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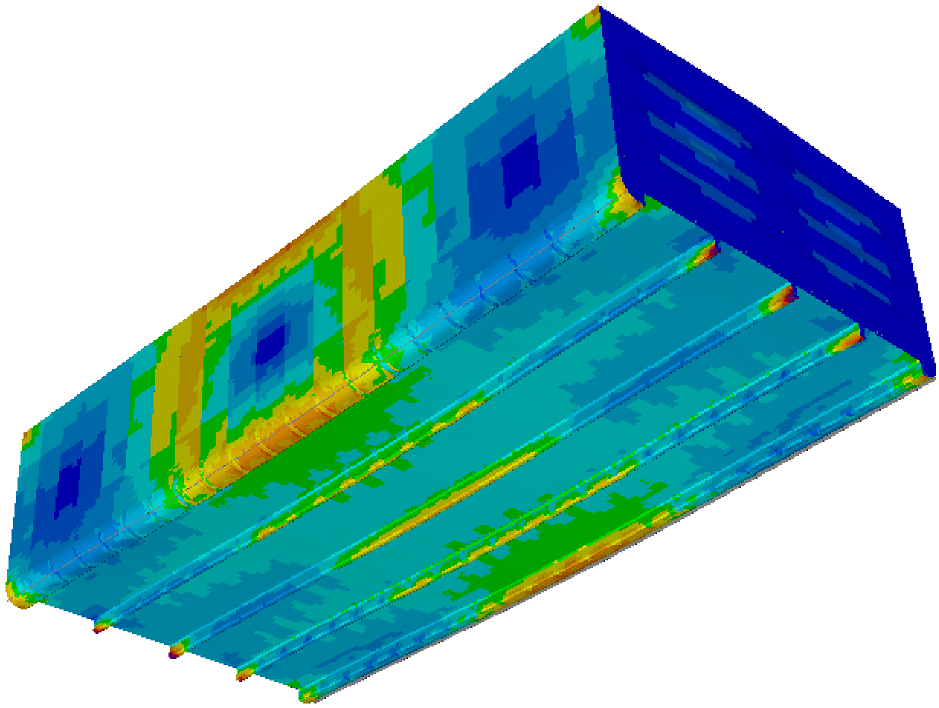

Consequence analysis for Air cavity design

A comparable structural and hydrostatic study between a conventional hull and an Air cavity hull

Master's Thesis in the International Master's Programme Naval Architecture and Ocean Engineering

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Göteborg, Sweden 2015

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FE Analysis of the Air cavity midship section, see chapter 4.4 - structural results.

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ABSTRACT

This thesis is a comparable study between a conventional tanker hull design and a tanker hull design when introducing the "Air cavity" bottom structure. The study is initiated by Stena Teknik to investigate the structural and hydrostatic effects. The concept "Air Cavity", were air pockets are introduced in the bottom structure, holds air kept under the hull so the vessel "floats" on cushions of air. The interface between the water and the air provides a lower viscous friction than the interface between water and the hull. Investigating this alternative hull design is an important step for the continuous work to reduce fuel consumption and environmental impact.

The objective is to identify and evaluate how the two designs relate to each other with respect to cargo loading properties, steel weight, critical structural areas and intact stability. The hydrodynamic data, from were the resistance reduction is taken, is a separate study and will not be covered in this thesis.

The conventional tanker hull have been be modelled to serve as a reference, while the new Air cavity hull design have been modelled from scratch, using the conventional hull as a design platform. The structure of the two designs was evaluated with respect to CSR (Common Structural Rules) as well as with the general stability requirements. A financial indication is done for the two concepts to further understand the impact of an Air cavity design.

The Air cavity design complies with the general stability requirements. Structural results indicate an increased steel weight of 6% and reduced cargo volume of 7% for the midship section. A shift in the stress distribution between the two designs is clearly seen and critical areas around the cavities are identified.

Based on these indicators the efficiency of the Air cavity system could be estimated. This was done with 15% friction reduction including loss for the air compressors/fans and systems for maintaining the air pressure in the cavity. Air cavity related systems and equipment is roughly estimated and included in the cost breakdown.

The bunker savings need to meet the added financial cost, "break-even" is met at bunker reduction of 19%, which equals the frictional reduction 26%.

Key words: Air Cavity design, tanker hull design, hydrostatic analysis, structural analysis.

Konsekvensanalys av luftkavitetsdesign
En jämförande strukturell och hydrostatisk studie mellan ett konventionellt skrov och ett luftkavitetsskrov
Examensarbete inom Naval Architecture and Ocean Engineering
PATRIK MOLANDER
MATHIAS LINDBÄCK
Institutionen för sjöfart och marin teknik
Avdelningen för Marin Teknik
Chalmers tekniska högskola

SAMMANFATTNING

Denna avhandling är en jämförande studie mellan en konventionell skrovdesign av ett tankfartyg och ett tankfartyg med en "Air cavity" bottenstruktur. Studien är utförd på uppdrag av Stena Teknik med fokus på att undersöka strukturella och hydrostatiska effekter. Begreppet "Air cavity", där luftfickor införs i bottenstrukturen, håller luft under skrovet så fartyget ”flyter” på luftkuddar. Gränssnittet mellan vattnet och luften ger en lägre viskös friktion än gränssnittet mellan vatten och skrov. Att undersöka denna alternativa skrovdesign är ett viktigt steg för kontinuerligt arbete att minska bränsleförbrukningen och miljöpåverkan.

Målet är att identifiera och utvärdera hur två designer relaterar till varandra när det gäller lastegenskaper, stålvikt, kritiska strukturella områden och intaktstabilitet. Hydrodynamisk data, varifrån resistansminskningen är hämtad, är behandlat i en separat studie och kommer inte att täckas i denna rapport.

Det konventionella skrovet har modellerats för att vara referens, medan det nya Air cavity-skrovet har modellerats från grunden, med det konventionella skrovet som en designplattform. Designen hos de två skroven utvärderades med avseende på CSR (Common Structural Rules) samt med allmänna stabilitetskrav. En finansiell indikation görs för de två koncepten för att ytterligare förstå effekten av en Air cavity design.

Air cavity design uppfyller allmänna stabilitetskrav. Strukturresultat indikerar en ökad stålvikt på 6 % och minskad lastvolym på 7 % för midskeppssektionen. En förändring i spänningsfördelningen mellan de två utföranden är tydlig och kritiska områden runt kaviteterna har identifierats.

Baserat på dessa indikatorer kan effektiviteten av Air cavity-systemet uppskattas. Effektiviteten beräknades med 15% friktionsminskning vilket även inkluderar förluster kopplat till pump- och fläktsystem. System och utrustning relaterade till luftkaviteten är uppskattade och inkluderade i kostnadskalkylen.

Bunkerbesparingar måste kompensera ökad nybyggndskostand och därmed finansiella kostnader, "break-even" nås vid bunkerminskning om 19%, vilket motsvarar friktionsminskning 26%.

Nyckelord: Hydrostatisk analys, tankerdesign, luftkavitet, struktur analys.

Acknowledgements

We would like to acknowledge and thank our supervisor, Professor of the Practice Bengt Ramne at the department of Shipping and Maritime Technology, for reading the technical report and supporting us through the project to improve our work. We would like to thank our examiner Professor Rickard Bensow. We would also like to thank our co-supervisors, Jörgen Andersson and Jacob Norrby at Stena Teknik, for excellent support and for offering us to study a genuine project. Last we, Mathias Lindbäck and Patrik Molander, would like to thank each other for a good co-operation.

Contents

Preface

This thesis is a part of the requirements for the master's degree at Chalmers University of Technology. It is carried out with support by the division of Marine technology at Chalmers together with Stena Teknik, Göteborg, who has offered an actual project to be studied.

The concept of Air Cavity design embraces a broad perspective of aspects, in order to indicate whether or not the concept would be feasible for further design process, we have penetrated deeper into some areas but also maintained a broad perspective, which we have come to appreciate.

Göteborg, 2015

Patrik Molander & Mathias Lindbäck

Abbreviations

BC	Boundary Conditions
CAD	Computer-Aided Design
CAPEX	Capital Expenditures
CSR	Common Structural Rules
deg	degrees
DNV	Det Norska Veritas
DWT	Dead Weight Tonnage
FWD	Forward
IMO	International Maritime Organization
IS	Intact Stability
m.t.	Metric ton
NA	Neutral axis
NH	Nauticus Hull
OPEX	Operating Expenditures
rad	radians

Notations

Roman upper case letters

A_{Flange}	Area of flange $[m^2]$
B_0	Centre of buoyancy
B_c	Block coefficient
B_c	Breadth in cavity [m]
B_{tank}	Breadth tank [m]
C_{wv}	Wave coefficient
d_c	Height in cavity [m]
L_c	Length cavity [m]
L_{tank}	Length tank [m]
P_c	Pressure in cavity [Pa]
P_{hys}	Hydrostatic sea pressure [kPa]
T_c	Draught in cavity [m]
T_{sc}	Scantling draught [m]
Z_{v-min}	Minimum section modulus $[m^3]$

Z_{v-req}	Required section modulus [m^3]
Z_x	Section modulus [m^3]
FE	Finite Element
G	Centre of gravity
GZ	Righting arm
K	Keel
L	Length [m]
LC	Load Case
M	Metacentre
P	Port
S	Starboard
T	Draught [m]
V	Immersed volume

Roman lower case letters

I_x	Moment of inertia [m^4]
p_0	Reference pressure [Pa]
y_1	Distance from neutral axis to flange [m]
b	Breadth[m]
d	Draught
g	Gravity [m/s^2]
l	Length [m]
p	Pressure [Pa]
z	Depth [m]

Greek case letters

ρ_{cargo}	Density cargo [kg/m^3]
ρ_{sea}	Density sea [kg/m^3]
ρ_{tank}	Density tank [kg/m^3]
ρ_{water}	Density water [kg/m^3]
γ	Density [kg/m^3]
ϕ	Angle of heel [deg]

1 Introduction

This thesis is a comparable study between a conventional tanker design and a tanker design when introducing air pockets in the bottom structure. The air pockets provide lower friction, which reduces the fuel consumption. The introduction of air pockets in the bottom of the hull leads to a change in the structural design of the hull. Understanding this unconventional structural design is an important step to realise the “Air cavity ship” and thereby reduced emissions.

The study is initiated by Stena Teknik to investigate the structural and hydrostatic effects of the Air cavity design. In this chapter the background, objective and the limitations for this thesis are accounted for.

1.1 Background

The world economy is dependent on large-scale transport, where waterborne transport represents a major part, (international transport forum, 2001). An important challenge to make maritime transport a long-term sustainable alternative is to reduce fuel consumption and environmental impact.

Resistance in the water is a critical factor, which further is subdivided into two major components, wave resistance and viscous resistance. In case of a tanker ship or equal hull design, the viscous resistance represents a major part of the two components (Ship resistance and Propulsion, 2011).

Stena, Chalmers and SSPA have investigated the possibility of introducing an alternative bottom design of the hull to reduce the viscous resistance. The concept is called "Air Cavity", where air pockets are introduced in the bottom structure, keeping the air under pressure. The water surfaces inside the air pockets substitute the hull plating, the interface between the water and the air provides a lower friction than the interface between water and the hull; this reduces the viscous resistance in the water.

Calculations and model tests, initiated by Stena Teknik, Chalmers and SSPA, have been carried out which show that the Air cavity design reduces resistance. A major challenge in such a hull design is to find a sustainable design that simultaneously meet the general design conditions that class societies and other generally accepted rules prescribe.

1.2 Objective

A conventional Suezmax tanker hull will be evaluated and compared with a Suezmax tanker with air cavity bottom structure. The conventional tanker hull will be computer modelled to serve as a reference, while the Air cavity hull will be modelled from scratch, using the reference ship as a platform. The two designs must be evaluated with respect to CSR (Common Structural Rules) as well as with the general stability requirements. The objective is to investigate and evaluate how the designs relate to each other in terms of intact stability, load capacity, steel weight and critical structural areas for different load cases. Based on these indicators the minimum efficiency of the

Air Cavity system will be estimated, i.e. the minimum efficiency improvement of such a system will be estimated with respect to economical impact.

1.3 Limitations

The thesis covers a broad perspective and the scope in its initial stage is wide. Limitations are established in order to define the work with respect to the prerequisites.

- This thesis covers 0.3L to 0.7L midships section structural and hydrostatic design and evaluation.
- This thesis does not cover changes in spacing of the web frames or outer main dimensions.
- The study is only taking normal operation conditions of the vessels into consideration. Special loading or strength cases such as collisions of fatigue loads are not included in this thesis.
- Ship specific details such as doors, hatches and other irregularities normally placed in connection to specific web frames are not taken into consideration. This means that the web frames under consideration are simplified before being analysed.
- The study will be conducted with CSR rules and 2008 IS code for initial stability, other rules or regulations are not considered in this thesis.
- Design for manufacturing will not be evaluated in this thesis.
- Only Von Misses stresses are analysed.
- Assuming a free liquid surface theory can be applied in the cavity.
- The financial implications cover costs, not earnings. This is based on the assumption that most shipments do not trade in a fully loaded condition.
- The propulsive efficiency as well as the affected hydrodynamics due to the cavity introduction are not accounted for. It is further assumed that the component of frictional reduction is directly related to fuel consumption.
  Fuel consumption for air supply in order to maintain the air cavity is not included.The two models were modelled with the following simplifications, this since the designs are to be compared with the first lop in a design phase.

- No stiffeners on the web frame were modelled.
- No stiffeners on the stringer planes were modelled.
- Air and drain holes were excluded.
- The corrugated transverse bulkheads were modelled as flat plates.
- No corrosion addition was added for comparison it is not necessary, same in both cases in accordance with CSR
- Systems and equipment for the air supply to the cavities are not studied or considered in this paper.

2 Background and Theory

This chapter introduces the most important theory that the report is based on. First the concept of the Air cavity design is introduced and the main particulars of the reference ship are presented. This is followed by basic structural and hydrostatic theory.

2.1 Concept of Air cavity

The concept of Air cavity is to reduce the hull resistance by introducing an air pocket, a cavity, in the bottom of a vessel. Underneath the bottom, the air-to-water interaction provides a lower friction than a traditional hull-to-water interaction. In turn this system decreases fuel consumption but on the drawback the application occupy space, which theoretically reduces space for payload, assuming unchanged main dimensions, length, width and draught. See figure 2.1.1 for illustration.

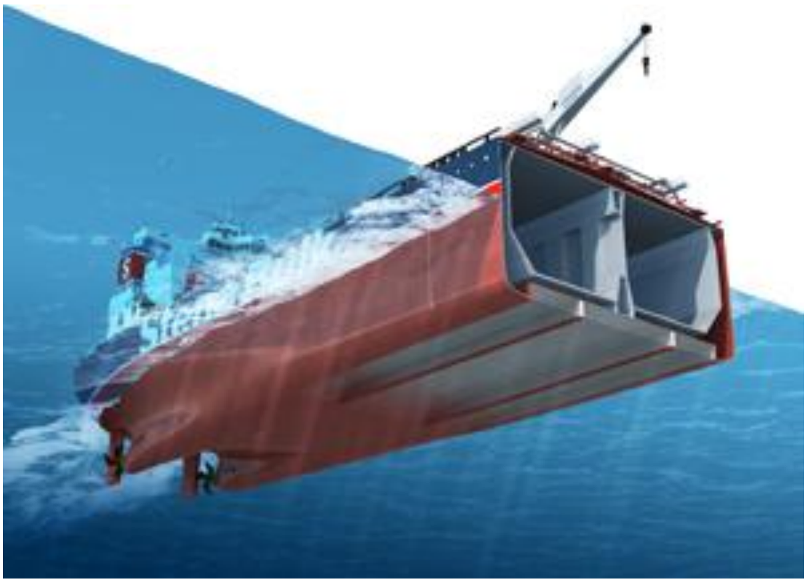


Figure 2.1.1 Illustration of Air Cavity

The air pressure in the cavities equals the pressure at the present deepest draught of the vessel. This means if more air is pumped in to the cavities, i.e. the pressure is increased leading to leakage of air. Design implications consider major parts of the ship. Structural arrangement and hydrostatic properties are covered in this thesis, but hydrodynamic stability and general arrangement for pumps, piping etc. is also affected, which is not discussed.

2.2 Reference ship

This thesis compares two different hulls, reference ship and Air cavity ship. The ship referred to as Air cavity ship is identical to the reference ship except for the midship section structural design. The reference ship is a Suezmax tanker, which is an existing design, built in 2010. The ship is subjected to the CSR (Common Structural Rules), which is a common set of rules related to the structural design. See table 2.2.1 for main data.

Table 2.2.1 main dimensions for the reference ship (Suezmax tanker)

Length over all	274 m
Length between perpendiculars	264 m
Width over all	48 m
Design draught	16 m
Summer DWT	158000 ton
Lightweight	24200 ton

The reference ship is built with a conventional tanker design; it has double bottom and double wall, see figure 2.2.1. The chosen section for the analysis is the midship section, between the web frames situated at 0.3L and at 0.7L. Frame spacing in this region is 4800 mm and watertight bulkheads are positioned between every cargo space, in this section at frame 63, 70, 77 and 84. The positions are visualised in Figure 2.2.2. The fore and aft ship is left unchanged in this thesis.

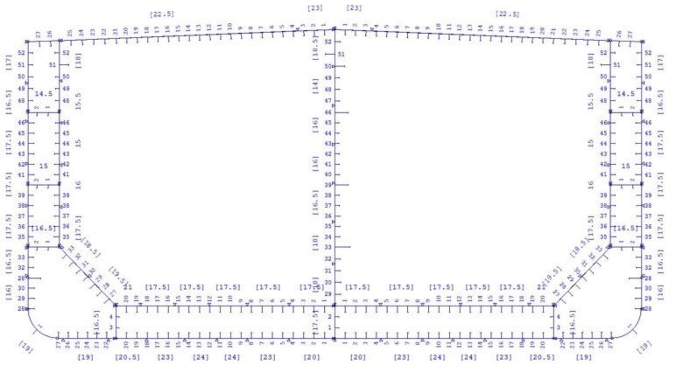


Figure 2.2.1 Cross section reference ship (Suezmax tanker)

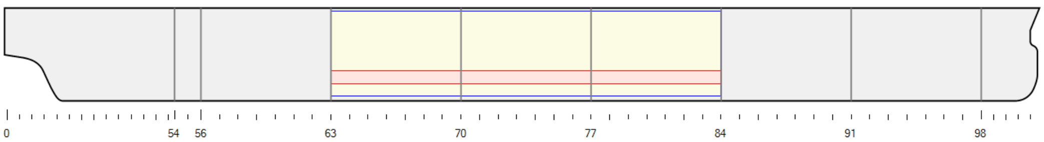


Figure 2.2.2 Stena Suezmax tanker – section of interest, position of frames and watertight bulkheads

2.3 Structure

During the lifetime of a ship the hull structure will be subjected to a significant amount of forces, the forces can be static or dynamic or a combination of both. For a ship in still water the static forces will be the dominating. The static forces from cargo loads and uneven structural distributions will contribute to an uneven weight distribution on the hull. Depending on where and how large these forces are and how they are distributed, the hull will be deformed in a sagging or hogging condition. (David, 2006).The ship will also be subjected to dynamic forces, when the ship is moving, waves will interfere with the hull, depending on if there is a crest or a trough the pressure distribution over the hull will shift. Dynamic forces from cargo and other fluids in tanks will also have an impact on the hull structure.

See figure 2.2.1 for illustration of sagging and hogging condition.

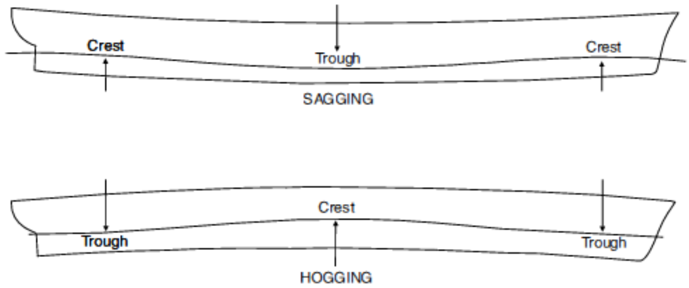


Figure 2.2.1 Sagging and Hogging of a hull. (David, 2006)

A ship is a complex structure with different types of structural elements working together to make the structure as light and strong as possible. Structural elements such as longitudinal bulkheads, transverse bulkheads and web frames are all stiffened longitudinally or transverse with stiffeners. When analysing the structure of a ship, basic beam theory can be applied. The hull girder can be seen as one single beam, this is illustrated in Figure 2.2.2. The hull girder in Figure 2.2.2 is in a sagging condition showing how the stresses are distributed when locking on it globally. (DNV, 2013)

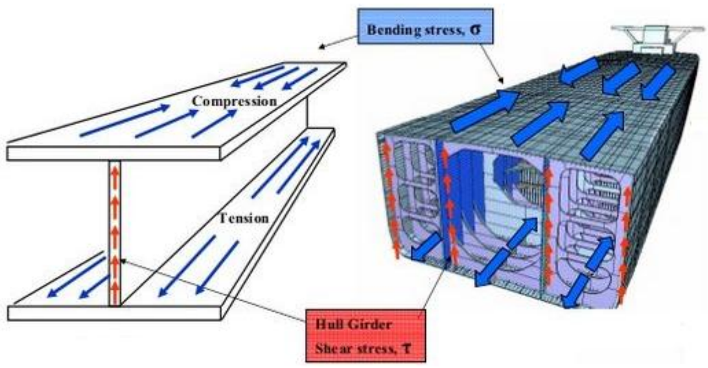


Figure 2.2.2 The hull girder as one single beam during sagging (DNV, 2013)

Depending on how the material is distributed the position of the neutral axis will shift and so also the stress distribution over the cross section of the beam. (DNV, 2013). The shift of stress distribution depending on the position of the neutral axis of a single beam can be seen in Figure 2.2.3.

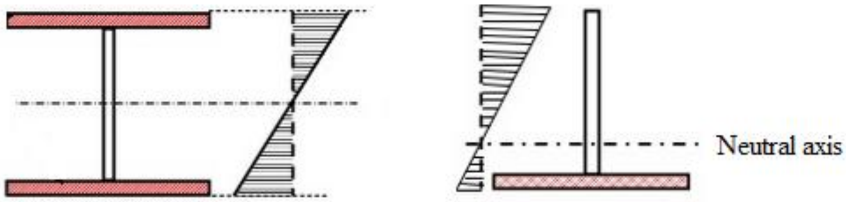


Figure 2.2.3 Neutral axis and stress distribution (DNV, 2013)

When adding or removing material over the cross section of the beam not only the neutral axis will be moved, the moment of inertia, Eq. 2.2.1, and the section modulus, Eq. 2.2.3, of the beam will also be affected. The section modulus can be seen as the beams ability to withstand bending and is therefore an important parameter when designing a hull girder. (DNV, 2013)

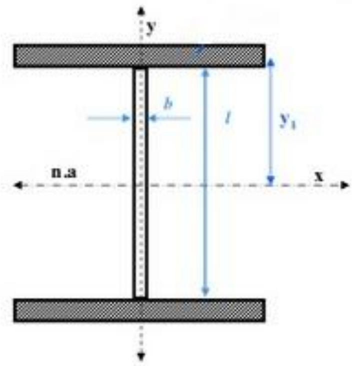


Figure 2.2.4 Dimensions of a beam with two flanges (DNV, 2013)

$$Moment\ of\ inertia\ I_x = \frac{1}{12}bl^3 + 2A_{flange}y_1^2 \qquad \text{(Eq 2.2.1)}$$

$$Section\ modulus\ Z_x = \frac{I_x}{y_1} \qquad \text{(Eq 2.2.2)}$$

2.4 Hydrostatics

This chapter presents the fundamental theory that the method in chapter 3.1 is based on. For a more detailed theory see reference literature, (Brian et. al. 2014).

Archimedes principle describes the theory of how a ship floats in the water. A body immersed in a fluid subjected to an upward force equal to the weight of fluid the body displace. The theory is further explained in figure 2.4.1 and equations 2.4.1-2.2.5.

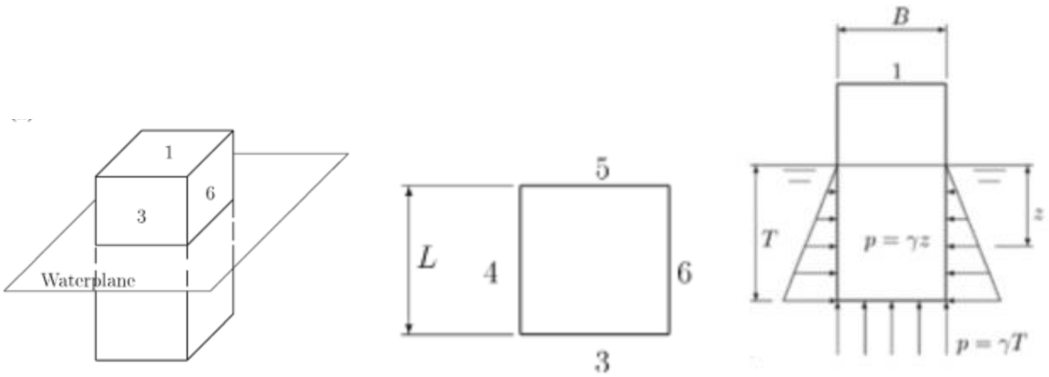


Figure 2.4.1 Illustrating a symmetrical box immersed in a fluid.

Assume positive to right direction:

$$F_4 = L\int_0^T \gamma z dz + p_0 LT = \frac{1}{2}\gamma LT^2 + p_0 LT \qquad \text{(Eq.2.4.1)}$$

$$F_6 = -L\int_0^T \gamma z dz + p_0 LT = \frac{1}{2}\gamma LT^2 + p_0 LT \qquad \text{(Eq.2.4.2)}$$

Conclude F4 and F6 cancel each other.

Forces on top and bottom, 1 and 2, is derived by:

$$F_1 = -p_0 LB \qquad \text{(Eq.2.4.3)}$$

$$F_2 = P_0 LB + \gamma LBT \qquad \text{(Eq.2.4.4)}$$

$$F_1 + F_2 = \gamma LBT = Volume\ of\ the\ immersed\ body \qquad \text{(Eq.2.4.5)}$$

Static stability is often illustrated with help of a GZ curve, which describes the relation between the angle of heel and the righting moment of the hull. With help of figure 2.4.2 a GZ curve can be derived. In figure 2.4.2 M is metacentre, G is the centre of gravity, B centre of buoyancy, K keel, Φ angle of heel, BΦ the new centre of buoyancy and Z is located on the line between M and BΦ to illustrate the distance between G and Z. GZ is the length of the righting arm.

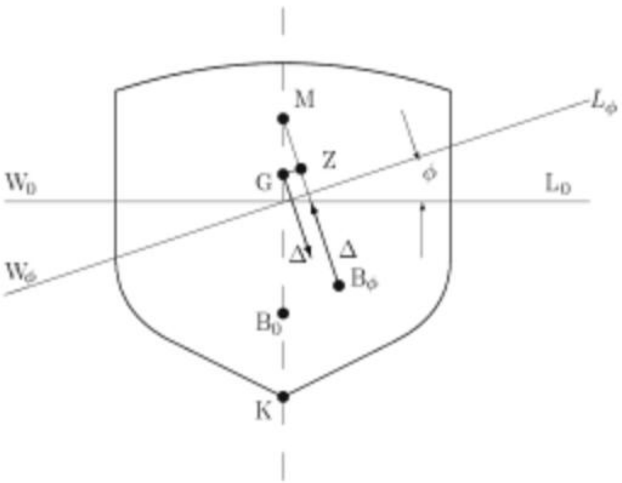


Figure 2.4.2 Righting arm GZ for small angle of heel

From figure 2.4.2 a GZ curve is derived, see figure 2.4.3. GZ curve is used to describe the static stability; from the curve several important stability parameters can be identified. Maximum GZ is the value where GZ reaches its maximum, in figure 2.4.3.

about 50 deg. Another important value is where GZ crosses zero, corresponding angle of heel is called “angle of vanishing stability”. GM, initial stability can be calculated from properties in figure 2.4.2 or estimated from the GZ plot in figure 2.4.3. GM indicates the initial stability, se equations 2.4.6-2.4.8.

$$BM = \frac{I}{V} = \frac{Moment\ of\ inertia}{Immersed\ volume} \quad \text{(Eq.2.4.6)}$$

$$KM = KB + BM \quad \text{(Eq.2.4.7)}$$

$$GM = KM - KG \quad \text{(Eq.2.4.8)}$$

In the GZ curve of static stability, at the heel angle 1 rad (approx. 57.3 deg.), a line will be drawn vertically, and the length will be equal to the GM derived from figure. 2.4.2, see eq. 2.4.8. Further a line will be drawn from the origin of the coordinate system the top of the line; this line is tangent to GZ curve. A ship with positive GM is said to be stable.

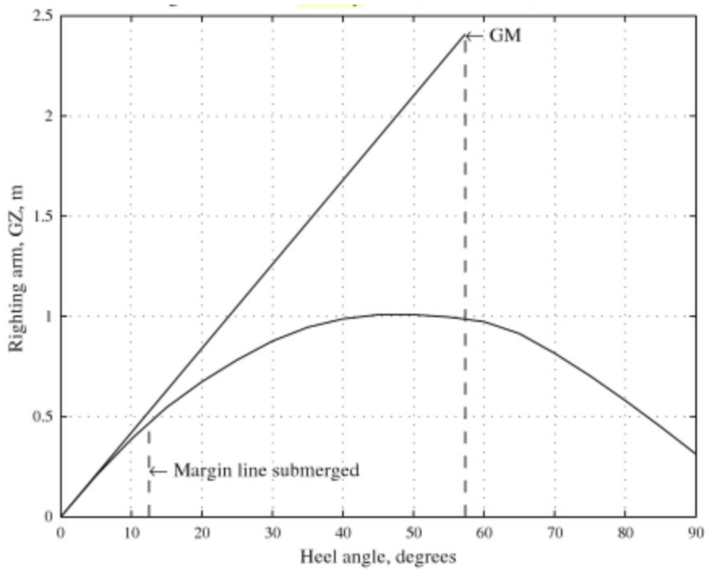


Figure 2.4.3 GZ curve, GM indicated in plot

The free liquid surface effect occurs if the ship is having a partially filled tank, also called slack tank. This will endanger the stability properties. When a ship is heeled the liquid will flow to the lower part of the tank, shifting the centre of gravity of the liquid, as well as for the ship. The stability properties will have to be corrected for this effect.

When the ship is heeled the underwater volume will change shape, which means the centre of buoyancy will move from B to B1. The liquid centre of gravity will move from g to g1, which in turn means the total centre of gravity will move from G to G_1. This means the distance decrease from GZ to G_1Z and GM is reduced to GvM, see figure 2.4.4.

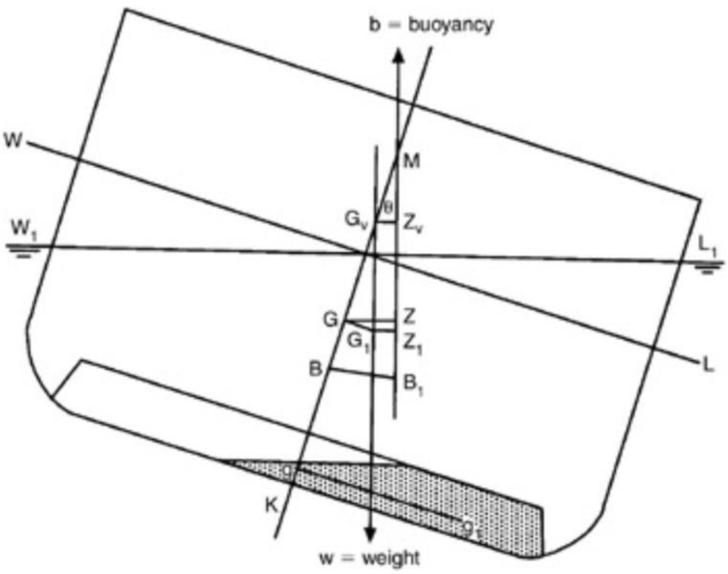


Figure 2.4.4 free liquid surface effect

The initial stability can be compensated for the free liquid surface effect, see eq. 2.4.9, where GM_solid is the initial stability without the free liquid surface effect, however with the same total weight.

$$GM_{new} = GM_{solid} - GM_{corr} = GM_{solid} - \frac{\frac{L_{tank}*{B_{Tank}}^{3}*\rho_{tank}}{12}}{displaced\ volume*\rho_{sea}} \qquad \text{(eq. 2.4.9)}$$

If more tanks are introduced in parallel to the ship x-axis, the structure centreline will have to be considered. Since the theory compensates for shifted centre of gravity, the same theory applies for shifted centre of gravity in the air cavities, see figure 2.4.10.

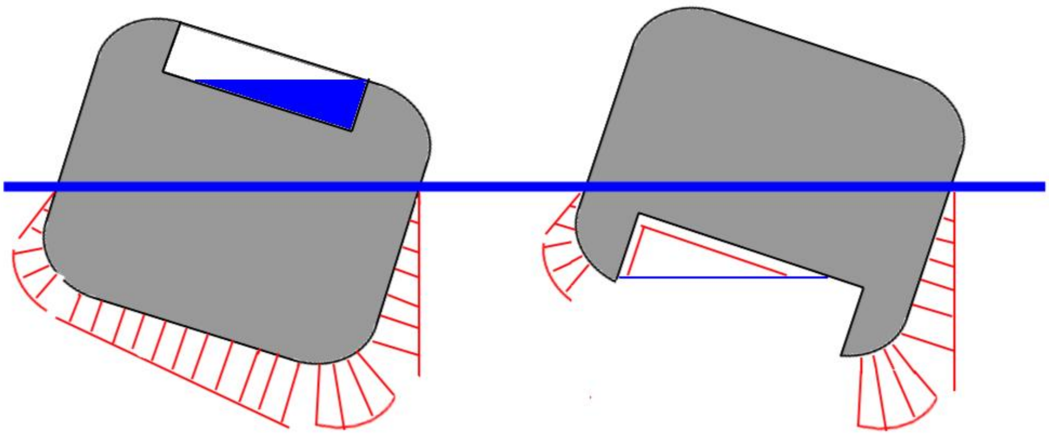


Figure 2.4.10 free liquid surface effect for tank and cavity, centre of gravity is equally moved

3 Method

In the following chapter, the work process and the method used for the comparative study will be presented. The work process starts with hydrostatics, followed by structural elements and the identification of critical structural areas.

Two models were created, one for the reference ship and one for the cavity ship. To make the models comparable, they were modelled to the same level of detail following the CSR (common structural rules).

The main software used to create and analyse the models were Rhino, Maxsurf, Nauticus Hull and GeniE. Rhino was used to create the hull shape from given poly-curves. With the hull created in Rhino, the hydrostatics for the cavity ship could then be analysed in Maxsurf. Nauticus Hull and GeniE were used to design and analyse the structure of the reference and the cavity ship, critical structural areas were identified both for rule based load cases and for docking.

All the different steps within the work process will be explained more in detail further on in the chapter.

3.1 Hydrostatics

In this chapter a new draught and scantling draught are derived for the Air cavity design. Further the hydrostatic properties are calculated, using computer software. There are general and specific requirements for stability, this is regulated in the 2008 IS Code. Additional loading condition is regulated by MARPOL.

Draught, T, is adjusted by cavity air pressure. In optimum operational condition the cavity submersion draught (Tc) equals zero, however this is hard to reach because small heel disturbances allows air to escape.

Scantling draught, Tsc, is the maximum design draught, at which the strength requirements for the scantlings of the ship are met. i.e. the maximum structural draft for the vessel, (DNV, 2010a). Maximum design draught is valid when there is 0% air (100% water) in the cavities combined with maximum load.

3.1.1 Draught

Model tests have been made from where the pressure difference between Po and Pc, figure 3.3.1, is measured to 100 Pa and 200 Pa, (Shiri et. al., 2012). The draught in the cavity model is calculated, derived from the pressure difference and scaled to the full size cavity application. See fig 3.1.1 and fig 3.1.2, equations 3.1.1 and 3.1.2.

$$P_0 = P_c + \rho g T_c \quad \text{(Eq. 3.1.1)}$$

$$P_0 = \rho g d \quad \text{(Eq. 3.1.2)}$$

Where d= hull draught

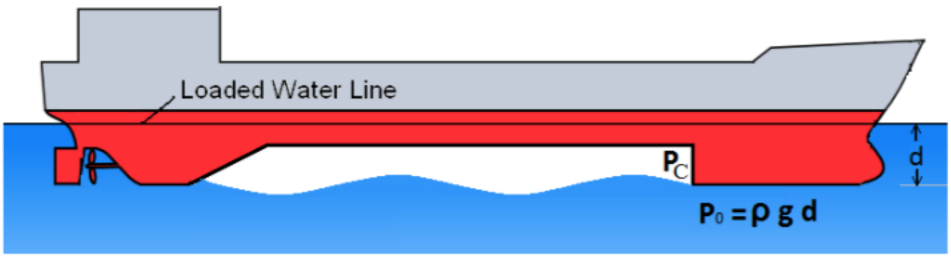


Figure 3.1.1 Illustration of pressure inside and outside of cavity

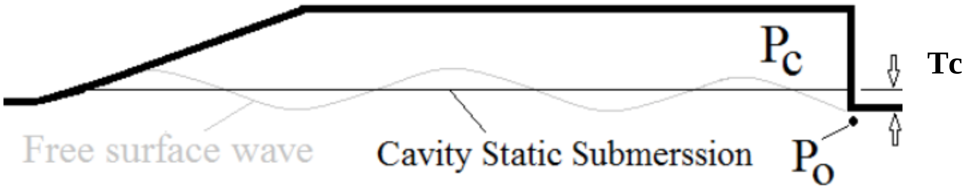


Figure 3.1.2 Illustration of pressure inside and outside of cavity

3.1.2 Scantling draught

Scantling draught refers to the maximum draught the ship is designed for from a structural point of view. The scantling draught for the cavity ship is calculated by substituting the loss of cargo space with the loss of buoyancy compared to the reference ship. This means the volume that the cavities occupies from the cargo volume will, in worst case, be filled with water and the scantling draught will increase compared to reference ship for the same cargo intake. The added difference of mass is based on subtraction of the cargo weight and addition of the weight of water, there is a net difference. Equation 3.1.3-3.1.6.

$$Volume_{cavity} = Lc * Bc * dc \qquad \text{(Eq. 3.1.3)}$$

$$\Delta m = Volume_{cavity}(\rho_{water} - \rho_{cargo}) \qquad \text{(Eq. 3.1.4)}$$

$$m_{ship} + \Delta m = L * B(d + \Delta d) * C_b * \rho_w \qquad \text{(Eq. 3.1.5)}$$

$$\Delta d = \frac{\Delta m}{L*B*C_b*\rho_{water}} \qquad \text{(Eq. 3.1.6)}$$

3.1.3 Intact stability – Calculation and rule evaluation

The purpose to evaluate the intact stability is to determine if the given Air cavity design meets the stability requirements. The stability properties are of importance because this is a fundamental step in the design phase. If the stability requirements can't be met, the initial design needs to be reconsidered. The intact stability properties of the conventional tanker design and the air cavity design will be compared and evaluated with respect to common stability requirements.

Computer calculations are performed to investigate how the designs affect the GZ-curves. Further the GZ curves are compared with common stability requirements to find if the Air cavity design meet the requirements and can be further developed. The computer software also generates actual structural hull girder loads for different loading conditions, which may differ from structural rule loads, used for structural design.

Two main software are used, Rhino and Maxsurf. Rhino provides tools for modelling curves, which represent the hull design, later used by Maxsurf stability. Maxsurf stability provides tools to evaluate the intact stability and compare the results to general requirements. Maxsurf will also generate “actual” global hull girder loads, later used to dimension the hull girders.

In order to model the cavities they are substituted with tanks with corresponding dimensions and free liquid surface in the tanks according to theory in chapter 2.3.

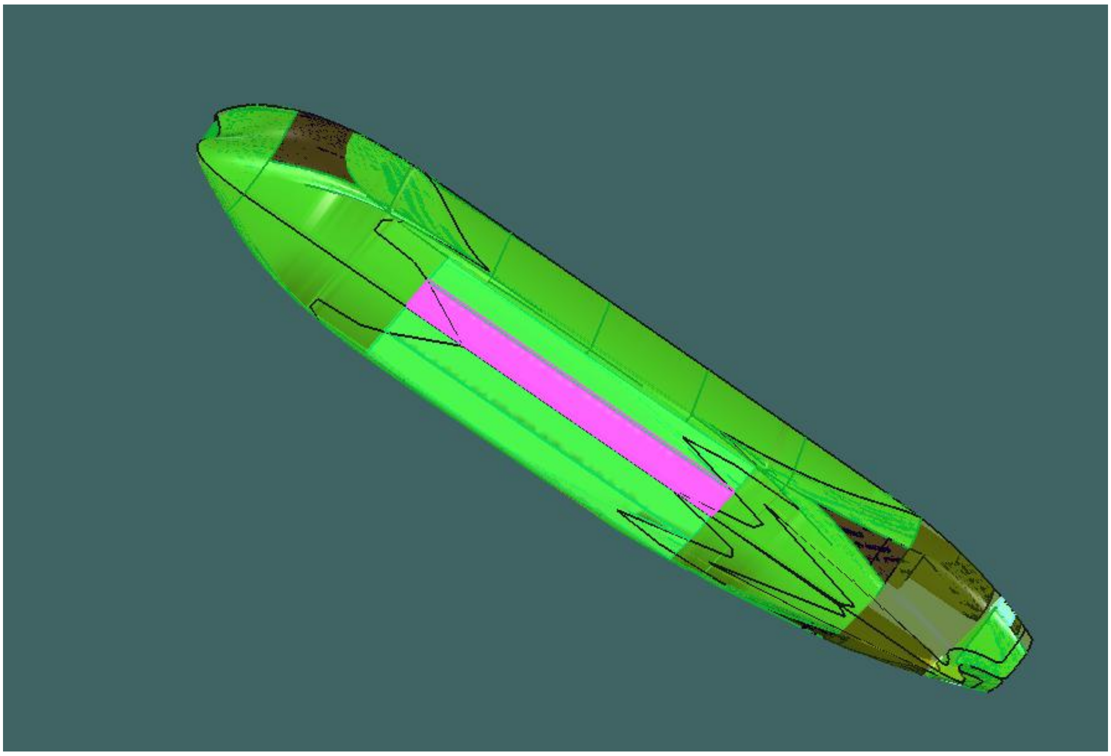


Figure 3.1.3 Hull from below, Air cavities

Requirements for intact stability for tanker ship is given by IMO (2008a), see table 3.1.2. Three load cases are tested, full cargo tanks and full ballast tanks and according to MARPOL regulation 27 the most severe free liquid surface loading condition, common referred to as “slack tanks”, (MARPOL, 2012). See table 3.1.1.

According to 2008 IS code all test should be performed for departure and arrival condition, this will have impact on the level of fuel tanks. In this thesis the fuel tanks are considered small, therefore full fuel tanks are assumed for all load cases, i.e. only three loading conditions are investigated.

The lightship weight is represented by evenly distributed local point loads every meter along the vessel.

It is recommended by IMO (2008b) that no tanks are considered empty due to the risk of free surface. Cargo tanks and ballast tanks are loaded to level of 1% minimum for all tests.

For the Air cavity hull the cavity draught, Tc, is 0.2 m during normal operation. However, when the hull is heeled the air will increasingly escape and be substituted with water, the total draft increases. The three most relevant stability cases are investigated. Initial stability normal operation (2), maximum free surface in cavities (3) and 0% air in cavities (4). Additionally the reference vessel is evaluated (1), see figure 3.1.4.

Table 3.1.1 Load cases

Load Case	LC1	LC2	LC3
Cargo tanks	Full (100%)	Empty (1%)	50%
Ballast tanks	Empty (1%)	Full (100%)	50%
Fuel tanks	Full (100%)	Full (100%)	Full (100%)

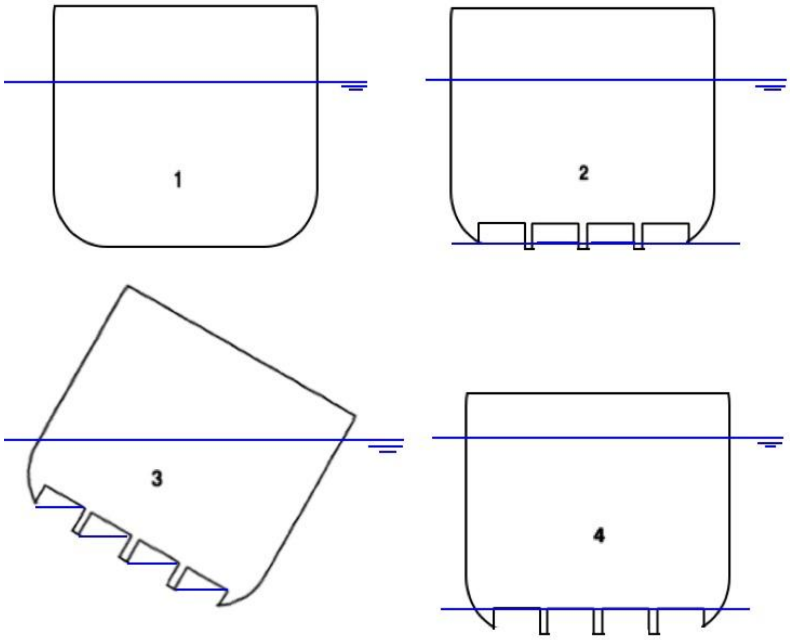


Figure 3.1.4 1(1) reference ship, (2)-(4) heeling angles for Air cavity hull illustrating how air escapes

	Criteria	Requirement	Unit	2008 IS Code
1	For angel of heel 0 to 30 the area under the GZ curve not less than	0.055	metre-radian	Part A, Ch 2.2
2	For angel of heel 0 to 40 the area under the GZ curve not less than	0.09	metre-radian	Part A, Ch 2.2
3	For angel of heel 30 to 40 the area under the GZ curve not less than	0.03	metre-radian	Part A, Ch 2.2
4	GZ at 30 or maximum greater than	0.2	m	Part A, Ch 2.2
5	Maximum GZ shall not occur before at angle of heel	25*	degree	Part A, Ch 2.2
6	The initial metacentric height GM0 shall not be less than	0.15	m	Part A, Ch 2.2
7	Angle of steady heel shall not exceed	16	degree	Part A, Ch 2.3
8	At steady wind, deck immersion less than	80	%	Part A, Ch 2.3
9	Area a/Area b shall be equal or greater than: (See fig. 3.1.5)	100	%	Part A, Ch 2.3

Table 3.1.2 2008 IS code general stability requirement

(* According to 2008 IS code 2.2.3 the maximum righting lever shall occur at an angle of heel not less than 25°. If this is not practicable, alternative criteria, based on an equivalent level of safety, may be applied subject to the approval of the Administration.)

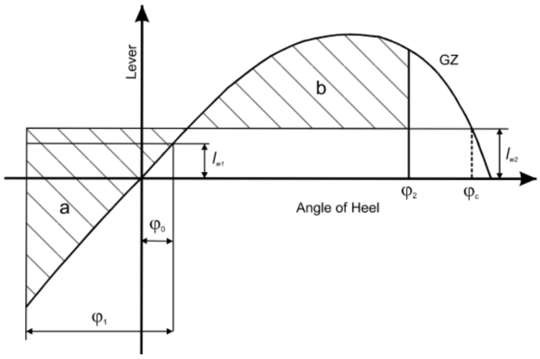


Figure 3.1.5y-axis righting arm, x-axis angle of heel, used for describing stability criteria nr 9

3.2 Strength elements

The ship is designed with respect to CSR (Common Structural Rules) for double hull oil tankers, valid for all tankers with length above 150m and where the class is member of IACS, International Association of Classification Society. The specific CSR reference used for design and evaluation of designs covered in this thesis is DNV CSR for double hull oil tankers, 2010. CSR 2010 is used because the reference ship complies with this version.

CSR serves as a guide during the design process, starting with identifying rules applicable for the specific design. Given general specifications such as length, draught, breadth etc., further loads acting on the structural elements are identified, both local and global. Based on this general ship data, structural limits and design criteria are calculated for both local and global elements. CSR also provides a description of the procedure, limits and suggestions for how to verify the model in an FE analysis.

CSR covers three main parts of the ship *aft end and machinery room*, *cargo area* and *fore end*. Different rules and limitations are valid for the different regions. Cargo area is defined as 40% of the length of the ship, the midship section. Since the given cavity design covers only the midship section, this thesis focuses on this region and leaves the other sections un-investigated. See figure 3.2.1.

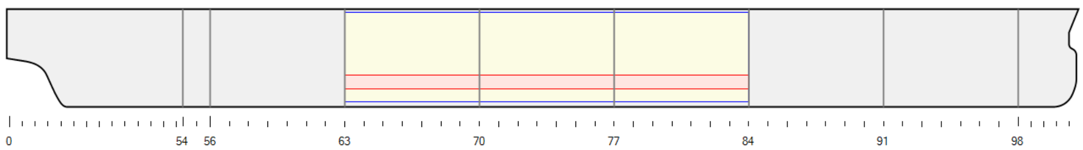


Figure 3.2.1 Illustration of midship section covered in this thesis

3.2.1 CSR Load Components

The CSR regulation loads are divided into six categories, DNV 2010. The categories are: static load components, dynamic load components, sloshing and impact loads, accidental loads and combination of loads. In this thesis static and dynamic loads are considered.

Static load is further divided into subcategories consisting of static hull girder loads (vertical wave bending moment and shear force) and local static loads as static sea pressure, static tank pressure and static deck load. Dynamic loads refer to dynamic loads on hull girders, horizontal wave bending moment, dynamic wave pressure and dynamic tank pressure.

Tools used in the dimensioning procedure are Nauticus Hull, Maxsurf, and Matlab. Nauticus Hull, further referred to as NH, is a software within DNV Sesam package. NH provides a user-friendly interface for calculating rule-based equations. Additionally NH provides a tool for design and evaluation of cross section design, which is compared to rule-based requirements. The evaluated cross section can be extruded to a homogeneous amidships section and exported to an FE analysis software. Maxsurf is described in chapter 3.1.3intact stability calculation and rule evaluation. Matlab is used for generating figures and comparing data from NH and Maxsurf.

3.2.2 CSR Static load components

The static load consists of static hull girder loads, vertical wave bending moment and shear force. Additionally static load components include local static loads as static sea pressure, static tank pressure and static deck load.

The CSR derives minimum allowable limits. The designer is to provide the calculated values based on load case model, such as a model in Maxsurf software. If any load from such a model test exceeds the CSR minimum prescribed load, the load from the model test is to be taken as minimum, (DNV 2010).

3.2.2.1 Vertical still water bending moment

Minimum permissible hull girder still water bending moment amidships for sea going operations is given by (DNV, 2010a). See equations 3.2.1-3.2.4

Hogging:

$$M_{sw-min-sea-mid} = f_{sea}\,(Z_{v-\min}\,\sigma_{perm-sea}\,10^3 - M_{wv-hog})\ \text{kNm}$$

which is identical to

$$M_{sw-min-sea-mid} = 0.01\,C_{wv}\,L^2\,B(11.97 - 1.9C_b)\ \text{kNm}$$

(Eq. 3.2.1 and 3.2.2)

(From NH calculations the value is 3700000 kNm. From Maxsurf design value is 5000000 kNm for ballast cond. The larger value will be considered bending as design bending moment.)

Sagging:

$$M_{sw-min-sea-mid} = f_{sea}\,(Z_{v-\min}\,\sigma_{perm-sea}\,10^3 + M_{wv-sag})\ \text{kNm}$$

which is identical to

$$M_{sw-min-sea-mid} = -\,0.05185\,C_{wv}\,L^2\,B(C_b + 0.7)\ \text{kNm}$$

(Eq. 3.2.3 and 3.2.4)

(From NH calculations the value is 2600000 kNm. From Maxsurf design value is 1800000 kNm for fully loaded cond. The larger value will be considered bending as design bending moment.)

To allow flexibility during loading and unloading operations the CSR minimum hull girder hogging and sagging still water bending moment for harbour operations is to be taken as equation 3.2.5 and figure 3.2.2.

$$M_{sw-min-harb} = 1.25\, M_{sw-min-sea} \quad \text{kNm}$$

(Eq. 3.2.5)

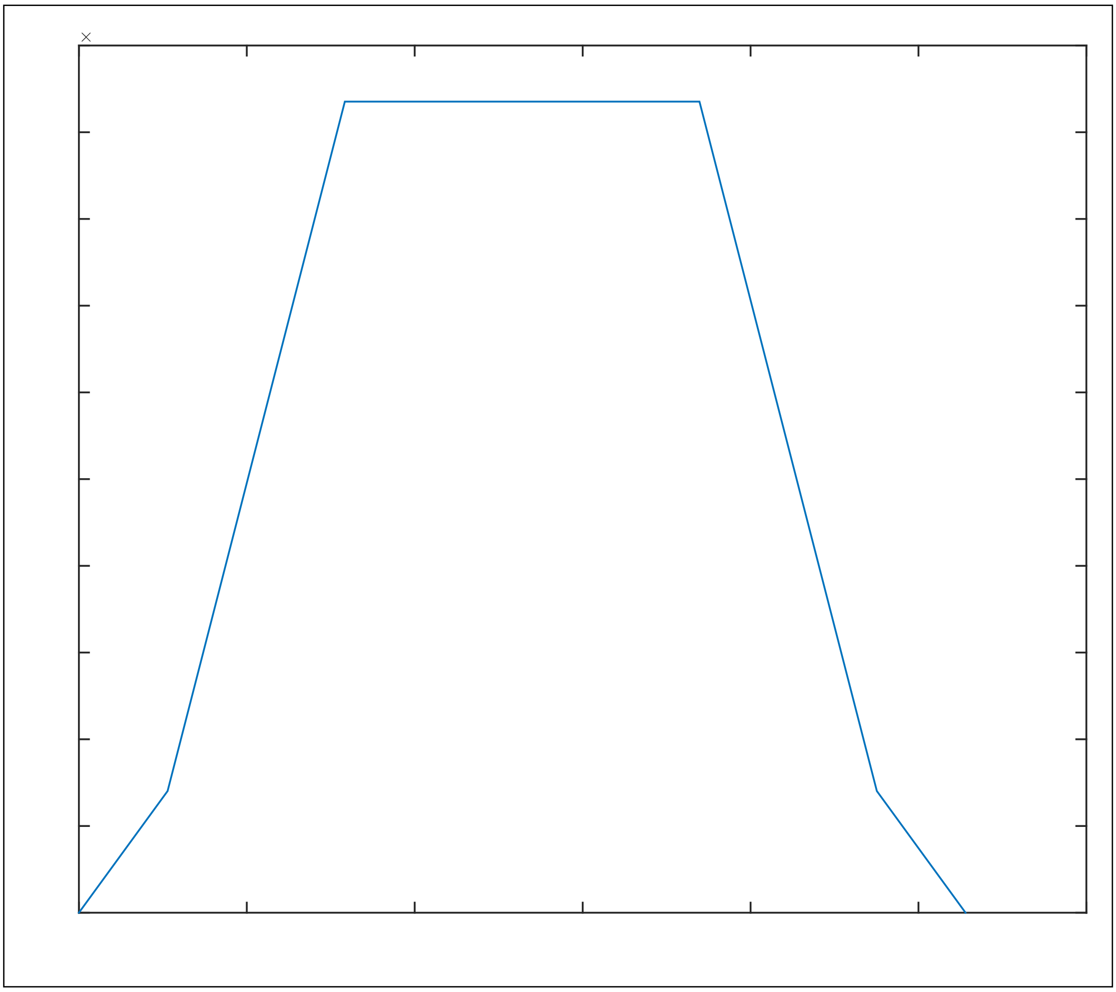


Figure 3.2.2 CSR restricted hogging still water bending moment for harbour operations

3.2.2.2 Still water hull girder shear force

According to CSR, the designer needs to provide the permissible hull girder still water maximum shear force. The sheer force calculated for sea going operation, Qsw-perm-sea, are to envelope the minimum hull girder still water shear forces, (DNV 2010).

For ships with centreline longitudinal bulkhead, the minimum hull girder positive and negative still water shear force for seagoing operation, $Q_{sw\text{-}min\text{-}sea}$, in way of transverse bulkheads between cargo tanks is to be taken as eq. 3.2.6 and eq. 3.2.7.

$$Q_{sw-min-sea} = \pm\, 0.4 \rho g B_{local} l_{tk} T_{sc} \quad \text{kN}$$

(Eq. 3.2.6)

For harbour operations:

$$Q_{sw-min-harb} = \pm\, 0.45 \rho g B_{local} l_{tk} T_{sc} \quad \text{kN}$$

(Eq. 3.2.6)

3.2.2.3 Static sea pressure

The static sea pressure is taken as eq. 3.2.8:

$$P_{hys} = \rho_{sw} g (T_{LC} - z) \quad \text{kN/m}^2$$

(Eq. 3.2.8)

The pressure distribution is illustrated in figure 3.2.3.

The static tank pressure is calculated from the highest point in the tank. This point is taken as eq. 3.2.9:

$$P_{in-tk} = \rho g z_{tk} \quad \text{kN/m}^2$$

(Eq. 3.2.9)

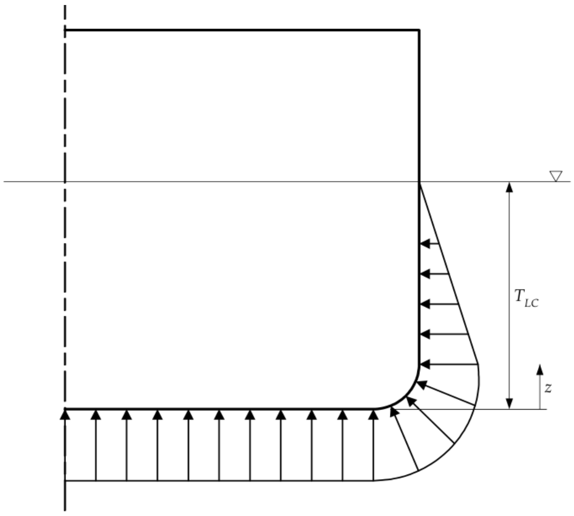


Figure 1.2.3 Static sea pressure

3.2.3 CSR Dynamic load components

Dynamic loads referrers to dynamic loads on hull girders, horizontal wave bending moment, dynamic wave pressure and dynamic tank pressure. For background information of the equations, see (DNV 2010).

Dynamic vertical hogging wave bending moment (kNm) eq. 3.2.10.

$$M_{wv-hog} = f_{prob}\, 0.19 f_{wv-v} C_{wv} L^2 B C_b$$

(Eq.3.2.10)

Dynamic vertical sagging wave bending moment (kNm) eq. 3.2.11 and figure 3.2.4.

$$M_{wv-sag} = -f_{prob} 0.11 f_{wv-v} C_{wv} L^2 B(C_b + 0.7)$$

(Eq.3.2.11)

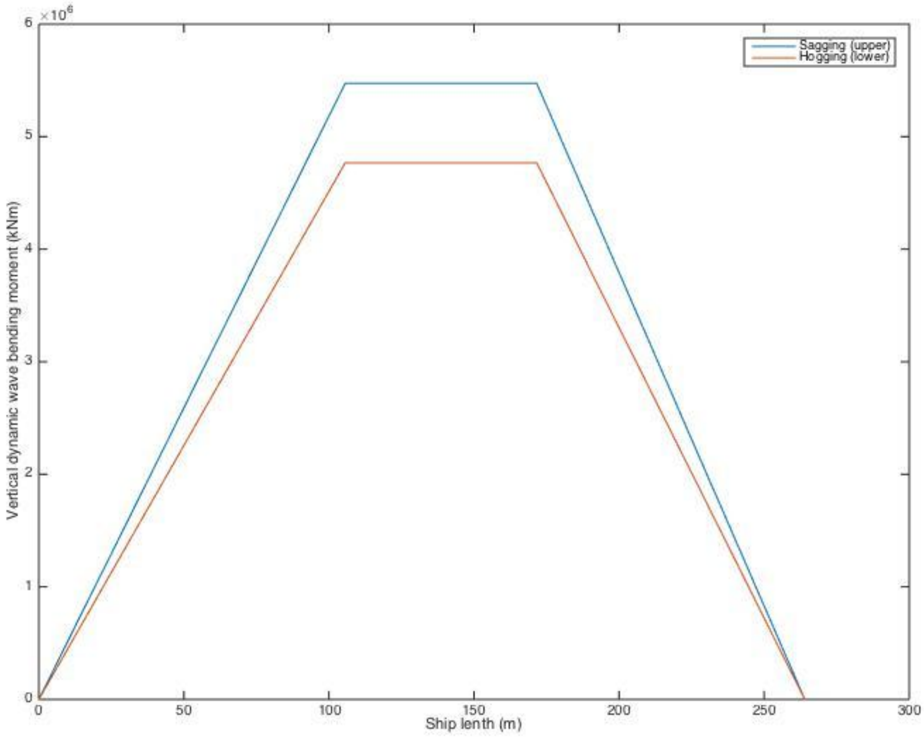


Figure 3.2.2 Vertical dynamic wave bending moment

Horizontal wave bending moment eq. 3.2.12 and figure 3.2.5.

$$M_{wv-h} = f_{prob}(0.3 + \frac{L}{2000}) f_{wv-h} C_{wv} L^2 T_{LC} C_b \quad \text{kNm}$$

(Eq. 3.2.12)

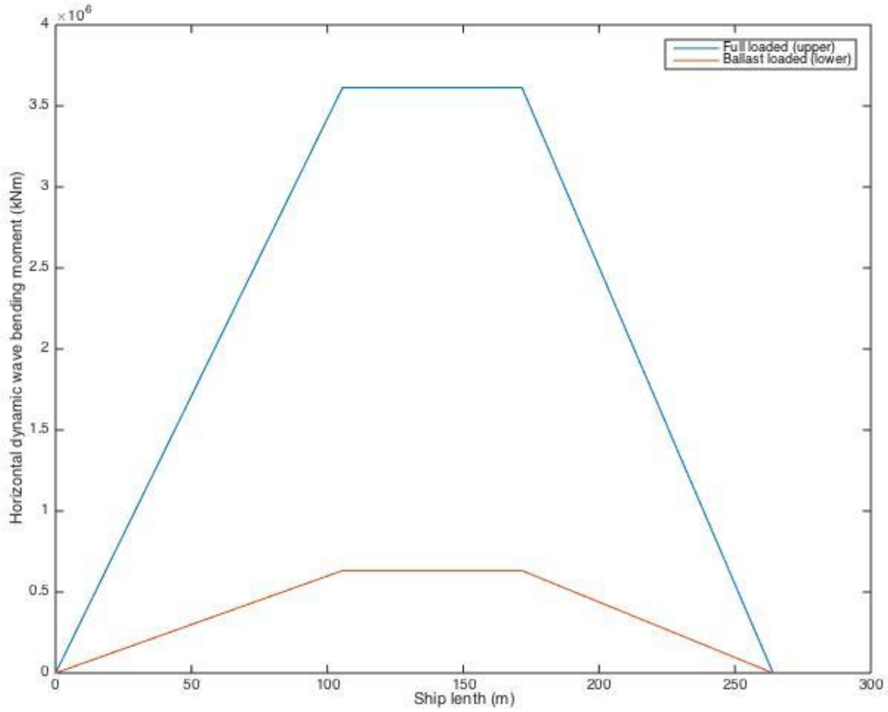


Figure 3.2.3 Horizontal dynamic wave bending moment

Positive vertical wave shear force (MPa) eq. 3.2.13 and fig 3.2.6.

$$Q_{wv-pos} = 0.3 f_{qwv-pos} C_{wv} L B (C_b + 0.7) \qquad \text{(Eq. 3.2.13)}$$

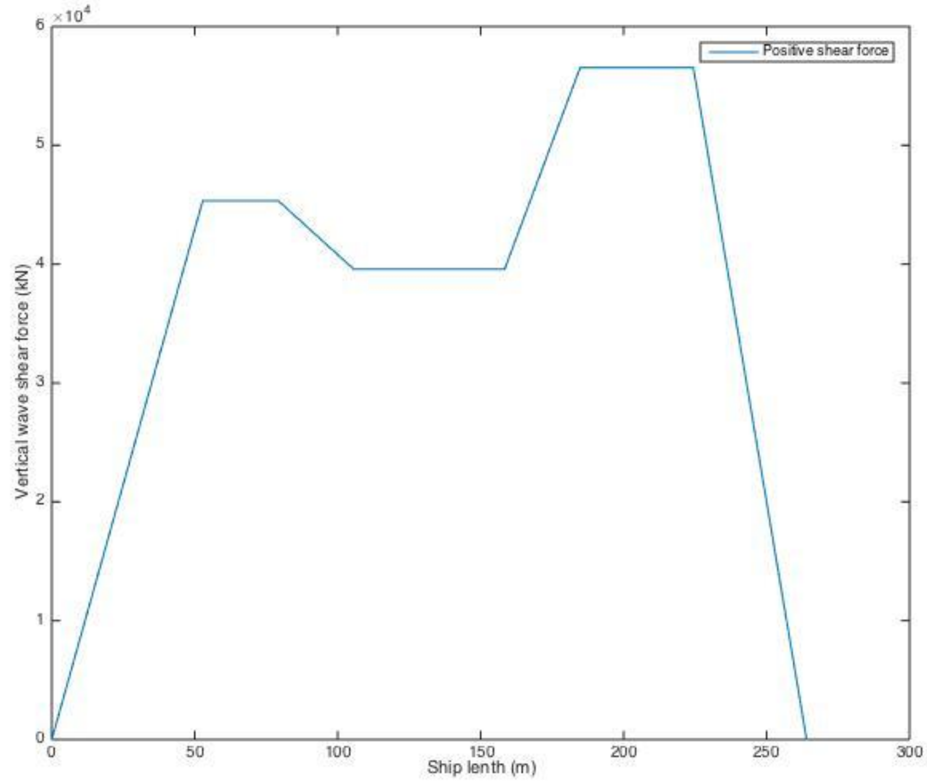


Figure 3.2.4 Positive vertical wave shear force

Negative vertical wave shear force (MPa) eq. 3.2.14 and fig 3.2.7.

$$Q_{wv-neg} = -0.3 f_{qwv-neg} C_{wv} LB(C_b + 0.7) \qquad \text{(Eq. 3.2.14)}$$

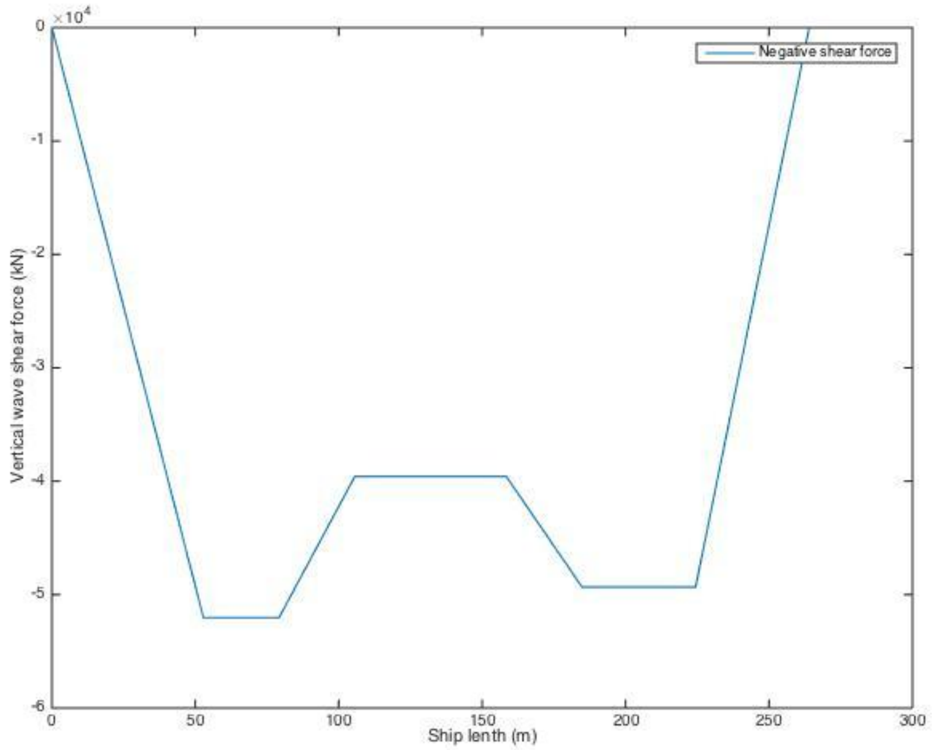


Figure 3.2.5 Negative vertical wave shear force

3.2.4 Longitudinal strength requirements 0.4L amidships

3.2.4.1 Hull girder bending requirements

At the amidships cross section the net vertical hull girder moment of inertia about the horizontal neutral axis, $I_{v\text{-}net50}$, is not to be less than the rule minimum vertical hull girder moment of inertia, $I_{v\text{-}min}$, defined as eq. 3.2.15.

$$I_{v-min} = 2.7 C_{wv} L^3 B (C_b + 0.7) \cdot 10^{-8} \quad \mathrm{m}^4 \tag{Eq. 3.2.15}$$

At the midship cross section the net vertical hull girder section modulus, $Z_{v\text{-}min}$, at the deck and is not to be less than the rule minimum hull girder section modulus, $Z_{v\text{-}min}$, defined as 3.2.16.

$$Z_{v-min} = 0.9 k C_{wv} L^2 B (C_b + 0.7) \cdot 10^{-6} \quad \mathrm{m}^3 \tag{Eq. 3.2.16}$$

The net hull girder section modulus about the horizontal neutral axis, $Z_{v\text{-}net50}$, is not to be less than the rule required hull girder section modulus, $Z_{v\text{-}req}$, based on the permissible still-water bending moment and design wave bending moment defined as eq. 3.2.17.

$$Z_{v-req} = \frac{\left| M_{sw-perm} + M_{wv-v} \right|}{\sigma_{perm}} 10^{-3} \quad \text{m}^3$$

(Eq. 3.2.17)

Msw_perm is the permissible hull girder hogging or sagging still water bending moment. Mwv-v is the hogging or sagging vertical wave bending moment. σperm is the permissible hull girder bending stress. See table 3.2.1.

Table 3.2.1 Permissible hull girder bending stress

Design load combination	*Still water bending moment, $M_{sw\text{-}perm}$*	*Wave bending moment, $M_{wv\text{-}v}$*	*Permissible hull girder bending stress, σ_{perm} (1)*	
(S)	$M_{sw\text{-}perm\text{-}harb}$	0	143/k	within 0.4L amidships
			105/k	at and forward of 0.9L from A.P. and at and aft of 0.1L from A.P.
(S + D)	$M_{sw\text{-}perm\text{-}sea}$	$M_{wv\text{-}v}$	190/k	within 0.4L amidships
			140/k	at and forward of 0.9L from A.P. and at and aft of 0.1L from A.P

The reference vessel is designed with three types of tensile steel, the three yield stresses used, 235, 315 and 355, see figure 3.3.3.1 for details. Mainly high tensile steel is used in bottom and deck structure while mild steel is used in the hull side structure. For calculation of permissible strength, the k value needs to be considered at each individual structural element.

3.3 Cargo hold

In this section the method for creating the cargo hold models are explained, three cargo holds from the mid ship region were modelled and a FE analysis were performed.

The hull strength were analysed in accordance with CSR 2.1.1.1 by creating two CAD models, one for the reference ship and one for the cavity ship. With these two models several finite element assessments were carried out and critical areas regarding stresses were identified. The stress distributions were also compared between the two models.

Two main software were used for this part, Nauticus Hull and GeniE. These two software, which is a part of the Sesam package from DNV, provides all the tools necessary to carry out the task.

Nauticus Hull which provides the rule based empirical tool were used to design and verify all the longitudinal material in the structure, which are deck, bottom, outer shell, inner bottom, inner sides, longitudinal bulkheads and longitudinal stiffeners. All the longitudinal parts created in Nauticus Hull and extruded in GeniE are shown in Fig3.3.1.

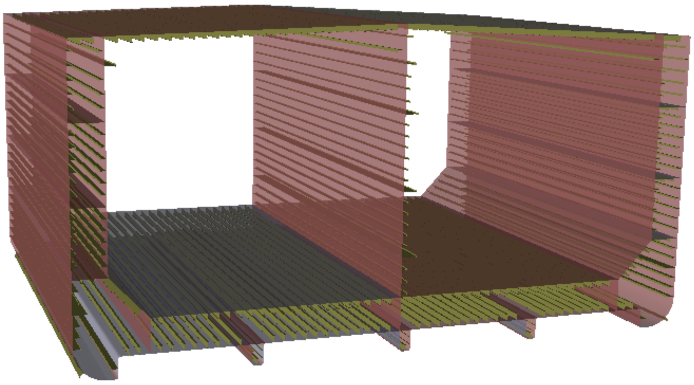


Figure 3.3.1 Longitudinal structure of the cavity ship created in Nauticus Hull

GeniE provides the tools needed to create the transverse web frames and the transverse bulkheads, it also included the tools needed to set up and carry out an FE-analysis.

The longitudinal materials designed in Nauticus Hull were extruded and imported into GeniE and the web frames, stringer planes and transverse bulkheads were modelled in GeniE using shell elements. The shell elements were assigned a specific thickness and material property. The structural arrangement of the web frames, stringer planes and the transverse bulkheads created i GeniE is show Fig3.3.2.

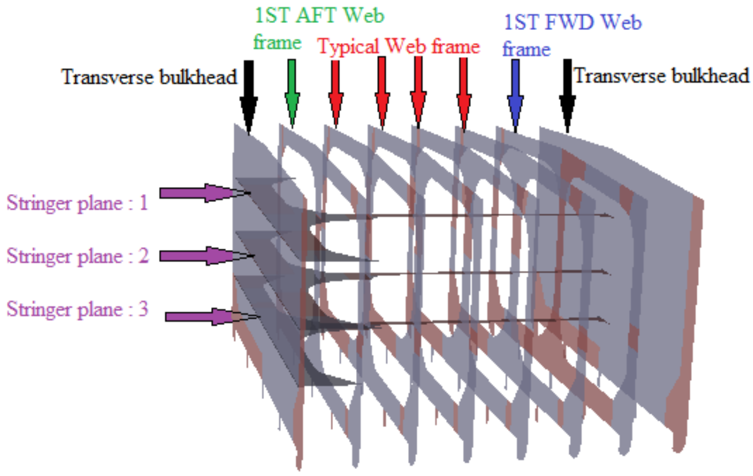


Figure 3.3.2 Structural arrangement of transverse webs, bulkheads and stringer planes for the cavity ship created in GeniE

To make a comparable analysis between the two hull designs, the importance of detail equivalence were highly prioritised. The reference and the cavity ship were modelled to the same level of detail.

The web frames and stringer planes for the two models have the same plate arrangement regarding material properties and thickness, all based on data from the reference ship.

The plate segments for the web frames, transverse bulkheads and stringer planes are colour coded to visualize the similarities between the two designs. Yield-stresses for respectively material is seen in Table 3.3.1.

Table 3.3.1 Material property by colour coding

Colour code/Material	σ_Y [MPA]	σ_{prem} [MPA]
A	253	143
AH	315	183
AH36	355	199

In Figure 3.3.4 the plate arrangement by material property for a typical web frame is seen. As seen below, the plate arrangement of the cavity ship follows the plate arrangement as the reference vessel. The 1ST AFT web frame are modelled as a typical web frame, this complies both for the reference and the cavity vessel. Figure3.3.5 and Figure3.3.6 and Figure 3.3.7 will show the plate arrangement by material property for the 1ST FWD Web frame, the transverse bulkheads and the stringer planes.

In figure 3.3.3 the material distribution for the longitudinal sections for the Cavity and the reference ship are showed.

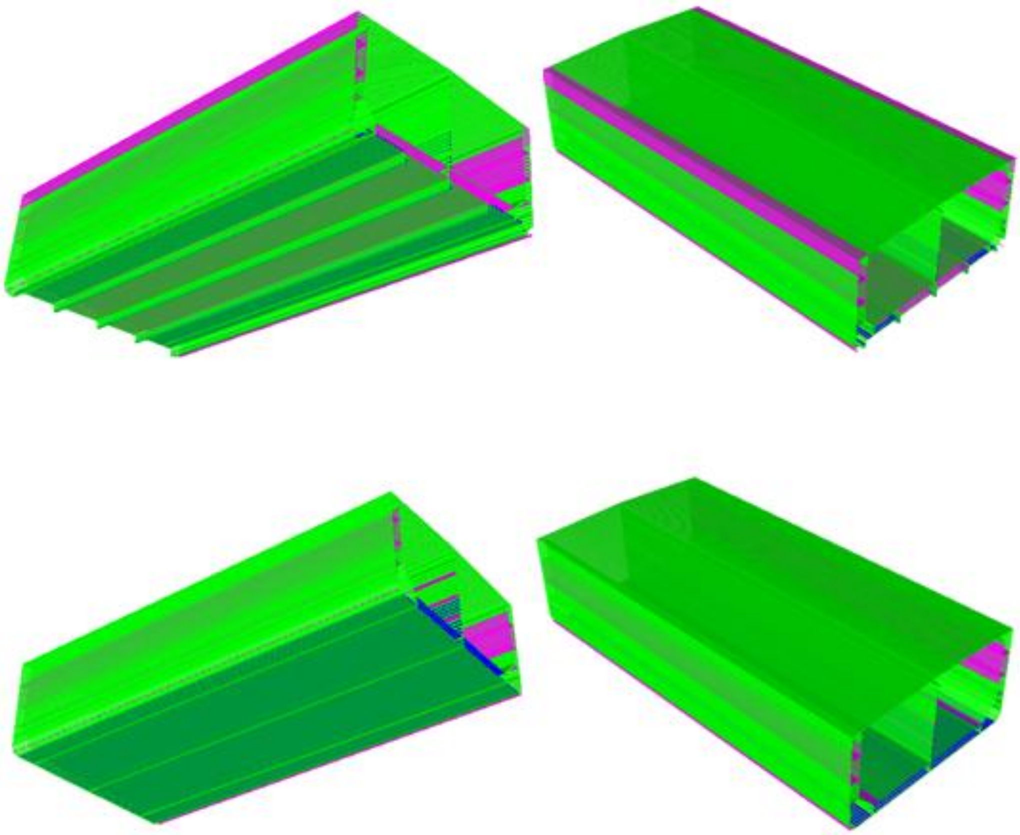


Figure 3.3.3 Colour coded material properties. Longitudinal material. (Upper figure are the cavity ship)

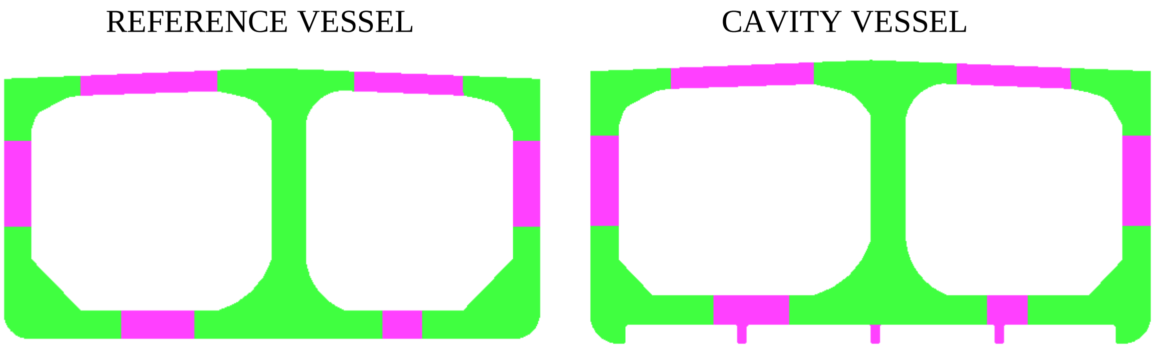


Figure 3.3.4Colour coded material properties. Typical Web frame

REFERENCEVESSEL CAVITYVESSEL

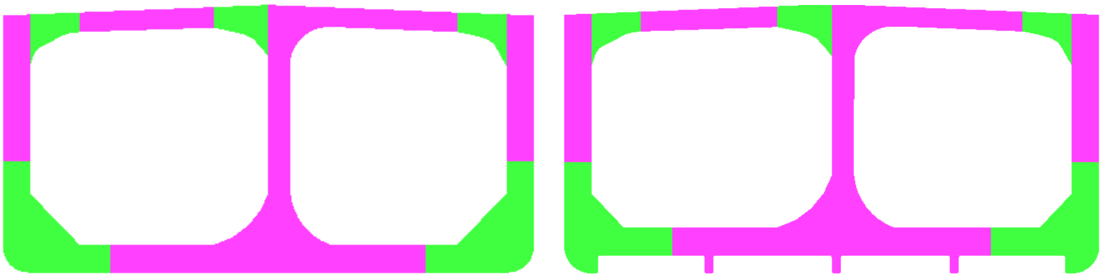


Figure 3.3.5 Colour coded material properties. 1ST FWD Web frame.

REFERENCE VESSEL CAVITY VESSEL

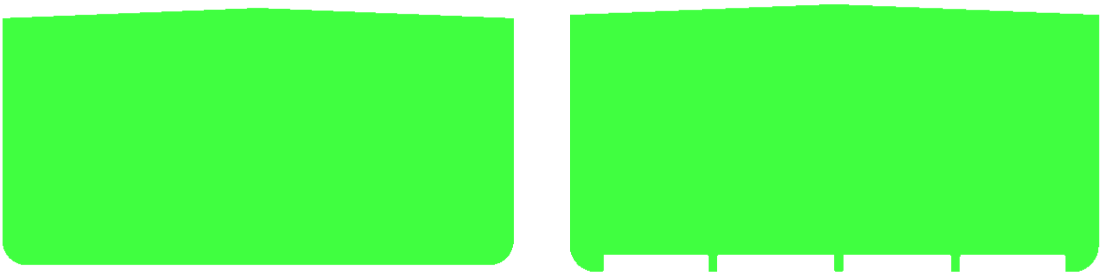


Figure 3.3.6 Colour coded material properties. Transverse bulkhead.

STRINGER: 1 STRINGER: 2 STRINGER: 3

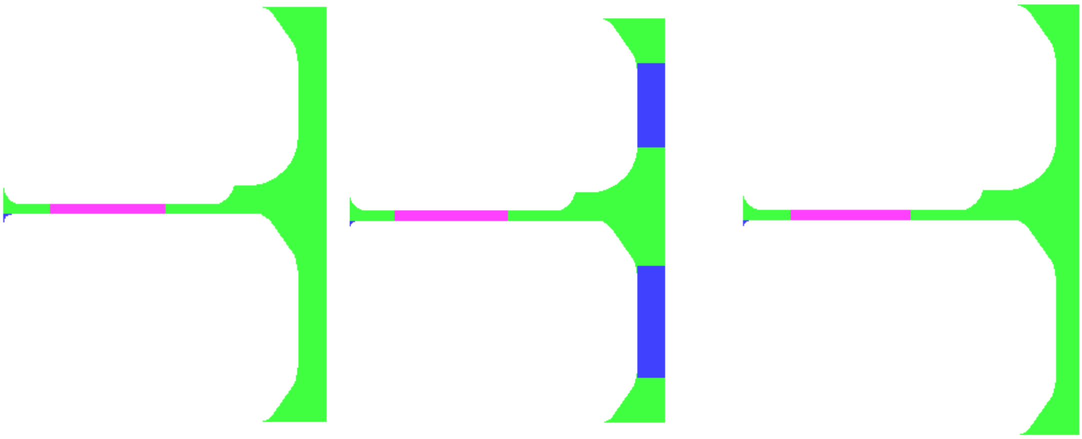


Figure 3.3.7 Colour coded material properties. Stringer plane.

The thicknesses of the different plate segments are also coded by colour to visualise the similarities, in Figure 3.3.8 the thickness arrangement for the 1ST FWD Web frame can be seen and a typical web frame can be seen Figure 3.3.9. As seen below the thickness arrangement of the cavity follows the reference vessel to the same extent. Since the 1st AFT web frame are modelled as a typical web frame the thickness of the plate arrangement are identical, which compiles both for the reference and the cavity vessel.

Figure3.3.10 and Figure 3.3.11 shows thicknesses of the different plate segments for the transverse bulkheads and the stringer planes.

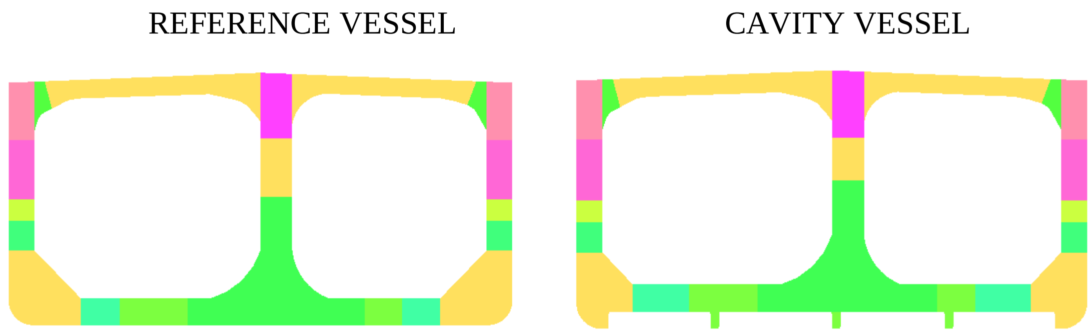


Figure 3.3.8 Colour coded thickness. 1ST FWD Web frame.

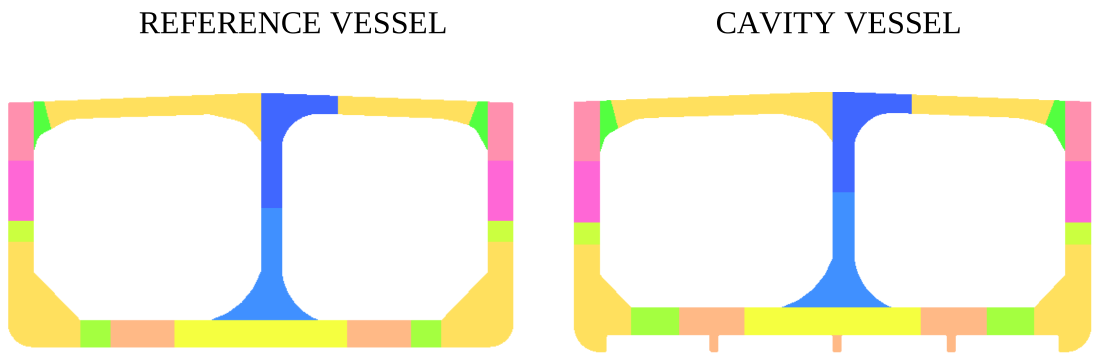


Figure3.3.9 Colour coded thickness. Typical Web frame.

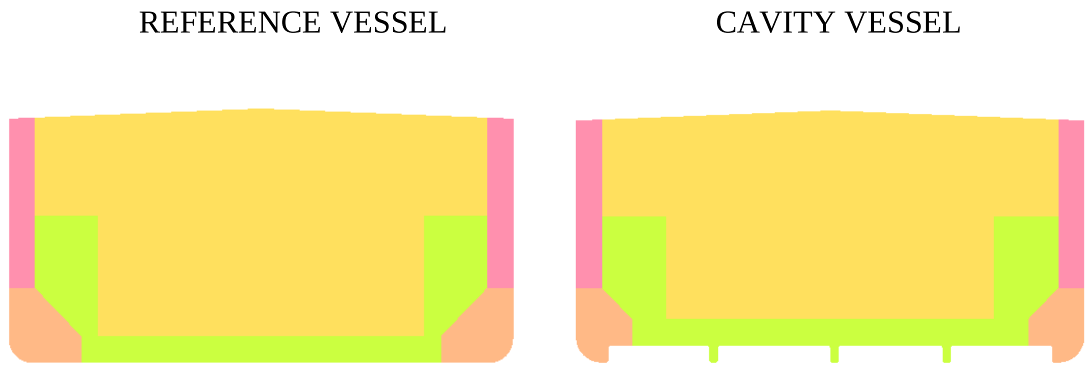


Figure 3.3.10 Colour coded thickness.Transverse bulkhead

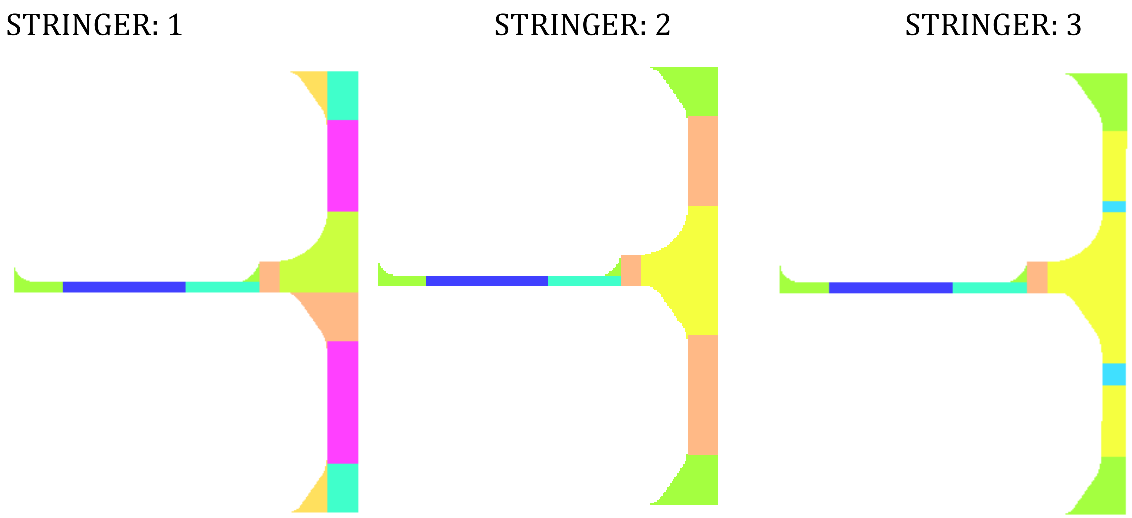


Figure 3.3.11 Colour coded thickness.Stringer planes

The following Figure 3.3.12, Figure 3.3.13 and Figure 3.3.14 shows a complete amidships cargo hold model for the reference, the cavity and a complete three cargo hold model.

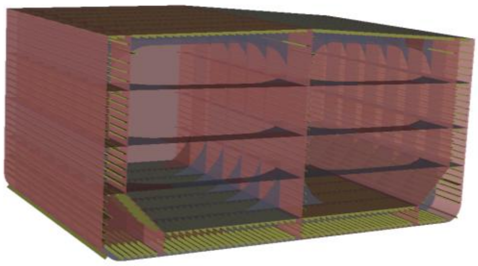


Figure 3.3.12Complete amidships cargo hold model of the reference

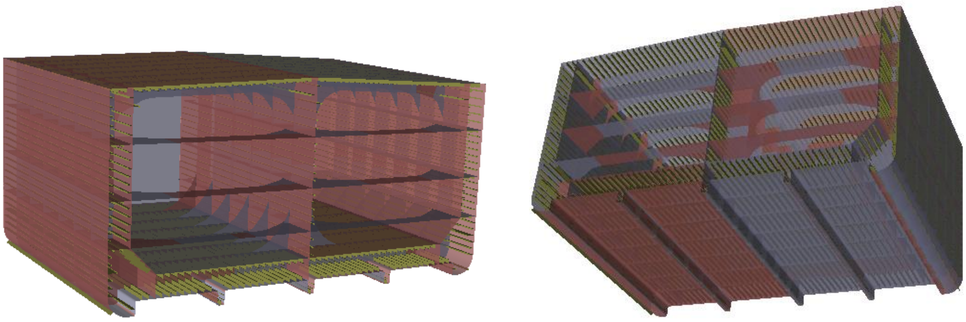


Figure 3.3.13 Complete amidships cargo hold model of the cavity

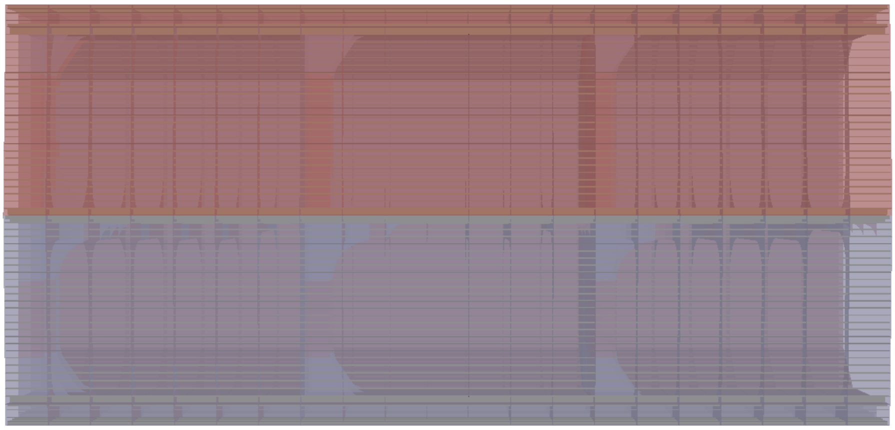


Figure 3.3.14 Three cargo hold model of the reference vessel seen from above.

3.3.1 Boundary conditions

According to CSR 2.2.2.3 the mesh size of the finite element model is to follow the size of the spacing for stiffening system of the structure as far as practical possible, this to represent the actual plate panels between the stiffeners, this was done except within the bottom structure. Due to the complex geometry of the bottom structure of the cavity vessel Figure 3.3.1.1 a more dense mesh 0.3m were used in that region.

1st order elements were used based on our limited computational capacity but due to a high number of elements the 1st order was to recommend.

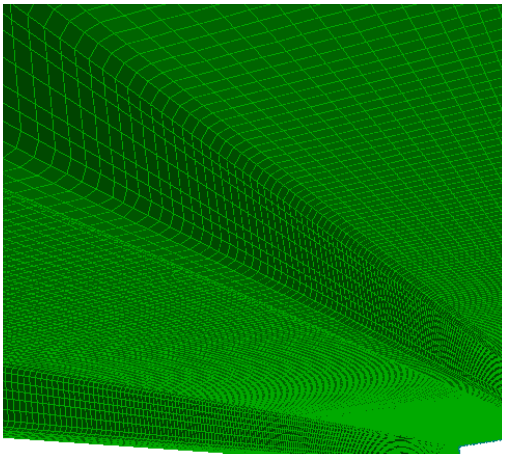


Figure 3.3.1.1 Mesh at complex Cavity bottom structure

The boundary conditions applied follow the CSR Table B.2.9 for one longitudinal centre line bulkhead.

To get the FE model to represent the reality as good as possible according to beam theory two rigid links were applied to the model. The rigid links are linked to an independent point at the neutral axis in the aft and fwd part of the cargo hold model which the end moments are applied to.

A region within all end elements, called end nodes, was connected to the rigid link at each end of the model.

Ground springs at the models ends were applied Figure 3.3.1.2, for translation in y-direction ground spring elements were applied to the deck, inner bottom and outer shell and for the translation in z-direction the ground springs were applied to the side, inner skin and longitudinal bulkheads.

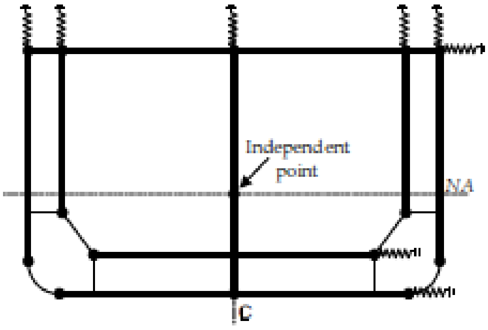


Figure 3.3.1.2 Spring constraints and independent point at models end(CSR Figure B 2.13)

The following boundary condition Table 3.3.1.1 Table 3.3.1.2 settings according to CSR for oil tankers are applied to the models in Nauticus Hull.

Table 3.3.1.1 Boundary conditions according to CSR oil tankers

Independent points	Dx	Dy	Dz	Rx	Ry	Rz
Aft end	Fix	Free	Free	Free	Free	Free
Fwd end	Free	Free	Free	Free	Free	Free

Table X3.3.1.2 Boundary condition conditions according to CSR oil tankers

Longitudinal elements	Dx	Dy	Dz	Rx	Ry	Rz
Aft end	RL	Free/Spring	Free/Spring	Free	RL	RL
Fwd end	RL	Free/Spring	Free/Spring	Free	RL	RL

3.3.2 Loads and loading condition

To verify the hull strength of the structure in accordance with the CSR, a set of pre defined load cases has to be tested and evaluated based on a FE analysis. These load cases consists of both static loads and combinations of load cases for both static and dynamic loads.

The load cases applied can be seen in Figure3.3.2.1and Figure 3.3.2.2.

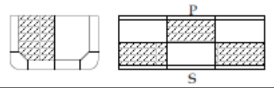
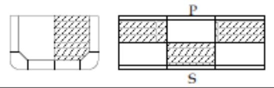
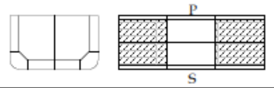
Table B.2.4 Load Cases for Tankers with One Centreline Oil-tight Longitudinal Bulkhead (Continued)							
Loading Pattern	Figure	Still Water Loads			Dynamic load cases		
		Draught	% of Perm. SWBM(2)	% of Perm. SWSF(2)	Strength assessment (1a)	Strength assessment against hull girder shear loads (1b)	
					Midship region	Forward region	Midship and aft regions
B9 (8)		$1/3T_{sc}$	100% (sag)	75% (+ve fwd) See note 4	Only applicable to strength assessment of midship region (see note 1(a))		
B10 (6, 8)		$1/3T_{sc}$	100% (sag)	75% (+ve fwd) See note 4	Only applicable to strength assessment of midship region (see note 1(a))		
B11(8)		T_{sc}	100% (Hog)	100% (-ve fwd) See note 5	Applicable to strength assessment of midship region (see 1(a)) and strength assessment against hull girder shear loads (see 1(b))		

Figure 3.3.2.1 Static load cases according to CSR

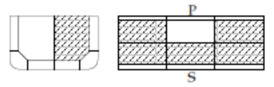
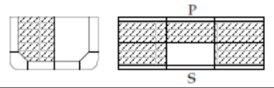
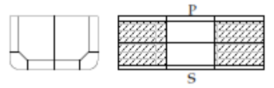
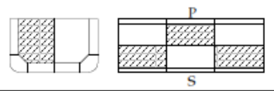
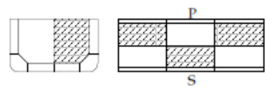
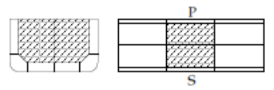
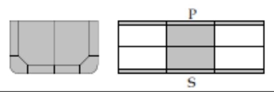
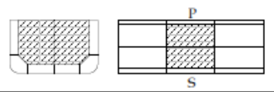
Table B.2.4 Load Cases for Tankers with One Centreline Oil-tight Longitudinal Bulkhead							
Loading Pattern	Figure	Still Water Loads			Dynamic load cases		
		Draught	% of Perm. SWBM(2)	% of Perm. SWSF(2)	Strength assessment (1a)	Strength assessment against hull girder shear loads (1b)	
					Midship region	Forward region	Midship and aft regions
Design load combination S + D (Sea-going load cases)							
B1		$0.9\ T_{sc}$	100% (sag)	See note 3	1	\	\
			100% (hog)	100% (-ve fwd) See note 4	2, 5a	\	\
B2 (6)		$0.9\ T_{sc}$	100% (sag)	See note 3	1	\	\
			100% (hog)	100% (-ve fwd) See note 4	2, 5b	\	\
B3		$0.9\ T_{sc}$	100% (hog)	100% (-ve fwd) See note 5	2	4	2
				100% (-ve fwd) See note 4	5a, 5b, 6a, 6b	\	\
B4		$0.6\ T_{sc}$	100% (sag)	75% (+ve fwd) See note 4	1, 5a	\	\
B5 (6)		$0.6\ T_{sc}$	100% (sag)	75% (+ve fwd) See note 4	1, 5b	\	\
B6		$0.6\ T_{sc}$	100% (sag)	100% (+ve fwd) See note 5	1	3	1
				100% (+ve fwd) See note 4	5a, 5b	\	\
B7 (7)		T_{bal-cm}	100% (sag)	100% (+ve fwd) See note 4	1	\	\
Design load combination S (Harbour and tank testing load cases)							
B8 (8)		$1/3T_{sc}$	100% (sag)	100% (+ve fwd) See note 5	Applicable to strength assessment of midship region (see 1(a)) and strength assessment against hull girder shear loads (see 1(b))		

Figure 3.3.2.2 Combined load cases according to CSR

Results from the FE analysis covers all loading conditions for static and dynamic load cases given in CSR. The main design differences between the reference ship and the cavity ship is the bottom structure. Plating, web spacing and other longitudinal structural elements are kept constant or increased dimension. The main changes to the cavity design are the bottom structure and the deck structure. The neutral axis is shifted but the cargo tanks contain less volume. With this background the results of interest are the maximum sagging and the maximum hogging load cases, B11 and B6.

3.3.2.1 Docking load

Docking a ship may be a severe loading condition for the bottom structure and it is of importance that the stresses in the structure do not exceeded the permissible limits. Docking is not regulated in the CSR, but it is recommended that the ship designer investigate this. An FE analysis was performed to investigate the differences in stresses which the two vessels are subjected to during a docking procedure. The following section will explain the procedure and setups for the models and how the analysis was carried out.

The following simplifications were made for the setup of the docking:

- The section for analysis is set to the space between two web frames 4800mm.
- The weight applied to the structure is only its self-weight.
- No ballast water was included.
- Only amidships section structural elements were included.

The docking were tested for both five and three keels supports, the first load case were when docking on a typical web frame and the second were when docking between two typical web frames on docking brackets.

The dimension of the keel supports was set to L=2000mm, B=800mm and was for the reference vessel placed in accordance to docking plan.

The keel supports were represented by ridged link connected to an independent point that was set to fix in all 6 degree of freedom.

Docking support on the web frame:

When docking the ship, the keel supports are placed on the web frames and between the web frames. In Figure 3.3.2.1.1 the simulation setup for the reference ship are showed, the setup are for docking on five and three keel supports which are represented by the green boxes.

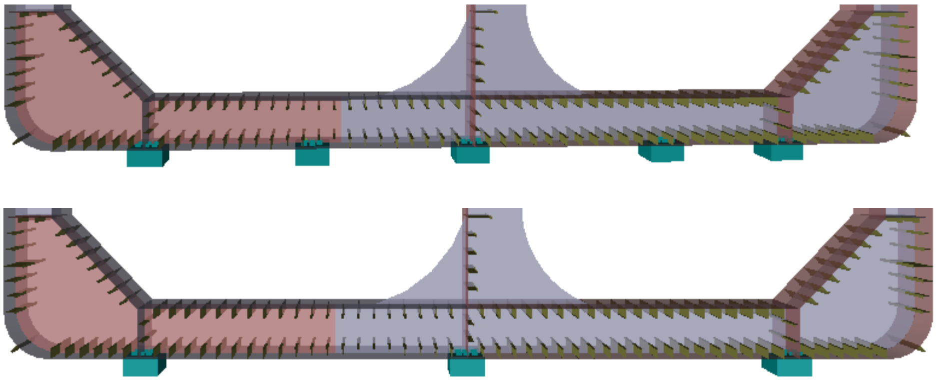


Figure 3.3.2.1.1. Docking setup for reference ship. Five and three supports

For the cavity ship, a similar setup was used. The supports were placed on the cavity walls which were done to simulate a worst case scenario. This can be seen in Figure 3.3.2.1.2.

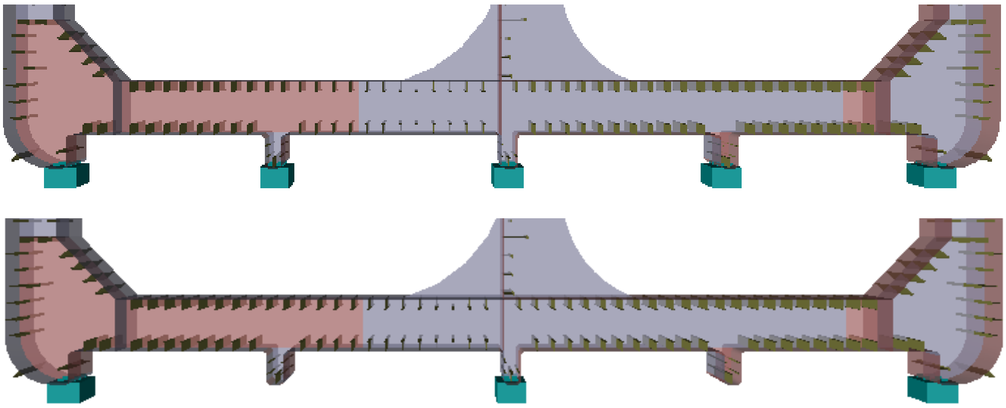


Figure 3.3.2.1.2 Docking setup for cavity ship. Five and three supports

Docking supports between two web frames:

When docking between the web frames, the keel supports are placed on docking brackets. The docking brackets which are marked in red in Figure 3.3.2.1.3 acts as a support between the double bottom. To get some extra support in the cavities docking brackets were added between the cavity and the bottom.

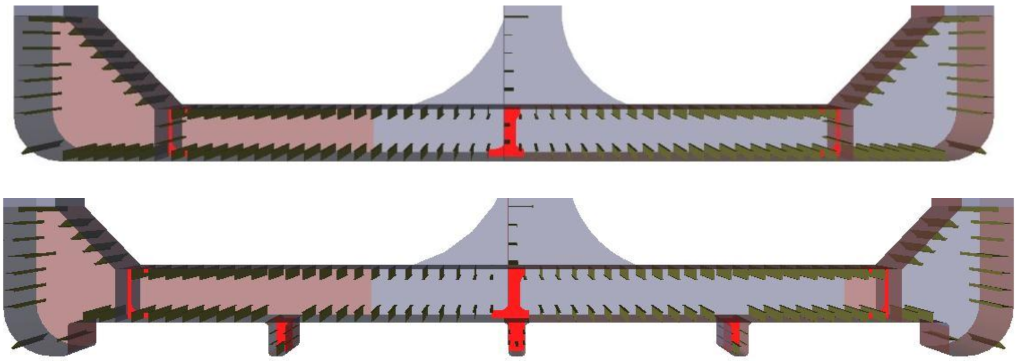


Figure 3.3.2.1.3 Docking brackets between web frames marked in red. Reference and Cavity ship

The docking setup for the reference ship can be seen in Figure 3.3.2.1.4and shows how the keel supports are placed in between the two web frames at the keel supports.

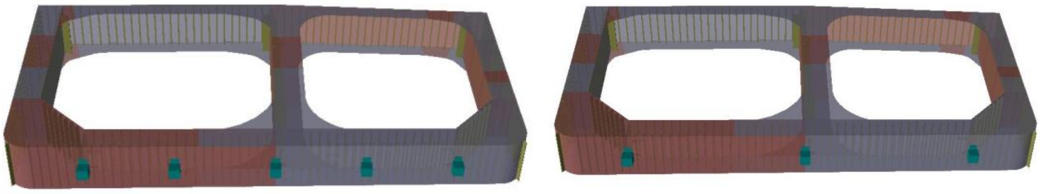


Figure 3.3.2.1.4 Docking setup for the reference ship. Five and three supports

The docking setup for the cavity ship can be seen in Figure 3.3.2.1.5 and shows how the keel supports are placed in between the two web frames, unlike the reference ship the cavity ship do not dock on the docking bracket at the longitudinal bulkhead at the hopper corner. The keel supports are here placed at the cavities at the bilges.

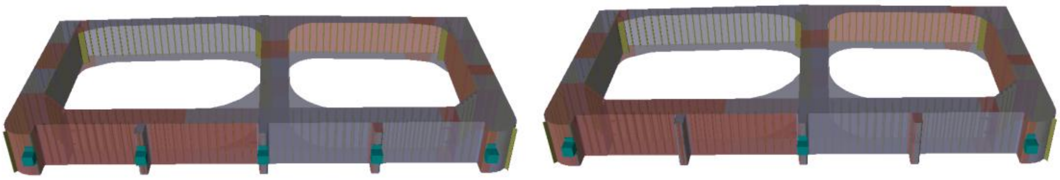


Figure 3.3.2.1.5 Docking setup for the cavity ship. Five and three supports

4 Results

4.1 Hydrostatic results

4.1.1 Draught

The maximum scantling draught and the draught in the cavities are given in table 4.1.1. The draught in the cavities is given at any loading condition and is set by the air pressure. The scantling draught is given at maximum loading condition for the reference ship, which will never be reached. The scantling draught is used as a reference for ship design.

Table 4.1.1 Scantling draught, Tsc for reference ship and cavity ship

	Reference ship	Cavity ship	
Draught in cavity Tc	-	0.2m	1.6m
Scantling draught, Tsc	17m	17.2m	

4.1.2 Stability-General stability criteria

A normal operation (90% air in Cavities) describes the initial stability and covers a valid range from -1.1 deg. To +1.1 deg (initial stability). The stability values for maximum free liquid surface, at 50% air in cavities, is valid from -50 deg. to +50 deg. The stability values for 0% air in cavities are valid for all heel angels. At 0% air in the cavities there are no free surface effects, however the draft is at maximum.

When the valid range is exceeded there will be a leakage of air. The effect is lost buoyancy, which leads to increased draught. The free surface effect will increase from 1.1 deg to maximum 50 deg. angle of heel, and reduce from 50 deg. to 90 deg. heel angle.

For GZ-curves and see Appendix A.

Table 4.1.2 Results general stability criteria 2008 IS code pass or fail matrix

	LC1	LC2	LC3
Reference ship	Pass	Pass	Pass
Air cavity ship (Normal operation 90% air)	Pass	Pass	Pass
Air cavity ship (Normal operation 90% air in cavities)	Pass	Pass	Pass
Air cavity ship (Maximum free surface effect 50% air in cavities)	Pass	Pass	Pass
Air cavity ship (Normal operation 0% air in cavities)	Pass	Pass	Pass

4.2 Strength elements

Rule loads are equal for reference ship and cavity ship, see table 4.2.1

Table 4.2.1 Rule loads results

Seagoing	Sagging	Hogging
Minimum Rule value (Msw_min_sea)	2580000 [kNm]	3741600 [kNm]
Actual value from (Maxsurf)	1755400 [kNm] (LC1)	4965500 [kNm] (LC2)

Table 4.2.2 Actual and rule required values for still water bending moments at 0.4L amidships section seagoing condition

Harbour	Sagging	Hogging
Minimum Rule value (Msw_min_harb)	3225000[kNm]	4676900[kNm]
Actual value from (Maxsurf)	2195300 [kNm] (LC1)	6206900 [kNm] (LC2)

The values chosen for $M_{sw\_perm\_sea}$ is 2580000 [kNm] for sagging and 4965500 [kNm] for hogging. $M_{sw\_perm\_harb}$ is 3225000 [kNm] for sagging and 6206900kNm] for hogging.

Table 4.2.3 wave bending rule loads, equal for reference and Air cavity

Wave bending CSR	Sagging	Hogging
Minimum rule value	5473600	4767300

Table 4.2.4 Shear forces rule loads, equal for reference and cavity ship

Shear forces	Positive	Negative
$Q_{sw\text{-}perm\text{-}sea}$	39600	39600 (Absolute value)
$Q_{sw\text{-}perm\text{-}harb}$	12400	12400 (Absolute value)
Q_{wv_max}	56500	52000 (Absolute value)

4.3 Primary results

4.3.7 Max cargo intake capacity, (Maxsurf hydrostatic model)

	Reference Ship	Cavity Ship
Cargo tanks volume [m^3]	176500	164000
Cargo tanks mass [ton]	157000	146000
Draught (simulation) [m]	15.7	15.3
Draught difference to reference ship [m]	0	0.4

Table 4.3.1 Primary results, steel weight and cargo volume for reference ship and cavity ship (note only midship region)

Midship section	Reference ship	Cavity ship
Relative steel weight	100%	105.6%
Relative volume	100%	93.2%

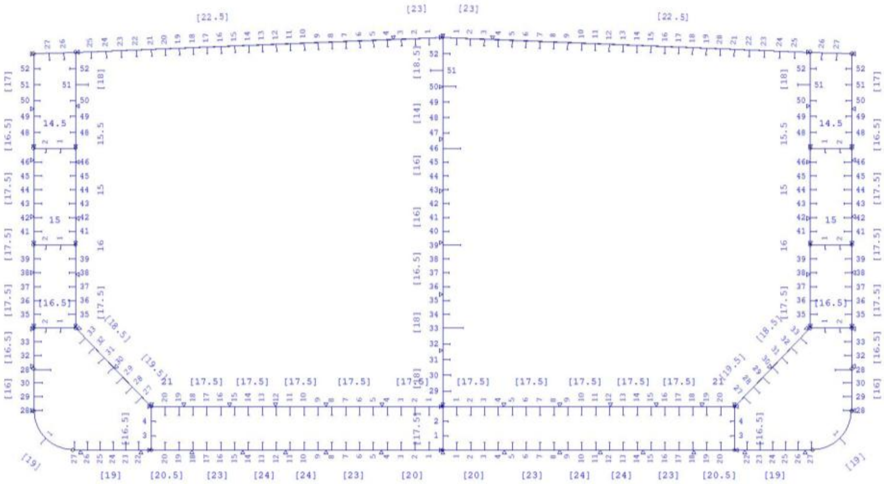


Figure 4.3.1 Cross section of reference ship

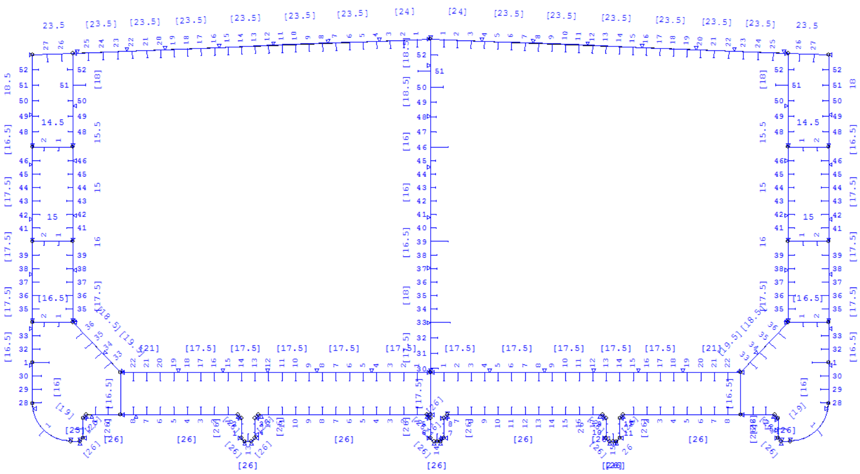


Figure 4.3.2 Cross section of cavity ship

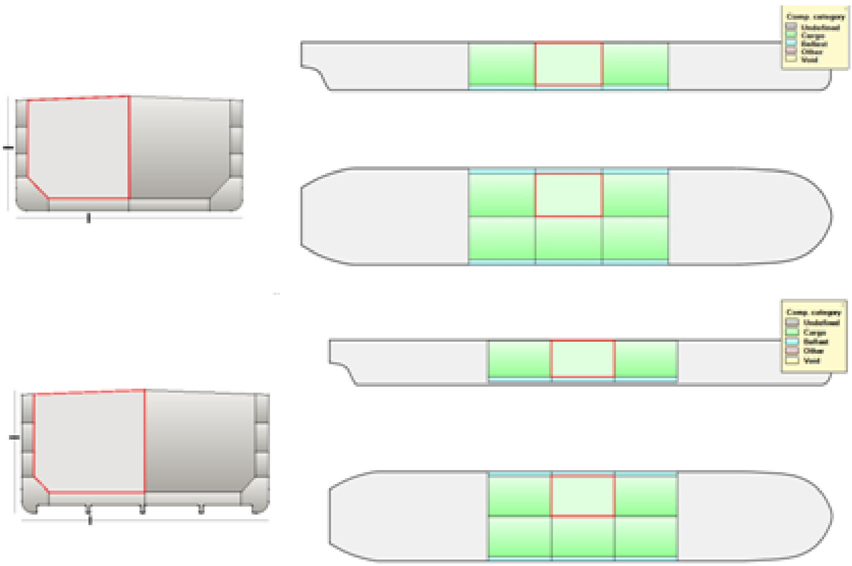


Figure 4.3.3 Tank arrangement

For the midship cross section the *net vertical hull girder section modulus* (Zv-min) is not to be less than the rule minimum hull girder section modulus, see table 4.3.2.

Table 4.3.2 Minimum requirements and actual values, moment of inertia and section modulus for the cross sectional design

	Inertia (m^4)	Zv-min Section modulus (m^3)
Minimum rule requirement	355	44
Reference ship actual 0.4L amidships value (margin)	546 (154%)	At deck: 43 (97%)
		At bottom: 52 (118%)
Cavity ship actual 0.4 amidships vessel (margin)	548 (155%)	At deck: 43 (97%)
		At bottom: 53 (120%)

The *net hull girder section modulus* about the horizontal neutral axis (Zv-net50) is not to be less than the rule required hull girder section modulus (Zv-req) see table 4.3.3. Neutral axis height is seen in table 4.3.4.

Table 4.3.3 Section modulus requirements for amidships section, based on static and dynamic bending moment

	Zv-req [m^3] (midship section)	
	Sagging	Hogging
Static	22.6	43.4
Static+Dynamic	56.3	68.1

Table 4.3.4 Height of neutral axis for Reference vessel and cavity vessel

	Neutral axis height (mm)	Difference (mm) Ref. vessel
Reference vessel	10454	0
Cavity vessel	10396	58

The placement of the neutral axis for the reference vessel are showed in fig 4.3.4.1.

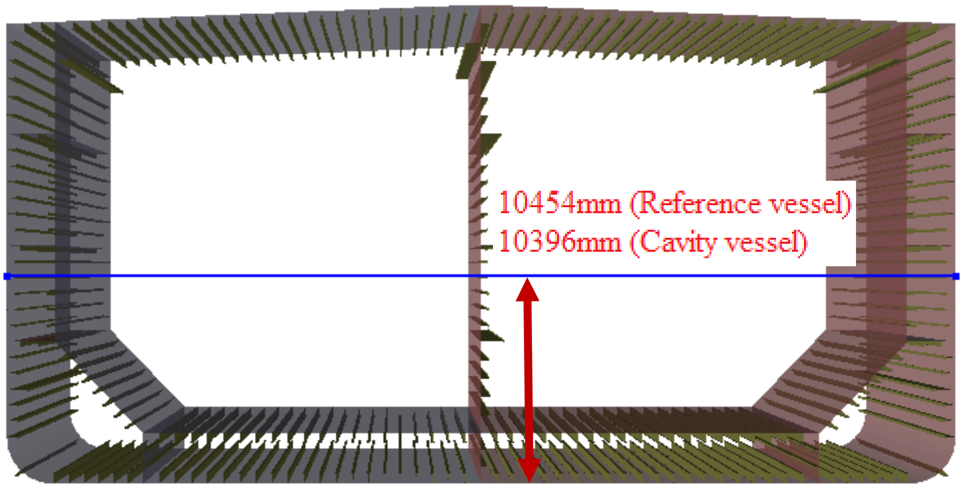


Figure 4.3.4.1 The location of the neutral axis for the reference vessel.

Rule based stress utilization of cross section midship design is studied. The values indicate the stresses relative the rule based acceptable stress limits. This means 100% (green) indicates the stress complies with the rules while above 100% indicates a margin and less than 100% there is a gap to the rule requirements that needs redesign. For stiffeners see figure 4.3.5, for plates see figure 4.3.6. The factor of utilization is seen in figure 4.3.4. The utilization comparison is based on equations from the adopted requirements, CSR 2010. The approved design needs to fulfil requirements for FE analysis, which might explain the low utilization indicated for the reference ship in figure 4.3.5 and figure 4.3.6. Note that plate, material and stiffeners differ for the two designs.

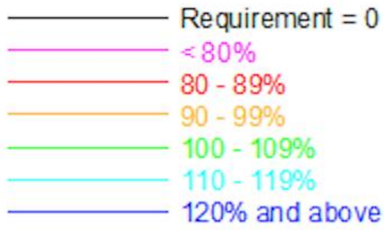


Figure 4.3.4 Utilization scale

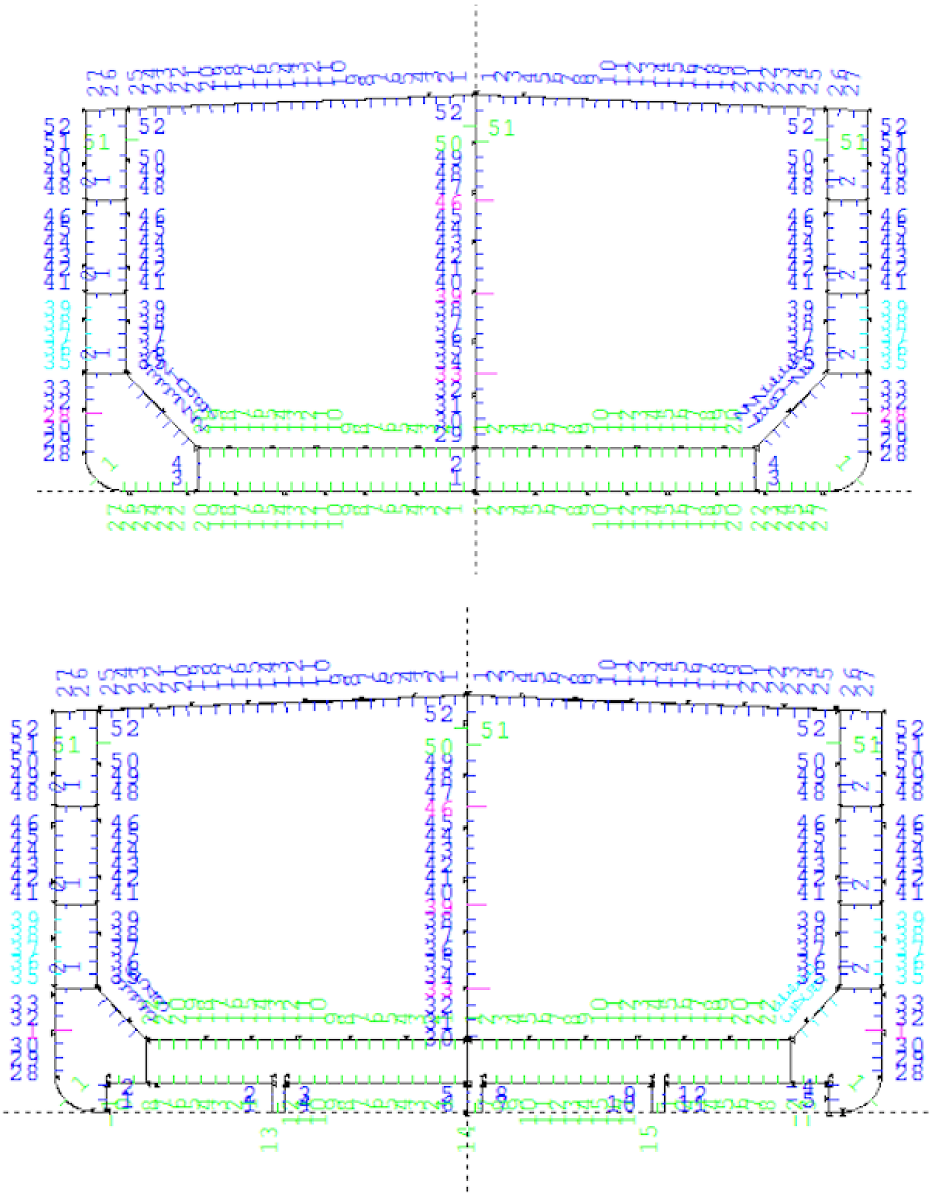


Figure 4.3.5 Utilization stiffeners

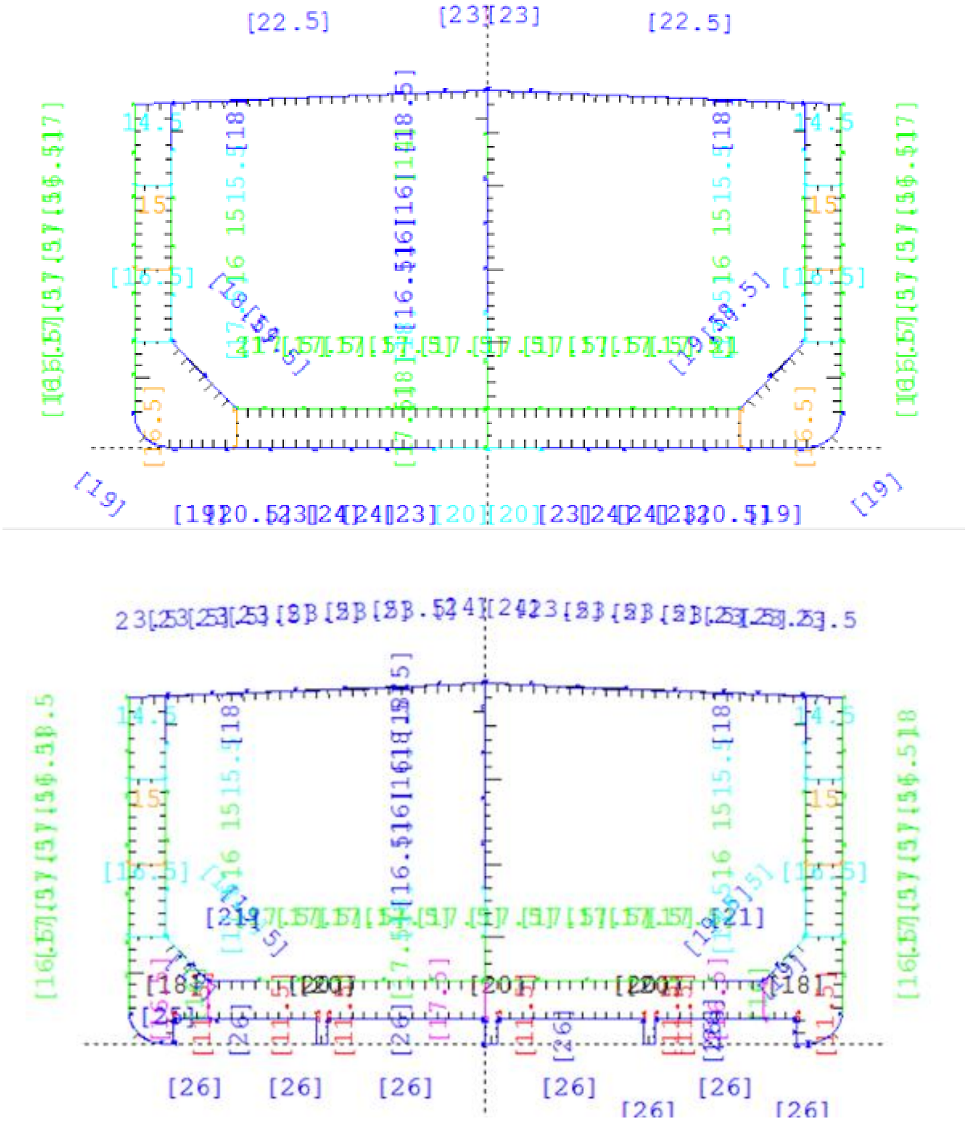


Figure 4.3.6 Utilization plates

4.4 Cargo hold FE Analysis

The following results are presented is the Von Mises stress for the structure, the region analysed is the middle tank within the three cargo hold model. This region has been analysed to minimise the effect from boundary disturbances within the FEA model.

The scale used for presenting the results can be seen in Figure 4.4.1 and are the same for load case B11_Harbour and B6_1. The scale reaches from 0 MPa to 280 MPa and above. All the permitted stress levels for each plate segment can be seen in chapter 3.3 table 3.3.1 which corresponds to the colour coding.

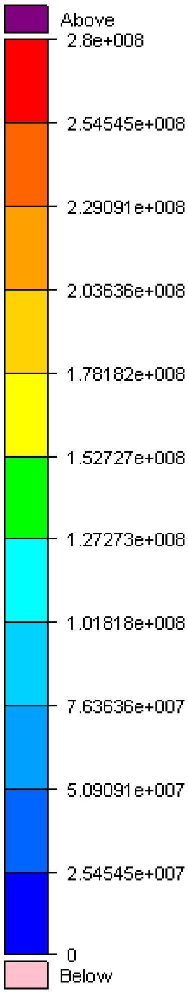


Figure 4.4.1

4.4.1 Maximum hogging, static load (B11_Harbour)

The stress distribution in the deck for the reference and the cavity ship can be seen in Figure 4.4.1.1. It can be seen that for the cavity ship the higher stresses are more located around the centreline of the ship. A uneven stress distribution at the deck centreline are also identified, since the mesh are well distributed and the correct boundary conditions are applied, this area will need to be further investigated. The uneven stress distribution might be one effect from that the cavity has a offset at 0.5m from the centreline, this might be a reason to the unsymmetrical stress distribution that also accrue in figure 4.4.1.3. This will need further investigation.

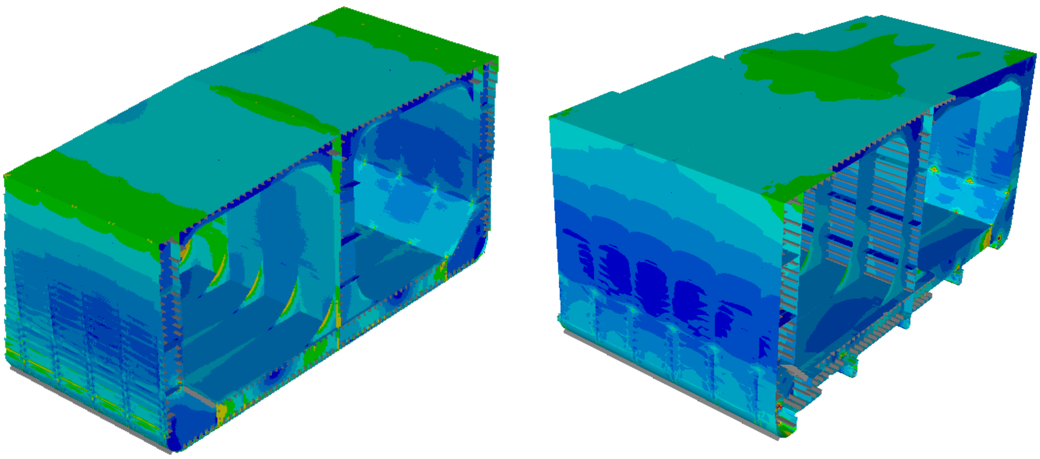


Figure 4.4.1.1 The stress distribution in the deck for the reference and the cavity ship

In the bottom structure the stress distribution differ from the reference and the cavity ship, in Figure 4.4.1.2 it can be seen that the stresses has decreased within the cavities and some higher stresses can be found on the cavity walls.

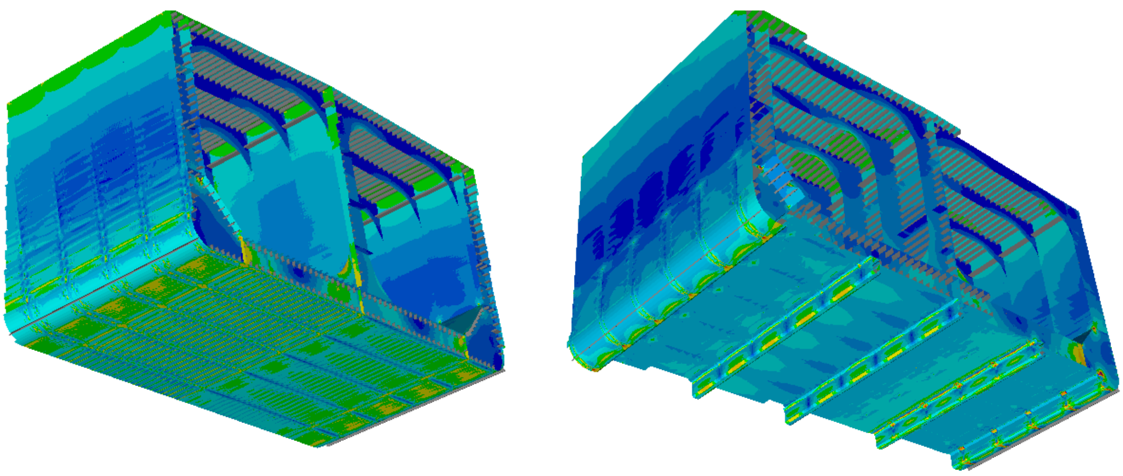


Figure 4.4.1.2 The stress distribution in the bottom structure for the reference and the cavity ship

The stress distribution in the web frames can be seen in Figure 4.4.1.3 and it can be seen that the stresses are focused around the hopper tank corners and at the starboard lower corner within the port tank. High stresses within the hopper tank corners are a well-known problematic area and it will also be that for the cavity, however unlike the high stresses within this area for the reference the cavity ship has high stresses within the cavity at the bilges.

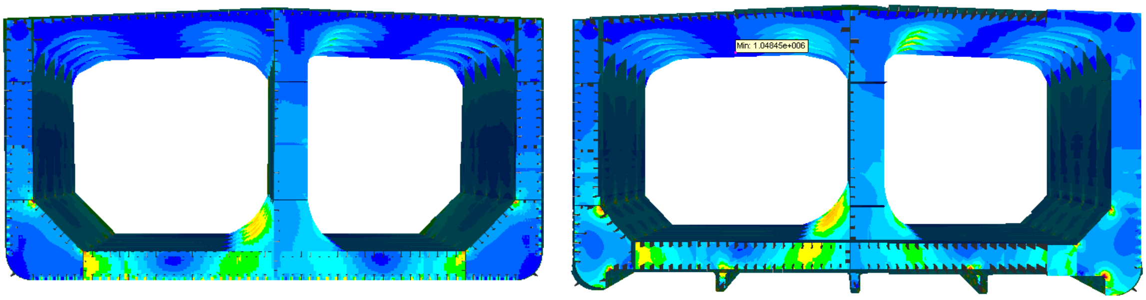


Figure 4.4.1.3 Stress distribution on the web frames

The highest stress within the hopper tank corners for the reference and within the cavity walls at the bilges for the cavity ship can be seen in Figure 4.4.1.4.These areas are identified as the most critical areas for the two vessels for this load case.

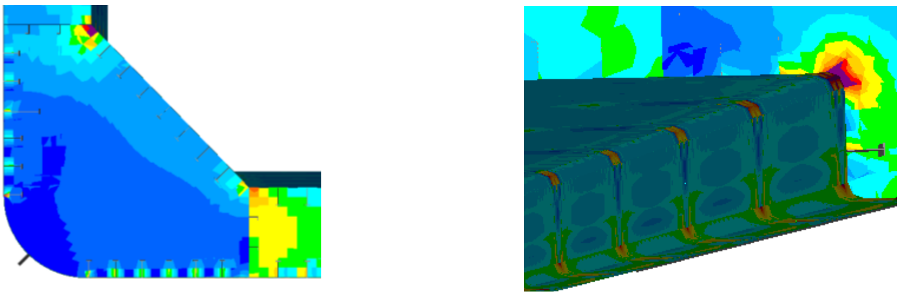


Figure 4.4.1.4 The most critical areas

4.4.2 Maximum sagging, dynamic load (B6_1)

The stress distribution in the deck for the reference and the cavity ship can be seen in Figure 4.4.2.1. It can be seen that for the cavity ship the higher stresses are more located around the starboard and port side and at the centreline of the ship, the stresses has also decreased in some well defined areas on the deck, a slightly different stress distribution can also be seen on the outer shell.

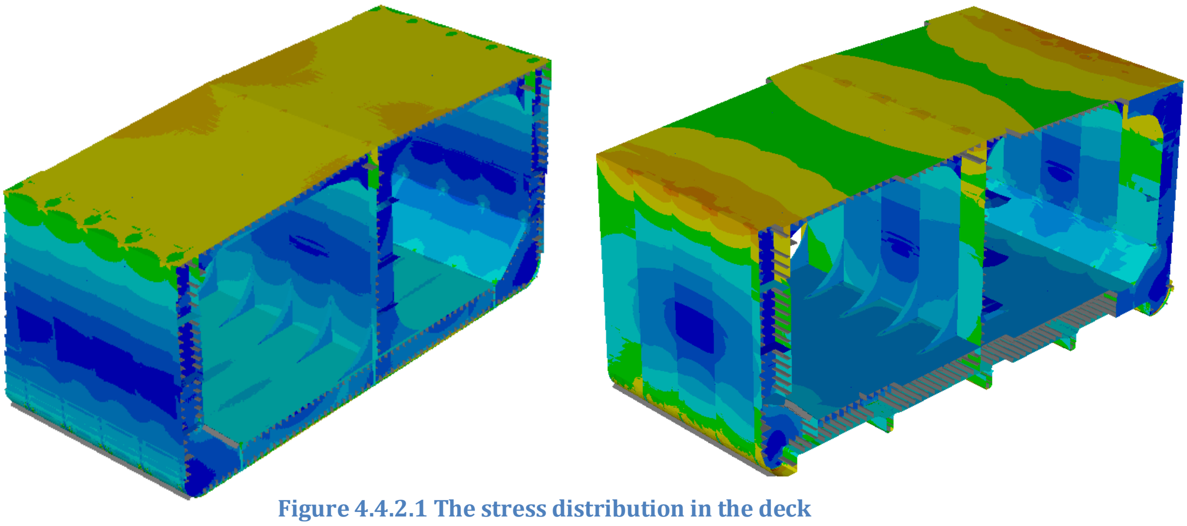


Figure 4.4.2.1 The stress distribution in the deck for the reference and the cavity ship

In the bottom structure the stress distribution differ from the reference and the cavity ship, in Figure 4.4.2.2.It can be seen that the stresses has decreased within the cavity and some higher stresses can be found on the cavity walls and at the bilges.

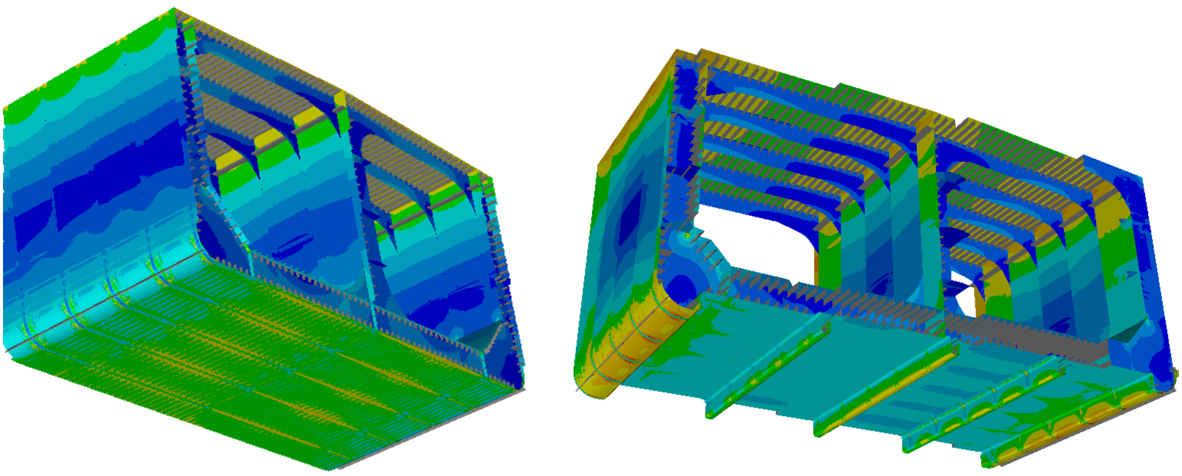


Figure 4.4.2.2 The stress distribution in the bottom structure for the reference and the cavity ship

The stress distribution in the web frames can be seen in Figure 4.4.2.3 and it can be seen that the stresses are focused around the hopper tank corners, however unlike the high stresses within this area for the reference the cavity ship has high stresses within the cavity at the centreline.

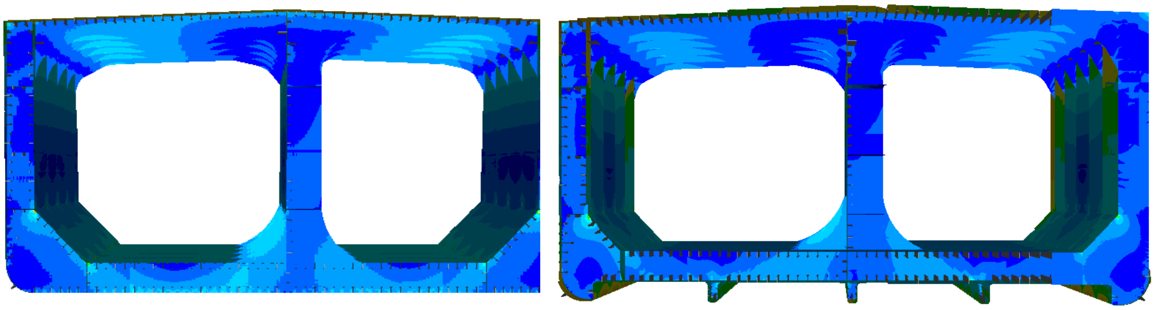


Figure 4.4.2.3 Stress distribution on the web frames

The highest stress within the hopper tank corners for the reference and within the cavity walls at the centreline for the cavity ship can be seen in Figure 4.4.2.4 this areas are identified as the most critical areas for the two vessels for this load case.

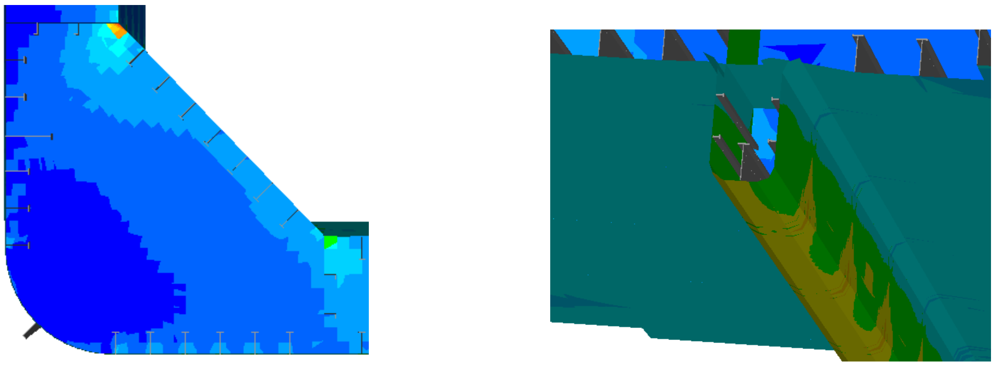


Figure 4.4.2.4 The most critical areas

4.5 Docking FE Analysis

The following results which are presented is the Von Mises stress for the structure, the region analysed is the space between two ordinary web frames.

The scale used for presenting the results can be seen in Figure 4.5.1 and it is the same for the following load case, docking on ordinary web frame at both five and three keel supports and for docking on docking brackets between two ordinary web frames on five and three supports

The scale reaches from 0 MPa to 200 MPa and above.

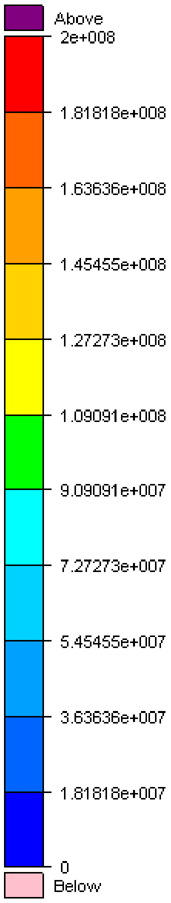


Figure 4.5.1

4.5.1 Docking on ordinary frame 5 keel supports

The stress distribution in the web frames can be seen in Figure 4.5.1.1 and it can be seen that the stresses at the reference ship are focused around the hopper tank corners, the stresses for the cavity ship are mainly located around the outer cavities and at the centreline.

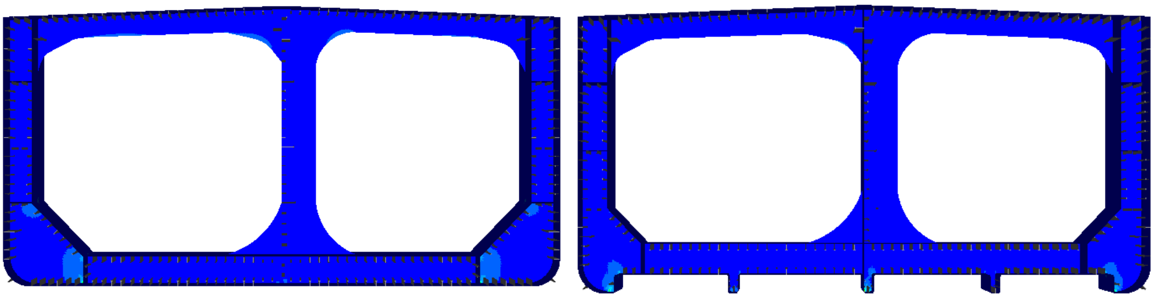


Figure 4.5.1.1 Global stress distribution

The highest stress within the hopper tank corners for the reference ship and within the cavity walls at the centreline for the cavity ship can be seen in Figure 4.5.1.2 this areas are identified as the most critical areas for the two vessels for this load case.

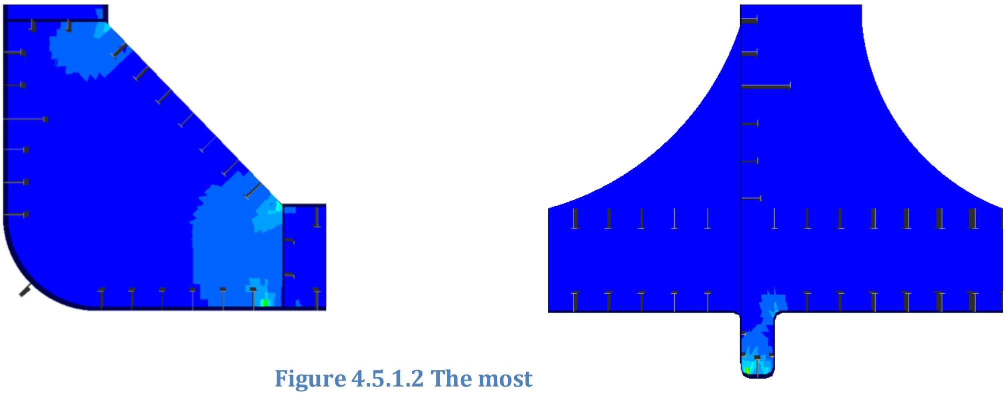


Figure 4.5.1.2 The most critical areas

4.5.2 Docking on ordinary frame 3 keel supports

The stress distribution in the web frames can be seen in Figure 4.5.2.1 and it can be seen that the stresses at the reference ship are focused around the hopper tank corners, the stresses for the cavity ship are mainly located around the outer cavities and at the centreline.

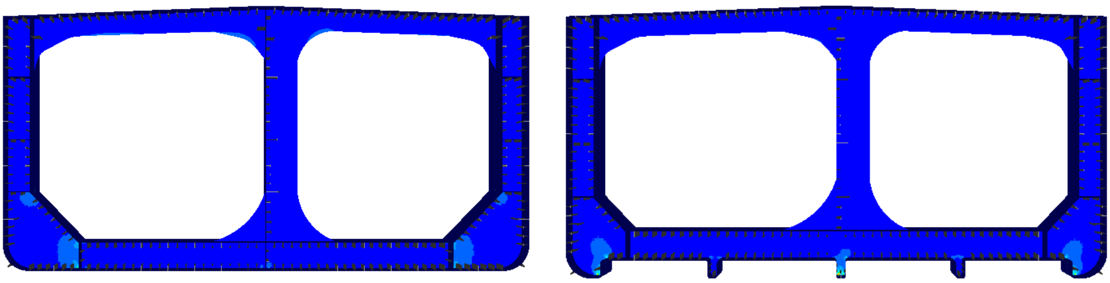


Figure 4.5.2.1 Global stress distribution

The highest stress within the hopper tank corners for the reference ship and within the cavity walls at the centreline for the cavity ship can be seen in Figure 4.5.2.2 this areas are identified as the most critical areas for the two vessels for this load case.

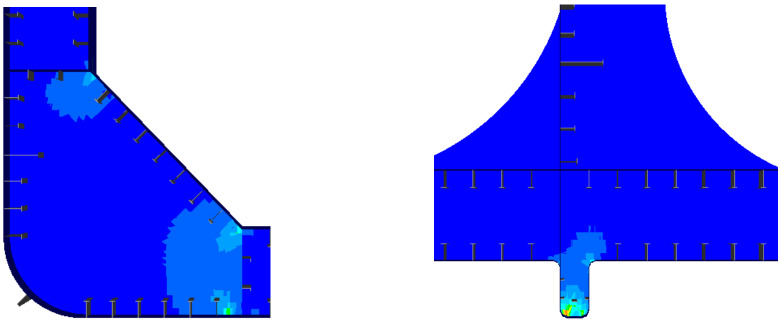


Figure 4.5.2.2 The most critical areas

4.5.3 Docking on docking brackets,5 keel supports

The stress distribution at the docking brackets can be seen in Figure 4.5.3.1 and it can be seen that the stresses for the reference ship are focused around the hopper tank corners, the stresses for the cavity ship are mainly located at the docking brackets at the cavities to the starboard and port side of the centreline.

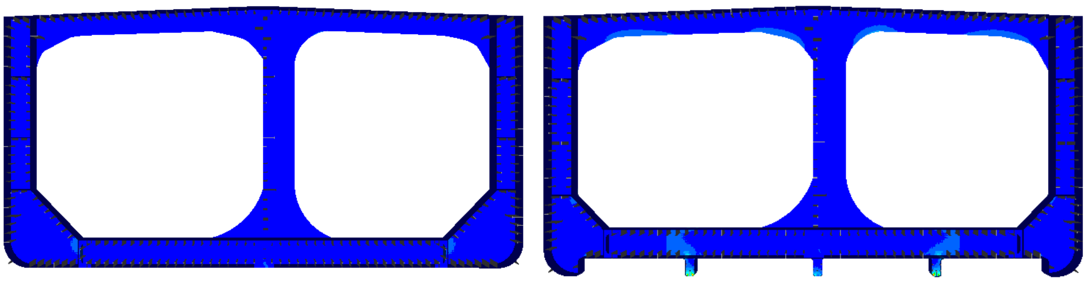


Figure 4.5.3.1 Global stress distribution

The highest stress within the hopper tank corners for the reference ship and the docking brackets at the cavities to the starboard and port side of the centreline can be seen in Figure 4.5.3.2. This areas are identified as the most critical areas for the two vessels for this load case.

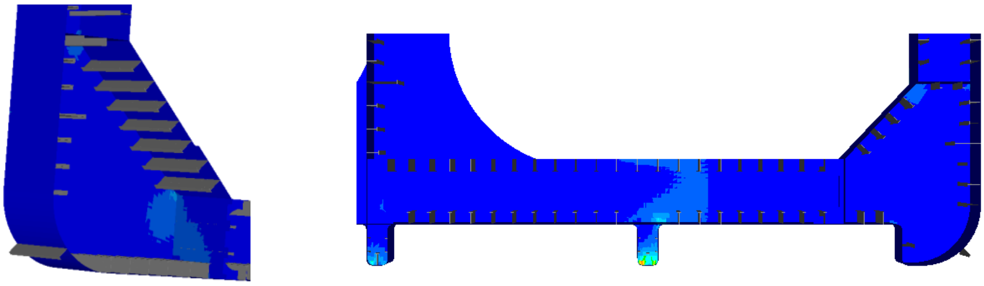


Figure 4.5.3.2 The most critical areas

4.5.4 Docking on docking brackets, 3 keel supports

The stress distribution at the docking brackets can be seen in Figure 4.5.4.1 and it can be seen that the stresses at the reference ship are focused around the hopper tank corners, the stresses for the cavity ship are mainly located at the docking brackets at the centreline. Increased stresses at the cavity web frame can also be seen.

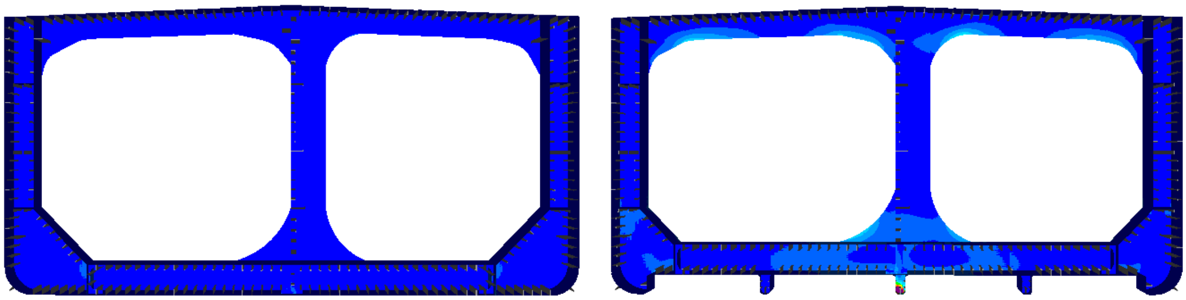


Figure 4.5.4.1 Global stress distribution

The highest stress within the hopper tank corners for the reference ship and the docking brackets at the centreline for the cavity ship can be seen in Figure 4.5.4.2. This areas are identified as the most critical areas for the two vessels for this load case. The stresses at the centreline docking bracket at the cavity ship are identified as the most critical area for all docking load cases.

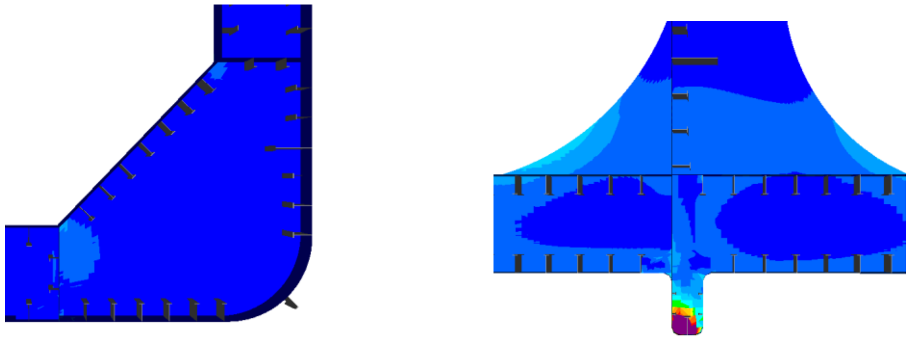


Figure 4.5.4.2 The most critical areas

5 Financial implication

The purpose to cover financial implications is to get a picture of how the Air cavity application impacts the economy. A financial case is given for the reference ship. Some facts are business confidential why some assumptions will have to be done.

The Air cavity design has an increased steel weight, more complex structure but a lower fuel consumption, which means increased investment cost but lower bunker cost. Air cavity ship cargo volume is lower compared to the reference ship. In fully loaded condition this would be a loss of income but wouldn't affect the income in a partially loaded condition. This assumption is related to market aspects, which is not elaborated on further. The efficiency of the cavities is affected by the load case because the wet surface fraction of the total resistance is affected by the draught.

The load case compared is 70% of fully loaded reference ship. Frictional resistance is based on this load case; however details regarding bunker consumption for the reference ship are not disclosed by Stena.

The load case compared is based on data from resistance and reference bunker cost

With this as background the results should be considered as an indication rather than a scientific investigation. Here follows a list of given as well as assumed data.

Given data:

- New building cost reference ship 70.000.000 USD
- Delivery cost reference ship 7.000.000 USD
- OPEX 9000 USD/day
- Bunker cost reference ship 12000 USD/day
- Bunker price 300 USD/m.t.(2015-06-08)

Assumptions:

- Both vessels operate at a draught corresponding to 70% of ref vessel
- Air cavity reduces the total frictional resistance by 15% including the power needed for maintaining the air pressure in the cavities, according to Stena Teknik.
- Wave resistance is unaffected
- Average speed for a Suezmax tanker is 13kn according to Stena Teknik.
- Resistance linear relation to bunker consumption
- Deprecation 20 years, Stena Teknik
- CAPEX 9%
- 6% increased steel weight for the amidships region means approx. total steel weight increase by 3%.
- Estimated steel weight of Suezmax, 18,800 ton, total light weight 24,000 ton. Assuming 3% increase would mean 550 ton. One could assume cost for additional steel at 1500$/ton, that would give an additional cost 550x1500=830,000$ for fabrication as a minimum.
- Additional piping, compressors and control system for maintaining the air pressure in the cavity is roughly estimated to $1,000,000
- The air cavity will involve new and different type of analyses and studies for the yard, they are estimated to $500,000.

- Since this is a novel design with associated uncertainties a normal yard will not just add a net cost for steel and equipment. It will mean additional design cost, for both design and more difficult production compared to a standard vessel. One can assume that they see all kinds of problems and that is a risk for a yard. They will not offer on same basis as a standard vessel with some different, more expensive, but known equipment or maker of equipment. They will see this as a risk and add maybe 5% to the standard contract price.
- Assumed additional costs for the Air cavity system, see table 5.1.

Frictional resistance see figure 5.1 and table 5.1

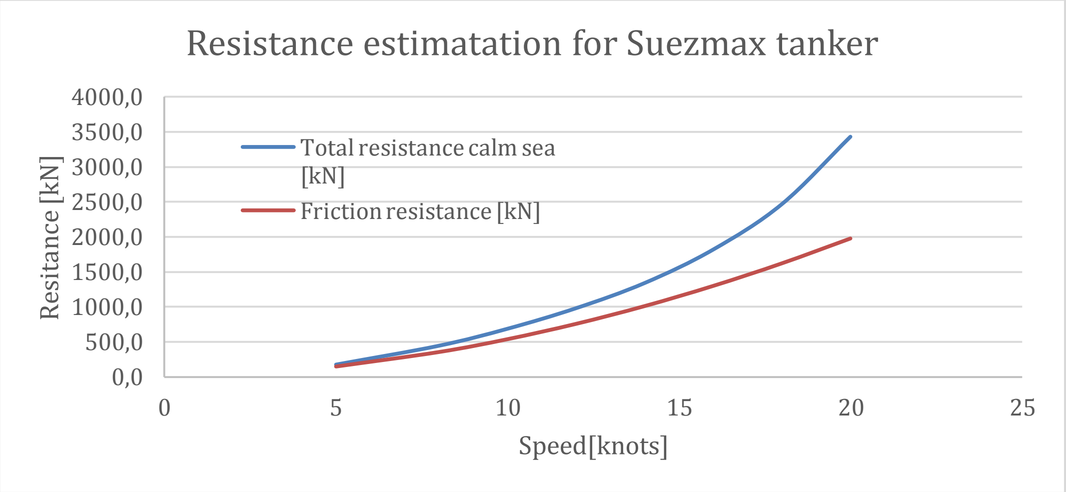


Figure 5.1 Resistance estimation for a Suezmax tanker

Table 5.1 Resistance estimation for a Suezmax tanker

Speed [knots]	Residual resistance [kN]	Friction resistance [kN]	Total resistance calm sea [kN]	Fraction of friction resistance
5	33	147	180	0,82
8	94	354	448	0,79
10	153	538	690	0,78
12	226	758	983	0,77
14	330	1012	1342	0,75
16	518	1301	1819	0,72
18	842	1624	2465	0,66
20	1447	1980	3427	0,58

Note this calculation does not take into account how the ship is loaded. Reasonable assumption is that the vessel seldom sails fully loaded, i.e. sails so that the volume decrease of the Air cavity design would impact the earnings. For in-depth study all the values assumed should be assured and loading statistics should be reviewed.

The result indicates that “break even” for the Air cavity design must meet an efficiency that lowers the ships total frictional resistance by 26%, which corresponds to a decreased bunker cost of 19%

In this study “break even” is met with a negative margin (-2%), equivalent to -982 USD / day, see calculations in table 5.2.

Table 5.2 Financial calculation for the reference ship and Air cavity ship

NB	Std ship	Diff comment	Diff %	Air cavity case
NB price ref ship	70000000			70000000
Delivery cost 10%	7000000			7000000
Additional steel	0			830000
Additional piping etc.	0			1000000
New design analyses	0			500000
Risk premium 5%	0			3500000
Tot building cost	77000000		8%	82830000
Finacial				
CAPEX	9			9
Ineterest / year	6930000			7454700
Write-off time [years]	20			20
Write-off/year	3850000			4141500
Financial cost/year	10780000			11596200
Days per year	350			350
Financial cost/day	30800		8%	33132
OPEX/day	9000		0%	9000
Resistance (e.g. aspect which will be affected)				
Original total recistance [kN]	1163	Linear interpolation		
% frictional	*75%*			
Frictional [kN]	872	Factor of fric res	85%	741
% wave	*25%*			
Wave [kN]	291	Factor of wave res	100%	291
Sum resistance	1163			1032
Bunker cost [USD/day]	12000	% reduction	11%	10650
Cost per day [USD]	51800			52782
Costsaving per day				-982
Costsaving per day %				-2%
Analysis break even				
Additional finacial cost				2332
Initial bunker cost/day				12000
Needed bunker saving to meet financial added cost				19%
Needed frictional reduction to meet added finacial cost				26%

6 Discussion

This section discusses the methodology used in this thesis, compares the results, differences and similarities followed. The discussion is divided into two parts, 6.1 hydrostatics and 6.2 structure.

6.1 Hydrostatics

The hydrostatic study is done with purpose to verify the initial design. The results indicates that both models (reference ship and Air cavity ship) complies with general requirements, 2008 IS code. As the hydrostatics theory claim, chapter 2.4, the introduction of Air cavities in the bottom structure equals introduction of free liquid surfaces. With this background the stability performance of the Air cavity ship should decrease in relation to the reference ship, which the results show.

The stability analyses performed are based on three loading conditions, ballast loaded, full loaded and slack tanks for the reference ship and the Air cavity ship. When the computer model of the reference ship is compared with real stability results from the reference ship, the stability results correspond to an 80% level. Given the prerequisites this correspondence is considered acceptable.

The tests are performed in the software Maxsurf and the models of the hulls are made in Rhino. The hull lines are provided by Stena Teknik and based on a similar vessel. The lines are modified to match the reference ship and the Air cavity ship as close as possible, however this means the model differ compared to the existing reference ship. There might be some difference between the model and the real reference ship regarding loading condition. In the computer model all tanks are left with minimum 1% level if they are considered empty to compensate for the risk of free liquid surface effect, this information is not available from the real test.

Reference ship maximum GZ-value occurs at a heeling angle of 25 deg., which tangents the minimum required value. It should be mentioned that the classification society could accept deviation from this value down to 15 deg. in some cases, (IMO 2008a).

The results show that the most severe loading condition, for maximum GZ and GM, for the Air cavity ship occurs at the 90% and 50% air in the cavities. Note the range of validation for these tests are limited with respect to heeling angle. The explanation is the free liquid surface effect in the cavities, why the ship becomes more stable with 0% air in cavities despite increased draught. In other words the computer tests indicate that the ship becomes more stable if the air escapes, which is a desirable property in an emergency situation.

In a failure mode there could be loss of air in one or in several cavities. In a severe case were the cavities fail and the ship is having a heel one can imagine a system were the two outer cavities are connected and controlled separate from the two inner cavities, i.e. the level of air is held on equal level in the two outer cavities. In an emergency situation were air is lost in one or more cavities a heel could be avoided. A more advanced control system could be designed to compensate for steady heel.

Further it has been discussed how a fast release of air from the cavities will work as an emergency brake, (Shiri. A. 2012). The effect from a failure, redundancy and the design of the systems is a challenge for future work.

6.2 Structure

The structural study is the main focus in this thesis. The purpose of this study is to investigate and evaluate the possibility and the effects of introducing the Air cavity design as an alternative bottom structure in the conventional Suezmax tanker.

By doing this investigation, a wider understanding of how the design process of a ship is gained. This includes the network of rules and requirements, effects of changes in the structure as well as working with commercial software have to model and simulate the strength and stresses of a marine structure.

The aim of this study was to compare a conventional Suez Max tanker with a Suez Max tanker with Air Cavity. By introducing an Air cavity bottom structure, the bottom was moved upwards in relation to the reference. Since most of the other dimensions was kept constant, it was expected that the volume of the cargo tanks would decrease, which the results indicates, chapter 4.3

The hull is subjected to heavy forces and bending moments, which could be, expressed as static and dynamic forces in accordance to the CSR. Therefore these forces have been calculated when the structure have been investigated. Limiting design parameters were both global and local according to the CSR. The major design difference is the cross sectional geometry, while local loads such as water pressure only were subjected to minor changes.

The major design difference for the rule requirements was the longitudinal section modulus. Section modulus is based on the material used and the thickness and its cross sectional geometry.

What might be a point of interest is how the neutral axis is moved upwards when the bottom structure is moved upwards, in turn, this affect loads acting on the deck structure, since the distance from deck to neutral axis has changed. Consider global and local requirements simultaneously as changes to thickness or geometry in one part of the structure affects other parts of the structure makes a complex equation.

By analysing the structure with an FE-analysis, which was well described in the CSR, the stresses for all different rule loads could be investigated.

The result from the FE-analysis, chapter 4.4 are highly dependent on the setup and its boundary conditions, these were applied strictly in accordance with the CSR and it was assumed that it was a fare god representation of the reality. The mesh size, the type of mesh and its distribution has also a great impact on the results. Regarding our results, the mesh size could be a possible source of error within our simulations; this due to the limited computational power which didn't allowed making a refined mesh within some specific regions in accordance to the CSR. However, since this is a comparable study it is not considered as a problem since the potential sources of error will be the same in both cases. According to this, this is considered an overall good representation of the reality was achieved.

That the global stress would be different for the two designs was expected but its distribution and its magnitude were unknown, this which now could be identified, chapter 4.4.

A major reason for a shift in the stress distribution between the two designs could be the material added to meet the rule requirements regarding the section modulus. The material was primarily added in the deck and bottom structure, this could also be a contribution to the higher weight for the cavity structure. Geometrical changes could also be contributed to a shifted stress distribution, especially in the bottom structure around the cavities, maybe more focus on designing the radius within the cavities could reduce stresses in that region.

Regarding the results from the docking, chapter 4.5, the placement of the keel supports might affect the stresses in the bottom structure. The reason why the supports were placed on the cavities was to investigate a worst-case scenario if a docking procedure should go wrong. An alternative way to place the supports might be in between the cavities.

A cavity structure has been introduced and some of the most critical areas has been identified with this study, but to figure out the optimal design for a Suezmax tanker with an air cavity hull, further work will be needed.

7 Conclusion

The static stability is affected by introducing an Air cavity system. However the effects are considered small, well within the requirements. Based on this knowledge and previous study regarding the reduction of friction, which has been carried out by Chalmers, Stena and SSPA the concept of Air cavity design may be a milestone within the development for the next generation of hull designs. In this thesis, some questions have been answered. The questions are regarding the steel weight and the cargo capacity, the 6% of midship weight increase and the 7% of midship reduced cargo volume are some of the issues future engineers has to struggle with when making the trade of between a light weight design, structural strength, cargo capacity and profitability. All this with respect to the environmental benefits which the lowering of the viscous resistance contributes to.

The structural design of the Air Cavity concept will need further development to reach an optimal and fully CSR approved design. However this thesis has highlighted the problematic areas on a global approach and we see no technical reason why this concept not should be taken to the next level.

The economic study is performed as a complement to the thesis, tanking the results from the technical study into consideration. The result indicates that “break even” for the Air cavity design must meet an efficiency that lowers the ships total frictional resistance by 26%, which corresponds to a decreased bunker cost of 19%. In this study “break even” is met with a negative margin (-2%), equivalent to -982 USD / day.

The new building cost is 8% higher compared to the reference ship, which can’t be met by the bunker savings. This implies there is no profitable business case taking this scenario into account. Additionally the cargo volume is decreased. A market analysis needs to be performed to study the effect of a ship operating with decreased cargo volume.

8 Future work

To find an optimal design for the Air Cavity, further work has to be carried out. It is not only the hydrostatic stability and structural issues for the design which has to be optimised, except of iterate the design to get safe and economical profitable structure some of the following will need to be investigated as well:

- Evaluate the risk and consequence of a failure of the Air cavity system.
- Further investigate the stresses within the cavities
- Investigate the uneven stress distribution on deck for load case B11_Harbour.
- Iterate the design loop with respect to weight and strength.
- Investigate the possibility to use a "rubber Air Cavity" which could be added or removed, this would add extra flexibility.
- Further investigate the arrangement for extra equipment needed for the cavity, compressors, pumps, piping and control system, etc
- Operability study, cargo intake and draft limitations in ports.
- Refine economic scenario, add market analysis

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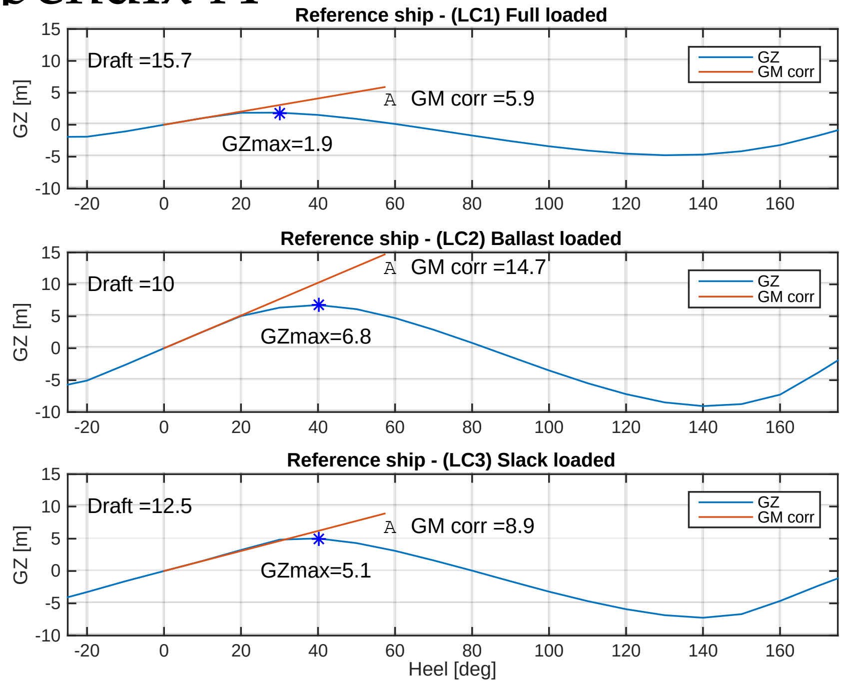
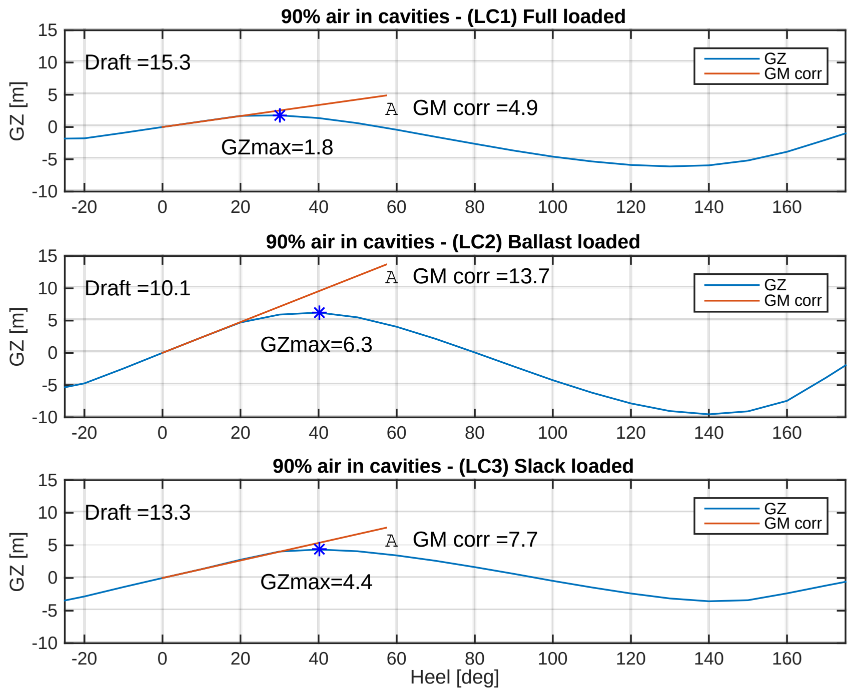
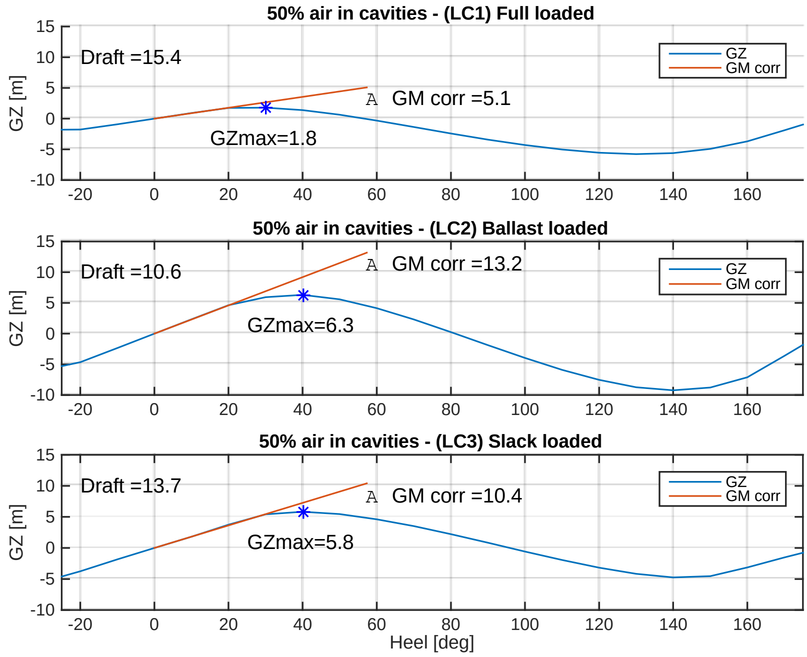
Appendix A



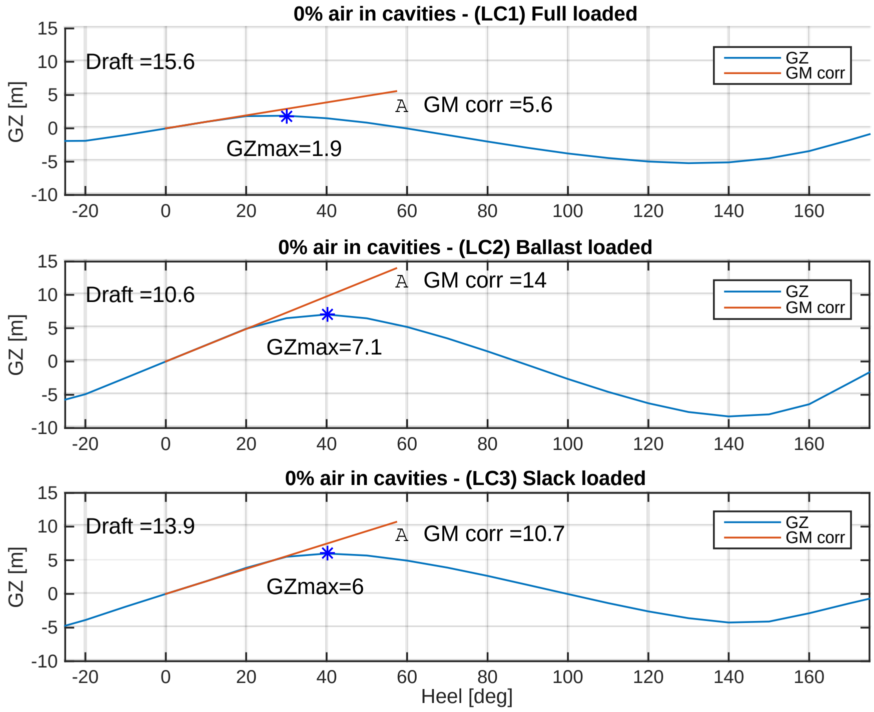
			Reference vessel						
			LC1		LC2		LC3		
Draught amidships [m]			15,7		10.0		12,5		
No.Cr	Require-ment	Unit	Actual	Margin %	Actual	Margin%	Actual	Margin%	Status
1	0.055	Metre-radians	0,7	1160	1,9	3370	1,3	2220	Pass
2	0.09	Metre-radians	1.0	1000	3,1	3310	2,2	2300	Pass
3	0.03	Metre-radians	0,3	920	1,2	3770	0,9	2850	Pass
4	0.2	m	1,9	850	6,8	3290	5,1	2470	Pass
5	25	Degree	25	0	39	56	36	45	Pass
6	0.15	m	5,9	3820	14,7	9670	8,9	5820	Pass
7	16	Degree	0,1	99,6	0,1	99,3	0,1	99,4	Pass
8	80	%	0,5	99,3	0,3	99,6	0,4	99,5	Pass
9	100	%	230	130	300	200	338	238	Pass



			Cavity ship [Normal operation 90% air in cavities] valid within -1.1<deg.> 1.1						
			LC1		LC2		LC3		
Draught amidships [m]			15,3		10,1		13,3		
No.Cr	Require-ment	Unit	Actual	Margin %	Actual	Margin%	Actual	Margin%	Status
1	0.055	Metre-radians	0,6	1050	1,8	3150	1,1	1930	Pass
2	0.09	Metre-radians	0,9	920	2,9	3080	1,9	1990	Pass
3	0.03	Metre-radians	0,3	850	1,0	3500	0,8	2450	Pass
4	0.2	m	1,8	810	6,3	3030	4,4	2120	Pass
5	25	Degree	26	1	38	53	35	38	Pass
6	0.15	m	4,9	3190	13,7	9010	7,7	5030	Pass
7	16	Degree	0,1	99,4	0,1	99,3	0,1	99,3	Pass
8	80	%	0,6	99,2	0,3	99,6	0,4	99,5	Pass
9	100	%	230	130	299	199	330	230	Pass



			Cavity ship [Maximum free surface 50% water in cavities] valid within -9.1<deg.> 9.1						
			LC1		LC2		LC3		
Draught amidships [m]			15,4		10,6		13,7		
No.Cr	Require-ment	Unit	Actual	Margin %	Actual	Margin%	Actual	Margin%	Status
1	0.055	Metre-radians	0,6	1070	1,8	3090	1,1	1960	Pass
2	0.09	Metre-radians	0,9	930	2,8	3060	1,9	2030	Pass
3	0.03	Metre-radians	0,3	850	1,1	3520	0,8	2510	Pass
4	0.2	m	1,8	800	6,3	3070	4,6	2180	Pass
5	25	Degree	26	2	38	53	35	38	Pass
6	0.15	m	5,1	3320	13,2	8730	10.4	6930	Pass
7	16	Degree	0,1	99,6	0,1	99,3	0,1	99,1	Pass
8	80	%	0,5	99,4	0,3	99,6	0,6	99,3	Pass
9	100	%	230	130	310	210	330	230	Pass



			Cavity ship [Fail case 100% water in cavities, no free surface]						
			LC1		LC2		LC3		
Draught amidships [m]			15,6		10,6		13,9		
No.Cr	Require-ment	Unit	Actual	Margin %	Actual	Margin%	Actual	Margin%	Status
1	0.055	Metre-radians	0,7	1130	1,9	3307	1,2	2080	Pass
2	0.09	Metre-radians	1	990	3,1	3320	2,1	2150	Pass
3	0.03	Metre-radians	0,3	910	1,2	3900	0,8	2670	Pass
4	0.2	m	1,9	850	7,1	3440	4,8	2310	Pass
5	25	Degree	26	2	40	60	36	42	Pass
6	0.15	m	5,6	3620	14.0	9220	10.7	7130	Pass
7	16	Degree	0,2	99,1	0,1	99,4	0,1	99,2	Pass
8	80	%	1,2	98,5	0,3	99,6	0,5	99,3	Pass
9	100	%	230	130	320	220	340	240	Pass