to obtain optimum was desirable.

Based on this, the best choice of algorithm for minimizing steel weight was found to be a heuristic as it is fast, minimizes amount of iterations/steps, good at avoiding local optimum and precise. Heuristic methods are also capable of handling non-linear and complex problems. All needed for the task at hand.

The specific type of heuristic chosen for the steel weight minimization was a particle swarm optimization (PSO) algorithm. This algorithm was chosen based on a number of reasons. PSO have constant computational cost and memory engagement at each iteration. An approximate solution was not required to initiate the optimization. The derivative of the objective function could be unknown and it manages non-convex problems (Pinto, 2007).

A particle swarm optimization is a global optimization technique originally introduced to solve unconstrained single objective optimization problems (Pinto, 2007). The PSO algorithm is an iterative search process through a given domain, trying to find the best global optimum. This is done by introducing a "swarm" of "particles" that communicate and scan the search/design space to find the best solution. Each particle has a velocity and a starting position. Where the particle moves/searches is influenced by that specific particle's best known solution and also the best solution found collectively by the swarm.

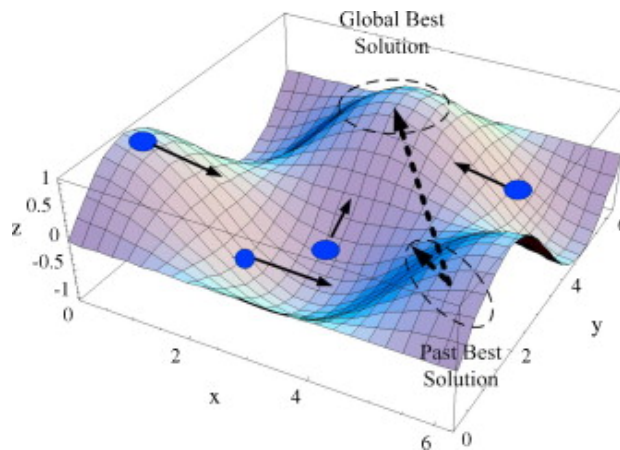


Figure 4.1: PSO illustration (Biswas, 2016)

The PSO algorithm is dependent on a set of inputs to obtain the correct/optimum solution (Engelbrecht, 2007).

- Number of particles,

The amount/number of particles that are set to search the design space will have to be determined. High numbers of particles result in a better covering of the domain. This again results in the computational time for each iteration to be more demanding, but a big swarm size may also result in fewer iterations needed to achieve the global optimum.

- The particles acceleration coefficients,

The particles direction of searching are dependent on information from the rest of the swarm. Two coefficients often called trust parameters need to be established, one

denoting the trust one particle has in itself/own solution and one denoting the trust a particle has in its neighbours solutions. The coefficients will have to be chosen such that the particles rely on both own and neighbours information to accelerate/search in the right direction.

- inertia weight,

The inertia weight is used to gain control over particles search velocity. The inertia value determines the velocity for particles, from one time step to the next.

- neighborhood size,

The neighborhood size determines the extent of social interaction within the swarm. Smaller neighborhoods increases the chance of convergence towards global optimum, but the time until convergence is larger.

- number of iterations, The amount of iterations set to be conducted before terminating the search. Not enough iteration can cause a non optimal solution to be final. Too many iterations cause the computational time to be unnecessarily large. (Engelbrecht, 2007).

4.3 Finite element method

Finite element method (FEM) is a numerical method used to approximate solutions for boundary value problems with partial differential equations. In structural analysis, FEM is used to calculate rotations and translations that are used to compute the stress and strain throughout the structure. FEM works by subdividing the structure into many smaller problems, finite elements. This process is often referred to as meshing and is crucial to get right in order for the calculations to be correct. The elements that are generated are solved as individual problems by using shape functions, before being reconnected to form the full picture.

4.3.1 Finite elements

There are a lot of different element types to choose from when using ANSYS. Choosing the right element types is crucial for the finite element analysis to return correct results. All the different element types have their own strengths and weaknesses and they are based on different theories, making them applicable for a variety of different structures and environments. (ANSYS, n.d.a)

The deck structure had to include two different element types, based on what purpose and function they had to serve. Plates were defined as shell elements, while stiffeners and girders were defined as beam elements. The important part of having the stiffeners and girders defined as beam elements was to ensure that axial, torsional, bi-directional shear and bending stiffness can be accounted for.

4.3.2 Meshing

The mesh is as mentioned an important part of the finite element analysis and it can also be an extremely tedious process. For a FEM analysis to return precise results, it is dependent on having a good mesh. A good mesh is a mesh that return results that are within an acceptable level of accuracy, given all inputs are correct (Hale, 2014).

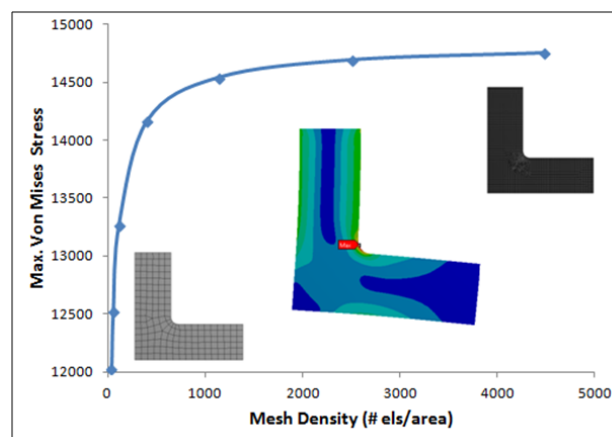


Figure 4.2: Stress sensitivity for different mesh sizes (Hale, 2014)

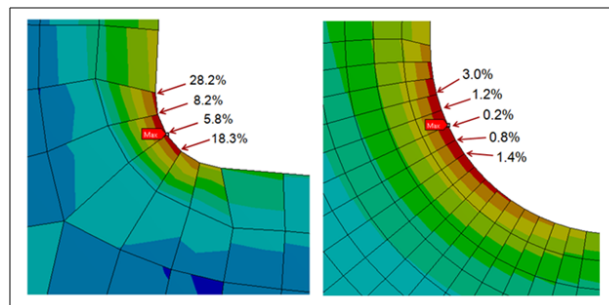


Figure 4.3: Shared nodes for coarse versus fine mesh (Hale, 2014)

A finer mesh will give a better representation of the structural behavior, but it comes at a cost. The finer the mesh the more time it will take to complete the mesh and hence cause the FEM analysis to require even more computational power and time. In order to know if the mesh size is dense enough to correctly project the actual situation, a test is carried out, to check that the peak stress converges for decreasing element sizes.

The time it takes for one optimization to be completed will increase non-linear with the computational time used in ANSYS. The computational time in ANSYS is directly dependent on the mesh size, hence it is desirable to use a mesh size as large as possible while still maintaining the accuracy of the analysis.

$$T = C \cdot SP \cdot (G + 1)$$

T = Optimization time

C = Computational time in ANSYS (4.1)

SP = Swarm particles

G = Number of generations

The mesh size does not require the same refinement throughout the whole structure. For areas where we know that the stresses will be relatively consistent or gradually increasing/decreasing, a coarser mesh can be used. For critical locations where high stress values and rapid changes are expected to occur, a finer mesh is required. This is important in order to reduce the computational time for large and complex structures, but as long as the structure is relatively basic and not too large, a constant mesh size can be used.

Chapter 5

FEM model for PSO

The FEM model used for the PSO experiment was chosen to be a deck structure as a deck structure has many of the components needed in a typical hull design. Girders, stiffeners and a plate are all represented, walls and other vertical support are examples of components not included for this model. The goal of the model was to provide information about any superior design if found. This meant analysing the results and check if there were some clear parameters that stood out as crucial for the steel weight to be as low as possible. A parametric relation was also to be created for the models, relating area to optimized steel weight. Much like the final test results from the midship section are transformed into parametric relations.

5.1 Model description

The chosen model was a section of a deck, supported by stiffeners and girders. The deck section was subjected to a uniform load and the deck was modeled as simply supported in four nodes. Three of the nodes were restricted against translation in x- and z -direction while one node was fully restricted in all directions.

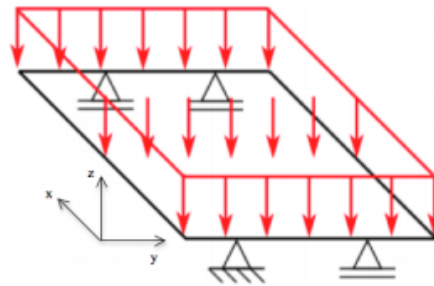


Figure 5.1: Model description

5.1.1 Parameters

The parameters for the model were the inputs that were set to be constant throughout the optimization. Most of these parameters were changed from one optimization to the next, while Young's modulus and gravity were the only parameters kept constant through all the different tests. Some of the parameters that changed were defined manually while the rest were defined as scalars. The parameters that were manually changed from one optimization to the next were plate length and plate width. All other parameters were defined as scalars of the plate length, plate width, Young's modulus, and gravity. It is also worth mentioning that the evenly distributed load was linearly increasing in compliance with the deck area.

Parameter	Definition	Unit
Plate length	L	m
Plate width	B	m
Number of girders	2	-
Gravity (g)	9.81	m/s^2
Youngs monulus (E)	$250 \cdot 10^6$	Pa
Length to first girder	$0.5 \cdot (\sqrt{2} - 1) \cdot B$	m
Length to second girder	$B - (\text{length to first girger})$	m
Load	$L \cdot B \cdot g \cdot 1466$	N

Table 5.1: Deck parameters

The girder placements were set as scalars of the plate width. The girder placements were not optimized in the MATLAB-script as they could be optimized by knowing that the moment peaks will have to be evenly distributed (fig. 5.2).

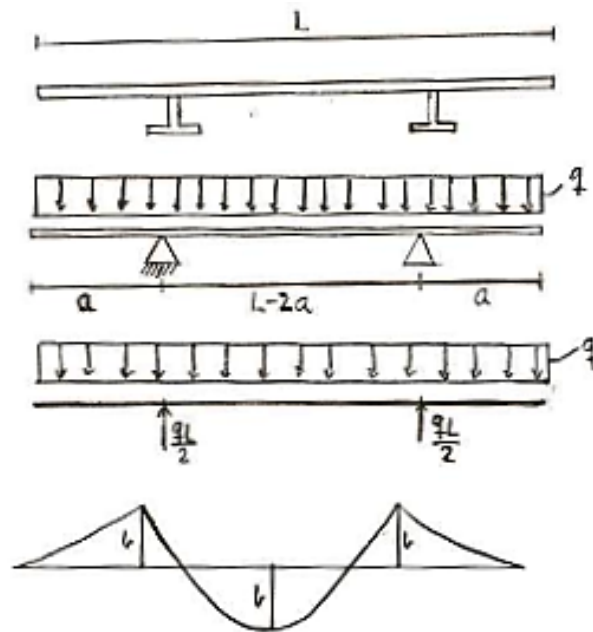


Figure 5.2: Girder placement

5.1.2 Variables

The variables formed the dimensions for the model and were set to be optimized for each deck configuration. The variables varied within a domain set for each variable. The final combination of these variables amounted to the optimum design and minimum steel weight. For some of the variables, the range was set to start below the minimum requirements set by (DNVGL, 2016) for the reason of testing whether or not sufficient strength could still be ensured, even outside of the DNV GL regulations.

Variables	Range	Unit
Plate thickness (pt)	0.005-0.030	m
Girder height (gh)	0.30-0.80	m
Girder thickness (gt)	0.005-0.035	m
Flange width (fw)	0.04-0.40	m
Flange thickness (ft)	0.005-0.05	m
Number of stiffeners (ns)	6-35	-
Stiffener height (sh)	0.03-0.2	m
Stiffener thickness (st)	0.005-0.030	m
Bulb height (bh)	0.005-0.050	m
Bulb width (bw)	0.005-0.050	m

Table 5.2: Variables

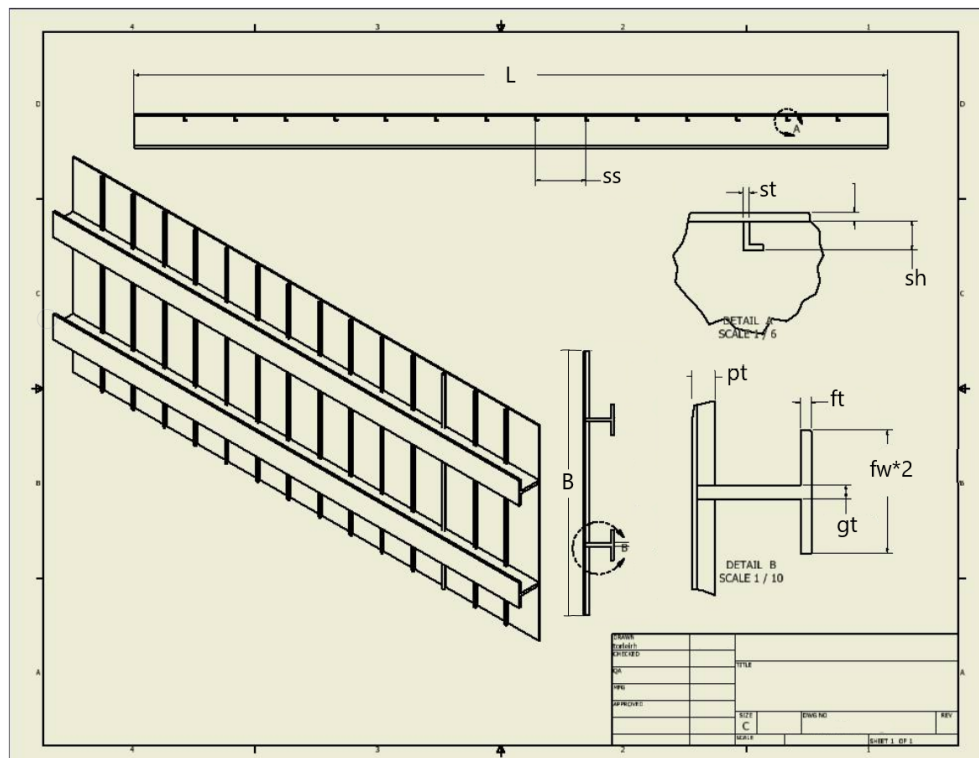


Figure 5.3: Technical drawing

5.1.3 Objective function

The objective of this task were to minimize the steel weight for the model. Hence the objective function was how to combine all the different parameters and variables that made up the deck section and then multiply that volume (V) with the steel density.

$$\text{Min(Steel weight)} = \text{Steel density} \cdot \text{Volume} \quad (5.1)$$

$$\begin{aligned} V = & (pt \cdot pw \cdot pt) \\ & + 2 \cdot ((pl \cdot gh \cdot gt) + (2 \cdot fw \cdot ft)) \\ & + ns((pw \cdot sh \cdot st) + (bh \cdot bw)) \end{aligned} \quad (5.2)$$

5.1.4 Constraints

The objective function was subjected to a set of constraints to ensure the deck section had sufficient strength. In all, there were three constraints, one limiting the deck height to no more than 5% of the length, the next constraint limited the deflection to maximum 1% of the length and the last constraint restricted the stress to be less than 80% of the Youngs modulus.

$$\begin{aligned} C_1 &= \frac{pt + ft + gh}{0.05 \cdot pl} \\ C_2 &= \frac{\text{max deflection}}{0.01 \cdot pl} \\ C_3 &= \frac{\text{max stress}}{0.80 \cdot E} \end{aligned} \quad (5.3)$$

5.1.5 Von-Mises

One of the three constraints was set to limit the stress. More specifically the constraint limits the Von-Mises stress to maximum 80% of the Youngs modulus. The decision of using Von-Mises stress was based on its properties being great for determining yield strength for ductile materials and handling complex loading conditions. The uni-axial tensile tests and

stress combination nature provided by Von-Mises is also highly desirable for a FEA like the one at hand.

5.1.6 ANSYS Elements and steel identities

The two elements used for the model were shell181 and beam188. Shell181 was used for most of the structure, except for the stiffener bulbs. Shell181 is based on Mindlin-Reissner theory (ANSYS, n.d.*b*), suitable when the thickness to length ratio is below $\frac{1}{10}$ (Peter Hansbo, 2011). Beam188 was used for the stiffener bulbs and is based on Timoshenko's beam theory, ideal for small height to length ratios (ANSYS, n.d.*a*). The theory behind the two elements (Beam188 and shell 181) builds on different theories as mentioned, and therefore it also introduces a potential source of error. The response for the two elements might deviate from each other and cause a wrong representation. This was tried minimized by analysing the structure thoroughly to validate the responses throughout the structure.

The steel used was set to be A36 steel as it is a commonly used steel which is easy to work with and quite cheap. A36 has a relatively high yield strength of 250 MPa, a Poisson's ratio about 0.3 and a density of $7850\text{kg}/\text{m}^3$. More information about different steel types and their strengths can be found in the chapter concerning the rules and requirements by DNV GL 9.1

5.1.7 PSO parameters

The optimization algorithm as mentioned earlier, a particle swarm optimization and this algorithms ability to correctly derive the optimal solution is based on a set of predefined inputs. The value for these parameters was based on the theory from (Engelbrecht, 2007).

Parameter	
Swarm size	60
Number of generations	30
Starting inertia	1,4
Beta (acceleration coefficient)	0,8
Beta k (rounds before decreasing inertia)	3
Penalty factor	2

Table 5.3: PSO Parameters

5.1.8 Mesh size

The mesh size used in the FEA was found to be 20 mm. This value was among the largest element sizes that still maintained acceptable results. This size was chosen in order to keep the computational time as low as possible and the results as correct as possible ref.4.3.2.

Chapter 6

Results of PSO approach

The parametric equation for estimating the steel weight of the deck was based on a number of different combinations of length and width. Changing these parameters simulated different deck sizes and were the basis for what was hoped to provide a guideline for a superior design. Based on these inputs, the optimization algorithm derived the other parameters such as the number of stiffeners, plate thickness, etc. that would result in the lowest possible steel weight, while not exceeding the boundary conditions.

Plate length	11,00	13,00	15,00	17,00	19,00	[m]
Plate width	3,00	4,00	5,00	6,00	7,00	[m]
Deck area	33,00	52,00	75,00	102,00	133,00	[m ²]
Number of girders	2,00	2,00	2,00	2,00	2,00	[-]
Length to first girder	0,62	0,83	1,04	1,24	1,45	[m]
Length to second girder	2,38	3,17	3,96	4,76	5,55	[m]
Plate thickness	0,0050	0,0050	0,0050	0,0050	0,0050	[m]
Girder height	0,3000	0,6000	0,6600	0,5000	0,8000	[m]
Girder thickness	0,0050	0,0050	0,0050	0,0340	0,0050	[m]
Number of stiffeners	32,00	35,00	33,00	22,00	35,00	[-]
Stiffener height	0,0300	0,0400	0,0300	0,0300	0,1000	[m]
Stiffener thickness	0,0050	0,0050	0,0300	0,0300	0,0050	[m]
Flange Width	0,4000	0,1000	0,0400	0,0400	0,1400	[m]
Flange thickness	0,0050	0,0230	0,0500	0,0050	0,0500	[m]
Bulb Width	0,0050	0,0050	0,0500	0,0050	0,0050	[m]
Bulb Height	0,0500	0,0500	0,0050	0,0050	0,0500	[m]
Weight	2530,30	4060,70	6113,30	9544,90	11955,00	[Kg]

Table 6.1: Results from test model

Plate thickness, a high number of stiffeners and small bulbs were some of the trends that were worth mentioning. With some exceptions, these parameters displayed a clear tendency for all the optimized designs. The irregularities and further discussion on this topic can be found in the next section 6.1.2.

All the different lengths and width combinations were tested more than one time to ensure that the weights that were found actually were the best possible solutions.

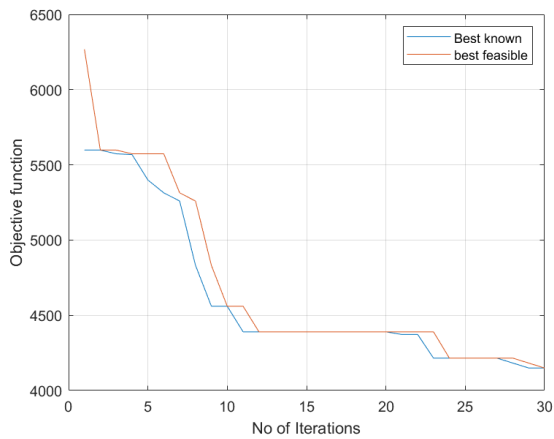


Figure 6.1: 30 iterations
with swarm size of 60 for deck: $L=13, B=4$

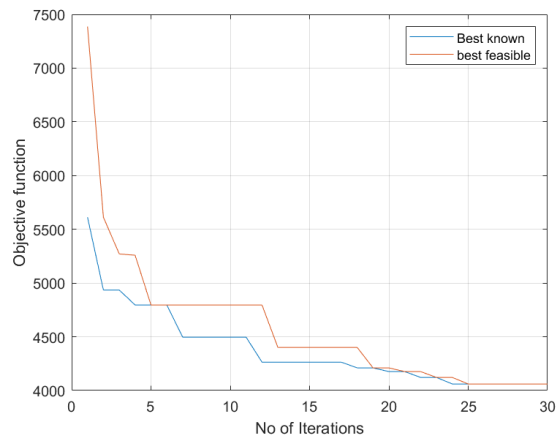


Figure 6.2: 30 iterations
with swarm size of 60 for deck: $L=13, B=4$

Figure 6.1 and 6.2 shows that the best feasible solution and the best known solution gradually converged to the same value for both runs. This indicated that the optimal solution should be very close.

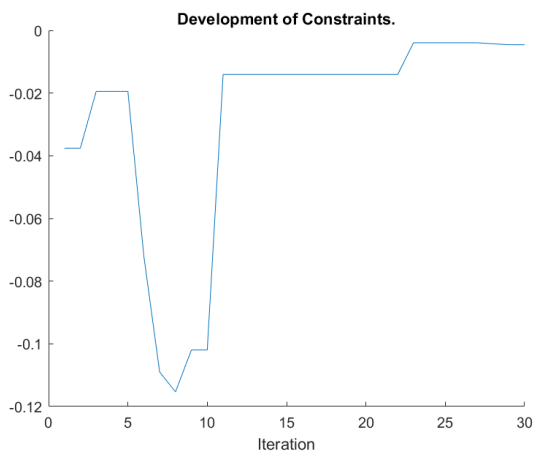


Figure 6.3: 30 iterations
with swarm size of 60 for deck: $L=13, B=4$

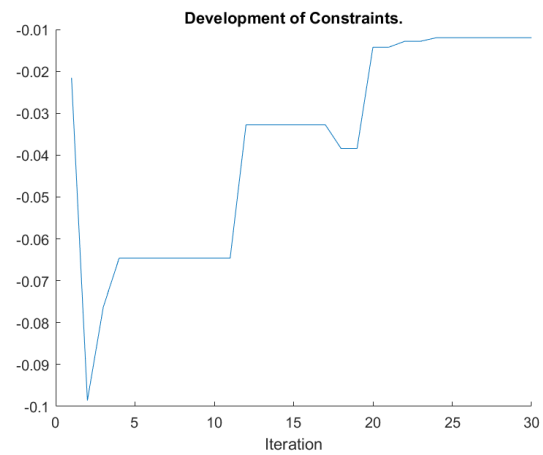


Figure 6.4: 30 iterations
with swarm size of 60 for deck: $L=13, B=4$

Figure 6.3 and 6.4 displays the development of constraints and how they changed throughout the optimization before converging when optimum was near. All figures seen above displays the results of optimizing the steel weight for a deck with length 13 meters and width 4 meters. The weight of the structure stabilized at 4,0607 tonnes. The design of the deck resulting in the lowest weight was.

Plate length	13,00	m
Plate width	4,00	m
Number of girders	2,00	-
Plate thickness	0.0050	m
Girder height	0.6000	m
Girder thickness	0.0050	m
Number of stiffeners	35.0000	-
Stiffener height	0.04000	m
Stiffener thickness	0.0050	m
Flange Width	0.1000	m
Flange thickness	0.02300	m
Bulb Width	0.0050	m
Bulb Height	0.0500	m
Weight	4060,70	kg

Table 6.2: Finished design for deck with L=13, B=4

This information was obtained for all the different deck sizes that were tested, ranging from 11-19 meters in length and 3-7 meters in width. The stress distribution, and deflection was calculated using ANSYS, which also produced a visualization of the actual design.

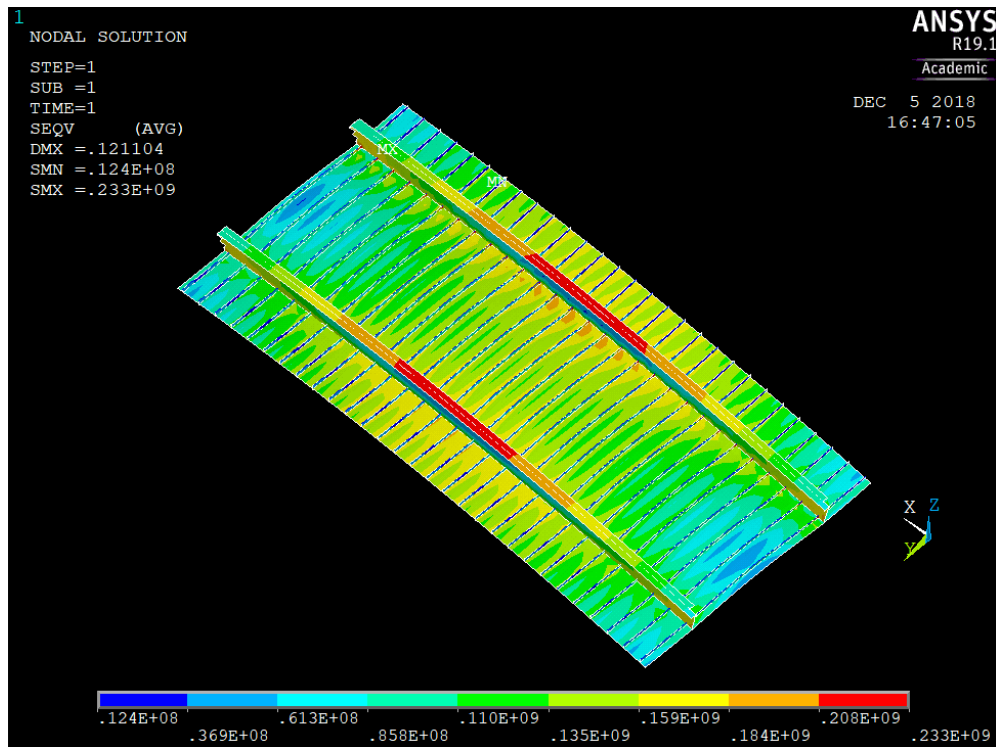


Figure 6.5: Stress distribution

The results from all the different length and width combinations were then plotted against the weight to form a graph displaying the estimated steel weight for decks ranging from 33-133 m^2 . This graph and the associated parametric equation was then to provide information regarding the best design and what parameters and structural supports to utilize in order to minimize steel weight. Preferably this kind of optimization analysis could be done for a complete midship section, but the reasons why that was not the case are discussed later.

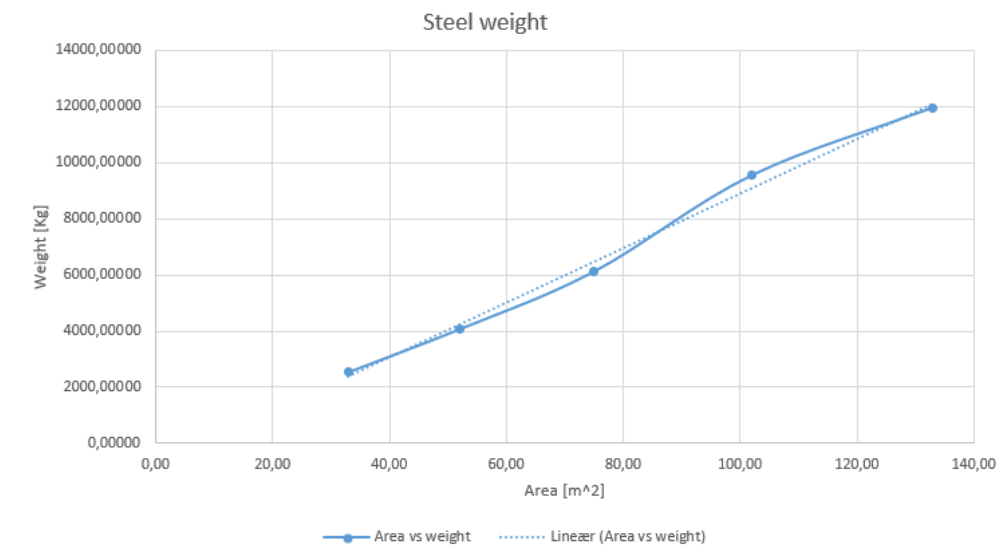


Figure 6.6: Steel weight estimate for test deck

$$W_{st} = 0.01X^2 + 95.56X - 794 \quad (6.1)$$

$X = \text{Area of deck}$

6.1 Discussion of PSO, FEM results and method

Finite element analysis (FEA) and particle swarm optimization (PSO) were used to optimize the steel weight for a deck structure. The deck structure that was chosen as the test model was chosen as it was one part of a midship hull section. However, there are some problems regarding how well the test model projects the more complex structure. Some assumptions and simplifications had to be done. The load was set to increase linearly as a function of the area, the deck was only supported in four nodes and not restrained against rotation and the load was set to be evenly distributed. If a similar structure optimization were to be done for a midship section, more predetermined parameters would have been needed, than for the test model.

6.1.1 Potential sources of errors

The structural model and optimization model are both crucial for the results to be correct, but they are also both sources for potential errors. The structural model and the calculation of stress, strain, and deformation for the structure was highly dependent on all the parameters and elements being modeled correctly. On top of this, the mesh size cannot be infinitely small and as a result, there will be a slight difference in what the analysis returns and what the actual values were. The PSO had two potential pitfalls. One being that the PSO is a heuristic that is not guaranteed to return the global optimum, there was a chance that the PSO returned a local optimum instead of the global. The chance of this happening was tried diminished by using large swarm sizes and many iterations to make sure as much of the domain as possible was evaluated. The optimization process was also carried out several times for each structure to make sure the results converged to the same weight. The second pitfall for the PSO is that the algorithm only searches through a predetermined domain, restricted by the analyst/me. This has both a positive and a negative side to it. The positive aspect of a predetermined design space was that infeasible design parameters could be excluded before even starting the optimization (minimum plate thickness etc.). The downside to the predetermined design domain was that desirable solutions can be deemed unobtainable if the domain was too tightly restricted.

The final result cumulated in a graph displaying the relationship between the steel weight and the area of the structure. This graph was set to be valid within the interval of 33 - 133m². The graph can not guarantee correct results beyond the set limits as there were no information to base the estimations on, only a trend line. There will also be a slight chance for the estimations within the limits to predict the wrong weight for a given area as the estimated weight curve was based on the FEM and PSO -analysis, which as explained earlier might return incorrect results. If given more time, the analyses could have been performed on a larger set of deck structures, making the regression even better and valid for a larger set of deck sizes as well as running even more thorough analyses (finer mesh, larger swarm, more iterations, etc.).

6.1.2 Observations from the results

The optimization process was performed several times for the different deck configurations to test that the steel weight converged towards the same steel weight in all runs, and it did in most of the runs. However, there are some curiosities regarding the parameter composition for each of the deck configurations. One would think that by doing this analysis, the results would have the same trend in all the optimized designs. Meaning that the parameters kept low for one deck configuration, would also be tried minimized for the next slightly larger deck, etc. This was not the case for all the optimized parameters. Some parameters displayed a clear tendency to want to be kept low, i.e plate thickness, while other parameters showed little to no trend from one deck configuration to the next. This leads me to believe that there might be some trouble with the particle swarm optimization and that the returned designs were not necessarily the global optimal design, merely a close to optimum design. This observation was therefore looked further into and tried fixed by alternating some of the PSO-parameters, but the results still did not show the clear tendency that I was hoping to find. This meant I had to look to other alternative ways of finding and showcasing the correlation between design parameters, strength, and steel weight.

6.1.3 Comments to the PSO derived parametric equations potential

Whether or not this method of estimating the steel weight at a preliminary stage will be better than other existing methods for estimating steel weight for cruise ships is hard to determine at this stage. The benefit of using a parametric equation that is based on FEM analyses is that the steel weight is minimized for each of the structures used to derive the parametric equation. This means that when a cruise vessel is to be built, the steel weight is estimated as a minimum, opposed to being based on other vessels that might have been designed using built-in margins, causing an unnecessary increase in the steel weight.

The results from this model showed that the use of FEM and PSO were able to derive close

to minimum steel weight for a deck structure, and that the results could be used to express a parametric relation between steel weight and deck area. The problems as explained in the earlier sections 6.1.2, indicates that for this to be a valid approach, further testing and perfecting is required. Another discovery made while testing the FEM and PSO approach was that the computation/simulation process was much more time consuming than initially thought. Hence further work if persecuted will require optimizing the method in terms of computational time. Presumably by altering the PSO code and if no sufficient improvement, look to other heuristics. Changing the mesh type from a same size mesh to an iso-mesh is another option. The time aspect of the analysis is especially important if the deck structure were to be switched out with a full scale midship hull section.

Chapter 7

Topology optimization

Finding new designs for hull structures was of course hard, knowing that cruise vessels have been around and designed for a long time. The problem was therefore to construct new and better designs and evaluate/compare them to old/conventional designs.

7.1 Topology approach

The first strategy (PSO approach) used to find a lighter deck design, was based on minimizing the steel weight for a conventional deck design. This topology approach, on the other hand, set out to find a completely new design. Not bound by any restrictions related to classic ship design. The chosen strategy for trying to find new and potentially better designs were by introducing topology optimization to a midship section. Topology optimization is a mathematical method that optimizes material layout based on a tradeoff between strength and material usage. The method is most commonly used in designing smaller parts and has helped produce breakthrough concepts related to airplanes i.e. (Uwe Schramm, 2006)

The technology is fully based on FEA to provide information about strength and stress distribution. FEA as known needs a structure, loads and restrictions to produce results. Topology optimization then uses the results produced from the FEA, and try to take away

as many finite elements as possible, while damaging the structural strength the least. This is an iterative process towards the lowest steel weight possible while maintaining the structural strength.

7.1.1 midship section

When exploring topology optimization for a design, one strives to not exclude any solutions. This means that at the start, very few limitations/boundary constraints were added to the section.

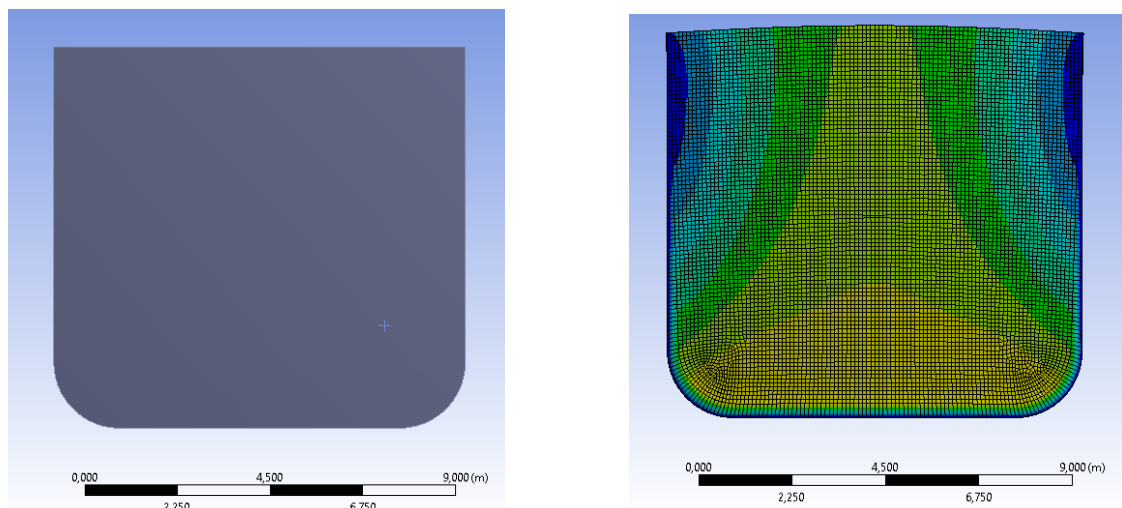


Figure 7.1: Midship section to be optimized **Figure 7.2:** Deformation of midship section

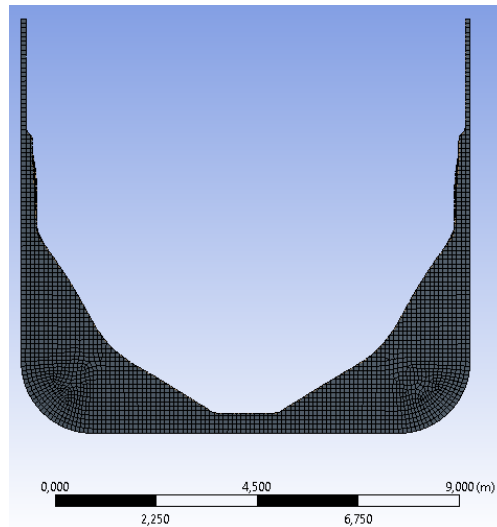


Figure 7.3: Initial topology optimization

The design and structure were developed using SpaceClaim, a CAD tool found in ANSYS workbench. The design was fairly basic and simple, but matching a typical midship section. The model and structure were given as one solid midship section block. No inner space was predefined, the only restriction was that the outer hull surface had to be kept intact (watertight). Hydrostatic pressure was added and then the topology optimization was set to develop a new and improved form.

The size was one part that quickly turned out to be a difficult aspect. The first model that was made was quite lengthy, similar to a full midship section length. This turned out to be difficult, when applying the topology optimization. The large structure meant that a large mesh and high computational time was needed in order to produce results. Another effect of using a full length midship section was that the structure reacting to the applied pressures/forces, reacted differently throughout the length of the structure, making it hard for the topology optimization to remove mass in an efficient way. As a result of this, the decision was to use more of a thick frame.

As can be seen from figure 7.3, the design is not a very efficient one. The topology optimization removed a significant part of the section, lowering the steel weight, but the design is not specially weight-efficient, nor is it a good design based on the hull utilization.

More constraints needed to be added for the topology optimization to produce more realistic and better design results.

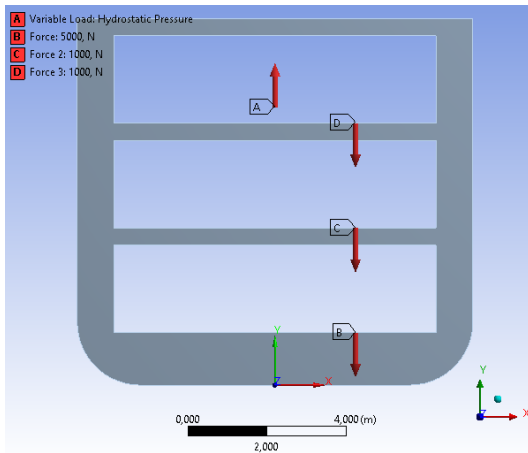


Figure 7.4: Midship section and loads

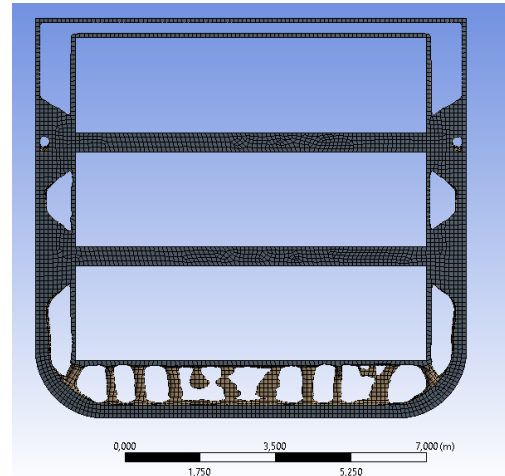


Figure 7.5: Topology optimized structure

The new and further refined frame structure to be topology optimized is illustrated in figure 7.4. It has the same hydrostatic pressure, acting on the outer hull, represented by the A tag in the figure. In addition to the hydrostatic pressure, three new decks were introduced as well as three new distributed loads, represented by B, C, and D in figure 7.4.

The model was restrained from alternation for all predetermined spaces as well as the outer hull surface. The rest of the structure was exposed to the topology optimization, and a new design was obtained. The design was closer to a conventional midship frame with respect to weight, compared to earlier, but yet again this method proved unable to develop a better design.

7.1.2 Design and process evaluation

Using topology optimization as a tool for discovering new and better designs did not provide any better new designs compared to conventional design. This was in many ways a long shot and the results were not that unexpected.

The positive aspect of this process was that a lot of different designs can be produced and tested fairly easy. The designs that were developed, were products of using the topology optimization tool found in ANSYS workbench. A tradeoff between strength and steel weight was then calculated by ANSYS, before returning a best possible design. The quality of the result is based on the model being produced correctly, and also highly dependent on the mesh refinement.

There were many troublesome aspects about using the topology optimization for hull design. For the topology optimization to explore as many options as possible, and not limiting the design space, the model should not be restrained too tightly. This posed problems when the design was to utilize the volume of the hull in a best possible way. The designs produced often tended to be "messy" and hard/impossible to realize and develop. Having stiffeners or girders going through/interrupting hull volume, where passengers or machinery is thought, will not give a good design.

Another problem and arguably the biggest problem in using topology optimization for hull design was all the different loading conditions and different wave loads that a vessel can encounter. The designs produced by topology optimization are specifically designed for a given loading condition, minimizing material in all locations deemed uncritical for that case. This means that a design although safe (sufficient strength) for one sea state, may be hazardous given a slight change in sea state.

The verdict regarding the use of topology optimization in hull design was that the usage might not be the best for a full midship section analysis. The approach is far better suited for optimizing more specific parts of the structure, such as single stiffeners. What could be taken away from these experiments and analyses, although it was not much or any form of new discovery, was that the hull structure was most exposed in the lower region.

Chapter 8

Cruise vessels

8.1 Volume vs Weight sensitivity

There are two categories of vessels in terms of cargo density and weight. One group being the weight sensitive group (tankers etc.) and the other being the volume sensitive group. The latter group of which just mentioned is also the group where cruise vessels are found. Volume being the critical factor, means that the cargo to be carried is light in comparison to the volume/space provided. Another problem to keep in mind when designing a cruise vessel is the center of gravity (COG). The COG is often located very high for cruise vessels compared with other vessel types (tankers, bulk carriers, etc.). Having a high COG means that the vessels stability might be a problem and should be kept in mind throughout the design phase. This COG is also the factor that determines the number of decks for the superstructure.

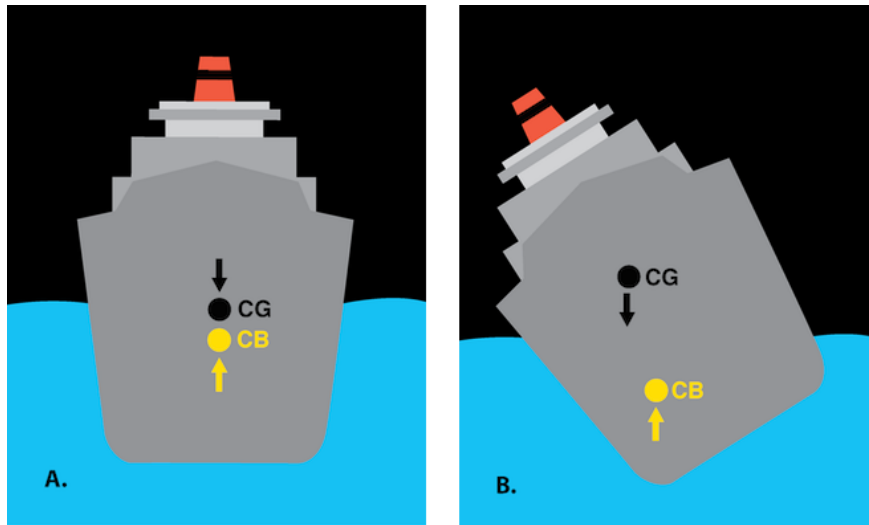


Figure 8.1: COG illustration

David G.M. Watson developed a way of using knowledge of the total volume of the vessel to be built, to relate the different main parameters. The method separated between weight critical and volume critical vessels. For a volume critical vessel such as a cruise vessel. The dimensions could be found by using the following equations from (Watson, 2002).

$$V_h = C_{bd} \cdot L \cdot B \cdot D_c \quad (8.1)$$

Using one of the following V_h equations.

$$V_h = V_m + V_o \quad (8.2)$$

or

$$V_h = \frac{(V_r - V_u)}{K_c} \quad (8.3)$$

D_c = Capacity depth in meters = $D + C_m + S_m$

D = Depth moulded in meters

C_m = Mean chamber in meters = $\frac{2}{3C}$ for parabolic chamber

C_{bd} = Block coefficient at moulded depth

V_h = Total moulded volume of the vessel below the upper deck, between the perpendiculars in m^3

V_r = Total required cargo capacity in m^3

V_u = Cargo capacity above the upper deck in m^3

S = Deduction for structure within the cargo space expressed as a proportion of the moulded volume

V_m = Moulded volume equivalent to required cargo capacity below upper deck = $\frac{(V_r - V_u)}{(1 - S)}$

K_c = Ratio of cargo capacity below the upper deck, to the total moulded volume = $\frac{(V_r - V_u)}{V_h}$

V_o = Other volume required for accommodation, stores, machinery, tanks and other non-usable space within the volume V_h in m^3

Equation 8.2 and 8.3 are adequate for different purposes, 8.3 being the one to use for cruise vessels. This is a quite troublesome process as it requires quite accurate values for the volumes included in V_o and V_h . The volumes needed can be calculated using a calculation sheet, adding all the volumes for the different groups together, before then using the values in the final formula 8.1 to calculate the total volume for the vessel.

8.1.1 Volume utilization

As a result of cruise vessels being volume sensitive, the goal is to obtain as much volume within the hull and superstructure as possible without compromising the seakeeping and operational ability beyond a significant penalty (Watson, 2002).

If the optimization of the dimensions for a cruise vessel is simplified, it can be seen as how to achieve the most volume within a box structure and use as little steel as possible in the process. In other words, reducing the surface area while maintaining the desired volume. The best utilization in terms of surface area compared to volume is found when all sides are the same length (again given that the vessel is simplified as a box) a quadratic cube. Figure 8.4 displays the consequence of going from quadratic box design to a more rectangular shape. The increase in surface area to obtain the same volume is increasing exponentially.

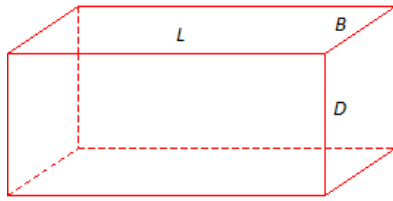


Figure 8.2: Box illustration

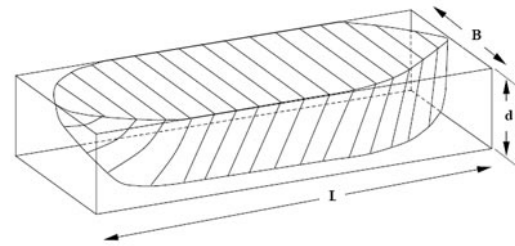


Figure 8.3: Moulded ship

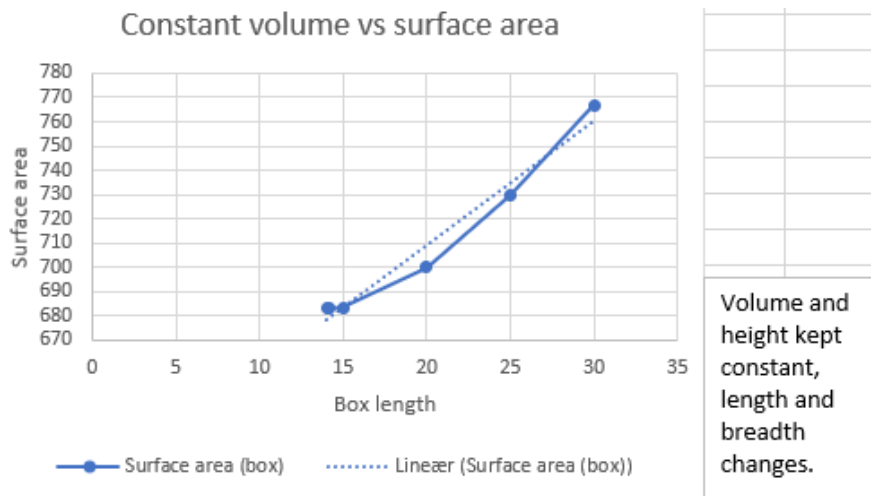


Figure 8.4: Surface area as a function of length and breadth. Height and volume kept constant

But this (quadratic box) is not a realistic design nor a good design for a cruise vessel. What we can take from this is that some dimensions should be tried maximized as much as possible, while not compromising the vessels operational abilities. For all vessels ¹, the length of the vessel is greater than the breadth, which again is greater than the dept and draft respectively ($L > B > D > T$). This means that there are in all three dimensions that can be enlarged and cause the volume to rise, while still keeping the surface area the same. The most influential parameter will be the draft as it is the smallest compared to the others. The next best parameter to address is the depth and breadth, respectively.

¹Might be possible to find rare exceptions where this does not apply

The alternation of the parameters should not be done in a manner that affects the vessels operational abilities. Hence the draft (T) should be taken as the maximum draft permitted by the length (L), breadth (B) and depth (D). Depth (D) should be the maximum permitted by length (L) and breadth (B). Breadth (B) as permitted by length (L).

8.1.2 Dimensional relationships

There are relationships between all global dimensions, which values speak to the quality and operability of the vessel. (Dr. Watson discussed some methods in his publication Watson (2002), from which I have drawn much inspiration.) The four main dimensions are as mentioned length, breadth, depth and draft. These dimensions are again coupled through six-dimensional relationships.

$$B = f(L)$$

$$D = f(B)$$

$$T = f(D)$$

$$D = f(L)$$

$$T = f(L)$$

$$T = f(B)$$

When designing a new vessel, these relationships can be used to establish whether or not the design is matching the required and desired qualities. What qualities and type of vessel it is will affect the different parameter relationships. Optimum for one vessel might not be optimum for the next, given different desired qualities.

8.1.2.1 Breadth/length (B/L)

There have been various opinions throughout the years regarding the L/B ratio. One of the first relationship formulas, addressing the length to breadth ratio, came in 1962.

$$B = \frac{L}{M} + K \quad (8.4)$$

Where M and K are varying depending on the vessel type. This again resulted in ratios varying from 6,2 for the smaller vessels, and up to 7,6 for the largest vessels. Later these values changed somewhat and new relationship ratios were established. It was also discovered that the vessel type did not affect the L/B ratio as much as earlier proposed. Trying to avoid low L/B ratios in itself was not a problem, but as a result of a low L/B ratio, the Froude number revealed that for a set displacement, a large breadth would demand an unbeneficial increase in the power required.

8.1.2.2 Breadth/depth (B/D)

Again back to the trouble with volume critical vessels, the stability. The B/D relation is especially important due to KM and KG being functions of the breadth and depth respectively. (Watson, 2002) states that back in 1962, a smaller vessel, with good stability, was to have a B/D ratio around 1,8 and if the vessel was larger, a ratio of 1,5 would denote that a moderate stability was obtained. Later these limits got redefined and more conservative and one required higher B/D ratios in order to be deemed safe. For weight sensitive vessels, stability is often not a problem. As a result, breadth and depth can be seen as individual variables and do not have to be evaluated against each other in the same way as volume carriers depend on the B/D relationship. This ratio will be further discussed in a later section 8.2.

8.1.2.3 Length/depth (L/D)

The length/depth ratio is a relationship that is of greater importance for weight carriers as the structural strength of the hull can be a problem when evaluating the vessel for different wave and load conditions. The L/D ratio has changed through time, specifically for tankers as new rules and standards are introduced. Other ship classes have seen less change over time.

8.2 Cruise vessels database

In order to be able to evaluate and determine the limits regarding cruise vessel design parameters, a database was developed to serve as a verification tool and guideline. Information about 447 cruise vessels have been gathered (Chanev, 2018) and processed to show trends and relationships regarding length, breadth, and draft. The list and information were checked against ship-info's ship base (Ship-Base, 2019) to validate the vessels dimensions.

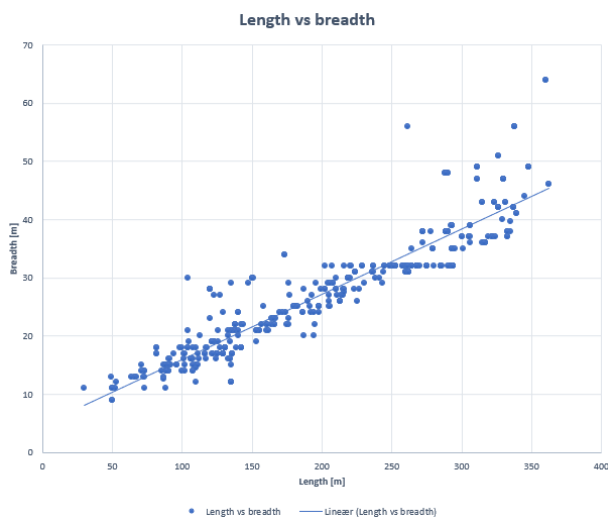


Figure 8.5: Length vs breadth

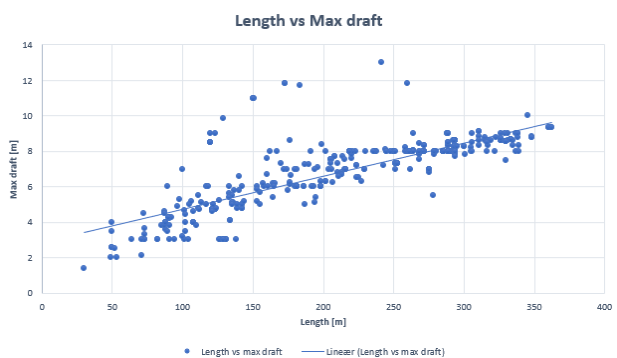


Figure 8.6: Length vs draft

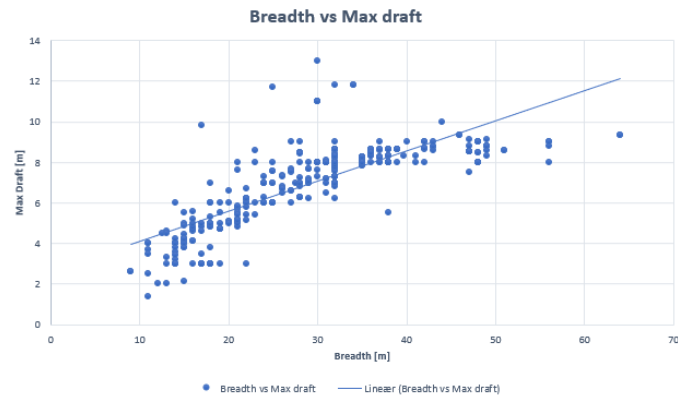


Figure 8.7: Breadth vs draft

From the graphs above, the relationship between length, breadth, and draft are displayed and one can see that there is a clear trend in all the plots, but there are also some irregularities that differ from the trend line. The vessels that differ the most from the trend lines are mostly due to the vessel being of a different type or operating area than the common cruise vessels.

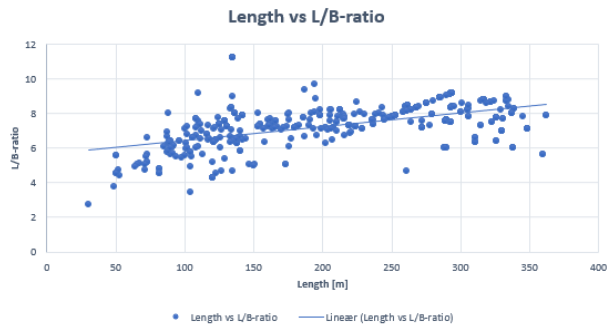


Figure 8.8: Length vs L/B-ratio

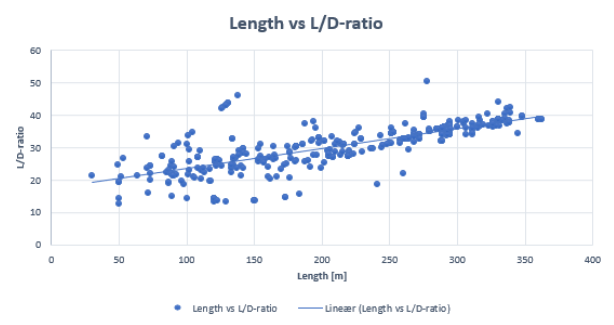


Figure 8.9: Length vs L/D-ratio

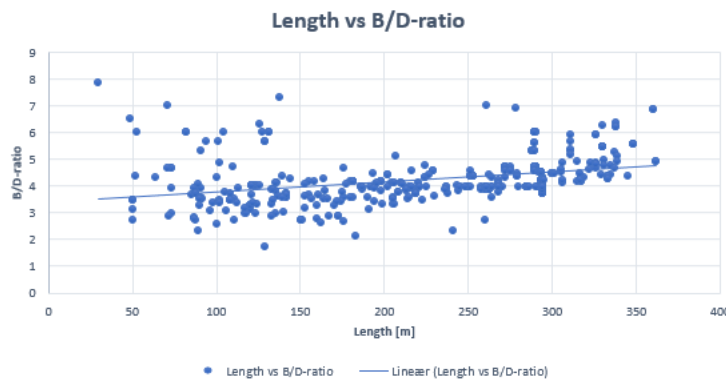


Figure 8.10: Length vs B/D-ratio

To display the tendency in different ratios, they have been plotted against the vessels length. The trend for all the different ratio plots tells us that there is a slight increase for both the L/B- and B/D-ratio as the vessel length increases. Whereas the L/D-ratio as a function of vessel length is much more prominent and increases quite a lot. When comparing the database ratios with the ones stated in the theory by Watson, we see some differences. The L/B ratio listed by Watson matches the average L/B ratio found from the ship database. The B/D ratio, on the other hand, was much larger for the cruise vessels included in the database, than the ones listed by Watson 8.1.2.2.

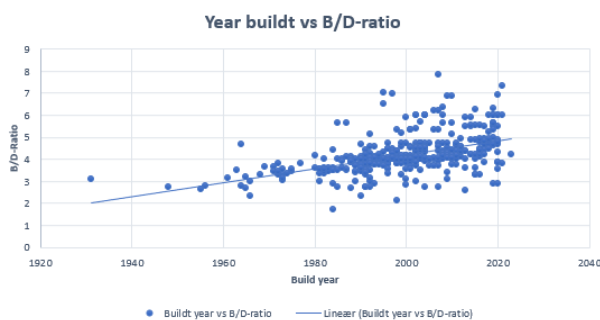


Figure 8.11: Year built vs B/D-ratio

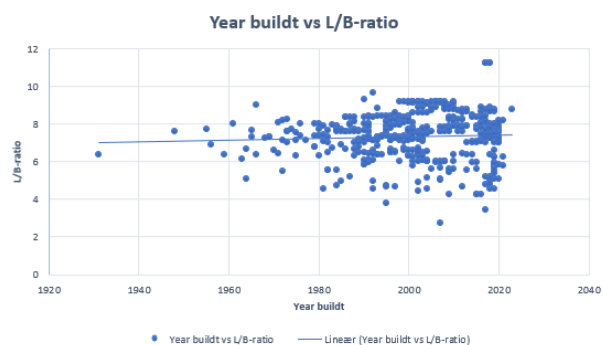


Figure 8.12: Year built vs L/B-ratio

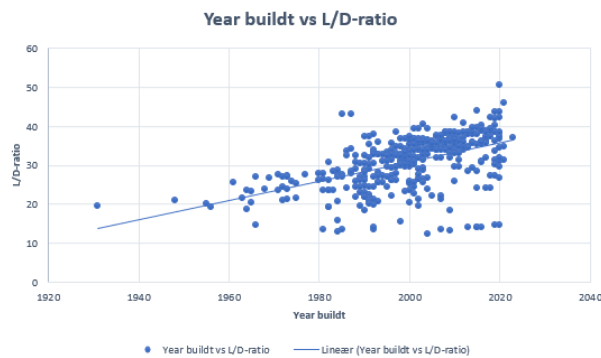


Figure 8.13: Year built vs L/D-ratio

The last set of graphs are displaying the relation between year built and the different ratios. This was done to evaluate how the trend changes throughout the history of built cruise vessels. The plots all show some form of inconsistency from the trend line, but the main mass of the vessels follow the trend pretty good. What is interesting to see is that both the B/D- and L/D-ratios increase for vessels being built in the later years compared to the older ones. The change in L/B-ratio, on the other hand, has seen close to zero change in the trend line, but the spread in L/B-ratio is quite large and increasing for newer vessels.

8.3 Effect of the global parameters

The parameters with the greatest impact, when calculating the steel weight for a vessel is the length (L), width (B), depth (D) and the block coefficient (C_B). If one of these parameters are altered, then that will affect structural parameters throughout the whole vessel.

If the length (L_{PP}) is increased, then that will affect almost all components within a ship. Hence the length of the vessel is the parameter with the highest impact in terms of steel weight. A longer ship will require more strength (steel) to withstand the increase in torque, which again will require an increase of parameters such as bottom plating, girders, etc. A longer vessel may also lead to more bulkheads being needed. All affecting the steel weight substantially.

The width (B) is the parameter with the second highest impact in terms of steel weight. The reason is quite the same as for the length, only now primarily in the transverse direction. Increasing the width of the vessel will however also contribute to a somewhat larger torque in the longitudinal direction which will have to be accounted for.

Depth and block coefficient are respectively the third and fourth parameters in terms of steel weight impact. The depth is linked to secondary parameters such as the number of decks, side plating, and ribs while the block coefficient affects the whole hull.

To understand how altering different global parameters affect the steel weight is of great importance when looking to develop a parametric equation for estimating the steel weight of cruise vessels. The impact of changing a global parameter will have to be quantifiable, meaning that if the length (L_{PP}) is scaled up, then that alternation will have to be directly transferable to some number of added steel weight. In order to make the changes of the global parameters quantifiable, all structural parameters affected will have to be accounted for. The relation between global parameters and added structural mass is key for the steel weight estimate. This relation is what the FEM models, coming later will try to quantify.

Chapter 9

Classification society

There are two ways of getting a classification society to approve a design. Either the design is based upon their requirements and follows their design specifications, or there will have to be sufficient proof that the design will "hold" and be within certain safety criteria. The classification society's requirements are set to be valid for a variety of ships, meaning it is not tailored to a specific vessel, which again means that the final design tends to be over-dimensioned due to built-in margins.

Some classes of constraints worth mentioning that are important when designing the structure for a new vessel are:

- Technological constraints in terms of manufacturing limitations
- Functionality constraints preventing discontinuity for design variables. Same bottom plate thickness throughout the length of the vessel i.e.
- Structural constraints, stress, strain i.e
- Functionality constraints, placement of cabins in regards to the machine room i.e.
- Global constraints.

Classification society's such as DNV GL, Bauer Veritas, etc, have all the requirements for all

new vessels being built. The structural requirements vary depending on the vessel type and can be very specific. These requirements concerns a broad scope of details, but the main aspect for most of the rules can be generalized to making sure stress, fatigue, and shear forces are within given limits. And this will have to hold for a set of loading conditions and sea states. The steel weight for a given ship will then be a result of having to satisfy all the constraints.

9.1 Rules by NDV GL

9.1.1 Still water forces

Formulas and notation for this section and subsections are in compliance with (DNVGL, 2015).

DNV GL specifies that the global design still water bending moments amidship, sagging and hogging should be computed as follows when used in stress and buckling tests.

$$\begin{aligned}
 M_S &= k_{sm} M_{SO} \quad [kNm] \\
 M_{SO} &= -0.065 C_{WU} L^2 B (C_B + 0.7) \quad (\text{sagging}) \\
 &= C_{WU} L^2 B (0.1225 - 0.015 C_B) \quad (\text{hogging}) \\
 C_{WU} &= C_W \quad \text{For unrestricted service}
 \end{aligned}
 \tag{9.1}$$

k_{sm} is dependent on what point along the ship you are interested in and can be found using this graph.

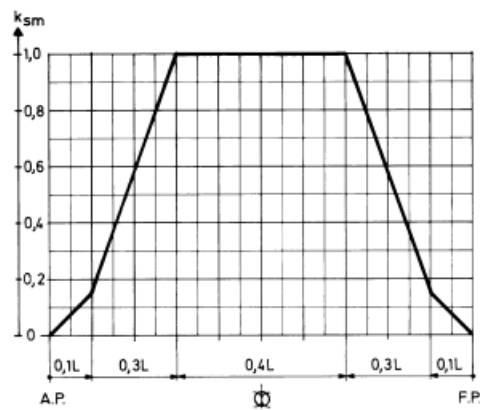


Figure 9.1: k_{sm} values along the length of the ship

Shear forces along the length of the vessel for still water are based on the bending moment for still water.

$$\begin{aligned} Q_S &= k_{sq} Q_{SO} \quad [kN] \\ Q_{SO} &= 5 \frac{M_{SO}}{L} \quad [kN] \end{aligned} \quad (9.2)$$

$k_{sq} = 0$ at A.P and F.P
 $= 1.0$ from $0.15L$ to $0.3L$ from A.P
 $= 0.8$ from $0.4L$ to $0.6L$ from A.P
 $= 1.0$ from $0.7L$ to $0.85L$ from A.P
 k_{sq} varies linearly between given k_{sq} values

The wave bending moments amidship are calculated for both sagging and hogging as for still water.

$$\begin{aligned}
M_W &= k_{wm} M_{WO} \quad [kNm] \\
M_{WO} &= -0.11\alpha C_W L^2 B (C_B + 0.7) \quad (\text{sagging}) \\
&= 0.19\alpha C_W L^2 B C_B \quad (\text{hogging}) \\
\alpha &= 1.0 \text{ for seagoing conditions} \\
&= 0.5 \text{ for harbour etc.} \\
&\text{and } C_B \text{ is not to be less than } 0.6 \\
k_{wm} &= 1.0 \text{ from } 0.4L \text{ to } 0.65L \text{ from A.P} \\
&= 0 \text{ at A.P and F.P}
\end{aligned} \tag{9.3}$$

Shear forces caused by waves can be calculated as:

$$\begin{aligned}
Q_{WP} &= 0.3\beta k_{wqp} C_W L B (C_B + 0.7) \quad [kN] \\
\beta &= 1.0 \text{ for seagoing conditions} \\
&= 0.5 \text{ for harbour etc.} \\
k_{wqp} &= 0 \text{ at A.P. and F.P.} \\
&= 1.59 \frac{CB}{CB + 0.7} \quad \text{from } 0.2L \text{ to } 0.3L \text{ from A.P.} \\
&= 0.7 \text{ from } 0.4L \text{ to } 0.6L \text{ from A.P} \\
&= 1.0 \text{ from } 0.7L \text{ to } 0.85L \text{ from A.P}
\end{aligned} \tag{9.4}$$

The stresses that are found must then be checked to determine whether or not they are within the allowable limits. In the test models that will be presented at a later stage in this report, there will not be included forces found by the DNV GL formulas. Hydrostatic pressure will be added in compliance with a still water test. The resulting response will still be checked against DNV GL's limits listed below.

Table B1 Allowable nominal stresses

Structure	Seagoing or harbour condition	Type of stress				Normal stress σ (N/mm ²)	Shear stress τ (N/mm ²)		Equivalent stress σ_e (N/mm ²)
		Hull girder stresses	Transverse bottom, side or deck girder stresses	Longitudinal bottom, side or deck girder stresses	Local stiffener bending stresses		One plate flange	Two plate flanges	
Longitudinal girders	Seagoing	X ¹⁾		X		190 f_1	90 f_1	100 f_1	
	Harbour	X ¹⁾		X		190 f_1	100 f_1	110 f_1	
Transverse and vertical girders	Seagoing		X			160 f_1	90 f_1	100 f_1	180 f_1
	Harbour		X			180 f_1	100 f_1	110 f_1	200 f_1
Girder brackets	Seagoing		(X)	(X)		200 f_1 ²⁾			
	Harbour		(X)	(X)		220 f_1 ²⁾			
Longitudinal stiffeners	Seagoing and harbour				X	160 f_1			
	Seagoing and harbour			X	X	180 f_1	90 f_1		
	Seagoing and harbour	X ¹⁾		X	X	245 f_1			
Transverse and vertical stiffeners	Seagoing and harbour		(X)	(X)	X	180 f_1			
Stiffener brackets	Seagoing and harbour		(X)	(X)	X	225 f_1			

X Stress component to be included
(X) Stress component to be included when relevant

1) Includes the hull girder stresses at a probability of exceedance of 10^{-4} , see 406.
2) Shows allowable stress in the middle of the bracket's free edge. For brackets of unproven design, additional stress analysis in way of stress concentration areas may be required. Reference is made to acceptance criteria for local structure analysis, F300.

Figure 9.2: Acceptance criteria given by DNV GL

f_1 is the material factor, and will change depending on what material and grade of steel that is used.

$f_1 = 1,0$ for normal structural steel with yield strength no less than $235 N/mm^2$. f_1 may go as high as 1,47 if the steel is high strength structural steel with yield strength no less than $390 N/mm^2$

DNV also have a set of minimum criteria regarding different deck parameters such as plate-, stiffener-, girder- and flange -thickness. Hence all deck structures should be designed using only parameters equal to or larger than the minimum values. The only way of being allowed to use smaller dimensions is if sufficient proof of strength can be submitted (DNVGL, 2016).

If the finite element analysis cant provide results that satisfy the given criteria, then the design must be reevaluated and improved until the results match the requirements. The vessel will have to meet the criteria in order for the vessel to be insured.

9.1.2 Minimum criteria by DNV GL

The different thickness criteria given by DNV GL is investigated and listed quite thoroughly as the effect of these parameters directly contribute to the steel weight of the structure and is important when constructing CAD models.

9.1.2.1 Steel deck thickness

Plate thickness for steel decks is not to be less than the lowest value found from:

$$t = t_0 + \frac{k \cdot L_1}{\sqrt{f_1}} + t_k \quad [mm] \quad (9.5)$$

or

$$t = \frac{15,8 \cdot k_a \cdot s \cdot \sqrt{p}}{\sqrt{\sigma}} + t_k \quad [mm] \quad (9.6)$$

$t_0 = 5\text{mm}$, minimum thickness for accommodation deck.

p = dependent on relevant design load, 11 different cases.

σ = Stress limit, $175 f_1$ for transverse stiffened and $120 f_1$ for longitudinal stiffened.

$k = 0,02$ for vessels with one continuous deck

$= 0,01$ for vessels with two continuous decks above $0.7 D$ from the baseline

$= 0,01$ as minimum for weather decks forward of $0.2 L$ from F.P.

$= 0$ for vessels with more than two continuous decks above $0.7 D$ from the baseline.

k_a = correction factor for aspect ratio of plate field

$$= (1.1 - 0.25 s/l)^2$$

$=$ maximum 1.0 for $s/l = 0,4$

$=$ minimum 0.72 for $s/l = 1,0$

s = stiffener spacing in m.

t_k = corrosion addition value = 0,5-3,0 mm depending on exposure.

Equation 9.5 and 9.6 are two ways of determining the minimum plate thickness. Equation 9.5 returns a constant thickness (for a given region) that does not depend on any other strengthening elements. Equation 9.6 on the other hand is dependent on the stiffener/girder layout. The selection of plate thickness can therefore be illustrated as a 3D-plot 9.3. The effect of decreasing or increasing stiffener spacing and/or span and its contribution to plate thickness is illustrated.

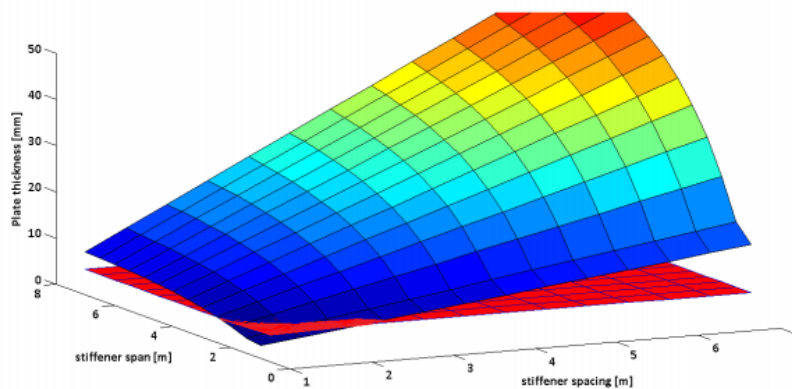


Figure 9.3: Deck plating thickness (Lausund, 2013)

A plot figure similar to the type displayed in figure 9.3 can be obtained for all coming structure sections, the only difference is the change in p , σ and t_k , that depend on the location of the plate and how exposed it is. A change in one or more of the mentioned parameters will only lead to a translation of the planes in the plot, the slope will remain the same.

9.1.2.2 Keel, bottom and bilge plate thickness

The keel plating and garboard strake go through the whole length of the vessel, reinforcing the bottom plating. The width of the keel plate will minimum have to be:

$$b = 800 + 5 \cdot L \quad [mm] \quad (9.7)$$

The thickness requirements for this keel plating or garboard strake is set to be minimum:

$$t = 7,0 + 0,05 \cdot L + t_k \quad [mm] \quad (9.8)$$

The keel section is never to be thinner than the bottom plating next to the keel.

The thickness for the rest of the bottom plating and bilge can be calculated corresponding to the lateral pressure.

$$t = \frac{15,8 \cdot k_a \cdot s \cdot \sqrt{p}}{\sqrt{\sigma}} + t_k \quad [mm] \quad (9.9)$$

p = the pressure as found from relevant pressure categories from p_1 or p_2 found in appendix B.2

σ = as described in appendix B.3

The minimum allowed thickness is:

$$t = 5,0 + 0,04 \cdot L + t_k \quad [mm] \quad (9.10)$$

The bilge plating should never be thinner than neither the wall plating or the bottom plating. If the bilge section is constructed without stiffeners or only one stiffener is added to the curved section, then the thickness of this section is to minimum be:

$$t = \frac{\sqrt[3]{R^2 \cdot l \cdot p}}{900} + t_k \quad [mm] \quad (9.11)$$

R = Radius of the bilge section [mm].

l = Length separating stiffeners closes to the bilge section [mm].

p = $2p_1 - 10h_0$ [kN/m²].

9.1.2.3 Inner bottom plating

The thickness for the inner bottom plating as given by lateral pressure:

$$t = \frac{15,8 \cdot k_a \cdot \sqrt{p}}{\sqrt{\sigma}} + t_k \quad [mm] \quad (9.12)$$

$p = p_3 - p_9$ as given in appendix B.2

$\sigma = 140 f_1$ for areas 0,4L perpendiculars.

Minimum thickness for the inner bottom plating:

$$t = t_0 + 0,03 \cdot L + t_k \quad [mm] \quad (9.13)$$

$t_0 = 5,0$ for holds covered by ceilings.

9.1.2.4 Floors and longitudinal girders

The thickness requirement for this section is the same as for inner bottom plating, but the pressure categories to be included are different, as well as the allowable stress (σ). Hence the calculation formula is the same:

$$t = \frac{15,8 \cdot k_a \cdot \sqrt{p}}{\sqrt{\sigma}} + t_k \quad [mm] \quad (9.14)$$

$p = p_9 - p_1$ as calculated and found in appendix B.2

$\sigma = 160$ for floors, 130 for girders.

The minimum requirements in terms of thickness for floors, longitudinal girders and other supporting plates, should be:

$$t = 6 + k \cdot L + t_k \quad [mm] \quad (9.15)$$

$k = 0,04$ for center girder

$= 0,02$ for other girders.

9.1.2.5 Web and flanges

The thickness requirements for the web and flanges must be minimum the largest of:

$$t = 4,5 + k + t_k \quad [mm] \quad (9.16)$$

or

$$t = 1,5 + \frac{h_w \cdot \sqrt{f_1}}{g} + t_k \quad [mm] \quad (9.17)$$

$k = 0,014L$

h_w = web height in mm

$g = 75$ for flanged profile webs

$= 41$ for bulb profiles

$= 22$ for flat bar profiles

The goal of the thesis was to develop a decision support for hull design with respect to steel weight. Earlier in this thesis, theory and experiments have been presented that enlightens different aspects of the design process and what to keep in mind when developing a hull design. The goal now was to use this information and develop a new test model that implements the findings made earlier and develop a model to evaluate the changes in steel weight as the design/parameters change.

9.2 Hull development

In order to test the effect of different designs and the effect of parameter changes, a base model was developed. The discussion was to use a midship hull section for this experiment.

By using a midship section the specific front and aft design could be excluded from the process. Different midship models were constructed and tested before landing on one model. The model had to serve as a basis and was the starting point for all designs tested. The model that was developed was a simplified hull of a midship section. The model was kept simple for a set of reasons. A simple model made it easier to modify and adjust parameters. Having a complex model would not serve the desired purpose of representing an arbitrary cruise vessel. Hence a stripped down midship hull section was thought to be the best choice. The midship section was set to be 10% of the ship length. Breadth and depth were obviously not set to be a certain scale of the main parameters as they are fully represented in a midship section.

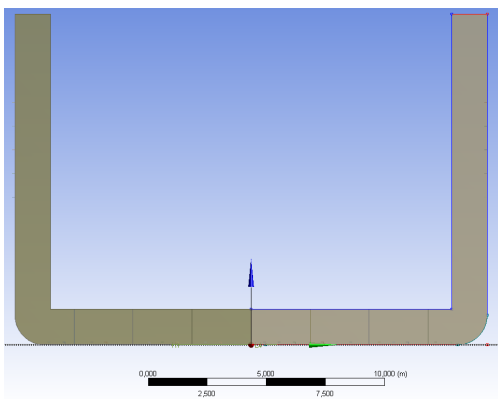


Figure 9.4: Midship frame

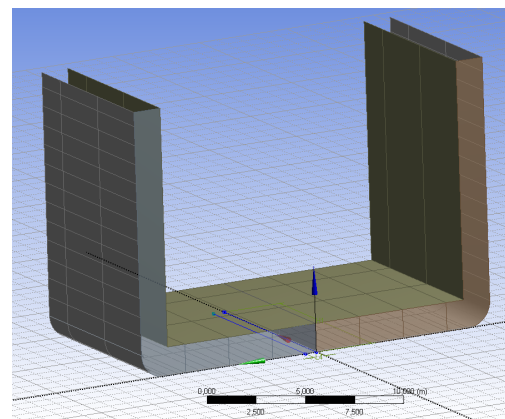


Figure 9.5: Midship basis

The midship frame 9.4 was the basis and the frame to serve as a starting point for different alternations off the midship section. At this stage, no stiffeners, girders or other supporting frames were added to provide additional support. The structure was merely just a shell.

9.2.1 Strengthening structure

Adding strength to the basic structure was crucial for the hull section to withstand the loads/forces that were to be introduced. Strengthening the structure had to be done in an

efficient way, not favouring one specific parameter change over others, in terms of weight and strength. This meant that changing/altering one parameter of the section (L, B, D) should not be a better option solely based on the strengthening design/solution for the base model. The knowledge obtained from the PSO approach was also to be implemented.

9.2.2 Stiffener decision

When deciding what type of stiffeners to use, safety, response and quality were evaluated. Both longitudinal and transverse stiffeners for the hull floor and walls had to be decided. The desired qualities for stiffeners are that they provide sufficient strength while not adding unnecessary weight. Meaning that the amount of structural stiffness per unit added weight should be profitable. In order to make sure the right and best stiffener solution was chosen, a correlation test was performed for a set of stiffeners.

The parameter correlation test was constructed by using CAD to develop a set of stiffener profiles. The stiffeners were then given boundary conditions and loads were added in order to be able to evaluate their strengths and weaknesses.

First stiffener profile that was created and tested was the T-profile, constructed with fixed supports for both ends as seen from tag B in figure 9.6. The load was added to the bottom part of the stiffener as a distributed load of 5000N seen on tag A. This loading situation simulates the interaction between the vessels shell and stiffeners.

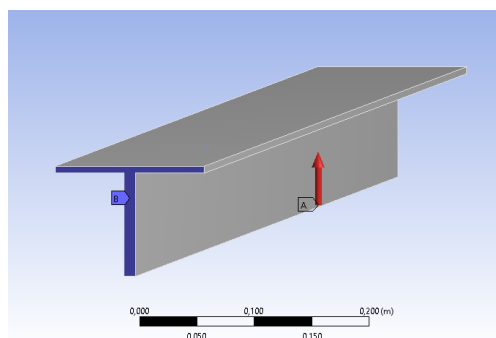


Figure 9.6: T-Stiffener

The next stiffener profiles that were tested were the L and flat profile as they are also commonly used in hull designs. The same loading and fixed supports were added for all the stiffeners, and can be seen in figure 9.7 and 9.8.

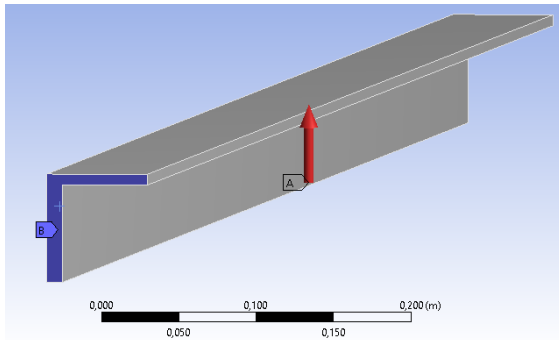


Figure 9.7: L-Stiffener

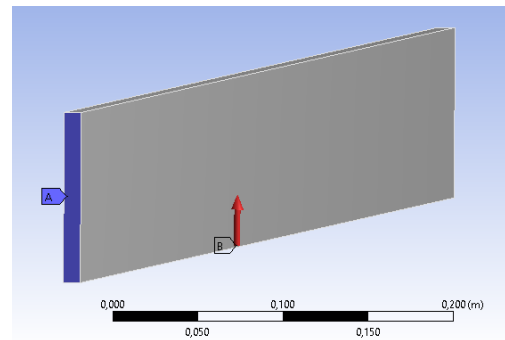


Figure 9.8: Flat-Stiffener

The stiffeners that were tested were all 5 meters long and had the same thickness for flanges and for the stiffener in general.

9.2.3 Parameter correlation test

The set of stiffeners that were created, were evaluated through the use of a parameter correlation analysis. The correlation analysis used FEA results obtained for a set of design points to evaluate how the different parameters were connected and how altering parameters contributed to stress, deflection, and weight.

The first design with the user-defined stiffener cross section served as design point 1 for the parameter correlation analysis. A total of 30 design points were created for each of the different cross-sections. These 30 design points were all put through FEM analyses to develop the relations between input parameters and output parameters.

The output parameters developed/calculated through FEM was maximum stress, weight of the structure and maximum deflection.

The input parameters and output parameters were then arranged as a matrix to visualize the effects in a systematically way.

9.2.3.1 T-stiffener

Four input parameters ensured the T-stiffener was fully constrained.

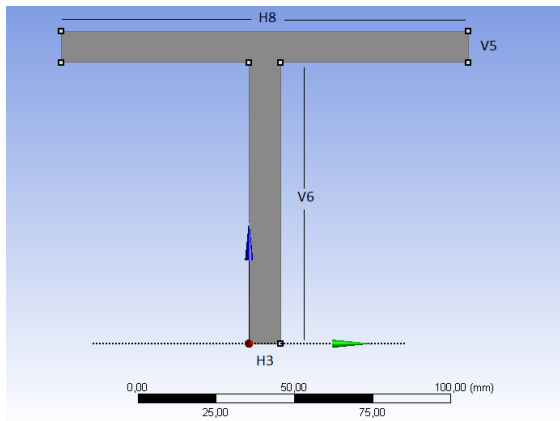


Figure 9.9: T-stiffener parameters

Name	Value	Type
✓ ThicknessBottomOfT.H3	10 mm	Length
✓ FlangewidthtotalT.H8	130 mm	Length
✓ FlangeThicknessT.V5	10 mm	Length
✓ HeightTbeam.V6	90 mm	Length

Figure 9.10: Parameters for T-stiffener

Each parameter and its respecting value can be seen from figure 9.9 and 9.10.

Running the analysis for all the 30 design points resulted in the following correlation matrix:

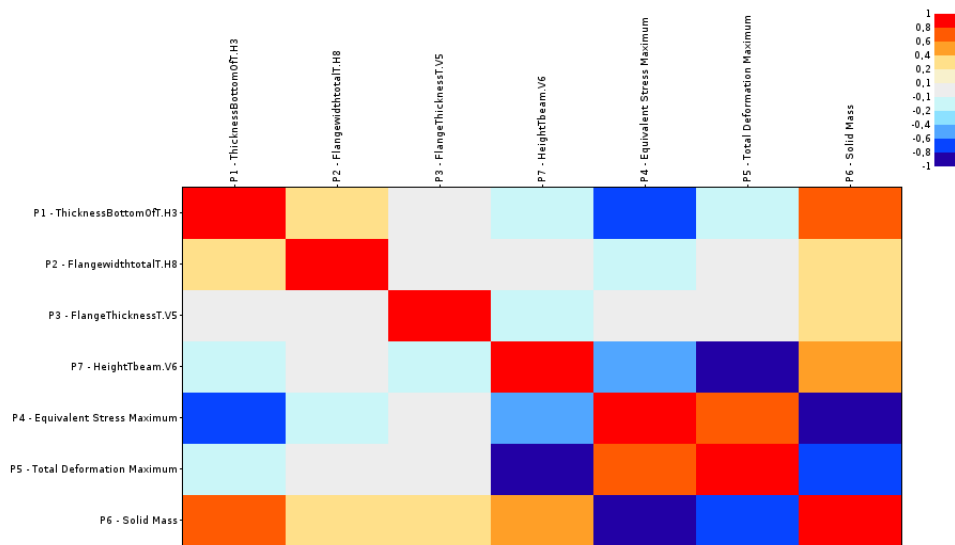


Figure 9.11: T-Stiffener correlation matrix

The color coding made the correlation matrix easier to interpret and evaluate. Each cross section between the parameters listed on the vertical side and the horizontal side had its own color grade. The diagonal of the matrix made up intersections between the same parameters interacting, and as a result, there was a 100% correlation. Dark red indicates that there was a strong correlation, while a dark blue color indicated that there was a clear negative correlation. The color grades in-between the two extremes indicated that there was a somewhat lesser correlation/negative correlation. These color nuances made it possible to evaluate the stiffener, and what parameters to exploit and which to neglect.

The correlation matrices were studied to find what parametric changes contributed to a stronger structure while adding as little steel weight as possible. This meant looking for input parameters that had a clear negative correlation with maximum equivalent stress and maximum total deformation. At the same time, the input parameter had to maintain an as low as possible correlation with the solid mass of the structure.

For the T-profiles correlation matrix (figure 9.11), there was a clear result regarding which parameter that had the most beneficial impact. The height of the stiffener proved to be the most influential parameter, having a direct negative correlation with maximum stress and

deflection. The weight added when increasing the height of the stiffener was correlated, but only at 0,3. The other parameters were somewhat harder to determine whether or not the improved strength was better than the gain in structural weight. Increasing the thickness of the T-stiffener contributed to lower stress, but the weight correlation made the trade off hard to determine. Flange thickness, on the other hand, indicates that the gain in weight was more substantial than the stress and deflection resistance gained.

9.2.3.2 I-stiffener

The flat-stiffener in figure 9.12 was a much simpler structure, and had no flanges.

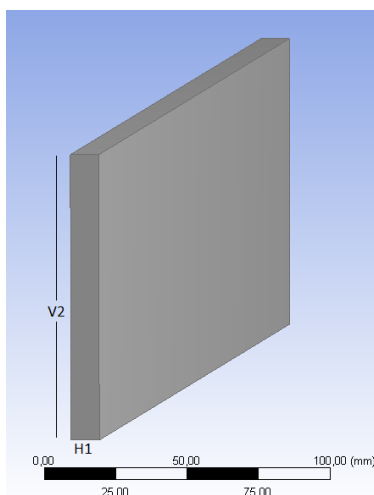


Figure 9.12: Flat-stiffener parameters

	Name	Value	Type
✓	Thicknessl.H1	10 mm	Length
✓	Heightl.V2	100 mm	Length

Figure 9.13: Parameters for flat-stiffener

This simple cross section meant not to many parameters to evaluate and the correlation matrix was also straight forward as seen in figure 9.14

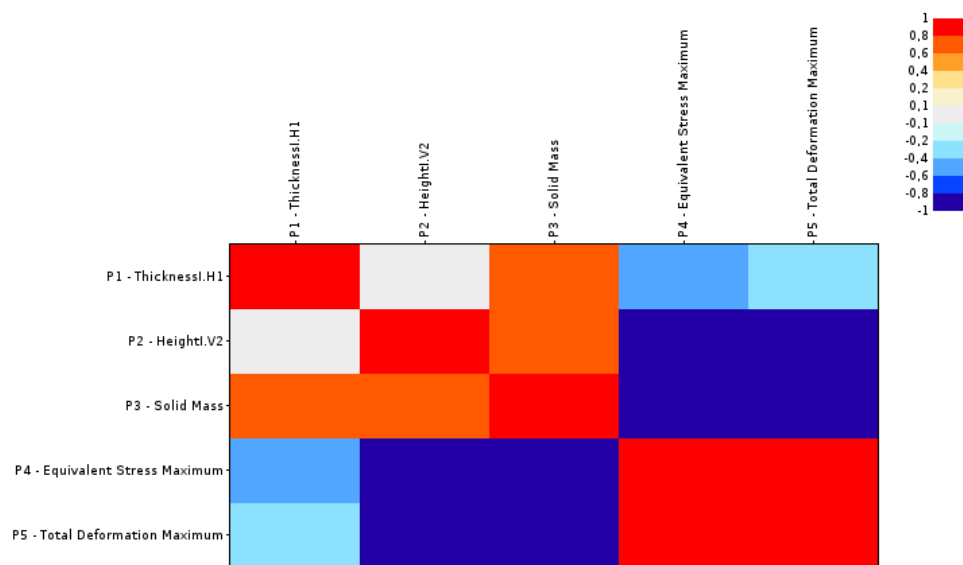
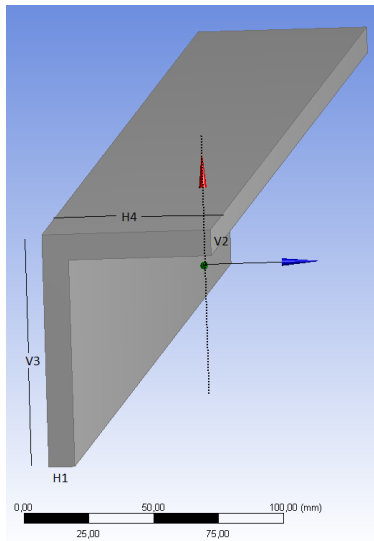


Figure 9.14: flat-Stiffener correlation matrix

The solid mass had a complete negative correlation with the maximum stress and deformation. Increasing the height of the cross section proved somewhat better compared to increasing the thickness, but the cost in terms of weight was naturally highly correlated for both input parameters.

9.2.3.3 L-stiffener

The L-stiffener had the same amount of input parameters as the T-stiffener.



Name	Value	Type
✓ ThicknessBottomOfL.H1	10 mm	Length
✓ FlangeWidthL.H4	65 mm	Length
✓ ThicknessFlangeL.V2	10 mm	Length
✓ HeightL.V3	90 mm	Length

Figure 9.16: Parameters for L-stiffener

Figure 9.15: L-stiffener parameters

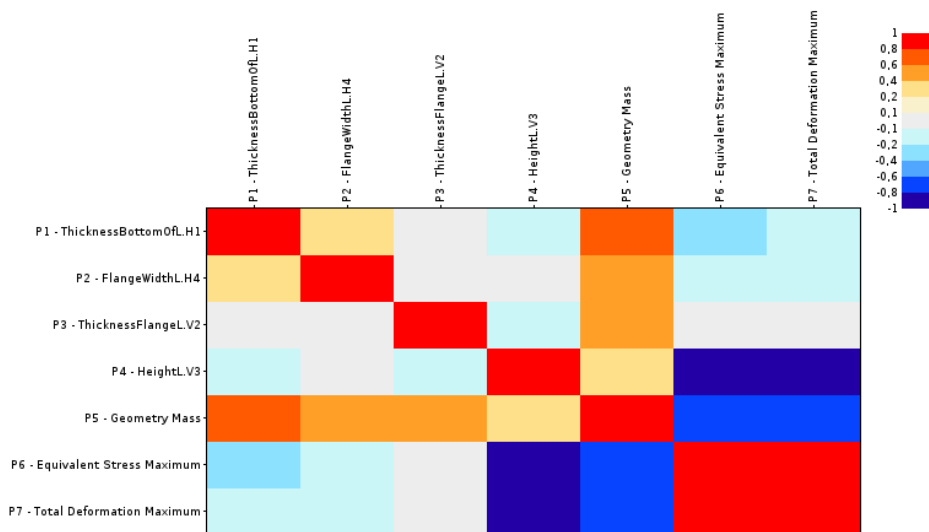


Figure 9.17: L-Stiffener correlation matrix

The correlation matrix (figure 9.17) for the L-stiffener revealed as for the T-stiffener that the best parameter to increase in terms of strength to weight trade-off was the height of the cross-section. The story was also the same for the other parameters as for the T-stiffener. No clear verdict in terms of how profitable the tradeoff between strength and weight was

for the other parameters.

9.2.4 Comparing stiffeners

The parameter correlation test was performed solely with respect to equivalent Von-Mises stress and deformation. The load situation was set to be for a one directional load. These decisions were made as they were thought to be the most critical factors and the best simulation of how the forces from the outer hull would be transferred to the stiffeners. This also meant that the cross sections with flanges did not get to show their full potential. Meaning that L and T stiffeners have a greater ability to absorb torsional stresses and withstand buckling than a straight stiffener.

Knowing what parameters to change in order to achieve a better strength to weight ratio for the stiffeners, meant new and improved dimensions could be introduced. Three stiffener models with new and improved dimensions were constructed, all exploiting their favorable parameter. The weight of the stiffeners was set to be the same for all stiffeners to evaluate which stiffener gave the best/most support compared to the weight of the stiffener.

Stiffener type:	L	I	T
Height [mm]	175	220	140
Thickness [mm]	10	10	10
Flange [mm]	55	0	90
Maximum stress [Mpa]	40,444	30,641	51,066
Maximum deflection [m]	0,0039446	0,00043685	0,0018618
Steel weight [kg]	86,35	86,35	86,35

Figure 9.18: Comparison of stiffeners

The test revealed that the flat-cross section returned the lowest stress and deflection maximums. The next best cross-section in this test was the L-profile, while the T-profile scored the worst.

Given the unexploited areas for the stiffeners with flanges, the decision was to go for a

combination of L and flat stiffeners. Both profiles tested well, and using the L stiffeners would make the structure more reliable for unexpected loading conditions and better at preventing buckling in the stiffeners. Using L stiffeners for the walls 9.19 also prevent water or other liquids from gathering and possibly affect the stability of the vessel.

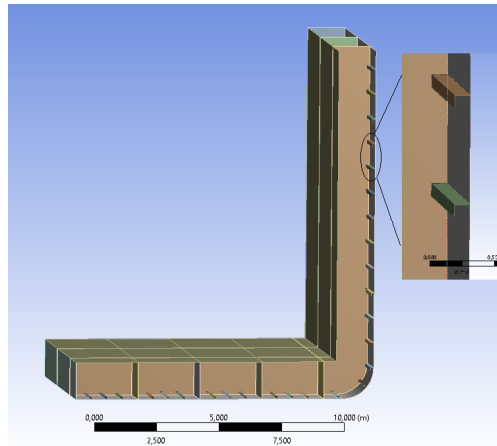


Figure 9.19: L-Stiffeners for the walls

The correlation test was initially also thought to be performed on the complete hull structure, and this was tried, but the amount of parameters made it impossible for ANSYS to handle. The computational time for only the stiffeners were quite tedious, and they only included a small fraction of the parameters needed for the hull model.

9.3 Hull parameters

The midship section that served as a basis before introducing the parameter changes, had to be constructed in compliance with the rules given by DNV GL, as explained and illustrated in chapter 9, section 9.1.

The frame for the model was the same basis as illustrated in figure 9.5. The Rules DNV GL required a set of minimum dimensions for the structure. The different zones with different minimum thicknesses are displayed in figure 9.20.

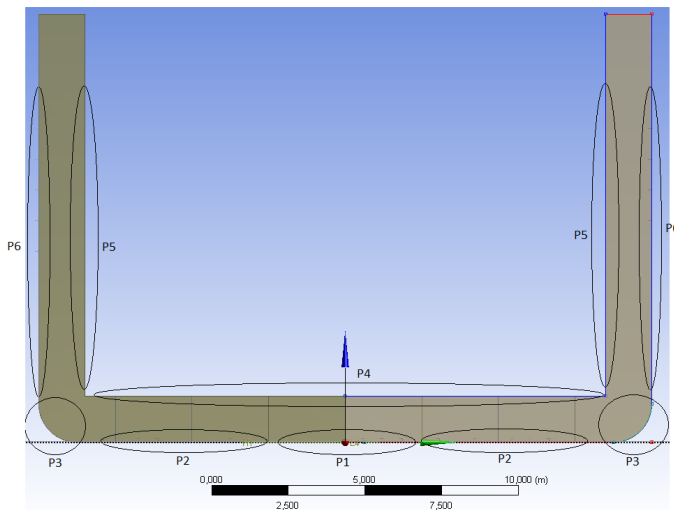


Figure 9.20: Zones for with individual thickness

Tag	section
P1	Keel plate
P2	Bottom plating
P3	Bilge
P4	Inner bottom plating
P5	Inner wall
P6	Outer wall

Figure 9.21: Zones

The required thickness for the different zones will vary according to the parameter changes of the midship section. The changes followed the formulas found in section 9.1 and new thicknesses had to be computed every time changes to the midship section were performed.

The total list of parameters used to determine the design for the midship section was (here for the basis model):

Model	1 (BASIS)
Length ship [m]	100
Length Midsection [m]	10
Width ship (B/2) [m]	10
Height hull (H) [m]	14
Depth (D) [m]	5
Keel plate width (P1) [mm]	1300
Keel plate thickness (P1) [mm]	14
Bottom plate thickness (P2) [mm]	11
Bilge thickness (P3) [mm]	14,67
Inner bottom plating (P4) [mm]	8
Inner wall thickness (P5) [mm]	5,5
Outer wall thickness (P6) [mm]	11
Stiffener height [mm]	150
Stiffener thickness [mm]	10
Stiffener flange [mm]	100
Flange thickness [mm]	7,5
Beam thickness floor [mm]	12
Beam thickness transverse [mm]	12
Longitudinal Beams wall (#)	0
Beam thickness wall [mm]	0
Longitudinal Beams floor (#)	4
Transverse beams (#)	3
Stiffeners floor (#)	11
Stiffeners wall (#)	14
Volume of section [m ³]	1400
Load [N]	30000

Table 9.1: Hull parameters for basis model

9.4 Framework for experiments (Design of experiments)

Finding a parametric relationship between the change in steel weight as a result of changing the main parameters, required a framework for the experiments.

Length, width, and depth were the parameters that were to be altered. Making a parametric relation for the three parameters and evaluating them against each other was the goal of the experiments/analyzes. A design of experiments approach was the solution.

In order to evaluate the effect of changing the parameters, a series of different models were created. In total seven models were made. The first and starting point was the basis model. Then two additional models were made for each of the parameters that were to be altered. This would result in three points that served as basis for the parametric equation, describing the relationship between steel weight and the parameter of interest. These individual relations could then be evaluated against each other to determine what yields the best design.

9.5 Ansys model development

9.5.1 Design Modeler

The rules and requirements from DNV GL along with the information gained from earlier tests had to be transferred to an ANSYS model, in order to evaluate the structure through FEM. Ansys workbench was the choice of analysis tool. The geometry and structure were developed in DesignModeler.

DesignModeler(DM) is a tool that lets you develop designs and structures, in other words, CAD. This was different from the method used earlier for the PSO approach, where the design/structure was created entirely using APDL script. No loads or restrictions were added in DM as opposed to writing the APDL script where everything was added throughout

the code.

The midship section was constructed using beam and shell elements. The stiffeners were added as beam elements to line bodies and the floors, bottom plating, etc. were set to be surface bodies, modeled as shell elements. The beam and shell elements used, were the same elements as used in the PSO approach, and for the same reasons (ref section 5.1.6). The material used was standard structural steel again as used for the PSO approach. What was important to keep in mind and to check as the FEM results were produced, were the connection between the beam elements and the shell elements. The theory of which the beam and shell are based upon, are different. Hence all the results had to be checked in order to validate that the response for both elements were correct.

The midship section was constructed as a half section since the hull is symmetric and there was no need to make the construction bigger, only adding computational time for the meshing and solving part.

The design of experiments required the design to be altered as the parameters were changed. This meant changing and reconstructing the basis model to obey the given requirements set by DNV GL. In total making seven different models.

9.5.2 Loads and restrictions

After the design and structure were completed, loads and restrictions were added. The structure was supported in both ends of the hull. The choice of support was based on trying to mimic a natural interaction between the rest of the vessel and the midship section. Hence the section was kept in place using fixed supports 9.22.

The hull was affected by two separate loads/pressures. The outer hull was set to be exposed to a hydrostatic pressure affecting the entire surface under the waterline 9.23. The waterline varied for the models that were constructed to evaluate the effect of changing the depth. The other models all had the same depth. The second load was a surface load added to the bottom floor. This load was added to represent ship equipment and other objects/installa-

tions adding weight to the inner hull bottom 9.24. The magnitude of the weight was scaled linearly in compliance with the volume of the hull.

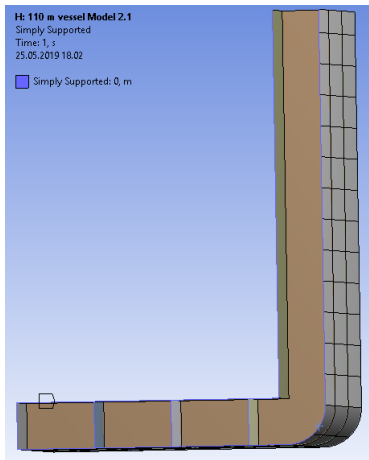


Figure 9.22:
Fixed support

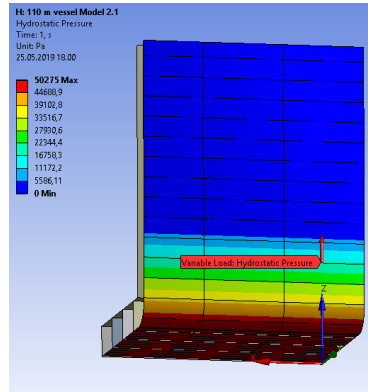


Figure 9.23:
Hydrostatic pressure

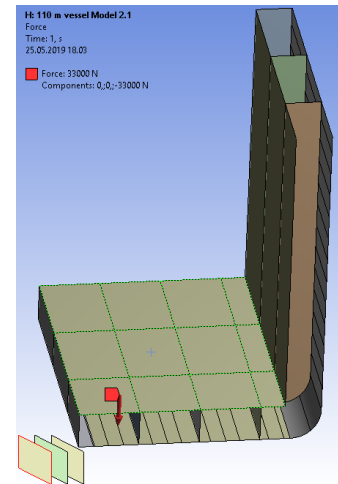


Figure 9.24: Internal load

Chapter 10

Hull evaluation

The main results and output that was extracted from the analyses, was information regarding each of the midship sections. More specific, stress, deflection, and weight were the quantifiable results. Another set of information provided by the FEM analyses was the graphic display of how the forces interacted and affected the structure. This made it easier to determine weak points and areas to be aware of.

10.0.1 Saint-Venant's principle

During the analysing process, different difficulties regarding the FEM model arose. For the analysis results to be correct, the model had to be constructed such that the forces interacting with the structure were correctly absorbed and distributed. For most of the structure, this was quite easy to achieve. Other areas proved more difficult. The transition between different plate thicknesses was one area that caused some problems for the FEM mesh and response.

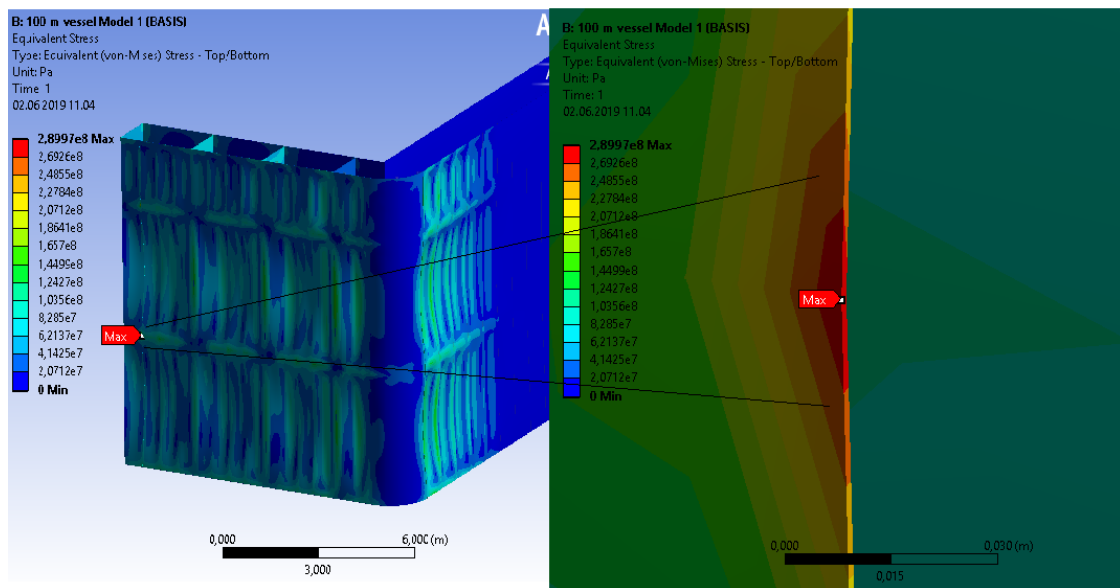


Figure 10.1: Stress singularity between keel plating and bottom plating

Figure 10.1 displays the first problematic area. The intersection between the keel plating and the outer bottom plating caused a sharp angle to occur as the thickness was larger for the keel plating. For a vessel that was to be built, this connection would have been smoothed in a way where there would be a seamless transition. In ANSYS, using surface bodies, this alternation was not possible. Hence, a singularity was produced. The model was tested with increased mesh refinement in order to validate that the stress spike, in fact, was a singularity.

Another area where a stress singularity occurred was in a transverse girder as seen from figure 10.2

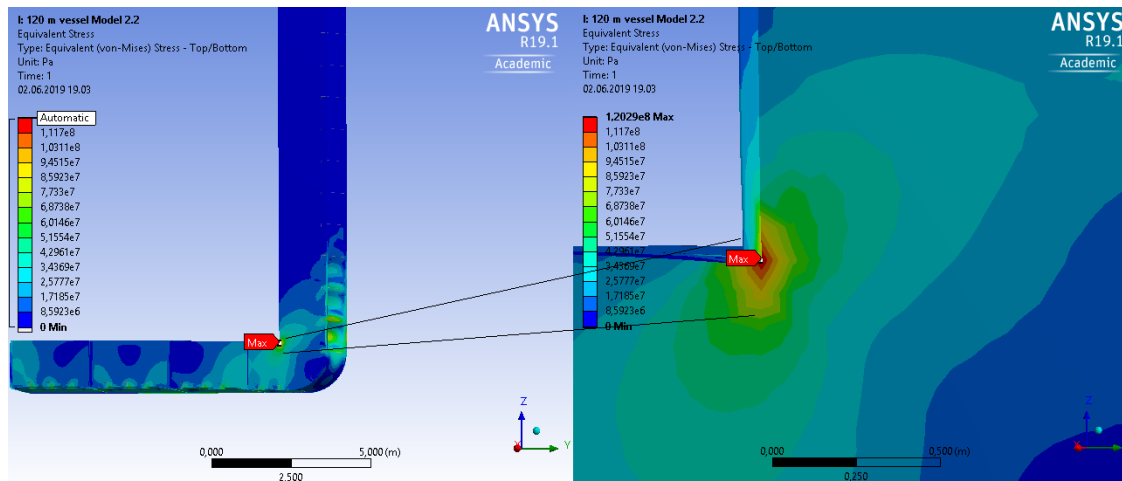


Figure 10.2: Stress singular

This stress singularity contrary to the first stress singularity was possible to avoid by adding a small curvature in the former perpendicular corner. The small curvature solved the problem by removing the possibility of having a singularity.

For the first singularity where no modification of the mesh or changes to the model could be done to prevent the singularity, Saint Venant's principle was introduced. The Saint-Venant's principle states that the difference between the effects of two different, but statically equivalent loads becomes very small as the distance increases. Explained in other words, a local disturbance to a uniform stress field will remain local and can be ignored/neglected as the distance from the area in question increases (Yang, 2018).

The singularity in question, again as seen in figure 10.1 was deemed to not be particularly more exposed to high stress spikes compared to the rest of the outer bottom plating. The singularity occurred in an intersection between two different plate thicknesses and was reinforced by a transverse girder from the inside of the hull structure. When comparing this region to other regions just like it, apart from the change in plate thickness, there are no such stress peaks. Through the use of the St. Venant's principle, the stress singularity was neglected. That meant when looking to find the highest stress areas, the singularity was

not considered. The rest of the structure was therefore inspected and analysed to find the real exposed areas, with high stress concentration.

10.1 Hull results

The first model to be tested was the basis model. This model was to represent a midship hull section of a 100 meter long vessel. The rest of the models were based on this model, but one of the main parameters were changed for each of the models. The full table including all the parameters and values for all the models can be seen from the table on the next page, table 10.1.

	BASIS	L	L	B	B	D	D
Model	1	2	3	4	5	6	7
Length ship [m]	100	110	120	100	100	100	100
Length Midsection [m]	10	11	12	10	10	10	10
Width ship (B/2) [m]	10	10	10	11	12	10	10
Height hull (H) [m]	14	14	14	14	14	15	16
Depth (T) [m]	5	5	5	5	5	6	7
Keel plate width (P1) [mm]	1300	1350	1400	1300	1300	1300	1300
Keel plate thickness (P1) [mm]	14	14,5	15	14	14	14	14
Bottom plate thickness (P2) [mm]	11	11,4	11,8	11	11	11	11
Bilge thickness (P3) [mm]	14,67	15,20	15,73	14,67	14,67	14,67	14,67
Inner bottom plating (P4) [mm]	8	8,3	8,6	8	8	8	8
Inner wall thickness (P5) [mm]	5,5	5,5	5,5	5,5	5,5	5,5	5,5
Outer wall thickness (P6) [mm]	11	11,4	11,8	11	11	11	11
Stiffener height [mm]	150	150	150	150	150	150	150
Stiffener thickness [mm]	10	10	10	10	10	20	
Stiffener flange [mm]	100	100	100	100	100	100	100
Flange thickness [mm]	7,5	7,65	7,8	7,5	7,5	7,5	7,5
Beam thickness floor [mm]	12	12,4	12,8	12	12	12	12
Beam thickness transverse [mm]	12	12,4	12,8	12	12	12	12
Longitudinal Beams wall (#)	0	0	0	0	0	0	0
Beam thickness wall [mm]	0	0	0	0	0	0	0
Longitudinal Beams floor (#)	4	4	4	4	5	4	4
Transverse beams (#)	3	3	4	3	3	3	3
Stiffeners floor (#)	11	11	11	12	13	13	13
Stiffeners wall (#)	14	14	14	14	14	14	14
Volume of section [m ³]	1400	1540	1680	1540	1680	1400	1400
Load [N]	30000	33000	36000	33000	36000	33000	36000

Table 10.1: Hull configurations

In order to evaluate the effects of increasing one parameter versus changing another parameter, a way of quantifying the gained volume versus the cost in terms of steel weight gained was developed. Determining the steel weight was easy, but finding a fair way to measure the volume was harder. Increasing the length or breadth of the hull sections resulted in a larger structure, and therefore also a larger total volume. When changing the depth, the structure does not necessarily have to be increased in terms of total volume. The solution was to demand the height of the structure to be the same above the waterline for all models. Meaning that when the depth was increased, the height of the wall structure would have to be increased by the same amount. This way, increasing depth would not be given a favourable starting point compared to the other parameters. Adding the extra height to the walls and therefore also adding extra steel weight when increasing the depth, can also be "justified" by the fact that an increased depth would allow the vessel to be higher as the COG is lowered.

10.1.1 Model results

The basis model was the smallest model. The stress and deflection pattern detected for this model was the same as experienced for many of the models with altered parameters. The structure description for the basis model can be seen from the table listed above 10.1, listed as BASIS.

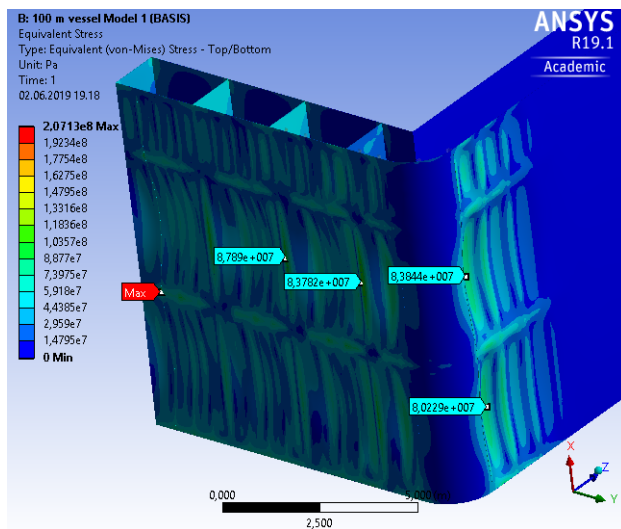


Figure 10.3: Stress for basis model

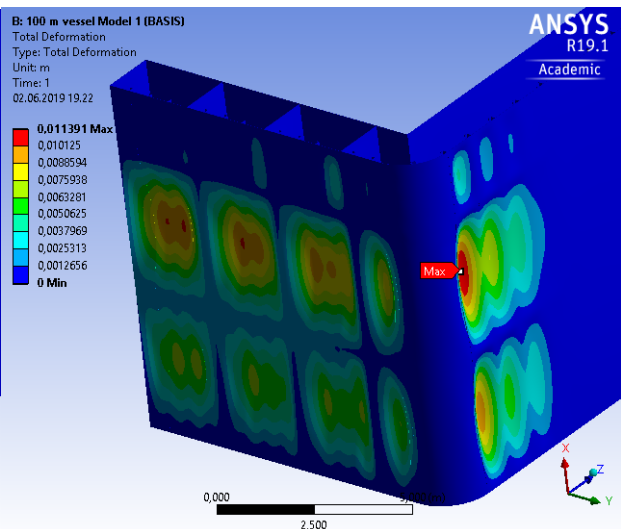


Figure 10.4: Deformation for basis model

Figure 10.3 illustrates the problem discussed in section 10.0.1. The stress scale on the left side in figure 10.3 displays a maximum stress much higher than the actual stresses experienced. The actual stress maximums can be found and seen by the tags added to the critical areas. The highest stress was found just around the connection between the floor girders and the bottom plating. Another exposed area in terms of stress is the lower part of the outer wall. The deflection is also at its largest for these areas. The response of the stress and deflection was scaled up in order to make it easier to visualize the response. The other models and their response can be found in appendix C.

The Keel plating and the bilge plating were both thicker than the rest of the outer plating, and this extra reinforcement can be seen from the two figures. The structure as a whole withstands the pressures and loads added in a good way. The stress was never in danger of exceeding the structural limits and a good margin was maintained. The results for the different models can be seen in table 10.2 below.

Results:	Model 1	Model 2	Model 3	Model 4	Model 5	Model 6	Model 7
Max steress (Exc. singularity)[MPa]	87,89	89,434	91,556	94,62	110,26	114,89	140,69
Max deflection [m]	0,011391	0,010778	0,012452	0,01136	0,011696	0,015012	0,018775
Weight of section [kg]	51078	56417,00	65271	53164	56663	52365,4	53652,8
Allowable stress [Mpa]	200	200	200	200	200	200	200
Percent of allowable stress [%]	43,945	44,717	45,778	47,31	55,13	57,445	70,345

Table 10.2: Hull Results

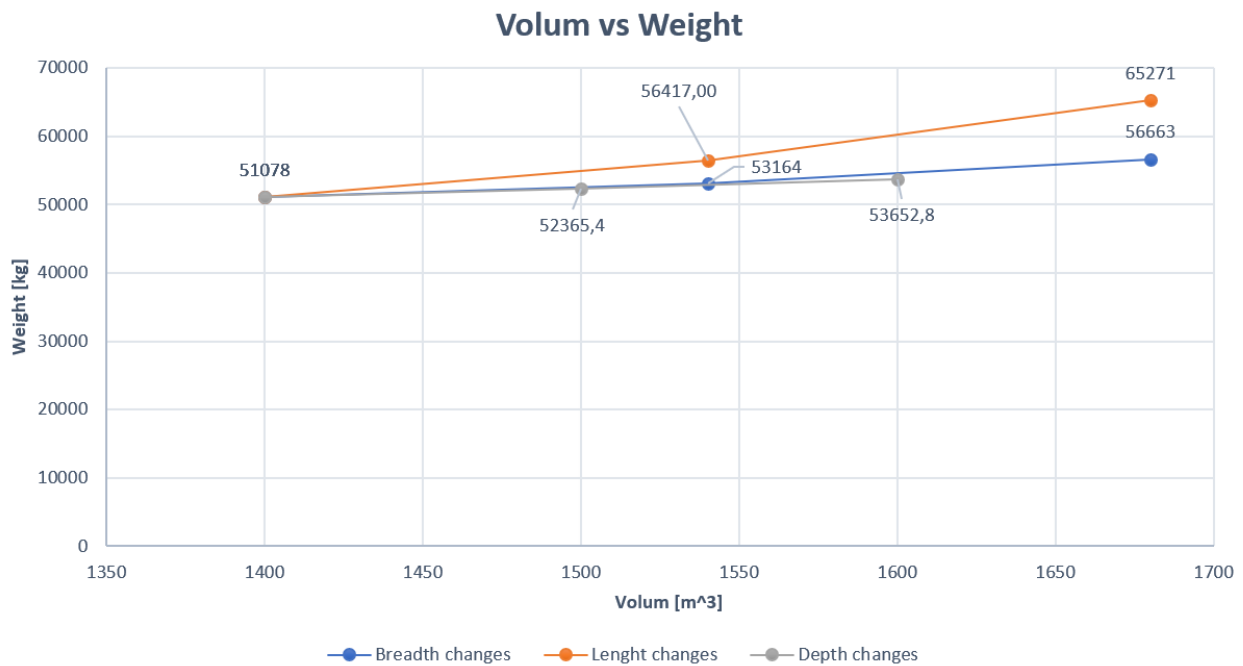


Figure 10.5: Parameter relation for steel weight

The graphs displayed in figure 10.5 show how the steel weight is affected by changing the main dimensions. Length was the parameter that caused the steepest increase in weight per gained volume. Breadth altering had the second steepest curve, while depth was slightly better than breadth.

The results from the graph were then transformed to parametric equations. The best regression for calculating/predicting the steel weight within the limits of the tested models was a second-degree polynomial. At least for the length and breadth parameter, depth, on the other hand, was best modeled in a linear way. The problem with using a second-degree polynomial for the length and breadth was that the weight prediction quickly became imprecise when venturing beyond the limits of the tested models. Hence in order to be able to use the parametric equations beyond the tested models' limits, linear regression was thought to be the best option. All the tested regressions can be seen in appendix D.

$$W_{st}(V(L)) = 50,6893 \cdot V(L) + 20472,8333 \quad (10.1)$$

V(L)=Volume given length(L) is the parameter to be changed

$$W_{st}(V(B)) = 19,9464 \cdot V(B) + 22917,5 \quad (10.2)$$

V(B)=Volume given breadth(B) is the parameter to be changed

$$W_{st}(V(D)) = 12,874 \cdot V(D) + 33054,4 \quad (10.3)$$

V(D)=Volume given depth(D) is the parameter to be changed

Based on the parametric equations, the effect of changing one parameter compared to changing one of the other parameters could now be quantified.

Increasing the volume by increasing the length was found to add 2,54 times more steel weight compared to increasing the breadth, and 3,94 times more than if the depth was increased.

10.2 Discussion of hull results

There were a couple of interesting things worth mentioning after looking at and analysing the results for all the models.

First of, all the stresses were found to be well below the structural limit of 200 Mpa. This can be explained by the way the model was constructed. The structure was designed using the requirements set by DNV GL and the knowledge obtained from previous tests. This meant that there should be a good safety margin. The models were not tested for dynamic situations, meaning that the vessels will encounter greater forces than experienced in these tests, and that extra strength might be crucial.

The highest increase in stress was naturally when the depth was increased. More interesting was the fact that increasing the width, caused the second largest impact in terms of stress increase. This discovery can be explained by the way the structure was built. When the bottom part of the hull structure was given a wider dimension, extra support in the longitudinal and transverse direction were added, but more support in the longitudinal direction compared to the transverse. This meant that the width ended up being a more sensitive parameter compared to the length in terms of stress.

The steel weight aspect of the models was the main goal of the analyses. The relation between the three parameters (L-B-D) displayed the same relation as earlier theory [ch.8] also suggested. The model did also provide a set of factors that quantified the relationship between the parameters. The clear winner was the depth parameter followed by the breadth. Increasing the best parameter first should be the strategy, but the performance of the vessel should not be compromised. This means that the favourable parameters should not be increased more than what suggested throughout chapter 8.

The mesh size/element size used to produce the result was kept at 0.049 meters. This element size ensured that the results were as precise as possible. The FEM analyses converged at larger element sizes but were kept lower to be in compliance with DNV GL. According to the FEM guidelines given by DNV GL, the element size in areas of interest should not be larger than 0.055m (DNVGL, 2017). This means that the FEM analyses were executed in compliance with DNV GL.

The results and the hull model was as earlier explained a simplified hull structure. The models' ability to predict accurate steel weights for genuine midship hull sections will prob-

ably not be the best use of these results. The results ability to provide information and the quantified relationship between the main dimensions is where the strength of these experiments lies. The relationship between transverse, longitudinal and vertical support was tried model as realistic as possible, meaning that the relationship between the main parameters in terms of steel weight should be valid.

Chapter 11

Evaluation of process and further work

The process towards finding a solution to the problems at hand was a tough process. The methodology and experiments constructed in the process were all based on my vision for this thesis. The goal was to use FEM, a precise but tedious process, to develop a quick and easy decision support for vessel design. Computing a full finite element analyses (FEA) at a concept stage in the design process is too time-consuming and expensive. By applying FEA for an arbitrary hull model, and using the results to produce a parametric relation between the main dimensions, the goal was to bridge the gap between concept stage and the contract design in terms of steel weight.

The minimising process was as earlier discussed, tried executed in three separate ways, all based of FEM. The overall goal of this process was to test if a superior design was possible to achieve. Meaning not a specific design for one particular structure, but in general terms, what affects the steel weight in a positive way, while maintaining the needed strength. The methods used for this purpose all had their own approach going at the problem in their own way. This resulted in a broad coverage of the problem which was the plan when deciding upon the methods. A drawback with the methods was that none of the methods included dynamic loads or pressures. Hence the results did not present information regarding a dynamic response. In all, the results provided the information needed, but further work for the weight minimizing aspect could be to introduce more advanced loading conditions.

Another aspect that could be interesting for further exploring is by applying the techniques to larger scale models.

The hull model and the global parameter relation that was developed was based on a midship hull section. The decision to use a midship section was based on a list of factors. A midship section is not affected by any specific or unique design for the forward and aft section. This was desirable as the results were to be valid for an arbitrary hull. The midship section represents the largest part of the vessel and served as a natural limitation for the FEM model, which was a necessity. The way the models were developed and tested served its purpose and produced the results needed to produce a parametric relation for the main dimensions in regards to the steel weight. The model was tried developed as realistic as possible, in order to make the results valid. Restrictions preventing access to comparable steel weight information made it hard to confirm or compare the results. Further work should therefore be focused on obtaining the desired steel weight information and comparing the results. A broader span in the models/more models is another point to add to the list for further work. Making more models would cause the results to be valid for a broader range of vessels, and help validate the existing results. If more time was available this would have been persecuted.

In all, the process have been very interesting and I have learnt a lot. Constructing my own approach and experiments have brought a new dimension to my way of thinking. Choosing the approach that I did and the tools needed for these tasks did also cause difficulties. Learning to use new programs/software and when stuck, having to struggle by your self, made me really appreciate the results when I finally succeeded.

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Appendix A

Codes

A.1 Optimization codes

A.1.1 StructOpt

```
1 function StructOpt
2
3 % How to get initial population?
4 % 1 = random or 2 = read from file.
5 source = 1; %source
6
7 % Number of continuous variables
8 continuous_variables = 0;
9
10 % Upper and lower bound for continuous variables
11 LowerB = [];
12 UpperB = [];
13
14 % PSO parameters
15 Algorithm = [60 -1 30 1.4 0.8 3 2 1];
```



```
16
17 % PSO-parameter
18 % swarm_size = 60;
19 % feasibles_initial_population = -1; % Feasible designs in intial
20 %population (-1 dont want to use).
21 % generations = 30; % Calculation rounds.
22 % inertia = 1.4; % Intertia at start.
23 % beta = 0.8; % Factor for dynamic inertia reduction
24 % beta_k = 3; % Number of rounds when it should improve,
25 %otherwise make inertia smaller
26 % penalty_factor = 2; % Penalty factor for violated constraints
27 % print_results = 1; % 0 not printed, 1 is printed (results)
28
29
30
31 % Set of discrete variables
32 Feasible_Set=[];
33
34 nvariables=10;
35 Feasible_Set=inf(nvariables,31);
36
37 %plate
38 Feasible_Set(1,1:26)=[0.005:0.001:0.03]; %plate thickness
39
40 %girders
41 Feasible_Set(2,1:26)=[0.3:0.02:0.80]; %girder height
42 Feasible_Set(3,1:31)=[0.005:0.001:0.035]; %girder thickness
43
44 %stiffeners
45
46 Feasible_Set(4,1:30)=[6:1:35]; %number of stiffeners
47 Feasible_Set(5,1:18)=[0.03:0.01:0.2]; %stiffener height
48
49 Feasible_Set(6,1:26)=[0.005:0.001:0.03]; %stiffener thickness
50
51 %flange
```

```
52 Feasible_Set(7,1:19)=[0.04:0.02:0.4]; %flange width
53 Feasible_Set(8,1:16)=[0.005:0.003:0.05]; %flange thickness
54
55 %bulb of stiffeners
56 Feasible_Set(9,1:16)=[0.005:0.003:0.05]; %bulb width
57 Feasible_Set(10,1:16)=[0.005:0.003:0.05]; %bulb height
58
59
60 % Carry out the optimization.
61 if source == 1
62     [best_f,best_x,best_g,history_f,history_x,history_g,iterations,...
63         particle,particle_history,t_history] = PSO('particle_fun',...
64 continuous_variables,Feasible_Set,LowerB,UpperB,Algorithm, []);
65 elseif source == 2
66     [best_f,best_x,best_g,history_f,history_x,history_g,iterations,...
67         particle,particle_history,t_history] = PSO('particle_fun',...
68 continuous_variables,Feasible_Set,LowerB,UpperB,Algorithm,...
69 'initial_feasible_population.txt');
70 end
71
72
73
74 %Best feasible particletion
75 format compact;
76 disp(' ')
77 disp(' ')
78 disp('Results:')
79 best_f,best_x,best_g
80 disp(' ')
81 disp(' ')
```

A.1.2 particlefun

```
1 function [f,g] = particle_fun(Xxx)
2
3 %Variables/parameters that are optimized
4
5 plate_t=Xxx(1);
6 girder_height=Xxx(2);
7 girder_t=Xxx(3);
8 n_stiffeners=Xxx(4);
9 stiffener_height=Xxx(5);
10 stiffener_t=Xxx(6);
11 flange_width=Xxx(7);
12 flange_t=Xxx(8);
13 bulb_width=Xxx(9);
14 bulb_height=Xxx(10);
15
16 %parameters not to be optimized
17 plate_length = 13;
18 plate_width = 4;
19 length_to_first_girder = 0.5*(sqrt(2)-1)*plate_width;
20 length_to_second_girder = plate_width-length_to_first_girder;
21 load = 9.81*(plate_length*plate_width)*1466;
22 m_size = 0.2;
23 stiffener_space = plate_length/(n_stiffeners+1);
24 offset_y = (bulb_width/2)+(stiffener_t/2);
25 offset_z = (bulb_height/2);
26
27
28
29
30 %making a file where the design variables and parameters are stored
31 fid = fopen('parameters.inp','w');
32 fprintf(fid,'plate_t = %f \n', plate_t);
33 fprintf(fid,'plate_length = %f \n', plate_length);
34 fprintf(fid,'plate_width = %f \n', plate_width);
35 fprintf(fid,'girder_height = %f \n', girder_height);
36 fprintf(fid,'girder_t = %f \n', girder_t);
```

```
37 fprintf(fid, 'n_stiffeners = %f \n', n_stiffeners);
38 fprintf(fid, 'stiffener_height = %f \n', stiffener_height);
39 fprintf(fid, 'stiffener_t = %f \n', stiffener_t);
40 fprintf(fid, 'flange_width = %f \n', flange_width);
41 fprintf(fid, 'flange_t = %f \n', flange_t);
42 fprintf(fid, 'load = %f \n', load);
43 fprintf(fid, 'bulb_width = %f \n', bulb_width);
44 fprintf(fid, 'bulb_height = %f \n', bulb_height);
45 fprintf(fid, 'm_size = %f \n', m_size);
46 fprintf(fid, 'length_to_first_girder = %f \n', length_to_first_girder);
47 fprintf(fid, 'length_to_second_girder = %f \n', length_to_second_girder);
48 fprintf(fid, 'stiffener_space = %f \n', stiffener_space);
49 fprintf(fid, 'offset_y = %f \n', offset_y);
50 fprintf(fid, 'offset_z = %f \n', offset_z);
51 fclose(fid);
52
53
54 !modeling.bat
55 fid = fopen('out_disp.txt', 'r');
56 disp_max = str2num(fgetl(fid))
57 fclose(fid);
58
59 fid = fopen('out_stress.txt', 'r');
60 stress_max = str2num(fgetl(fid))
61 fclose(fid);
62
63
64 youngs = 250*10^6; %sigma yield in ANSYS
65
66 volume = plate_length*plate_width*plate_t + 2*plate_length*...
67     (girder_height*girder_t + flange_t*2*flange_width)...
68     + n_stiffeners*plate_width*...
69     (stiffener_height*stiffener_t + bulb_height*bulb_width);
70
71 mass = 7800*volume;
72
```

```
73 objective=mass; %Lowest possible mass is the objective
74
75 c1 = ((plate_t+flange_t+girder_height)/(0.05*plate_length))-1%limit height
76 c2 = (disp_max/(0.01*plate_length))-1 %limiting the deflection
77 c3 = (stress_max/(0.80*youngs))-1 %max 20% of youngs modulus
78
79 %constraint must be between -1 og 1
80
81 constraint=max([c1,c2,c3]);
82 if constraint > 1
83     constraint = 1;
84 elseif constraint < -1
85     constraint = -1;
86
87 end
88
89 %constraint and objective function value
90 f=objective %objective
91 g=constraint %constraint feasible -1 to 0 and infeasible is from >0 to 1
```

A.1.3 Particle swarm optimization

Obtained from the course TMR 4320

```
1 function [best_f,best_x,best_g,history_f,history_x,history_g,iterations,...
2         particle,particle_history,t_history] = PSO(fun,counter_number,...
3         Feasible_Set,LowerB,UpperB,Algorithm,Init_Popu)
4
5 % PSO-parameter.
6 swarm_size = Algorithm(1);
7 feasibles_initial_population = Algorithm(2);
8 generations = Algorithm(3);
9 inertia = Algorithm(4);
```

```

10 beta = Algorithm(5);
11 beta_k = Algorithm(6);
12 penalty_factor = Algorithm(7);
13 print_results = Algorithm(8);
14 c1 = 2;
15 c2 = 2;
16
17 % CPU-time at beginning.
18 t0 = cputime;
19
20 % Number of analysis.
21 particle = 0;
22
23 % No of discrete variables
24 if isempty(Feasible_Set)==1
25     discrete_var_number = 0;
26 else
27     discrete_var_number = length(Feasible_Set(:,1));
28 end
29
30 % Generate automatically the initial population randomly so that it
31 % contains desired amount of feasible particletion
32 if isempty(Init_Popu)==1
33     [position,objective_function,constraint,particle,history_best_f,...
34         best_individual,history_f,best_current_individual,history_g] = ...
35         new_population(fun,swarm_size,(counter_number+discrete_var_number),...
36         feasibles_initial_population,Feasible_Set,penalty_factor,...
37         counter_number,LowerB,UpperB);
38
39 % Read init pop from file.
40 else
41     [position,objective_function,constraint,particle,history_best_f,...
42         best_individual,history_f,best_current_individual,history_g] = ...
43         read_population(fun,Init_Popu,particle,swarm_size,...
44         (counter_number+discrete_var_number),feasibles_initial_population,...
45         Feasible_Set,penalty_factor);

```

```
46 end
47
48 % I pop objective and constraint values.
49 f = objective_function;
50 g = constraint;
51
52 % Best global position (feasible or infeasible) and objective function
53 % value
54 best_glob = best_individual;
55 best_glob_f = history_best_f;
56
57 % Best global position (feasible) and objective function
58 % value and constraints.
59 best_glob_current_f = history_f;
60 best_glob_current = best_current_individual;
61 best_glob_current_g = history_g;
62
63 % Each particals best position equals the intial position.
64 best_location = position;
65
66 % Calculate penaliest objective function values
67 penalized_f = lower_penalized(f,g,penalty_factor);
68
69 % Each particals best known position equals initial position
70 best_location_f = penalized_f;
71
72 % Calculate max speed
73 if counter_number > 0
74     for i = 1:counter_number
75         v_min(i) = LowerB(i);
76         v_max(i) = UpperB(i);
77         maximum_v(i) = 0.3*(v_max(i)-v_min(i));
78     end
79 end
80 if discrete_var_number > 0
81     for i = (counter_number+1):(counter_number+discrete_var_number)
```

```

82     v_min(i) = Feasible_Set(i-counter_number,1);
83     v_max(i) = max(Feasible_Set(i-counter_number,:));
84     maximum_v(i) = rounding(0.3*(v_max(i)-v_min(i)),...
85         Feasible_Set(i-counter_number,:));
86     end
87 end
88
89 % Set 0 as init speed.
90 velocity = zeros((counter_number+discrete_var_number),swarm_size);
91
92 % Round calculator, size calculator and inertia calculator=1
93 round_counter = 1;
94 size_counter = 1;
95 inertia_counter = 1;
96
97
98 % Loop
99 while round_counter <= generations
100
101     % Lets make inertia smaller, if best feasible objective
102     %function value has not improved during certain amount of cycles
103     if inertia_counter > beta_k
104         inertia = beta*inertia;
105         inertia_counter = 1;
106     end
107
108     % Best feasible particletion number of analysis and store the time
109     history_f(round_counter) = best_glob_current_f;
110     particle_history(round_counter) = particle;
111     t_history(round_counter) = cputime-t0;
112
113
114     % Print to the screen the progress of calculation
115     disp(['Iter.: ' sprintf('%4i',round_counter) ...
116 ' /' sprintf('%4i',generations) ...
117 ' Obj.: ' sprintf('%8.2f',best_glob_current_f)...

```



```

118 ' Analysis: ' sprintf('%4i',particle) ...
119     ' Inert.: ' sprintf('%2.4f',inertia)...
120     ' Feasibles: ' sprintf('%4i',feasibles_initial_population)...
121     ' /' sprintf('%3i',swarm_size)]
122
123
124     % Speed and position for each particle
125     for i=1:swarm_size
126         % randomized r1 and r2.
127         r1 = rand;
128         r2 = rand;
129
130         % calculate new speed per particle
131     new_velocity(:,i) = inertia*velocity(:,i) + c1*r1*(best_location(:,i) - ...
132         position(:,i)) + c2*r2*(best_glob_current-position(:,i));
133
134     % Is the maximum speed exceeded?
135     for j = 1:(counter_number+discrete_var_number)
136         if new_velocity(j,i) > maximum_v(j)
137             new_velocity(j,i) = maximum_v(j);
138         elseif new_velocity(j,i) < -maximum_v(j)
139             new_velocity(j,i) = -maximum_v(j);
140         end
141     end
142
143     % Calculate new position for particle
144     new_position(:,i) = position(:,i) + new_velocity(:,i);
145
146     % Did we exceed upper and lower boundaries
147     for j = 1:counter_number
148         if new_position(j,i) < LowerB(j)
149             new_position(j,i) = LowerB(j);
150         end
151         if new_position(j,i) > UpperB(j)
152             new_position(j,i) = UpperB(j);
153         end

```

```
154     end
155
156     % Standardisation of discrete variables
157     for j = (counter_number+1):(counter_number+discrete_var_number)
158         new_position(j,i) = rounding(new_position(j,i),...
159             Feasible_Set(j-counter_number,:));
160     end
161 end
162
163 % Update position and speed
164 position = new_position;
165 new_position = [];
166 velocity = new_velocity;
167 new_velocity = [];
168
169 % Calc objective and constraints for all particals in new positions
170 f = [];
171 g = [];
172 for i=1:swarm_size
173     % Objective fct and constraints
174     [new_f,new_g] = feval(fun,position(:,i));
175
176     % Make number of analysis one higher
177     particle = particle+1;
178
179     f(i) = new_f;
180     g(:,i) = new_g;
181 end
182
183 % Calc penalized objective fct
184 penalized_f = lower_penalized(f,g,penalty_factor);
185
186 % Set no of feasible particles to 0
187 feasibles_initial_population = 0;
188
189 % Update the best position and value of each particle in memory
```

```
190     % global best position and value and best feasible position and value
191     % plus constraint values
192     for i=1:swarm_size
193         % check if best known value is improved
194         if penalized_f(i) < best_location_f(i)
195             best_location_f(i) = penalized_f(i);
196             best_location(:,i) = position(:,i);
197         end
198
199         % check if best global value is improved
200         if penalized_f(i) < best_glob_f
201             best_glob_f = penalized_f(i);
202             best_glob = position(:,i);
203         end
204
205         % check if best global feasible value is improved
206         if penalized_f(i) < best_glob_current_f & all(g(:,i)<=0)
207             best_glob_current_f = penalized_f(i);
208
209             best_glob_current = position(:,i);
210             best_glob_current_g = g(:,i);
211         end
212
213         % record the number of feasible particles.
214         if all(g(:,i)<0)==1
215             feasibles_initial_population = feasibles_initial_population+1;
216         end
217     end
218
219     % safe the best known and feasible value
220     if isempty(best_glob_current_g) == 1
221         history_x = [];
222         history_g = [];
223     else
224         history_x(:,round_counter) = best_glob_current;
225         history_g(:,round_counter) = best_glob_current_g;
```

```
226     end
227     history_best_f(round_counter) = best_glob_f;
228
229
230     % check if size and inertia calculators should equal to 1
231     if round_counter>2 & history_f(round_counter)<history_f(round_counter-1)
232         inertia_counter = 1;
233         size_counter = 1;
234     else
235         inertia_counter = inertia_counter+1;
236         size_counter = size_counter+1;
237     end
238
239     % increase round calc by 1
240     round_counter = round_counter+1;
241
242 end
243
244 % no of calculation rounds
245 iterations = round_counter-1;
246
247
248 % return best feasible particletion
249 best_f = history_f(iterations);
250 best_x = best_glob_current;
251 best_g = best_glob_current_g;
252
253 if (find(g==best_g)~=0)
254     itt = find(g==best_g);
255     best_f = min(f(itt));
256 end
257
258 % print graphs if requested
259 if print_results==1
260     figure;
261     plot([1:round_counter-1],history_best_f,[1:round_counter-1],history_f)
```

```
262     legend('Best known','best feasible')
263     grid on;
264     ylabel('Objective function')
265     xlabel('No of Iterations')
266
267     figure;
268     plot(particle_history,history_best_f,particle_history,history_f)
269     legend('Best known','best feasible')
270     grid on;
271     ylabel('Objective function')
272     xlabel('Analysis')
273
274     figure;
275     hold on;
276     for i=1:length(history_g(:,1))
277         plot([1:round_counter-1],history_g(i,:));
278     end
279     title('Development of Constraints. ');
280     xlabel('Iteration');
281 end
```

A.2 ANSYS code

A.2.1 ADPL code for deck

```
1 finish
2 /clear
3 /prep7
4 /input,parameters.inp
5
6
7 !.....making plate.....
```

```
8
9 numstr,area,100
10 numstr, kp, 1
11 k,,0,0,0
12 k,,plate_length,0,0
13 k,,plate_length,plate_width,0
14 k,,0,plate_width,0
15
16
17 A,1,2,3,4
18
19
20 !...making stiffeners (counterclockwise)....
21
22 *do,i,1,n_stiffeners,1
23 k,,i*stiffener_space,0,0
24 k,,i*stiffener_space,plate_width,0
25 k,,i*stiffener_space,plate_width,stiffener_height
26 k,,i*stiffener_space,0,stiffener_height
27
28
29 A,(1+4*i),2+(4*i),3+(4*i),4+(4*i)
30
31
32 !Siste knutepunktet = 24 hvis
33 *enddo
34
35
36 !...Making the frame for the first girder,
37 ! the upper part, above the stiffeners (clockwise)..
38
39 numstr,kp,200
40
41
42 k,,0,length_to_first_girder,0
43 k,,0,length_to_first_girder,girder_height
```

```

44 k,,plate_length,length_to_first_girder,girder_height
45 k,,plate_length,length_to_first_girder,0
46 k,,(plate_length-stiffener_space),length_to_first_girder,0
47 k,,(plate_length-stiffener_space),length_to_first_girder,stiffener_height
48 k,,stiffener_space,length_to_first_girder,stiffener_height
49 k,,stiffener_space,length_to_first_girder,0
50
51
52 A,200,201,202,203,204,205,206,207
53
54
55 !..Second girder, same as for the first girder (clockwise) .....
56 numstr,kp,240
57
58 k,,0,length_to_second_girder,0
59 k,,0,length_to_second_girder,girder_height
60 k,,plate_length,length_to_second_girder,girder_height
61 k,,plate_length,length_to_second_girder,0
62 k,,(plate_length-stiffener_space),length_to_second_girder,0
63 k,,(plate_length-stiffener_space),length_to_second_girder,stiffener_height
64 k,,stiffener_space,length_to_second_girder,stiffener_height
65 k,,stiffener_space,length_to_second_girder,0
66
67
68 A,240,241,242,243,244,245,246,247
69
70 !Making the boxes under the first girder, between the stiffeners(clockwise) ..
71
72 numstr,kp,504
73
74 *do,i,1,(n_stiffeners-1),1
75 k,,(i*stiffener_space),length_to_first_girder,0
76 k,,(i*stiffener_space),length_to_first_girder,stiffener_height
77 k,,((i+1)*stiffener_space),length_to_first_girder,stiffener_height
78 k,,((i+1)*stiffener_space),length_to_first_girder,0
79

```

```
80 A, 500+(4*i), 501+(4*i), 502+(4*i), 503+(4*i)
81
82
83
84 *enddo
85
86 !.... Same as above but for second girder...
87
88 numstr, kp, 704
89
90 *do, i, 1, (n_stiffeners-1), 1
91 k, , (i*stiffener_space), length_to_second_girder, 0
92 k, , (i*stiffener_space), length_to_second_girder, stiffener_height
93 k, , ((i+1)*stiffener_space), length_to_second_girder, stiffener_height
94 k, , ((i+1)*stiffener_space), length_to_second_girder, 0
95
96 A, 700+(4*i), 701+(4*i), 702+(4*i), 703+(4*i)
97
98
99
100 *enddo
101
102
103 ! f rste knutepunkt = 132
104 !siste knutepunkt = 147
105
106
107 !....Making the flange for the first girder...
108 numstr, kp, 1000
109
110 k, , 0, (length_to_first_girder-flange_width), girder_height
111 k, , 0, (length_to_first_girder+flange_width), girder_height
112 k, , plate_length, (length_to_first_girder+flange_width), girder_height
113 k, , plate_length, (length_to_first_girder-flange_width), girder_height
114
115 A, 1000, 1001, 1002, 1003
```



```
116 !A,148,149,150,151
117
118
119 !.....Making the flange for girder 2.....
120 numstr,kp,1200
121
122 k,,0,(length_to_second_girder-flange_width),girder_height
123 k,,0,(length_to_second_girder+flange_width),girder_height
124 k,,plate_length,(length_to_second_girder+flange_width),girder_height
125 k,,plate_length,(length_to_second_girder-flange_width),girder_height
126
127 A,1200,1201,1202,1203
128
129
130 !....Gluing everything together....
131 aovlap,all !hindrer at flens og plate limes
132 aglue,all
133
134
135 !...Selecting and naming the plate....
136 asel,s,loc,z,0,0
137 cm,plate,area
138
139
140 !.....selecting and naming the stiffeners .....
141
142
143 asel,s,loc,z,0,stiffener_height
144 asel,u,loc,z,0,0
145 *do,i,1,n_stiffeners-1,1
146 asel,u,loc,x,(stiffener_space*i+0.0001),stiffener_space*(i+1)-0.0001
147 *enddo
148 cm,stiffeners,area
149
150
151
```

```
152 !...selecting and naming the girders.....
153
154
155 asel, s, loc, y, length_to_first_girder
156 asel, A, loc, y, length_to_second_girder
157 cm, girders, area
158
159
160
161
162 !...selecting and naming the flanges.....
163
164
165 asel, s, loc, z, girder_height, girder_height
166 cm, flange, area
167
168
169 allsel,all
170 cmsel,all
171
172
173 !...marking the toplines of the flanges...
174
175 lsel, s, loc, x, stiffener_space
176 *do, i, 1, (n_stiffeners-1)
177 lsel, a, loc, x, stiffener_space*(1+i)
178 *enddo
179
180
181 !.....bulb.....
182
183 lsel, r, loc, z, stiffener_height
184 cm,bulb,line
185
186
187
```

```
188 !...selecting everything so it shows in ANSYS (early point).....
189 asel, all
190
191 !.....Adding the element types being used.....
192
193
194 ET, 1, shell181
195 ET, 2, beam188
196
197 MPTEMP,1,0      !hvordan metallet p virkes av temperatur
198 MPDATA,EX,1,,2E+11      !E-modul
199 MPDATA,PRXY,1,,0.26      !poisson (tverrsnittskonstant)
200 MPDATA,DENS,1,,7800 !density
201
202
203 !.....Assigning element types to the different parts ....
204
205
206 sect, 1, shell,,plate
207 secdata,plate_t,1,0.0,3
208 secoffset,top,
209
210 sect, 2, shell,,stiffeners
211 secdata,stiffener_t,1,0.0,3
212
213 sect, 3, shell,,girders
214 secdata,girder_t,1,0.0,3
215
216 sect,4,shell,,flange
217 secdata,flange_t,1,0.0,3
218
219 sect, 5, beam, rect, bulb, 0
220 secdata, bulb_width, bulb_height
221 secoffset, user, offset_y, offset_z
222
223
```

```
224 !.....Meshing.....
225
226 ESIZE, m_size
227
228 !.....plate.....
229 type,1
230 secn,1
231 cmsel,s,plate
232 amesh,all
233
234 !.....stiffeners.....
235 type,1
236 secn,2
237 cmsel,s,stiffeners
238 amesh,all
239
240 !.....girders.....
241 type,1
242 secn,3
243 cmsel,s,girders
244 amesh,all
245
246 !.....flange.....
247
248 type,1
249 secn,4
250 cmsel,s,flange
251 amesh,all
252
253 !.....bulb.....
254 type,2
255 secn,5
256 cmsel,s,bulb
257 lmesh,all
258
259
```

```
260
261 !..Boundary conditions..
262
263 nsel,s,loc,z,0
264 nsel,r,loc,x,0
265 nsel,r,loc,y,length_to_first_girder
266 D,all,UZ,0
267 D,all,UX,0
268 D,all,UY,0
269 D,all,rotz
270 asel,all
271
272 nsel,s,loc,z,0
273 nsel,r,loc,x,0
274 nsel,r,loc,y,length_to_second_girder
275 D,all,UZ,0
276 asel,all
277
278 nsel,s,loc,z,0
279 nsel,r,loc,x,plate_length
280 nsel,r,loc,y,length_to_second_girder
281 D,all,UZ,0
282 asel,all
283
284 nsel,s,loc,z,0
285 nsel,r,loc,x,plate_length
286 nsel,r,loc,y,length_to_first_girder
287 D,all,UZ,0
288 asel,all
289
290 nsel,all
291 asel,all
292
293 !...Applying gravity and load...
294 ACEL,0,0,-9.81 !defining gravity in negativ Z-direction
295
```

```
296 SF,plate,PRES,(load/(plate_width*plate_length)) !defining the pressure
297
298
299 !.....Solve.....
300
301 /SOLU !enter the solution processor
302 solve
303 /sho, file
304 esel,s,cent,x,(plate_length*0.1),(plate_length*0.9)
305
306 !Enter the database result postprocessor
307 /post1
308 /efacet,1
309
310
311
312 !....Max stress and displacement.....
313
314 plnsol,u,z
315 *GET,max_disp,PLNSOL,0,MAX !Finding maximum displacement
316 plnsol,s,eqv
317 *GET,max_stress,PLNSOL,0,MAX !Finding maximum stress
318
319
320 !....Writes into two files.....
321 *cfopen,out_disp.txt
322 *vwrite,max_disp
323 (f10.5)
324 *cfclos
325
326 *cfopen,out_stress.txt
327 *vwrite,max_stress
328 (f15.2)
329 *cfclos
```

Appendix B

DNV GL

B.1

A 200 Definitions

201 Symbols:

- L = rule length in m ¹⁾
B = rule breadth in m ¹⁾
D = rule depth in m ¹⁾
T = rule draught in m ¹⁾
C_B = rule block coefficient ¹⁾
V = maximum service speed in knots on draught T
L₁ = L but need not be taken greater than 300 m
t = rule thickness in mm of plating
Z = rule section modulus in cm³ of stiffeners and simple girders
k_a = correction factor for aspect ratio of plate field
= $(1.1 - 0.25 s/l)^2$
= maximum 1.0 for $s/l = 0.4$
= minimum 0.72 for $s/l = 1.0$
s = stiffener spacing in m, measured along the plating
l = stiffener span in m, measured along the topflange of the member. For definition of span point, see Sec.3 C100. For curved stiffeners l may be taken as the cord length
w_k = section modulus corrosion factor in tanks, see Sec.3 C1004
= 1.0 in other compartments
σ = nominal allowable bending stress in N/mm² due to lateral pressure
p = design pressure in kN/m² as given in B
f₁ = material factor ²⁾
= 1.0 for NV-NS steel ²⁾
= 1.08 for NV-27 steel ²⁾
= 1.28 for NV-32 steel ²⁾
= 1.39 for NV-36 steel ²⁾
= 1.47 for NV-40 steel ²⁾
f_{2b} = stress factor below the neutral axis of the hull girder depending on surplus in midship section modulus and maximum value of the actual still water bending moments:
$$f_{2b} = \frac{5.7(M_S + M_W)}{Z_B}$$

Z_B = midship section modulus in cm³ at bottom as built
M_S = normally to be taken as the largest design still water bending moment in kNm. M_S shall not be taken less than 0.5 M_{SO}. When actual design moment is not known, M_S may be taken equal to M_{SO}
M_{SO} = design still water bending moment in kNm given in Sec.5 B
M_W = rule wave bending moment in kNm given in Sec.5 B. Hogging or sagging moment to be chosen in relation to the applied still water moment.

B.2 Design loads

B. Design loads

B 100 Local loads on bottom structures

101 Generally applicable local loads on bottom structures are given in Table B1. In connection with the various local structures, reference is made to this table, indicating the relevant loads in each case.

Table B1 Design loads		p (kN/m ²) ¹⁾
Structure		
Outer bottom:	Sea pressure	$p_1 = 10 h_0 + p_{dp}$
	Liquid cargo in tank above	$p_2 = g_0 \rho h_s$
Inner bottom:	Dry cargo in cargo holds	$p_3 = k g_0 \rho_c H_C$
		$p_4 = k g_0 \rho h_s$
		$p_5 = 0.67 (\rho g_0 h_p + \Delta p_{dyn})$
	Liquids in tank above ²⁾	$p_6 = \rho g_0 h_s + p_0$
		$p_7 = \rho g_0 (h_s + 0.3 b)$
		$p_8 = \rho g_0 (h_s + 0.1 l)$
Inner bottom, floors and girders:	Pressure on tank boundaries in double bottom	$p_9 = 0.67 (10 h_p + \Delta p_{dyn})$
	Flooded condition	$p_{10} = 10 h_s + p_0$
		$p_{11} = 10 h_b$

1) For ships with service restrictions, p_3 , p_4 and the last term in p_1 may be reduced as given in Sec.3 B1103.
 2) p_7 and p_8 refer to tank sides and ends, respectively.
 Adjacent structures to be reinforced accordingly.

The pressure p_{dp} is taken as:

$$p_{dp} = p_l + 135 \frac{y}{B + 75} - 1.2 (T - z) \quad (\text{kN/m}^2)$$

$$p_l = k_s C_W + k_f$$

$$= (k_s C_W + k_f) \left(0.8 + 0.15 \frac{V}{\sqrt{L}} \right) \text{ if } \frac{V}{\sqrt{L}} > 1.5$$

$$k_s = 3C_B + \frac{2.5}{\sqrt{C_B}} \text{ at AP and aft}$$

$$= 2 \text{ between } 0.2 L \text{ and } 0.7 L \text{ from AP}$$

$$= 3C_B + \frac{4.0}{C_B} \text{ at F.P. and forward}$$

Between specified areas k_s is to be varied linearly.

- a = 1.0 for ship's sides and for weather decks forward of 0.15 L from F.P., or forward of deckhouse front, whichever is the foremost position
 = 0.8 for weather decks elsewhere
 k = 1.3 aft of 0.2 L from F.P.
 = 1.5 within 0.2 L from F.P.
 h_0 = vertical distance from the waterline at draught T to the load point (m)
 z = vertical distance from the baseline to the load point, maximum T (m)
 y = horizontal distance from the centre line to the load point, minimum B/4 (m)
 C_W = 0.0792 L
 k_f = the smallest of T and f
 f = vertical distance from the waterline to the top of the ship's side at transverse section considered, maximum 0.8 C_W (m)
 h_s = vertical distance in m from load point to top of tank
 h_p = vertical distance in m from the load point to the top of air pipe
 h_b = vertical distance in metres from the load point to the deepest equilibrium waterline in damaged

condition obtained from applicable damage stability calculations. The deepest equilibrium waterline in damaged condition should be indicated on the drawing of the bulkhead in question. The vertical distance shall not be less than up to the bulkhead deck

- H_C = stowage height in m of dry cargo. Normally the height to 'tween deck or top of cargo hatchway is to be used in combination with a standard cargo density $\rho_c = 0.7 \text{ t/m}^3$
- ρ_c = dry cargo density in t/m^3 , if not otherwise specified to be taken as 0.7
- ρ = density of liquid cargo in t/m^3 , normally not to be taken less than 1.025 t/m^3
- p_0 = $0.3 L - 5$ (kN/m^2), minimum 10 generally
 = 25 kN/m^2 in cargo tanks
 = pressure valve opening pressure when exceeding the general value
- b = breadth of tank in m
- l = total length of tanks in m.
- Δp_{dyn} = calculated pressure drop according to Pt.4 Ch.6 Sec.4 K201.

B.3 Stress allowed

Table C1 Values of σ	
	Allowable stress σ ¹⁾
Transverse stiffening:	$60 \frac{Z_B}{Z_R}$, maximum 120 within 0.4 L
	160 within 0.1 L from the perpendiculars
Longitudinal stiffening:	120 within 0.4 L
	160 within 0.1 L from the perpendiculars
1) Between specified regions the σ -value may be varied linearly.	

Appendix C

Hull models

C.1 Length altered 1

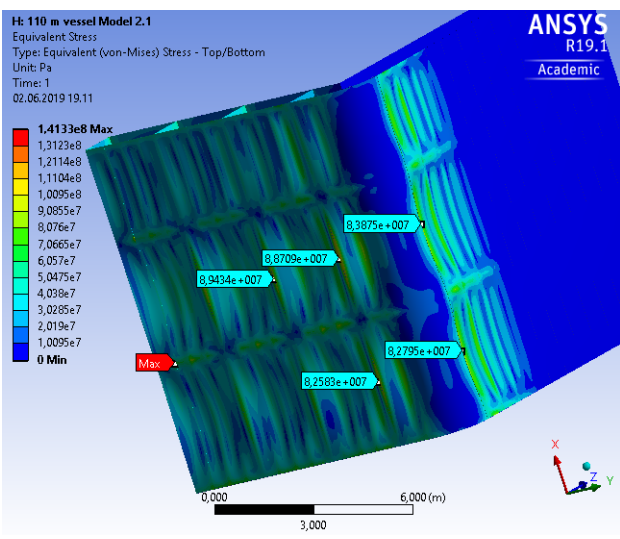


Figure C.1: Stress (Length=11m)

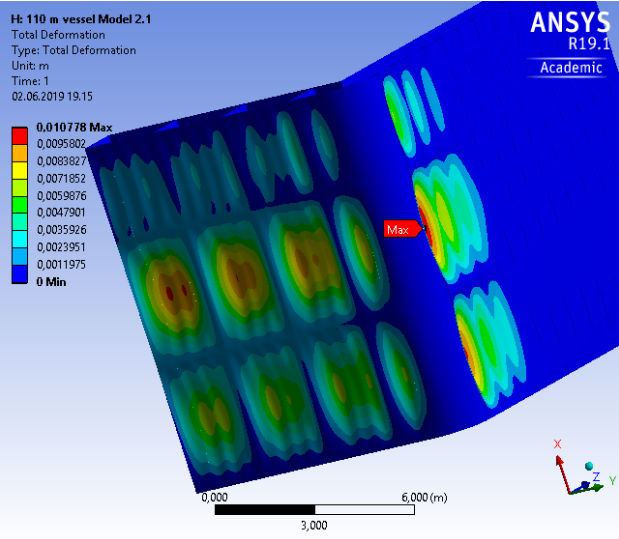


Figure C.2: Deformation (Length=11m)

C.2 Length altered 2

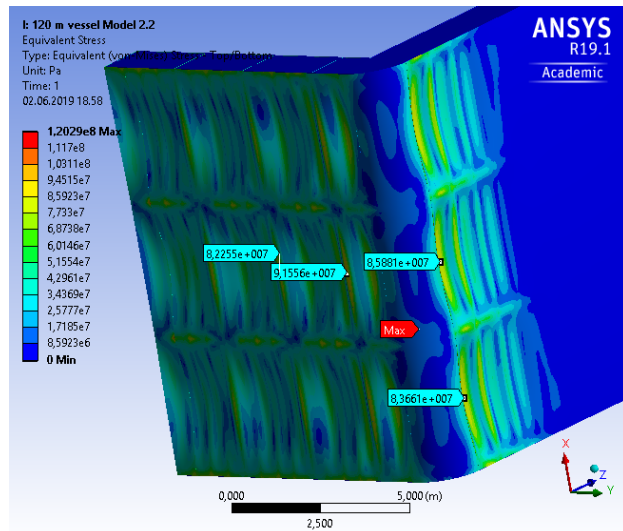


Figure C.3: Stress (Length=12m)

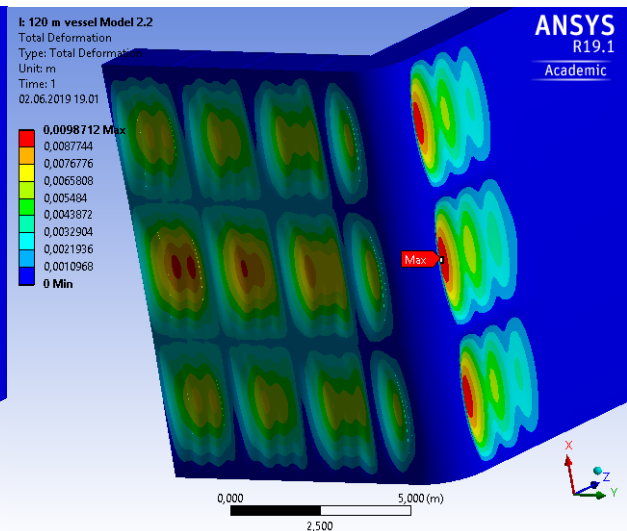


Figure C.4: Deformation (Length=12m)

C.3 Width altered 1

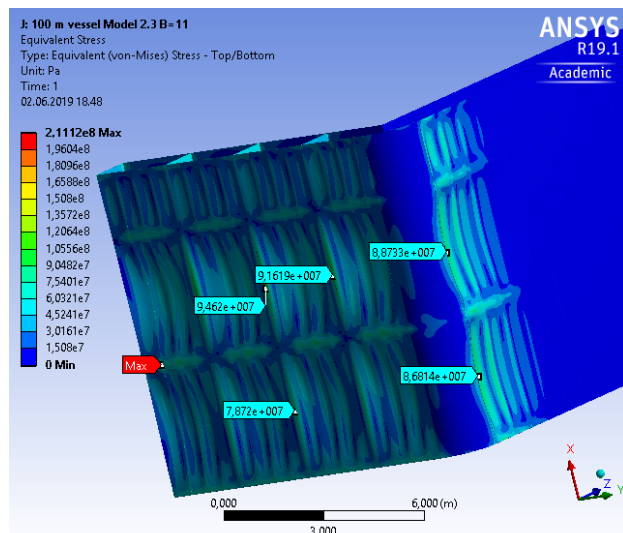


Figure C.5: Stress (Width=11m)

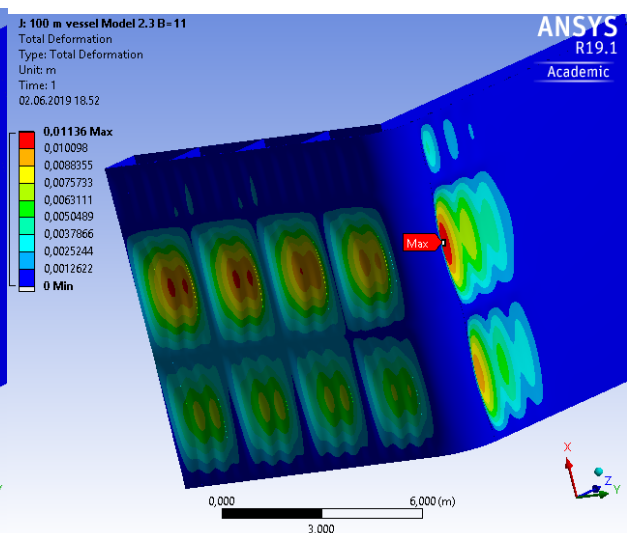


Figure C.6: Deformation (Width=11m)

C.4 Width altered 2

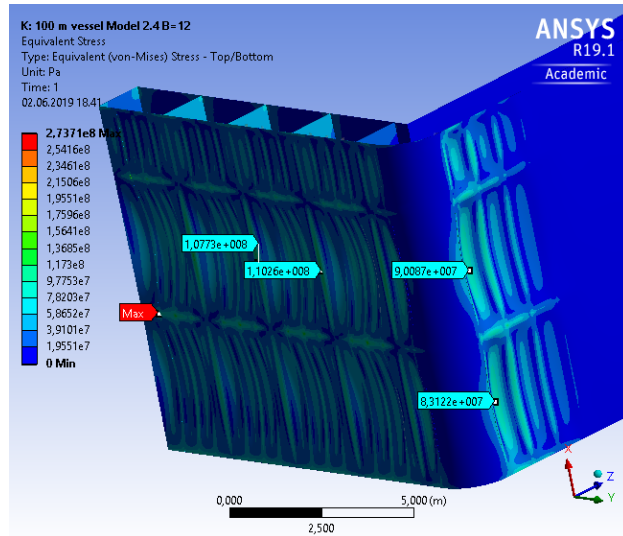


Figure C.7: Stress (Width=12m)

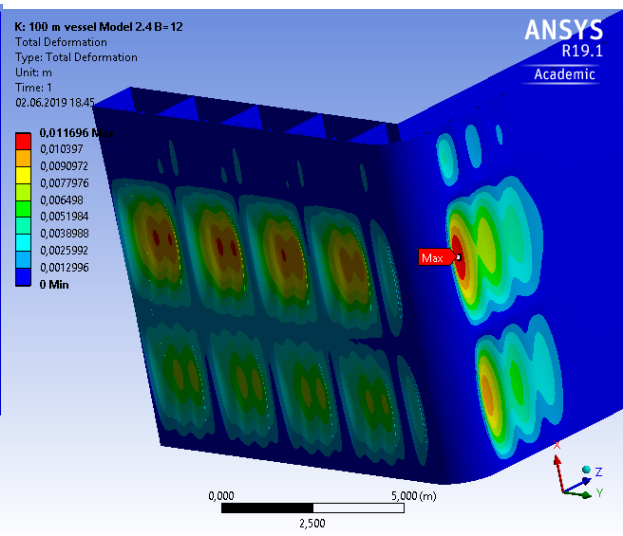


Figure C.8: Deformation (Width=12m)

C.5 Depth altered 1

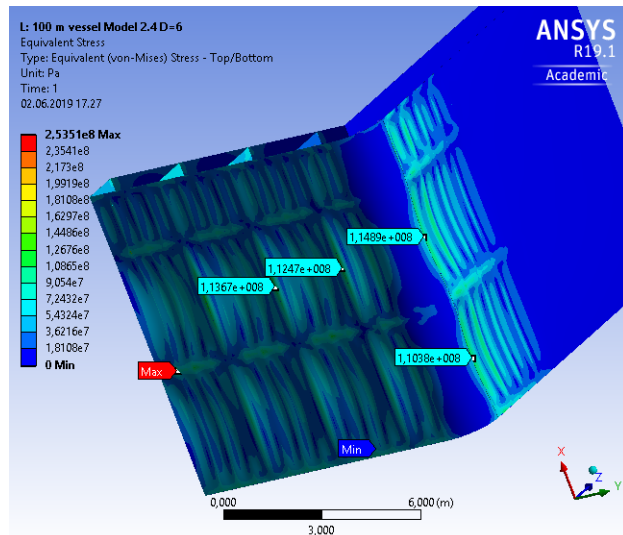


Figure C.9: Stress (Depth=6m)

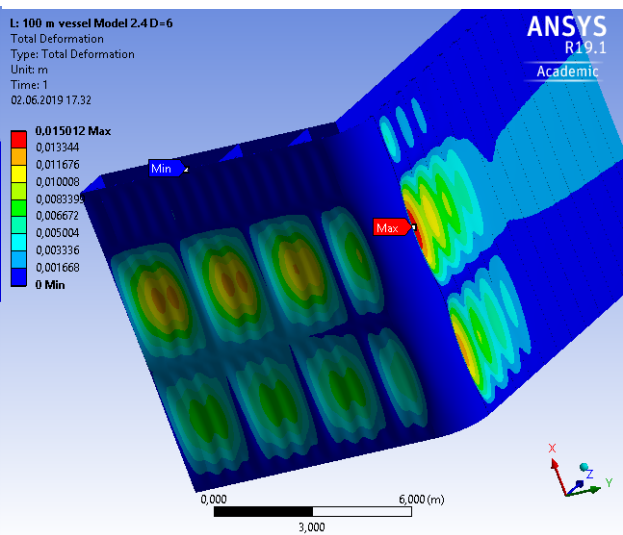


Figure C.10: Deformation (Depth=6m)

C.6 Depth altered 2

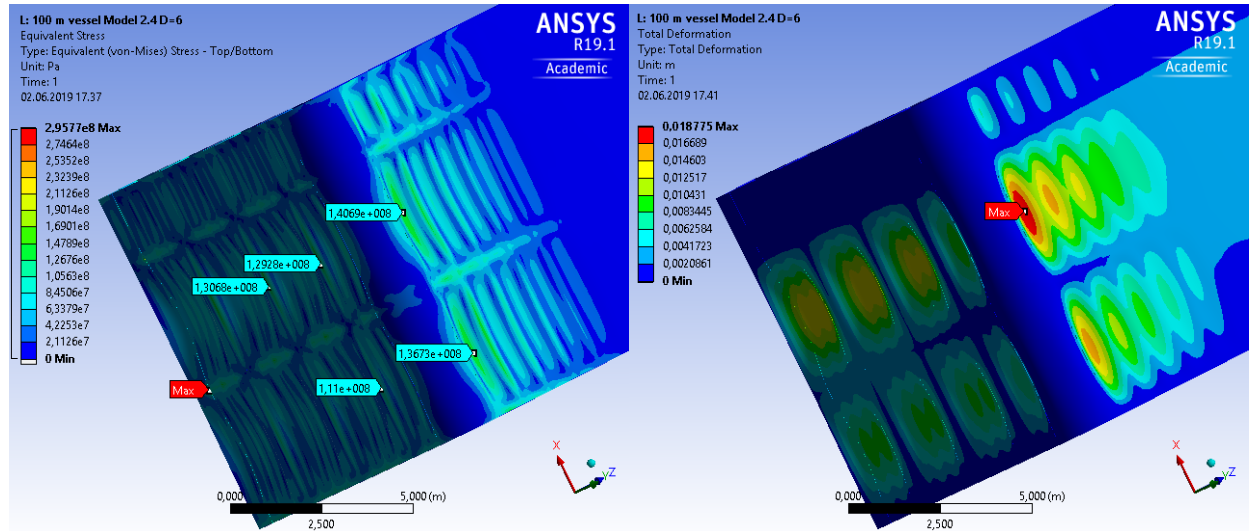


Figure C.11: Stress (Depth=7)

Figure C.12: Deformation (Depth=7m)

Appendix D

Regression tests

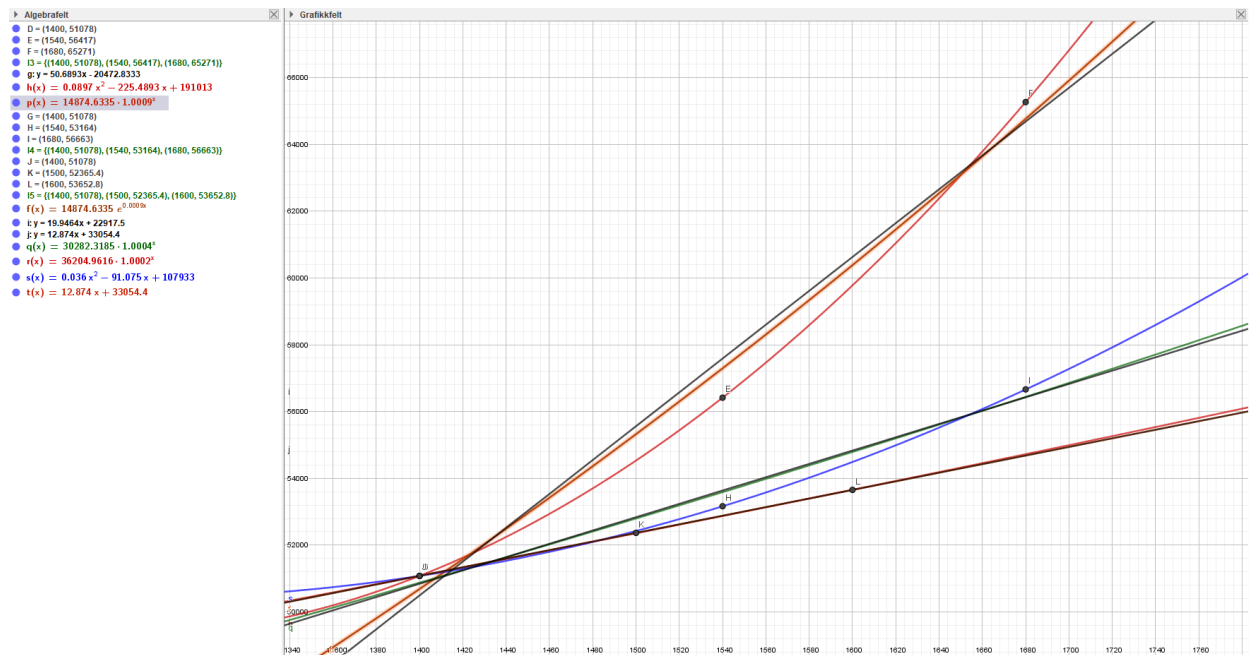


Figure D.1: Regression tests

Appendix E

Ansys student community



torleirh posted this 2 weeks ago

- Quote
- Delete
- Edit
- Attach

Hi,
Hope someone here can help me with a problem described bellow.

Construction:

I have a problem with stress singularities or at least that is what I think it is. I have designed a midsection of a ship hull and strengthened it using stiffeners and girders (see picture 1). The L stiffeners are added as cross sections to line bodies while the girders are added as surfaces with specified thicknesses. The outer plating of the hull is divided using the slice function in DM and was done to create different thicknesses for different parts (keel, walls, etc).

Loads and supports:

Tere are two loads added. One distributed force applied to the inner floor, and a hydrostatic pressure to the outer plating. The hull is simply supported in the two ends, for the whole frame (bottom, walls and girders).

Meshing:

Have tried different mesh sizes and different mesh types (triangular and quadratic). The resulting stress increases as mesh size decreases, as one would expect in a singularity.

Questions:

When testing the structure, to check the equivalent (Von-Mises) stress, I get high peaks (assuming singularities) in the following regions (depending on the mesh size)

Outer plating at the intersection between longitudinal girder and outer plating.

Intersection between keel plating and regular (thinner) bottom plating. This also goes for other regions where two different thicknesses of plates meet.

How can I avoid getting these singularities and get correct results?

And follow up: is it possible to smooth-out a transition from one plate thickness to another. (Tried the blend function in DM unsuccessful)

I also find it weird that the stress maximums change position from one mesh size or type to another.





jj77 posted this 2 weeks ago

Interesting. Which Uni are you studying in?

#Permalink 0 Like 0 Dislike 0 Favourite Is Solution Reply



torleirh posted this 2 weeks ago

NTNU Marine Technology in Norway

#Permalink 0 0 0 Is Solution



jj77 posted this 2 weeks ago - Last edited 2 weeks ago

There are a couple of things.

First make sure it is a singularity so keep refining the mesh and see that the stresses do not change much. If it is changing during the refinement process at this location then it could well be a singul.

If it is one (normally not very common in plates/shells), then just let it yield locally so assign a NL material curve for the steel. If the strains are small then stress will redistribute and it will be OK.

Below is a good discussion on singularities:

<http://www.acin.net/2015/06/02/stress-singularities-stress-concentrations-and-mesh-convergence/>

Have in mind that when designing, the classification societies will dictate how to carry out the FEA, and what mesh size to use, see e.g., (LR will different criteria)

There are also extrapolation techniques (e.g., hot spot) recommended by these societies to get stresses there (ask some professor in your Uni. that has actually done ship or platform design according to DNV on shell structure - he or she should be able to guide you because it is important for structural engineers to understand this practice)

<https://rules.dnvgl.com/docs/pdf/DNVGL/CG/2015-10/DNVGL-CG-0127.pdf>

<https://dmsonline.us/trust-your-singularity/>



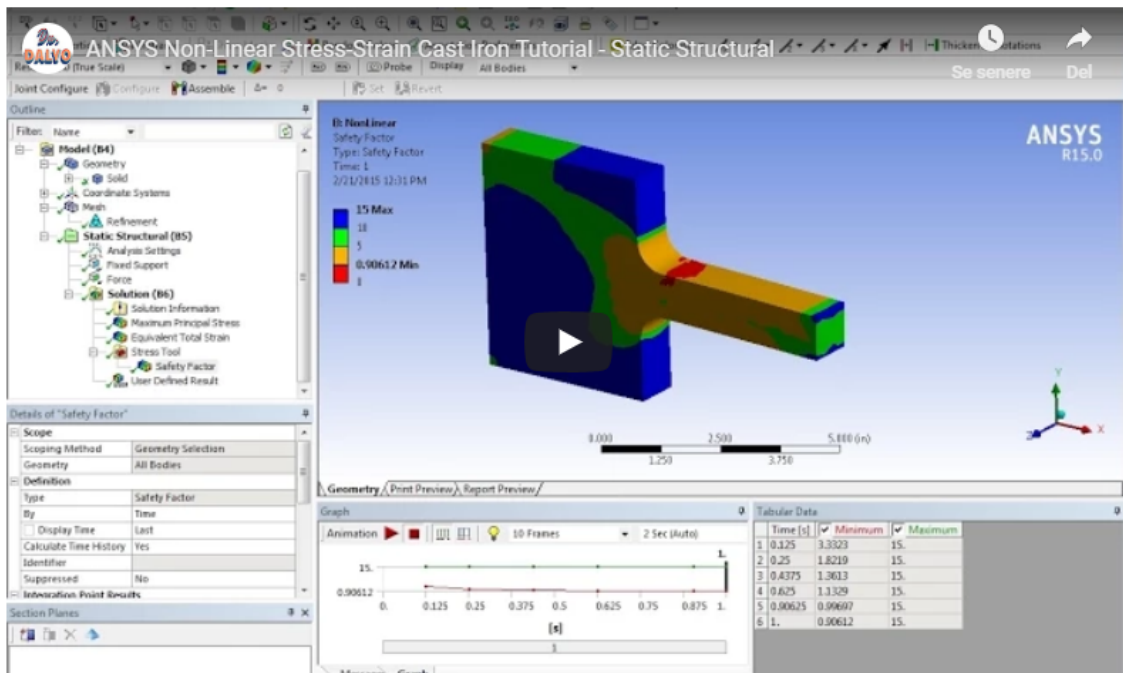
torleirh posted this 2 weeks ago

Hi again,

Thank you so much for replying so fast!

When refining the mesh, I find that the stress increases for the specific point in the intersection between the keel plate and the rest of the bottom structure (intersection between different plate thicknesses). Same point as displayed in the picture added in original post. This has made me to believe that it is in fact a singularity.

When you say to assign a nonlinear (NL) material curve for the steel. How can I do this in Ansys, and can I do it just for specific regions of the structure. I found a video on YT



this how to do it, or is there a way to add a more standardized NL material curve in the Engineering Data folder.

I have been reading up on the DNV GL guidelines for FEM analyses and I try to follow them.



Again, thanks for taking the time to help!

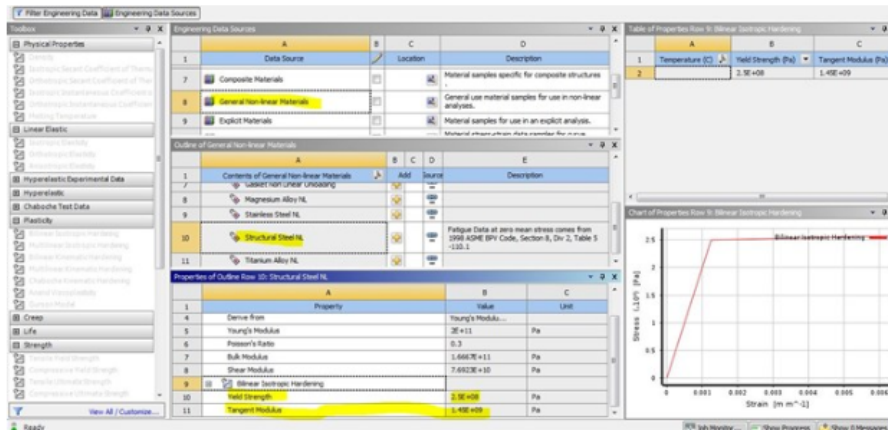


jj77 posted this 2 weeks ago - Last edited 2 weeks ago

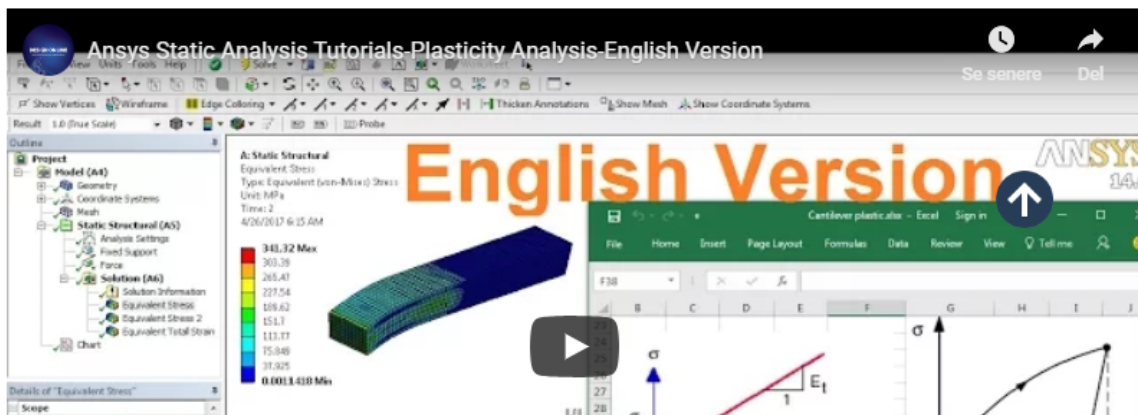
No worries - good that you are looking at DNV - many times also naval engineers know the critical areas so an area like this they will know if it is important or not - I assume that comes with experience and after having designed many ships

The video you have is for nonlinear cast iron which is not what you want.

In engineering data, one can select a bilinear isotropic hardening curve, e.g., go to engineering data sources and nonlinear materials and pick one (StrSteel NL) - one would need to set the yield stress and tangent modulus. See below where it is set to 250MPa. Just press the yellow plus button and it will be added to your material - you can then choose to assign it in Mechanical (For these surface bodies or for all).



See also this video:





torleirh posted this 3 days ago

Well, I'm back again. Beaten by ansys yet another time.

I added NL structural steel for the regions where the singularity occurred (and for the whole structure, just to try), still I got the same high stress peak. Is this indicating that there must be something wrong with the way this is modelled, or do you have any other tricks or tips that might help me solve this problem?

#Permalink 0 0 0 Is Solution



jj77 posted this 3 days ago

It is very likely due to the extrapolation - NL things are enforced on the Gauss Points (GP), but when we extrapolate depending on the mesh, and the stress variation between elements we might go above yield stress.

First check that we actually do get yield and that it is OK. So check plastic strains (results strains).

Then if that is OK, we can look at non extrapolated results - inert a command in solution saying ERESX,NO (this command forces copy of GP values to nodes).

