Structural design of Platform Supply Vessel less than 90m

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ABSTRACT

The concept behind Structural design of ship is making sure that the design satisfies a number of objectives: withstand the loads it is subjected to over its life span, fulfill the classification society requirements, and become economically viable. It is not a onetime process; rather an initial design is made and reiterated repeatedly until an adaptable solution that strikes a balance among conflicting objectives is achieved.

Demand for fuel has attracted significant investment in offshore exploration more than ever. This has created demand for offshore support vessels in general and platform supply vessel in particular. The offshore activities are moving to deeper waters and afar from shore which arise need for larger sizes of platform supply vessels that can come up with challenges due to it. Hence a design of large vessels with reliable features has made the task worth it.

The objective of this work is to do a structural design of platform supply vessel by identifying the responses of the hull under the given loading condition near the mid ship area. The work of the thesis is substantiated by the internship session attended in Crist Ship Yard, Gdynia. During the time, construction aspect of self-elevating offshore wind turbine installation vessel at outfitting stage was attended. It somehow, augmented the areas that have to be critically dealt with in structural design of offshore vessels.

In the first section of the thesis, general characteristics of offshore support vessels and their operational features are discussed. On the following, the mid ship section structural materials (plates and supporting members), dimensions and orientations are assumed as a starting point to begin the scantling of hull structure by the Det Norske Veritas (DNV) Software, Nauticus Hull. The corresponding dimension of the structural members and the hull girder strength that satisfies the rule is obtained. Furthermore, buckling check on critical structural members is also dealt with. The structure is then analysed by finite element method and proved to have stress not beyond the acceptable bounds.

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LIST OF SYMBOLS

σ_L	Normal stress, N/mm²
τ_L	Shear stress, N/mm²
σ_V	Von Mises stress, N/mm²
σ_X	Normal stress component in the longitudinal direction, N/mm²
σ_Y	Normal stress component in the transversal direction, N/mm²
AP	Aft Perpendicular
B	Molded breath of the ship, m
C_B	Block coefficient of the ship
D	Depth of the ship, m
f_1	Material factor
FP	Forward Perpendicular
g	Gravitational acceleration, m/s²
l	Stiffener / plate panel length [m]
L_{OA}	Length over all of the ship, m
L_{PP}	Length between perpendicular of the ship, m
M_{SW}	Still water vertical bending moment, N·m
M_{WH}	Wave horizontal bending moment, N·m
M_{WV}	Wave vertical bending moment, N·m
M_{YY}	Bending moments around the Y-axis, N·m
M_{ZZ}	Bending moments around the Z-axis, N·m
pi	Pressure load [kPa] (i – number of relevant rule formula of maximum pressure)
Q_s	Still water shear force, KN
Q_w	Wave shear force, KN
R_{eH}	Yield stress, N/mm²
s	Stiffener spacing / plate panel width [m]
T	Scantling draught, m
t	Thickness, mm
t_k	Corrosion addition [mm]

V	Service speed, m/s
Z	Section modulus, m^3
t_{act}	Actual plate thickness [mm]
t_{req}	Required thickness [mm]
A_{req}	Required web area [cm^2]
A_{act}	Actual web area [cm^2]
t_{min}	Minimum thickness (plate, stiffener web) [mm]
z_{act}	Actual stiffener section modulus [cm^3]
z_{req}	Required stiffener section modulus [cm^3]
z_{min}	Minimum stiffener section modulus [cm^3]
$\sigma_{perm,}$	Permissible stresses [MPa]

LIST OF ACRONYMS

AHT	Anchor Handling Tug
AHTS	Anchor Handling Tug Supply
CAGR	Compound annual growth rate
CPP	Controllable Pitch Propeller
DNV	Det Norske Veritas
DP	Dynamic Positioning
DWT	Dead Weight tonnage
ERRV	Emergency Response and Rescue Vessel
FEM	Finite Element Method
HP	Holland Profile
IMO	International Maritime Organization
OSV	Offshore Support Vessel
PSV	Platform Supply Vessel

LIST OF ABBREVIATIONS

Main Deck at height	Z = 75000 mm	D
Tween Deck at height	Z =35000 mm	TD
Inner side	Y = 82000 mm	IS
Stringer	Z=4300 from bottom	S
Longitudinal girder	Y=4200 mm from centerline	LG
Stiffener		S
Longitudinal Bulkhead		LGBD
Inner Bottom		IB
Strength Deck		SD
Bottom of Vessel		Bottom
Side of Vessel		Side

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1. INTRODUCTION

Structural design, in any field of engineering, is all about coming up with a robust structure that at least meets the performance expectation by all parties involved, if not exceed, in supporting the loads it is subjected to over its life span and ship structural design is no exception.

The customary approach in ship structural design is to design a structure that fulfils the performance expectation of the owner with least possible cost, manufacturability with ease for the ship yard and the stringent requirement set by the classification societies that conforms to environmental and other regulations. To harmonize, the interests of these parties, classification societies have coined rules based on experience accumulated over a long time; numerical analysis drawn from experiments or a combination of one or more in a simplified manner.

In fact, the rules may not have detailed formulae set forth for all type of ships; when the need arrives, the classification society works in collaboration with the client considering the peculiar nature of the design. That is why the rules are kept updated covering more areas as well as refining the existing once in order to improve the design process.

The classification rule ensures that the design satisfies the minimum requirement for safety, speed, strength, and other crucial parameters at feasible cost. The loading on the vessel including the structure and associated environmental loads are used as a main ingredient for the design objective.

The structural design process encompasses broad three phases: initial or conceptual design, technical design and detailed design. The initial design is the starting point where, the hull, general arrangement, stability resistance and the likes are performed to see if the initial idea is worth going for with a number of alternatives. Once this stage is believed to be successful, the technical design follows. In the technical design, the structural strength of main elements that make up the structure is compared with the standards set by the classification societies. The final stage is the detail design where the manufacturing process of the elements, assemblies and subassemblies are dealt with in detail.

The increasing fuel price has triggered a momentum in offshore exploration more than ever. This in turn has created demand for offshore support vessels in general and platform supply vessels in particular. This trend in demand is also believed to follow the same suit in the future. Furthermore, the variety of cargo types carried and the multifarious tasks executed by the PSVs unlike most other vessels has drawn the attention of the work.

The task of this thesis is to undertake, structural design of 83.3 m vessel in midship section area considering one full cargo hold. The conceptual structural scantling of the PSV in midship area is done and an optimum is obtained that satisfies the DNV rule. Then the strength of the primary structural members are calculated and compared against the allowable.

Furthermore, Finite Element Method is used to get the stress levels at different members to compare against the strength value of the material selected in the scantling process.

To substantiate the thesis work, internship was attended in Crist ship yard in Gdynia on self-elevating offshore wind turbine installation unit named VIDAR that was at outfitting stage. The overall task of the internship was planning the manufacturing stage of pipes and routine follow up of the installation inside and outside jack house as well as planning of pipes installation in cofferdam. Even though the internship work is not part of the thesis work, it gave a global picture of how the manufacturing is made at the detail level that helped me gain where and how the focus areas on the design are emphasized.

2. OVERVIEW OF OFFSHORE DEVELOPMENT ACTIVITIES

2.1. Development of Offshore Activities

There are numerous claims up to where and when the offshore drilling was started. Some studies claim that the first is thought to have begun around 1891, in the fresh waters of the Grand Lake St. Mary's (Mercer County Reservoir) in Ohio. [13]

Others claim that offshore drilling for oil began off the coast of Summerfield, California, south of Santa Barbara, in 1896. The distance from the shoreline was limited as the drilling structure was supported by platforms connected to shoreline. The drilling activity showed a leap around 1947 when the first well was developed at around 17 km off Louisiana coast. The well was still a shallow one at a depth of about 5.5 m. The advance in drilling technology from conventional pile movers to rotary rigs has promoted the offshore industry at large [14]. Nonetheless, we can ascertain that offshore drilling was started at the end of the 19th century.

In offshore drilling activity, after the oil is drilled offshore, it can either be temporarily stored in the drilling facility itself or directly piped to refinery located onshore. Storage of the oil at the facility has grown significantly because of the growing size of the drilling facility itself. The increased size has also made the facility able to accommodate more crew and machineries that aid the function. The drilling facility also known as oil rig can be mounted in the sea in different forms. It can be of stationary on the sea floor, floating or stationed on artificial island created by pumping pile around it.

In Fig 2.1 different forms of mounting oil platforms and the depth to which they are mounted to are shown.

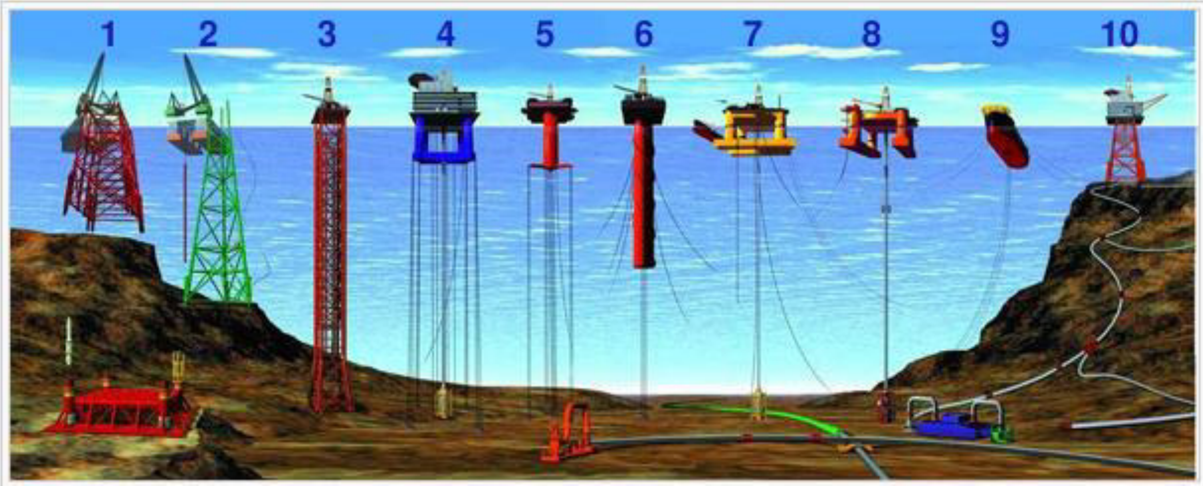


Figure 2.1 Types of offshore oil and gas structures [17].

Available from:http://oceanexplorer.noaa.gov/explorations/06mexico/background/oil/oil.html

[Accessed 10 August 2013]

In Fig 2.1, the specific configuration of mounting of the platforms and numbers are:

- 1, 2) Conventional fixed platforms (Shell's Bullwinkle in 1991 at 412 m);
- 3) Compliant tower (ChevronTexaco's Petronius in 1998 at 534 m);
- 4, 5) Vertically moored tension leg and mini-tension leg platform (ConocoPhillips' Magnolia in 2004 1,425 m);
- 6) Spar (deepest: Dominion's Devils Tower in 2004, 1,710 m);
- 7, 8) Semi-submersibles (deepest: Shell's NaKika in 2003, 1920 m);
- 9) Floating production, storage, and offloading facility (deepest: 2005, 1,345 m Brazil);
- 10) Sub-sea completion and tie-back to host facility (Shell's Coulomb tie to NaKika 2004, 2,307 m).

The configurations of the platforms listed in Fig 2.1 are evolved through times. A brief pictorial over view of the evolution of offshore platforms are depicted in Fig 2.2. It shows offshore drill connected to the coast around 1896 to the one in 2010 which can be deemed as a state of the art in the industry.

Figure 2.2 Pictorial metamorphoses of oil platforms [14].

Available from: http://oceanexplorer.noaa.gov/explorations/06mexico/background/oil/oil.html

[Accessed 10 August 2013]

The proportion of offshore oil and gas to the global output is growing. This is depicted in Fig 2.3. From this figure, around 30% of oil and gas extracted in the world is from off-shore oil & gas production, and most of that offshore production is from shallow water wells. Only 9% of that oil and gas is being recovered from deep-water wells, but this is changing rapidly.

Over the next 15 to 20 years the industry expects offshore oil and gas production will equal on-shore production. Most onshore fields have matured, but the ocean holds vast untapped potential, and the oil industry is betting its future on deep-water drilling. The industry is spending some $27 billion annually on subsea facilities--for wells at depths of 7,000 feet/ 213.5 m or more. This number is set to grow almost five times to $130 billion in 2020 [15].

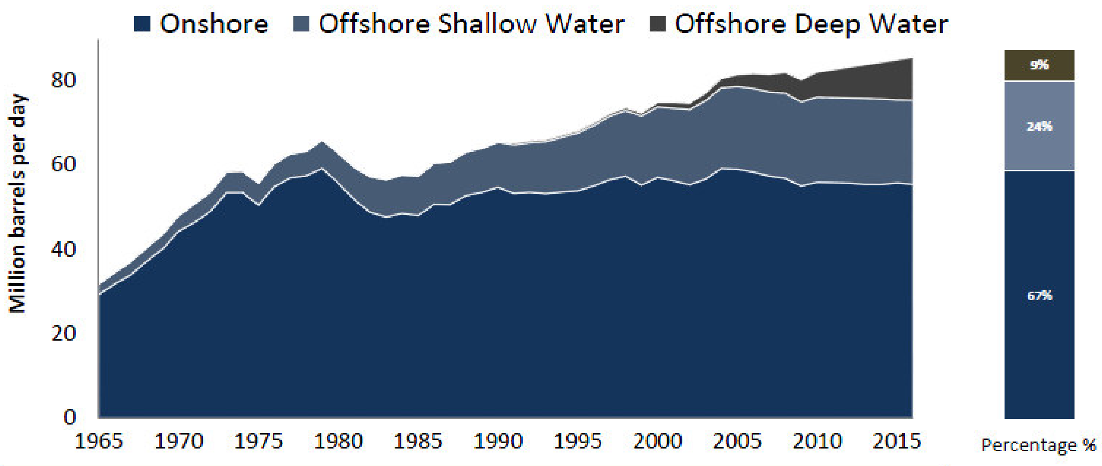


Figure 2.3 Proportion of onshore and offshore oil production [15].

Available from: John Ferentinos, 2013. Global Offshore Oil and Gas Outlook– Infield Systems Gas/Electric Partnership

2.2. Development of Offshore Supply Vessels

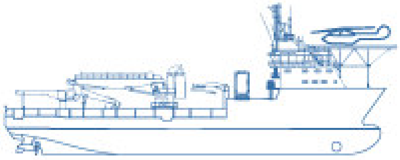
Offshore development activities have grown over the last six decades and are continuing to grow at higher rates to satisfy the insatiable appetite of the developing economies. Due to increased demand for offshore exploration and development, the need for supplies for the drilling activity such as cement, mud, pipes and miscellaneous marked the need for a specialized type of offshore vessels called offshore support vessels.

Among the pioneers of the supply vessels, arguably, the first one is the *Ebb Tide*, shown in the Fig 2.4.

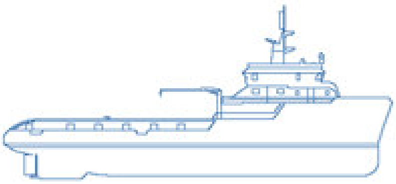
Figure 2.4 The first supply vessel, Tidewater's *Ebb Tide* [18]

The surplus of vessels from World War II had significantly boosted the offshore industry by serving the platforms. The pilot house of these vessels was situated at the aft of the vessel. This trend changed around 1954 where the pilot house moved to the front of the vessels to save wide deck area for the loads and cargo to be transported [8].

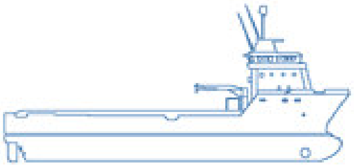
The advancement and diversity of the offshore industry has brought specialization of vessels that serve specific area of need, though some address multiple needs. The generic name offshore support vessel applies to every vessel engaged in the industry while a specific name is given corresponding to the sector of engagement. Fig 2.5 shows the common types of offshore support vessels and the purpose they serve.



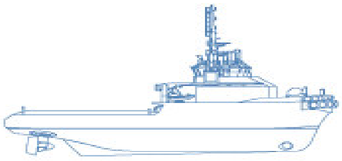
Multi-Purpose Supply Vessels (MPSV) - carryout various types of tasks in the offshore industry. They are outfitted with facilities and equipment to execute functions as fire-fighting, helipads, dynamic positioning, subsea construction activities and transportation of equipment and personnel.



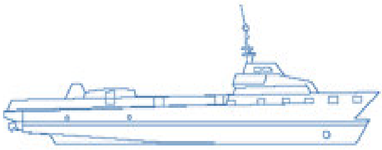
Anchor handling Tug Supply vessels (AHTSV) - are used to install and maintain oil platforms as well as towing non-self-propelled drill rigs or other non-self-propelled vessels. Like PSVs, they have large deck cargo spaces. These vessels are at times may be equipped with appendages to provide emergency services as well.



Platform Supply Vessels (PSV) –replenish the platform with crew and supplies and bring crew and cargo back to shore. They are used to transport various cargo types in tanks and on their deck.



Terminal tugs (TS) - assist tanker operations in oil and gas terminals and mainly identified by their bollard pull. These are also equipped with firefighting and rescue provisions.



Fast Supply Intervention Vessels (FSIV) - are engaged in supplying emergency provisions. They are also used to transport crew members to offshore facilities.

Stand-by vessels - have been compulsory for attendance of all manned offshore installation.

Well stimulation vessels - are specially equipped units which can perform well stimulation methods like "*fracturing*" and "*acidizing*" to facilitate and improve production.

Seismic Vessels - are vessels used for surveying purposes to locate potential areas for drilling.

Figure 2.5 Common types of offshore vessels [4]

2.3. Offshore Support Vessels Trends and Forecasts

Early supply vessels are used solely for the purpose they were designed for. The supply vessel initially was used only to supply the deliverables to drilling activities until it was later realized around 1960 that it can serve other purposes with the addition of some appendages. For instance, a supply vessel can be adapted to serve mooring activity by appendage of winch at the front of the deck.

These days, supply vessels are performing multi tasks contrary to most other ships, which commonly have one type of hold usually transporting one type of cargo. [1]

The need for energy of the developed and developing nations has attracted significant investment in the offshore drilling to amass the opportunity. To tap increased production, contractors are heavily investing in offshore vessels. For instance, in 2013 the offshore support vessel market is valued at $69.3 billion; this figure is estimated to reach $91,228.8 million by 2018, with a CAGR of 5.7% from 2013 to 2018 [16].

The increased production has called for improved production techniques to lessen the overall production cost as well as investing in research and development in order to develop new and improved vessels for offshore platforms.

PSVs are designed with a forward deckhouse and large open aft cargo deck on which offshore containers and deck cargo such as casing and drill pipe is transported and internal tanks for fuel, portable water, drilling water, mud, brine, chemicals and dry bulk. PSVs usually stay out for a day or two in operation but in deep waters, they may stay for a week. [6].

Since the beginning of 2000, the supply vessel fleet has increased with 950 vessels, which is more than tripling of fleet. In the Fig 2.6 platform supply vessel market in main offshore drilling zones are shown.

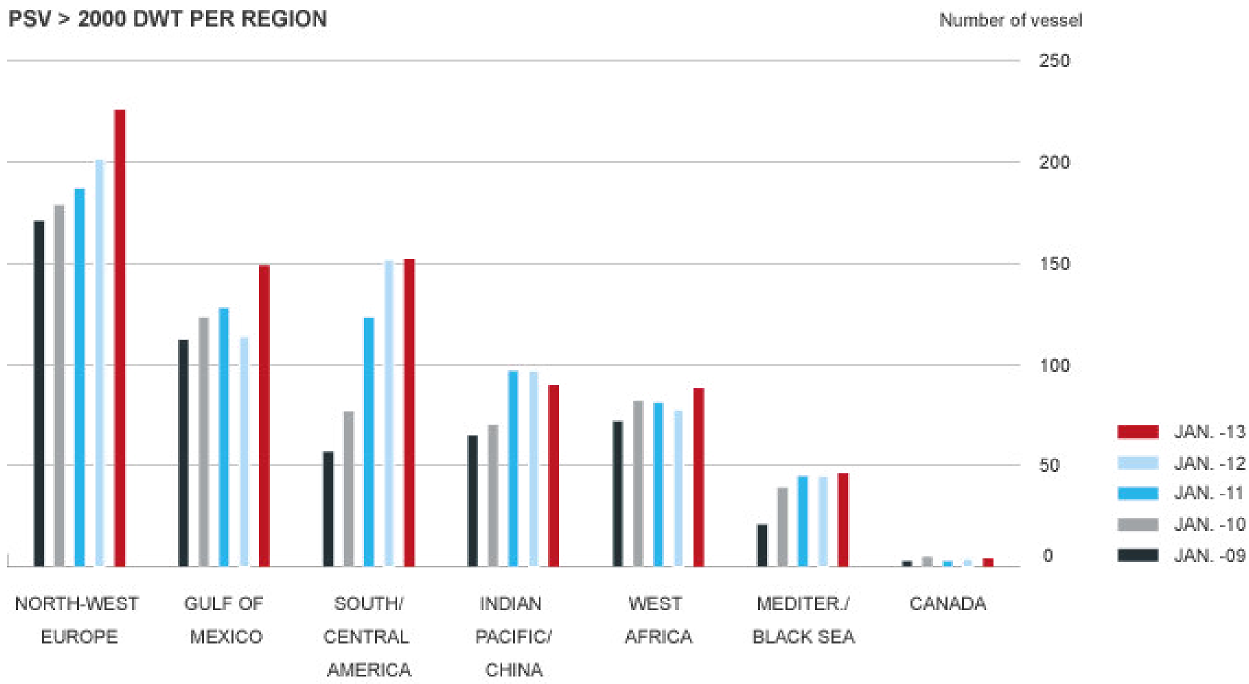


Figure 2.6 Offshore supply vessel markets [12]

From Fig 2.10, one can see that North West Europe, Gulf of Mexico and south Central America still cover the largest of Platform supply vessel market greater than 2000 dead weight tonnage (DWT). In fact the revenue obtained from PSVs per day depends on many factors such as age of the vessel, DWT, area of operational condition and the likes.

The number of PSVs in operation, scrapped or lost in number as well as the order book in years starting from 1967 to 2013 is given in the Fig 2.7. It shows that the market for PSV increased from until the peak in year 1982 and started to decline until 1985. Between 1985 and 1997, it had witnessed ups and downs. Around year 2000, it had gained momentum to kick back and steadily increase. In the Fig 2.7, the number of PSVs in operation is less because this is the time when the data is documented and produced.

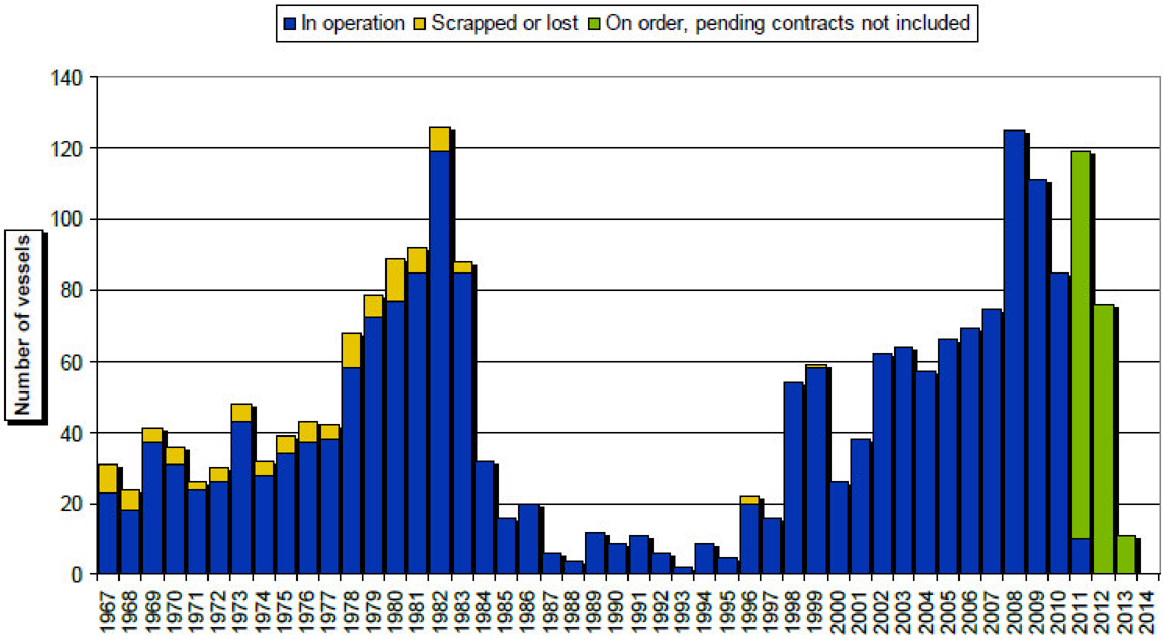


Figure 2.7 Number of PSVs in years and in operation [11]

As the demand for offshore exploration is increasing, so is the demand for PSVs as well as their size to cope with the increased need for materials and supplies movement. Besides the size, the trend is pushing for more efficient and less fuel consuming PSVs to counter the cost and make use of economies of scale. In a push to have efficient PSVs, the mere task takes its step from the design stage to the manufacturing up to servicing. Therefore, it is up to the designer to come up with the utmost design that delivers the requirements. The minimum aspect a client expects is the nature of the ship, the features it has to be equipped with, the area and extent of operation and some special considerations. The designer then plays on the fine line of making a design that is effective, efficient and safe not to mention conformance to the stringent requirements of the class societies. The effectiveness links to the behavior of the vessel in meeting the client's deemed performance requirement while the efficiency is related to the ability of the vessel in performing its sought functions at utmost reliability. The vessel should also perform with no or very small margin of risks to the crew onboard, near operation area and the environment as a whole.

This all steps are not something that can easily be attained. Rather it requires a sustained and meticulous steps of repetitive routines in an attempt to strike an optimum design performance features. Hence, it is an iterative process, where early estimates are made based on the requirement of the client, repeatedly adjusted until an optimal point is reached. The iterative process goes through many conflicting objectives and makes effort to come up with an adaptable solution that balances the objectives.

2.4. Contemporary Ship Structural Design

The structural design of ships is primarily rule based where rules are formulated on the basis of experience, empirical studies as well as scientific investigation. As such when the sizes and complexity of the ships started increasing, there were no existing rules to guide the designer. In such cases, the Finite Element Method (FEM) proved to be a valuable tool for the analysis and design of ship structures [10].

FEM, an analytical method, is used to determine how well structural designs survive different loadings. Its basic idea is division of a complex structure in to finite elements subjected to mathematical models of known elastic and geometric properties. It reduces the time and costs associated with prototyping and physical testing.

A number of ship structural analysis have also been carried out by the help of Finite Element Method to enhance the ease with which complexity is minimized and estimated with fine details.

Quite a number of structural analyses have been conducted in ships using finite element methods. Anthony [2] analysed the structure of Autonomous Underwater Vehicle (AUV) regarding its strength against sea pressure at depth. FEM is also used for buckling analysis of ship like structure and it proved to have significantly reduced the weight of the structural component [3].

It is used as a methodology for modeling bulk carriers in order to assess the structural integrity [5]. Moreover, a structural analysis of a model of war ship is carried out using FEM. [7].

Overall, FEM has found wide area of usage in analyzing the strength of different type ships and has proved to be an indispensable tool. It is based on this notion that this work is using the FEM for the analysis of the mid ship section of Platform supply vessel.

3. DESCRIPTION OF THE VESSEL

3.1. General Characteristics

The platform supply vessel used in this thesis mainly serves offshore platforms by transporting; pipes, cement, liquid and cargo to and from mainland and offshore installations used on oil and gas production platforms. The main characteristics of the PSV are given in Table 3.1.

Table 3.1 Main particulars of ship

PARTICULARS:	Symbol	Value	Unit
Length Overall	L_{OA}	83.3	[m]
Length between Perpendiculars	*Lpp*	76.8	[m]
Draught	*T*	6.2	[m]
Depth To Main Deck	*D*	7.5	[m]
Moulded Breadth	*B*	18	[m]
Speed	*V*	14	[kn]
Block Coefficient	*CB*	0.721	-

The draught mentioned in Table 3.1, is the summer load draught and the speed is set at design draught.

3.2. The Vessel and Purposes

The vessel is a mono hull having double skin throughout its length. The storage arrangement is made in such a way that cargoes are stored in the main deck and below the main deck. There are provisions for double bottom and wing tanks in its length. Accommodation is situated at forward of the vessel. There are basically five decks in the vessel: tween deck, main deck, A- deck, B-deck and C-deck.

The vessel has five bulkheads and four of them ware fitted by water tight doors.

The hull structure scantling and general arrangement design is carried out according to DNV rule and the construction is made of welding.

The view of the vessel is depicted as in Fig 3.1.

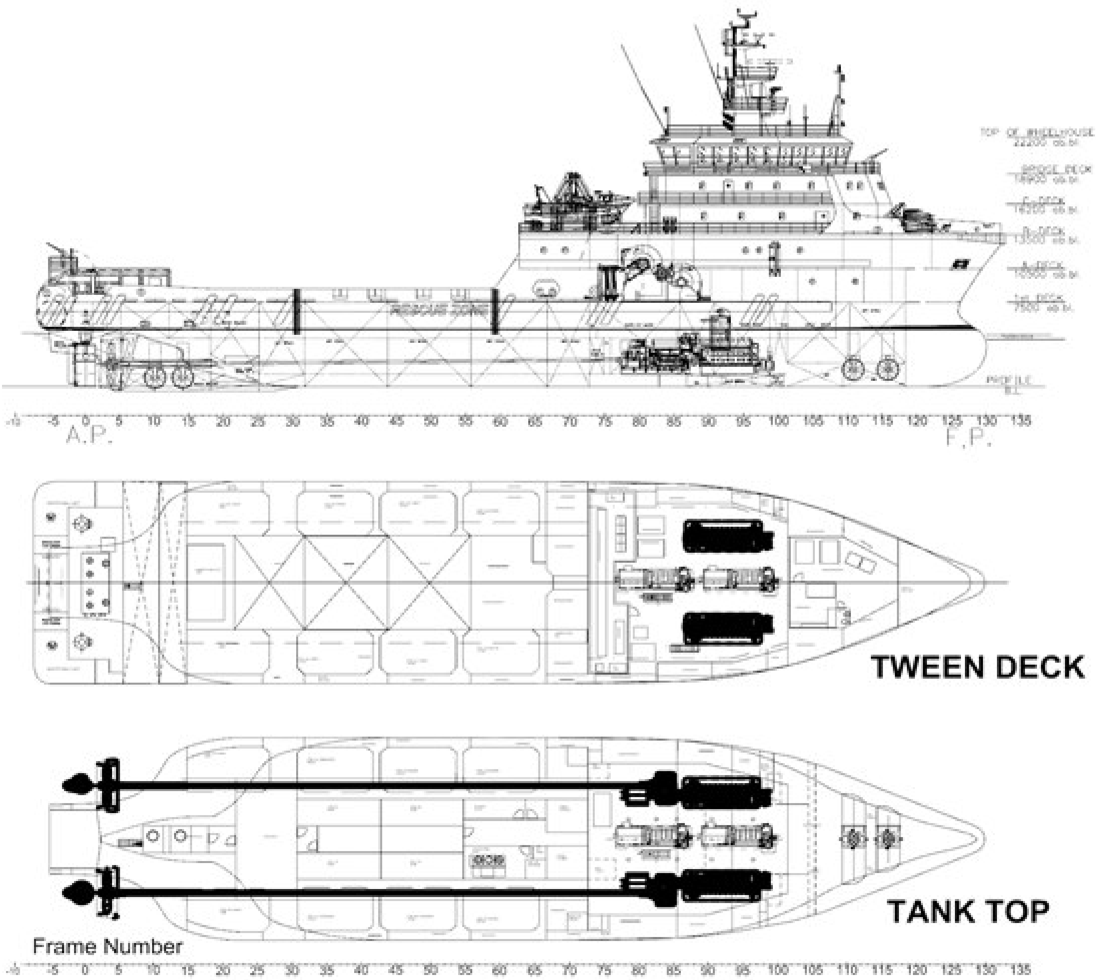


Figure 3.1 Platform supply vessel

Source: Crist shipyard and Author

The vessel is design to operate worldwide except Areas with ice, Arctic zones, US inland waters, Caspian Sea and similar areas with special restrictions and requirements. The environmental condition for the operation is:

- Ambient air temp, summer: + 35^0C
- Ambient air temp, winter: - 10^0C
- Engine room air temp.: + 50^0C
- Relative humidity temp, summer: 70%
- Seawater temp.: -4°C to + 35°C

The PSV is sought to address the following purposes:

- Supply of the required materials and provisions to the platform vessel,
- Dynamic positioning,

- Anti-pollution control / Oil Recovery,
- Movement of personnel's and cargo to and from platform
- Accommodations for special & crew personnel.
- Rescue and standby operations

The vessel has a number of loads as liquid and bulk cargo in its cargo hold and on the main deck. The major loads are given in Table 3.2.

Table 3.2 Main characteristics and loads on the Vessel

Dead-weight at T= 6.20m	approx. 2700 tonnes
Working deck area	approx. 700 m^2
Working deck load bearing	10 tonnes/m^2
Fuel oil, bunkers	Approx 700 m^3
Fuel oil, cargo bunkers	1350 m^3
Fresh water, cargo	200 m^3
Ballast water	1000 m^3

All cargo systems loading and discharge arrangements are completely remote controlled, in addition to local manual control.

In designing the frame structure according to the DNV rule, the current vessel is supposed to carry a load of 10 tonnes /m^2 throughout its main deck.

The propulsion system is driven by two four stroke in line diesel engines connected to two (2) propellers via shafting and gearboxes. The engine has rating of approximately 3000 kW each. The maximum speed is 750rpm (medium speed).

The propeller is also equipped with:

- Two (2) CP propellers as main propulsion,
- Two (2) high lift flap rudders
- Two (2) electrical driven tunnel thrusters forward with CPP and fixed rpm
- Two (2) electrical driven tunnel thrusters aft with CPP and fixed rpm

The dynamic positioning of the vessel is aided by dual computer dynamic positioning system according to IMO DP-II and classification notation.

4. STRUCTURAL MODELING ACCORDING TO DNV RULES

4.1. Introduction

DNV software Nauticus Hull is chosen for the structural scantling of platform supply vessel in this thesis because it is a powerful software package for strength assessment and it is in use by more than 200 shipyards and ship design offices around the world. [14]

Having entered main ship data in the Nauticus hull package, sections of the model are entered in the section scantling program that helps in designing the cross sections and transverse bulkheads.

After modeling the section scantling, the rule check was used to verify whether the section made conforms to DNV rule. The features that are mainly checked by the rule check analysis include; hull girder longitudinal strength, local strength and buckling of plates and stiffeners.

4.2. Material Selection and Assumption

Appropriate material selection aids lengthening the life of vessel. When a material is selected for ship design, the physical and chemical properties of the material that gives adequate strength considering the loading and working condition have to carefully be selected.

Material selection plays a pivotal role in ship design. Some of the failures of ships are attributed to the failure of the materials they are made of or the way they are manufactured. One example of failure due to material could be the brittle failure of liberty ships [9].

It is widely believed that if a ship fails catastrophically, the study of a sister ship may give a clue to how it failed. This notion is widely observed in auto and aerospace industries; when accidents happen, similar models are usually called for for further examination.

The material selected for the ship design also dictates the way it should be fabricated. Hence, it is right to say that at the core of designing and manufacturing, the material selection stands out to be of paramount importance.

Considering the working condition, size and loading, in this thesis, grade A mild steel is selected for all the structural parts as it possesses the necessary characteristics. The selected material property is shown in Table 4.1.

Table 4.1 Properties of structural material selected for the vessel.

Grade	A	Shell, Transversals , Longitudinals, Transverse bulkheads, etc.
Yield Point, R_{eH}	235	MPa
Density, ρ	7.85	t/m^3
Young's modulus, E	$201E^3$	MPa
Poisson ratio, v	0.3	-

Conventionally vessels are framed either transversally, longitudinally or mixed. The type of framing system to use has long been an issue still on the table as to which type best serves what type of ships under specific conditions. Early ships were built transversally; however, as ships became larger, longitudinal framing has enjoyed wide suitability due to advance in manufacturing from riveting to welding. In this particular case, a combination of longitudinal and transverse framing is used with a frame spacing of 600 mm. Longitudinal is used on the main deck whereas bottom, sides, tween deck, bilge and others are framed transversally. The span of each longitudinal is made at each four frame spacing (2400 mm).

The scantling is done at four cross-sections starting from aft perpendicular (AP):

- Frame 13 (7.8 m) -31 (7.8-18.6 m),
- Frame 55-67, frame 64 being midship section.

The type of stiffener chosen for the scantling is HP as they are made for plate stiffening. The shape they possess makes them more attractive during construction as well as working life of the vessel.

A typical HP bar with height *w* and thickness *t* is given in Fig 4.1.

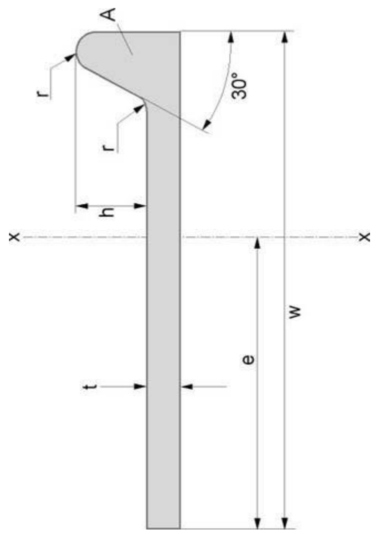


Figure 4.1 HP stiffener cross-sectional area

In the scantling that follows the far aft (frame 22) and the mid ship section are made to enable the comparison and to figure out how the structure behaves as the section goes far aft. The cargo holds are planned between 12 frames (7.2 m) longitudinally and the bulkheads are planned to take two frames.

To make a scantling for a ship structure, it is essential to have a concept sketch of the appropriate cross-section where all the divisions and subdivisions with in the structure are clearly identified. By the same token, concept sketch of the mid ship section is made as a spring board for starting the scantling and is shown in Fig 4.2.

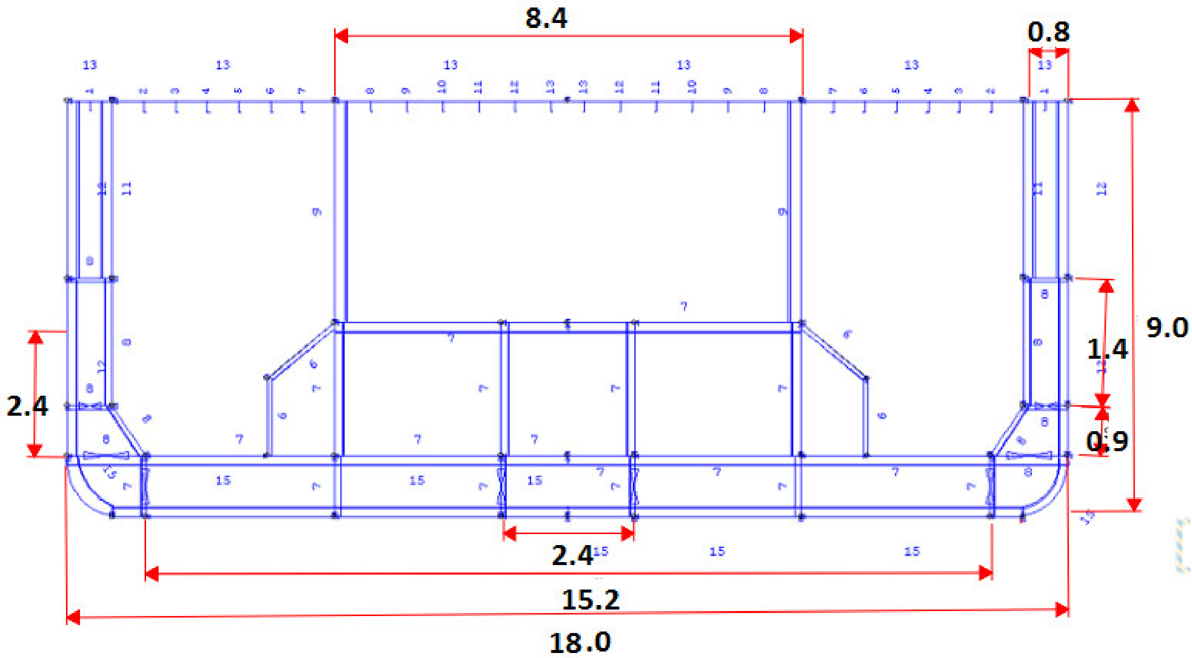


Figure 4.2 Concept sketch of cross-section at midship section (frame 62 = 38.4 m)

The structure is designed to carry different cargo types. The double bottom part is allotted for water ballast and the top part of the double side also used as a tank for water ballast.

Similar approach was taken to draw the concept sketch of cross-section at frame 22 and the initial assumption as with the thickness and member's arrangement in DNV Nauticus hull software prior to optimization is portrayed in Fig 4.3.

As one moves from midship section towards aft of the vessel, the cross-section gets narrower. In fact the height from the base line to hull of the vessel also increases to give way to the propellers that drive it. The double bottom parts also start to diminish as the distance gets farther aft from the midship section.

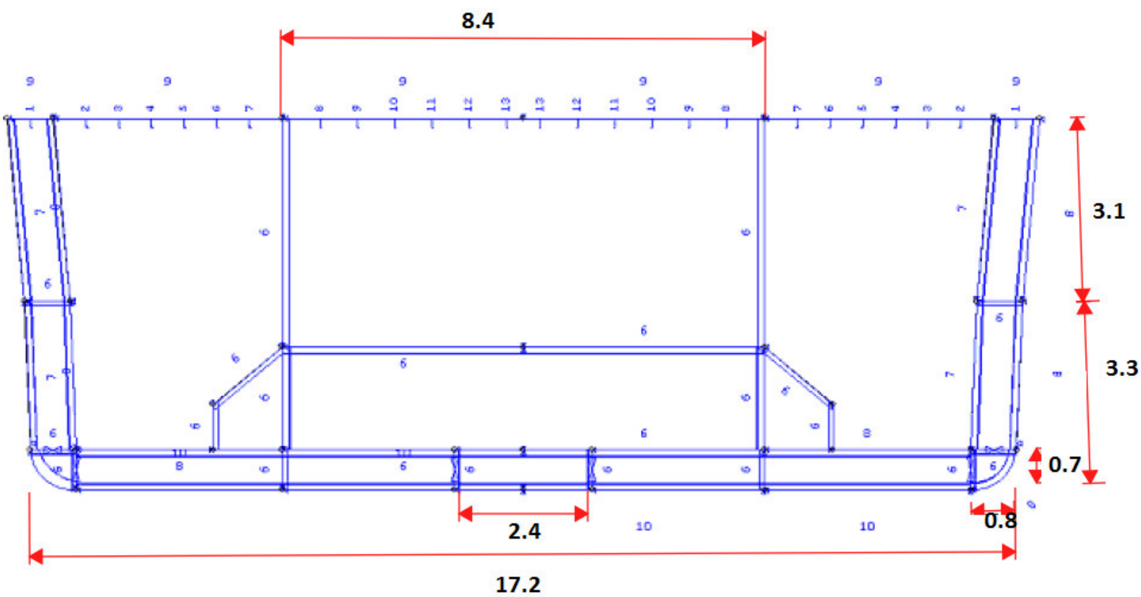


Figure 4.3 Concept sketch of cross-section at frame 22.

The midship section structure concept sketch, presented in Fig 4.2, is extruded to have one cargo hold around the same area and presented in Fig 4.4. The three dimensional presentation typical features including; the double bottom, double skin and tween deck with adjourning cargo hold are explicitly defined.

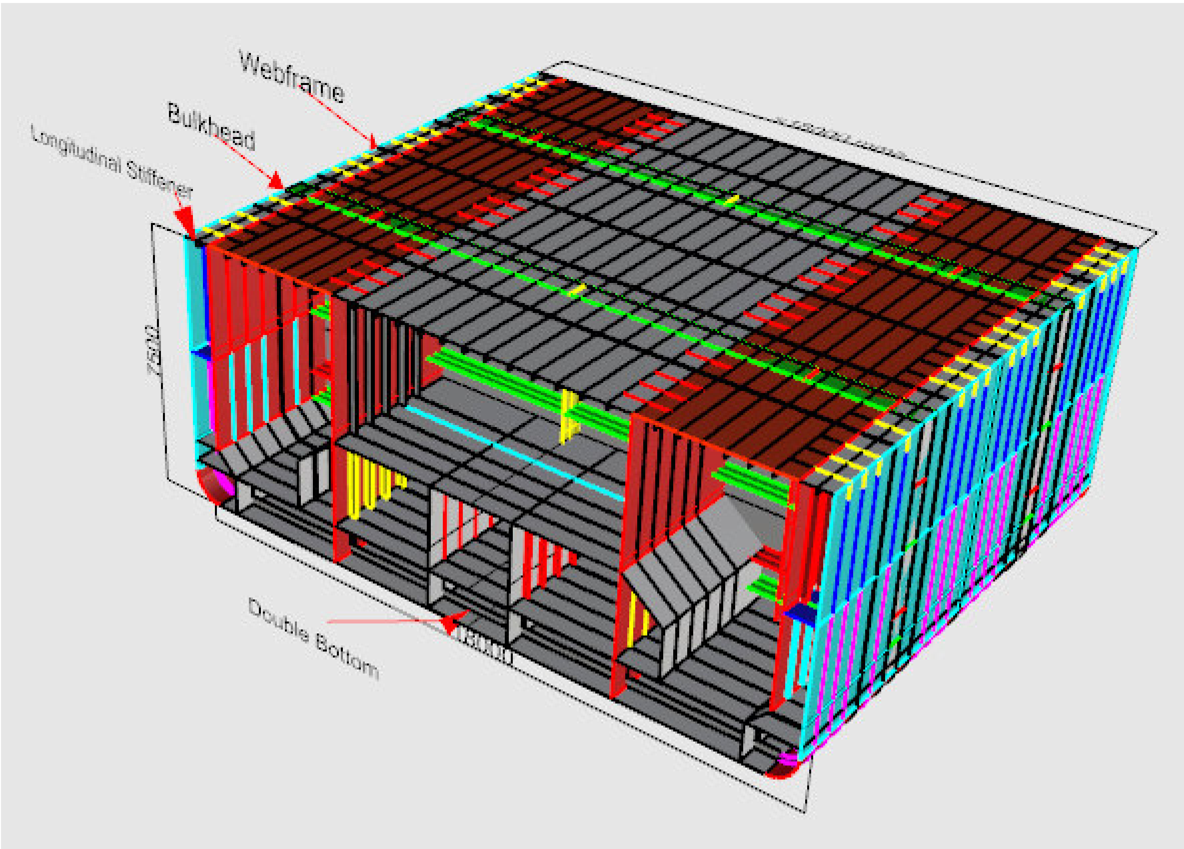


Figure 4.4 Sketch of three cargo hold at mid ship area for FEM

In the concept sketch of cross-section at frame 64 (mid ship section) and cross-section at frame 22, the double skin and double bottom of the vessel are visible, with the latter having a reduced size.

4.3. Compartmentation and Loads

The compartmentation is made based on the General arrangement of the vessel. The type of loads and the density are set along with the frame spacing the specific cargo occupies. The cross-section and initial section scantling is set before the compartmentation was made. Based on the cross-section made, the user interface made the structure available so that each and every spacious volume is assigned some kind of load.

Fig 4.5 shows the partial view of the compartmentation made for the midship section.

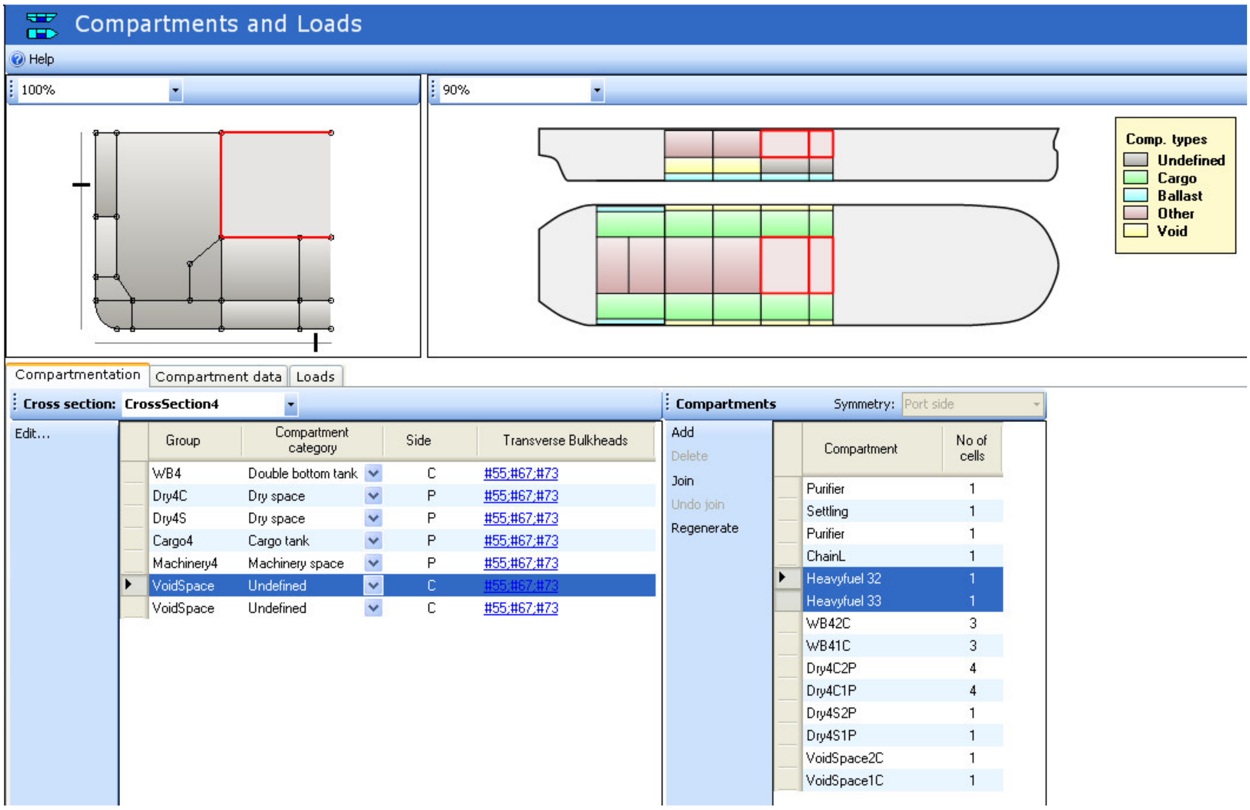


Figure 4.5 Definition of compartment in mid ship section

From Fig 4.5, it can be observed that, the cargo/load group, category of the compartment, the position (whether it is port side or star board) and the available transversal bulkheads bounding the cargo are presented. The length of each cargo hold is 6.6 m along the length of the vessel and 0.6 m empty space for separation of cargo are provided at the end of each cargo length.

After the compartmentation was accomplished, the respective densities of the loads with in the hold are filled in order to account for the total loading in each hold. The rule check was made to easily spot elements that need some attention.

4.4. Design Bending Moments and Shear Force

Longitudinal strength loads are loads which affect the overall strength of the ship's hull girder, in which the ship is regarded as a beam or girder, because of its slender profile. They are represented in terms of longitudinal bending moment, shear force and torsional moment. The longitudinal strength loads may be divided in to static longitudinal bending loads and dynamic one. Static Longitudinals emerge from local in equalities of weight and buoyancy in still water condition.

Difference between weight and buoyancy along the longitudinal is a main source of static bending moment and static shear force. It is referred as static because, the consideration is a still water scenario. Asymmetric loading of the vessel along longitudinal could also be a major source for static torsional moment.

When the vessel is subjected to waves, the waves induce a bending moment and shear forces that are regarded as dynamic loadings. Waves also create dynamic torsional loading on the structure depending on the approach of the wave. When waves approach the vessel at oblique angle, there is high probability that some kind of dynamic torsional loads are applied.

The magnitude of dynamic longitudinal load used in strength calculation of wave bending moment and shear force is calculated according to IACS rule. The midship section bending moments for still water and wave bending moments according to DNV rules for ships less than 100 m Part three Chapter 2, section 4, are presented in equations 2, 3 and 4:

$$C_w = 0.0792L \quad (1)$$

$C_w = 6.08256$

Stillwater bending moments within 0.4 L (DNV rule July 2011, Pt. 3 Ch 2, section 4 B)

$$M_{so} = 0.0052L3B(CB + 0.7)\ (KNm) \quad (2)$$

C_B= 0.721, from Table 3.1,

M_{so}= 60250KNm

Wave load conditions (DNV rule July 2011, Pt. 3 Ch 2, Section 4 B)

The sagging and hogging bending moments are obtained by equations (3) and (4) respectively.

$$M_{wo} = 0.11Cw\ L2B(CB + 0.7)\ (KNm) \quad (3)$$

Sagging bending moment $M_{wo} = 100941\ KNm$

$$M_{wo}= 0.19Cw\ L2\ B\ (KNm) \quad (4)$$

Hogging bending moment M_{wo}= 88465 KN.m

The above results obtained in equations 2,3 and 4 for bending moments with hand calculation turn out to be equivalent with the values obtained in the Nauticus hull. The summary of the bending moments for still water and wave loads according to DNV rules obtained from Nauticus hull is given in Table 4.2.

Table 4.2 Design bending moments in still water and wave condition

	Sagging(KNm)	Hogging (KNm)
Standard values according to Rules, Mso	60250	60250
Given as input (actual cargo/ballast conditions)	60250	60250
Design still water bending moments, Ms	60250	60250
Design wave bending moments, Mw	100941	88465
Design wave bending moments, Mw for buckling check	100941	88465

The input values used for further analysis based on rule check is therefore taken equal to the rule shown in Table 4.2. From the same Table, the design wave bending moment for buckling is also taken as the same value obtained by the rule check.

The bending moment diagram obtained from the hull girder loads subsection in Nauticus hull at midship section in still water in the case of sagging is shown in Fig 4.6.

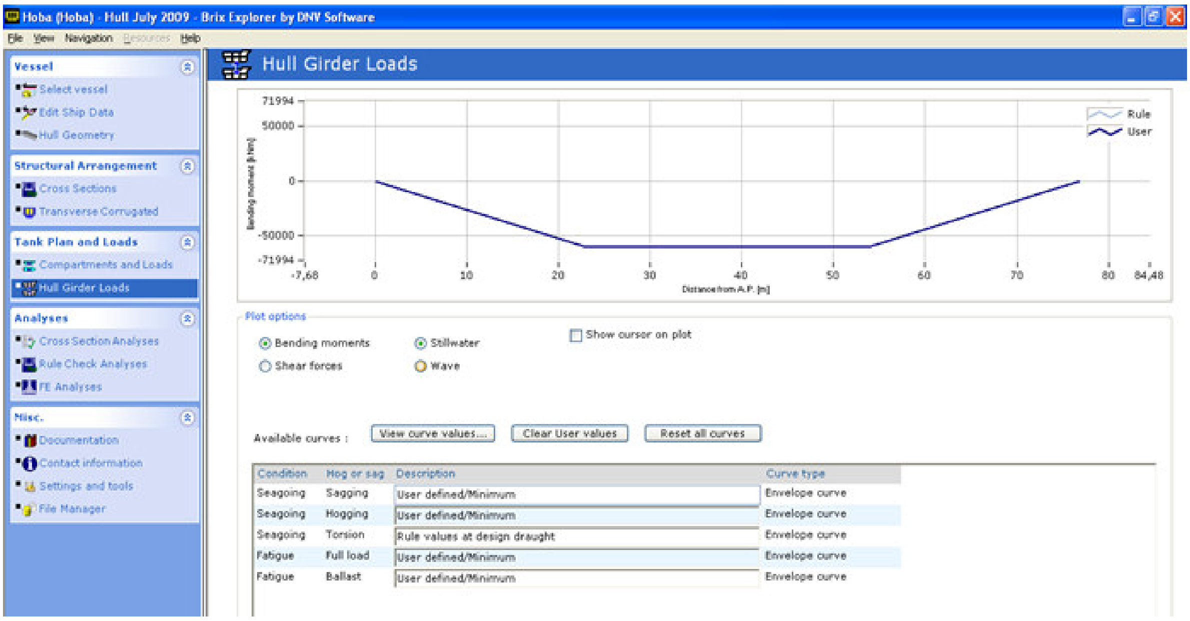


Figure 4.6 Still water bending moment at midship section

The effect of shear force on the global strength, for a ship less than 100m is not significant and sometimes is neglected even by the rules. Nevertheless, for the ship considered in this thesis, the shear force along its entire length is shown in Tables 4.3, and 4.4, in seagoing and harbor conditions respectively.

Table 4.3 Minimum global shear force-sea going condition

Longitud.	**Longitud.**	**Design global shear forces [kN], Rule min. - Seagoing condition**					
position	**position**	**Hogging**			**Sagging**		
x/L	x_AP [m]	Qs	Qw	Qs+Qw	Qs	Qw	Qs+Qw
0,00	0,00	0	0	0	0	0	0
0,10	7,68	402	14459	14861	-14411	-9179	-23589
0,15	11,52	603	21689	22292	-21616	-13768	-35384
0,20	15,36	603	28918	29522	-21616	-18357	-39973
0,30	23,04	603	28918	29522	-21616	-18357	-39973
0,40	30,72	483	13967	14450	-17293	-13967	-31260
0,50	38,40	0	0	0	0	0	0
0,60	46,08	-483	-13967	-14450	17293	13967	31260
0,70	53,76	-603	-31465	-32068	21616	19953	41570
0,80	61,44	-603	-31465	-32068	21616	19953	41570
0,85	65,28	-603	-31465	-32068	21616	19953	41570
0,90	69,12	-402	-20976	-21379	14411	13302	27713
1,00	76,80	0	0	0	0	0	0

The sagging shear force for seagoing condition, from Table 4.3 is 41570 KN at a length 75 to 85% from aft perpendicular while the maximum hogging shear force is -31465 KN. These two are the minimum that are allowed for the design.

In table 4.4, the shear force in harbor condition is figured.

Table 4.4 Maximum allowable still water shear force- harbor condition.

Longitud.	Longitud.	Max. allowable still water shear forces acc. to Rules [kN] - Harbour condition					
position	position	Hogging			Sagging		
x/L	x_AP [m]	Qs	Qw	**Qs max**	Qs	Qw	**Qs max**
0,00	0,00	0	0	**0**	0	0	**0**
0,10	7,68	402	7230	**7632**	-14411	-4589	**-19000**
0,15	11,52	603	10844	**11448**	-21616	-6884	**-28500**
0,20	15,36	603	14459	**15062**	-21616	-9179	**-30795**
0,30	23,04	603	14459	**15062**	-21616	-9179	**-30795**
0,40	30,72	483	6984	**7466**	-17293	-6984	**-24277**
0,50	38,40	0	0	**0**	0	0	**0**
0,60	46,08	-483	-6984	**-7466**	17293	6984	**24277**
0,70	53,76	-603	-15732	**-16336**	21616	9977	**31593**
0,80	61,44	-603	-15732	**-16336**	21616	9977	**31593**
0,85	65,28	-603	-15732	**-16336**	21616	9977	**31593**
0,90	69,12	-402	-10488	**-10890**	14411	6651	**21062**
1,00	76,80	0	0	**0**	0	0	**0**

From Table 4.4, maximum shear force summation in still water and wave on harbour condition in hogging is found to be -16336 KN at a distance 70 to 85% of the length from aft perpendicular. The corresponding maximum shear force in sagging condition for both still water and wave is also found to be 31593 KN, at similar distance from aft perpendicular.

The graphical representation of still water shearforce in Fig 4.7 shows the values given in Table 4.3.

In still water the shear significant amount of shear force would be sagging due to the loading on the vessel as shown in Fig 4.7. however, hogging load could also be experienced due to loading on extremes and empty or less cargo hold in the mid ship area. On the other hand, hogging and sagging are reality in seagoing vessel due to the phase and wave length compared to the length of the vessel.

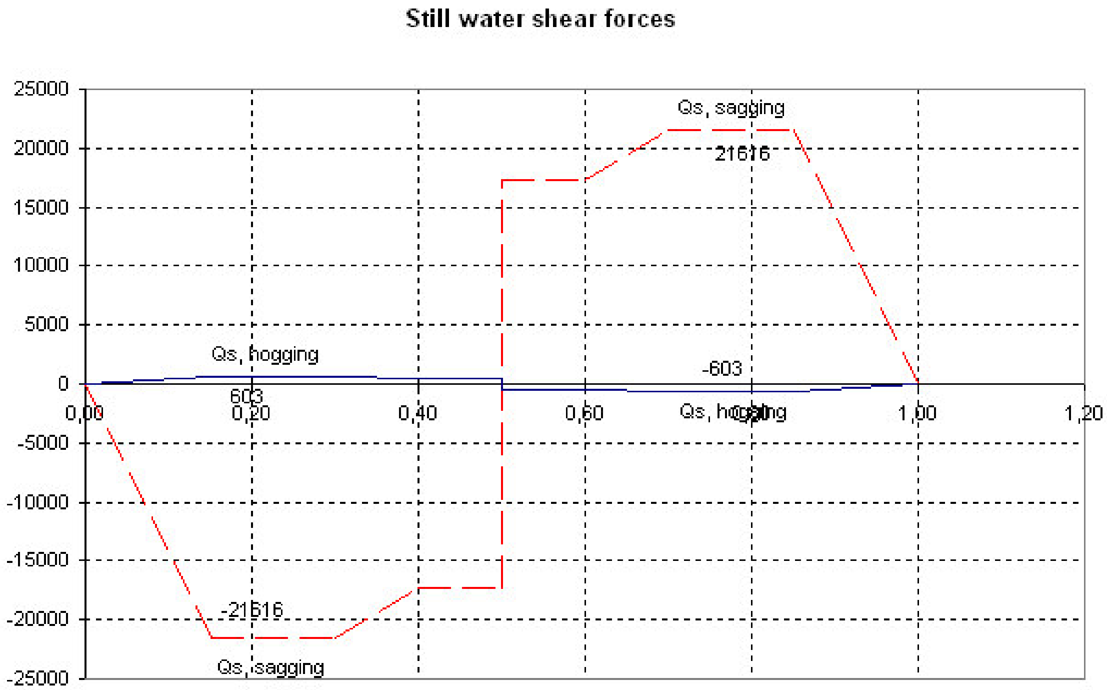


Figure 4.7 Still water shear force

For this vessel under consideration, the rule shear force in wave is as in Fig 4.8.

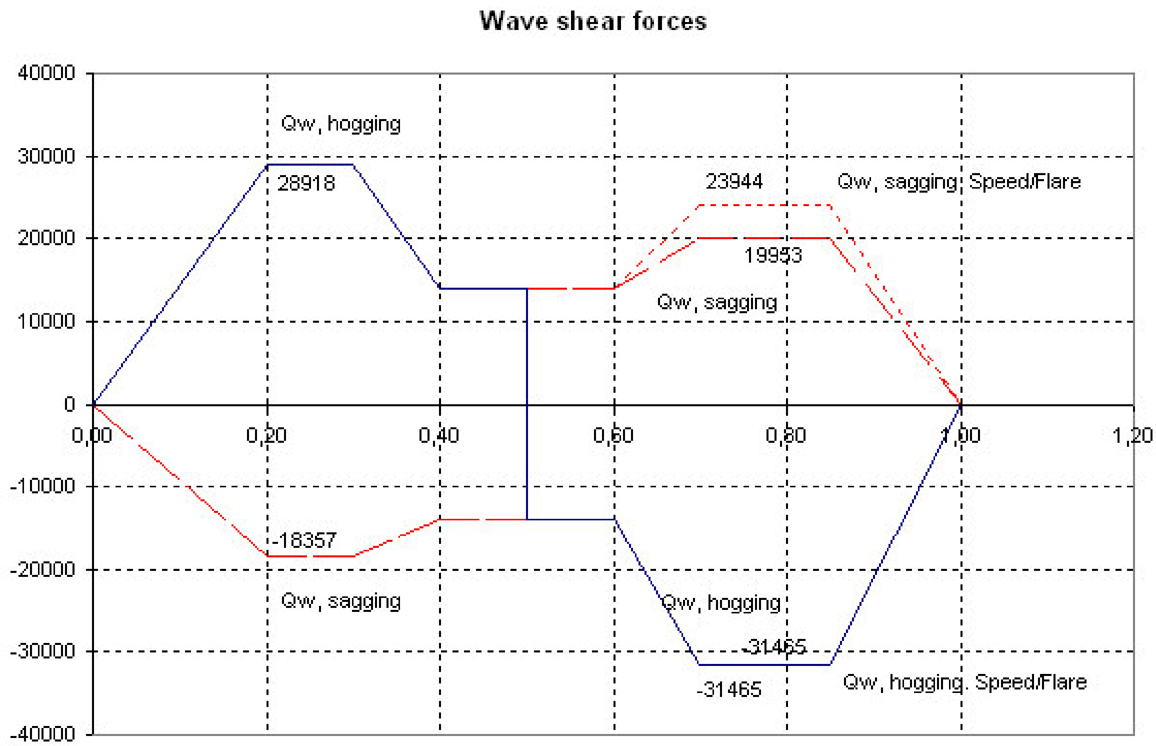


Figure 4.8 Wave shear force

The section modulus according to DNV rule for the vessel amid ship is calculated by the following equation. (DNV rule July 2011, Pt. 3 Ch 2, section 4 C)

$$z = \frac{Ms+Mw}{175}\, 10^3 \tag{5}$$

The rule section modulus focuses mainly on the physical structure of the vessel in consideration rather than the loading it is designed to support. Based on equation 5, the section modulus is calculated and presented in Table 4.3. In fact the result obtained is analogous to the Nauticus hull result.

Table 4.5 Hull girder section modulus at midship section

	Bottom	**Deck**
Section Modulus ratio *Za/Zr*	2.693	2,488
Minimum Section Modulus	1.11483	1.11483
Sagging (60250 kNm)	0.92109	0.92109
Hogging (60250 kNm)	0.84980	0.84980
Rule section modulus	1.11483	1.11483

From Table 4.5, the minimum section modulus ratio of the deck and bottom are higher than the minimum requirement.

The scantling that results the section modulus in Table 4.5 have; the cross-sectional area, height to the neutral axis from bottom and moment of inertia coupled with the percentage comparison of the section modulus against the DNV rule is presented in Table 4.6.

Table 4.6 Hull girder strength summary at midship

	Actual	Rule	STATUS(%) 100=Rule
Cross-sectional area (cm^2)	11607		
Height to the neutral axis (mm)	3602		
Moment of Inertia (m^4)	9.699		
Section modulus, bottom (m^3)	2.693	1.115	241.5
Section modulus, deck line (z = 7500 mm), (m^3)	2.488	1.115	223.2

From Table 4.6, the current scantling of the midship section well satisfies the minimum requirement for the hull girder.

The bending moments of the vessel in sagging and hogging condition in still water and wave of cross-section at frame 22 is also obtained and portrayed in Table 4.7.

Table 4.7 Hull girder section modulus of cross-section at frame 22

	Sagging (KNm)	Hogging (KNm)
Standard values according to Rules, *Mso*	36087	36087
Given as input (actual cargo/ballast conditions)	36087	36087
Design still water bending moments, *Ms*	36087	36087
Design wave bending moments, *Mw*	45345	39740
Design wave bending moments, *Mw* for buckling check	45345	39740

Table 4.7 shows that it significantly declined from 60250 KNm at mid ship section to 36087 KNm. The design wave bending moment also followed the same reduction even if not in line with the still water case It is because, the initial bending moment formula for still water and wave cases were no more usable in cross-section at frame 22, because it is beyond 0.4 L amid ship hence another formula were employed.

The cross-sectional area the height of the neutral axis and the section modulus of cross-section at frame 22 is depicted in Table 4.8.

Table 4.8 Hull girder strength summary of cross-section at frame 22

	Actual	Rule	STATUS(%) 100=Rule
Cross-sectional area (cm^2)	9328		
Height to the neutral axis (mm)	4093		
Section modulus, bottom (m^3)	1.979	0.668	296.3
Section modulus, deck line (z = 7500 mm), (m3)	1.797	0.668	269.1

After the bending moments for both still water and waves are seen for cross-sections at frame 64 and 22, plate and stiffeners in the longitudinal and transversal directions were observed in accordance with the requirement for local and buckling strengths.

4.5. Strength of plates and stiffeners

4.5.1. Longitudinal Plates

4.5.1.1. Deck Loads

The design loads according to DNV are given for each part of the structure in terms of pressure value.

The pressure at the deck is calculated by (DNV rule July 2011, Pt. 3 Ch 2, section 7B):

$$p1 = a(p_{dp} \text{-} (4+0.2k_s)h_0) \quad (6)$$

$a = 0.8$

h_0 = vertical distance in m from the waterline at draught T to the deck
= 1.3 m

(DNV rule July 2011, Pt. 3 Ch 2, section 5B)

$$k_s = 3kC_B + \frac{2.5}{\sqrt{C_B}} \quad (7)$$

= 7.711

$$pl = k_s C_{w+} k_f \quad (8)$$

k_f = the smallest of T and f
f = vertical distance from the waterline to the top of the ship's side at transverse section considered, maximum 0.8 C_W (m)

k_f=5

pl =17.166KN/m^2

$$p_{dp} = pl + 135\,\frac{y}{B+75} - 1.2(T - Z) \quad (9)$$

y=4.5 m,

= 24.7KN/m^2

Based on equations 7, 8 and 9, the deck pressure at the selected section was found to be:

$P1$ = 16.49KN/m^2

The pressure at tween deck is calculated by equations 10 and 11 (DNV rule July 2011, Pt. 3 Ch 2, section 7 B)

$$p_2 = kg_o q \quad (10)$$

q is deck cargo load in t/m^2 and equal to 1.

k=1.3,

Hence, $P2$ =12.68 KN/m^2

$$p_3 = k\rho_c g_o H_c \tag{11}$$

ρ_c= dry cargo density taken 0.7 t/m^3

Hc is the tween deck height = 4m

Finally, p_3 = 35.51 KN/m^2

Ka, correction factor for aspect ratio of plate field is obtained by eauation 12 (DNV rule July 2011, Pt. 3 Ch 2, section 3 B):

$$Ka=(1.1-0.25*s/l)^2 \tag{12}$$

s = 0.6 m stiffener spacing

l= length of stiffener span = 2.4 m

Ka =1.076

The deck plate thickness is calculated by equation 13(DNV rule July 2011, Pt. 3 Ch 2,section 7C):

$$t = \frac{15.8k_a s\sqrt{p}}{\sigma}+t_k \tag{13}$$

σ=120 N/mm^2

t= 5.28 mm

$$t = t_0 + k\,L + tk\;(mm) \tag{14}$$

t_0=5.5

k=0.02

t=5.5+0.02*76.8+1.5=8.536 mm

The minimum thickness of the plating of tween deck is given in equation15.

$$t = t_0 + t_k \tag{15}$$

t= 5.5mm+1.5mm = 7mm

4.5.1.2. Side Loads

Sea pressure below summer load water line is calculated by equation 16, (DNV rule July 2011, Pt. 3 Ch 2, section 6 B):

$$p_1 = 10h_o + p_{dp} \tag{16}$$

P_{dp} is from equation 9,

=36.7 KN/m^2

Sea Pressure above summer load waterline

$$p_2 = (p_{dp} - (4 + 0.2k_s)h_o \quad (17)$$

P_{dp} is from equation 9,

Ks from equation 7,

= 17.98KN/m^2

Pressure at ballast, bunker or liquid cargo is obtained by equations 18 and 19.

$$p_3 = k\rho g_o h_s \quad (18)$$

h_s distance from load point to top of tank = 2.6 m

k=1.3

g_0= 9.81kgm/s^2

P_3=39.2 KN/m^2

$$p_4 = \rho g_o h_s + p_o \quad (19)$$

$$p_o = 0.3L - 5 \quad (20)$$

P_0= 18.04 KN/m^2

P_4 = 51.14KN/m^2

The side plate thickness below the water line can be calculated using equation 13 and the pressure and stress of the side from equation 2 and σ =120 N/mm^2:

t= 5.833 = 6mm,

The thickness can also be obtained from equation 13;

$t = t_0 + k\,L + tk$ (mm)

t= 8.536 mm = 9mm

The minimum thickness for side structure is 9 mm.

4.5.1.3. Bottom Loads

The loads at the bottom of the vessel are calculated using equations 21, 22 and 23:

The outer bottom sea pressure is given as (DNV rule July 2011, Pt. 3 Ch 2, section 6 B):

$$p_1 = 10T + p_{dp} \quad (21)$$

P_{dp} is from equation 9,

P_1 = 86.7 KN/m^2

Pressure at a bottom structure having a liquid cargo in tank above is obtained by equation 18.

$$p_2 = \rho g_o h_s \tag{22}$$

P_2 = 30.16 KN/m^2

In inner bottom having a liquid tank above, the pressure is:

$$p_3 = 1.3\rho_c g_o h_s \tag{23}$$

P_3=39.215475KN/m^2

The minimum thickness of the bottom using equation 13 is;

σ= 130 N/mm^2

t= 8.55 mm= 9mm

The thickness of the plates at deck, tween deck, side and bottom structures calculated by the DNV rule, is the minimum required thickness for the specific vessel. The thickness values obtained from the Nauticus hull, also close to the above thicknesses. The thickness values for all the plates including the longitudinal plates rule requirement for mid ship section is given in Table 4.9.

Initial scantling gave rise to the requirement with color values that shows whether the plate fulfilled the requirement or not. Under the umbrella of the rule check, the thicknesses of the plates are changed until the required color, ‘green’ is observed which means it is good enough to withstand the current loading condition.

Table 4.9 Rule requirement longitudinal plates at midship

Position	Plate number	Thickness (mm)		Loc%	Buck%	Final%
		Rule	Actual			
Bottom	1	11.84	12	100	120	100
Bottom	2	9.94	12	133	120	120
Bottom	3	9.02	12	150	133	133
Bottom	4	8.07	9	113	0	113
Side	5	8.07	9	113	164	113
Side	6	9.51	10	111	105	105
SD	7	8.04	10.0	125	222	125
SD	8	10.45	11.0	105	169	105
SD	9	10.59	11.0	105	183	105
Stringer 1700	1	7.2	8	114	200	114
LGBD 4200	1	8.04	9	120	113	113
LGBD 4200	2	7.36	8.0	107	145	107
LGBD 4200	3	8.58	9.0	150	106	106
TD 3500	1	5.88	6.0	100	400	100
TD 3500	2	5.5	6.0	109	171	109
TD 3500	3	5.95	6.0	109	100	100
LGBD 1200	1	8.54	9.0	120	106	106
LGBD 1200	2	5.77	6.0	100	109	100
Stringer 4300	1	8.20	8.0	100	229	100
IB 1100	1	6.55	8	114	145	114
IB 1100	2	8.02	8.0	100	100	100
IB 1100	3	8.45	9.0	106	106	106
IB 1100	4	8.3	9.0	106	129	106
IS	1	6.83	8.0	114	145	114
IS	2	6.85	8.0	114	178	114
IS	3	10.57	11.0	157	105	105
LGBD7600	1	7.04	7	117	100	100

From the plate requirement, it was observed that, all the plate thicknesses used satisfy the local and buckling requirement. Therefore, the scantling obtained for longitudinal plates are adequate and acceptable. The maximum thickness of the plate is found to be 12 mm and is at the bottom of the hull. The next maximum is at the deck with a thickness of 11 mm.

4.5.2. Longitudinal Stiffeners

The section modulus of longitudinal stiffeners on deck (DNV rule July 2011, Pt. 3 Ch 2, Section 7C)

$$z = \frac{83\, l^2 s p w_k}{\sigma} \quad (cm^3) \qquad (24)$$

P=16.49 from equation 6, w_k=1.15,σ = 95 N/mm^2

Z= 57.27 cm^3

Longitudinal stiffeners are situated all on the strength (main) deck. The Rule requirement values and the given scantling for longitudinal stiffeners are given in Table 4.10.

Table 4.10 Rule requirement of longitudinal stiffeners midship

Position	Stiff. ID	Prof Type	Dimensions	Sect. mod. (cm^3)		Decisive req	Loc%	Buc%	Final%
				Rule	Actual				
SD	1	20	180*10	166	172	Zr	103	291	103
SD	2	20	200*12	243	252	Zr	109	293	104
SD	3	20	200*12	243	252	Zr	109	293	104
SD	4	20	200*12	243	252	Zr	109	293	104
SD	5	20	200*12	243	252	Zr	109	293	104
SD	6	20	200*12	243	252	Zr	109	293	104
SD	7	20	200*12	243	252	Zr	109	293	103
SD	8	20	200*11.5	248	248	Zr	100	292	100
SD	9	20	200*11.5	248	248	Zr	100	292	100
SD	10	20	200*11.5	248	248	Zr	100	292	100
SD	11	20	200*11.5	248	248	Zr	100	292	100
SD	12	20	200*11.5	248	248	Zr	100	292	100
SD	13	20	200*11.5	248	248	Zr	100	292	104

From Table 4.8 the longitudinal stiffeners used are mainly three types of HP and all satisfy the rule requirements. The maximum size of the stiffener is 200*12. Besides, the buckling strength witnessed is almost three fold to the requirement because the longitudinal stiffeners are mainly subjected to bending than buckling. This stiffeners support the longitudinal strength of the hull girder as they directly take the loads of cargo on the deck.

4.5.3. Rule Transverse stiffness

The section modulus of transverse stiffeners at tween deck is found by equation 25.

$$Z = 0.63 l^2 s p w_k \tag{25}$$

P is the pressure of at tween deck = 35.5 KN/m^2
Z= 77.3199922 cm^3
The minimum section modulus for the side is the greater of equations 25 and 26.

$$Z = 0.5 l^2 s p w_k \tag{26}$$

P = 36.70 KN/m^2
Z = 72.93cm^3

$$Z = 6.5\sqrt{L} \tag{27}$$

Z=56.96 cm3
The section modulus taken is Z=72.93 cm^3

The section modulus for the bottom of transverse stiffeners is calculated by equation 25 with the attributes of the bottom.

The value of Z is found to be:

P=39.21 KN/m^2

Z = 98.19 cm^3

The section modulus calculated from equations 25 to 27 are satisfied as the section modulus found for the transverse stiffeners According to DNV rules from Nauticus are greater than these values.

Table 4.11 Rule requirement of transversal stiffeners midship

Stiff.No.	ACT ACT	Pos Za cm^3	K c	Type Type (mm)	H T (mm)	bf tf	Y Z (mm)	σ_f F1 N/mm^2	M wk	Tkw Tkf (mm)	Tpl (mm)	Span Spac (mm)
LOC			Zr cm^3	Excess (%)	Tmin (mm)	Load Ref.		$\sigma N/mm^2$	Ang_PL deg	P KN/m^2	Comp Ref.	Aconn Cm^2
OS1	ACT	bottom	0,0	20	220	0	2700	235	10,0	1,5	12	3000
	ACT	292		HPbulb	10	0	0	1	1,09	1,5		600
	LOC		278	5	8	sea		160	90	75,1		0,0
OS2	ACT	bottom	0,0	20	240	0	5900	235	10,0	0	12	3000
	ACT	361		HPbulb	10	0	0	1	1	0		600
	LOC		337	7	6,5	sea		160	90	80		0,0
OS3	ACT	side	0,0	20	160	0	9000	235	10,0	0	9	2300
	ACT	109		HPbulb	7	0	3150	1	1	0		600
	LOC		108	1	5,8	sea		160	90	53,9		0,0
OS4	ACT	side	0,0	20	200	0	9000	235	10,0	1,5	10	4000
	ACT	216		HPbulb	9	0	5900	1	1	1,5		600
	LOC		196	10	7,3	sea		160	90	29,7		0,0
TD 1	ACT	twsk	0,0	20	200	0	2700	235	10,0	0	6	3000
	ACT	215		HPbulb	10	0	3500	1	1	0		600
	LOC		209	3	5,8			160	90	61,5	5	0,0
TD 2	ACT	twsk	0,0	20	80	0	4800	235	10,0	0	6	1562
	ACT	23		HPbulb	6	0	3000	1	1	1		600
	LOC		15	56	6,5	oil			90	0	1	0,0
IBIS1	ACT	Hoptk	0,0	20	100	0	7900	235	10,0	0	8	1801
	ACT	37		HPbulb	6	0	1550	1	1	0		600
	LOC		33	10	5,8	oiltst		160	90	75,8	1	0,0
IBIS2	ACT	InSid	0,0	20	140	0	8200	235	10,0	0	8	1800
	ACT	79		HPbulb	7	0	3150	1	1	0		600
	LOC		76	4	5,8	oiltst		160	90	62,1	1	0,0
IBIS3	ACT	InSid	0,0	20	80	0	8200	235	10,0	1,5	11,0	4000
	ACT	264		HPbulb	13	0	8200	1	1,09	1,5	11	600
	LOC		255	4	8	oiltst	5900	160	90	38,7	1	0,0

In Table 4.11, for all the transversal stiffeners at cross-section of frame 64, the excess percentage is positive; hence, it satisfied the rule minimum requirements.

The plate thickness requirement of cross-section at frame 22 and its outcome is presented in Table 4.12.

Table 4.12 Rule requirement of plate of cross-section at frame 22

Position	Plate number	Thickness (mm)		Loc%	Buck%	Final%
		Rule	Actual			
Bottom	1	10.84	11	100	183	100
Bottom	2	8.07	9	113	120	113
Bottom	3	8.07	9	113	120	113
Side	4	8.07	9	113	180	113
Side	5	8.07	9	113	180	113
Side	6	9.07	9	100	113	100
SD	7	6.64	9	138	222	138
SD	8	8.68	9	106	180	106
SD	9	9.17	9.0	100	180	100
IBIS	1	7.3	8.0	107	123	107
IBIS	2	7.8	8.0	100	114	100
IBIS	3	6.9	8.0	114	160	114
IBIS	4	7.43	8.0	114	107	107
LGBH 4200	1	5.77	6.0	100	133	100
LGBH 4200	2	6.90	7.0	100	140	100
LGBH 4200	3	5.88	7.0	117	280	117
LGBD 6900	1	7.2	8.0	114	178	114
Stringer 4300	1	8.2	9.0	113	360	113
TD 3500	1	5.82	6.0	100	171	100
ST	1	8.07	8.0	100	178	100
TD	2	7.20	8.0	114	200	114
Stringer 1700	1	7.2	7	100	156	100
BG 1200	1	7.54	8.0	107	178	107

From Table 4.12, the tabular values show that the plates satisfy the minimum requirement for local and buckling strength. The final three columns show values equal and greater than 100. The maximum buckling strength was observed at stringer 4300.

Table 4.13 Rule requirement of longitudinal stiffener at deck

Position	Stiff. ID	Prof Type	Dimensions	Sect. mod. (cm^3)		Decisive req.	Loc%	Buc%	Final%
				Rule	Actual				
SD	1	20	200*10	198	218	Zr	110	420	110
SD	2	20	220*10	259	282	Zr	109	421	109
SD	3	20	220*10	259	282	Zr	109	421	109
SD	4	20	220*10	259	282	Zr	109	421	109
SD	5	20	220*10	259	282	Zr	109	421	109
SD	6	20	220*10	259	282	Zr	109	421	109
SD	7	20	220*10	261	282	Zr	108	421	108
SD	8	20	220*11.5	296	305	Zr	103	421	103
SD	9	20	220*11.5	296	305	Zr	103	421	103
SD	10	20	220*11.5	296	305	Zr	103	421	103
SD	11	20	220*11.5	296	305	Zr	103	421	103
SD	12	20	220*11.5	296	305	Zr	103	421	103
SD	13	20	220*10	216	278	Zr	129	422	129

The requirement and outcome based on final scantling according to DNV rule, for transversal stiffeners at cross-section frame 22 is shown in Table 4.14. The values at the last three columns clearly show that the stiffeners live up to the requirement. The value that needed to be stressed at were those associated with the bending as their main purpose is to substantiate the longitudinal strength.

Table 4.14 Rule requirement of transversal stiffeners at cross-section frame 22

Stiff.No	ACT ACT	Pos Za cm^3	K c	Type Type (mm)	H T (mm)	Bf tf	Y Z (mm)	σf F1 N/mm^2	M wk	Tkw Tkf (mm)	Tpl (mm)	Span Spac (mm)
LOC			Zr cm^3	Excess (%)	Tmin (mm)	Load Ref.		$σN/mm^2$	Ang_PL deg	P KN/m2	Comp Ref.	Aconn Cm^2
OS1	ACT	bottom	0,0	20	220	0	2700	235	10,0	1,5	12	3000
	ACT	292		HPbulb	10	0	0	1	1,09	1,5		600
	LOC		278	5	8	sea		160	90	75,1		0,0
OS2	ACT	bottom	0,0	20	240	0	5900	235	10,0	0	12	3000
	ACT	361		HPbulb	10	0	0	1	1	0		600
	LOC		337	7	6,5	sea		160	90	80		0,0
OS3	ACT	side	0,0	20	160	0	9000	235	10,0	0	9	2300
	ACT	109		HPbulb	7	0	3150	1	1	0		600
	LOC		108	1	5,8	sea		160	90	53,9		0,0
OS4	ACT	side	0,0	20	200	0	9000	235	10,0	1,5	10	4000
	ACT	216		HPbulb	9	0	5900	1	1	1,5		600
	LOC		196	10	7,3	sea		160	90	29,7		0,0
TD 1	ACT	twsk	0,0	20	200	0	2700	235	10,0	0	6	3000
	ACT	215		HPbulb	10	0	3500	1	1	0		600
	LOC		209	3	5,8			160	90	61,5	5	0,0
TD 2	ACT	twsk	0,0	20	80	0	4800	235	10,0	0	6	1562
	ACT	23		HPbulb	6	0	3000	1	1	1		600
	LOC		15	56	6,5	oil			90	0	1	0,0
IBIS1	ACT	Hoptk	0,0	20	100	0	7900	235	10,0	0	8	1801
	ACT	37		HPbulb	6	0	1550	1	1	0		600
	LOC		33	10	5,8	oiltst		160	90	75,8	1	0,0
IBIS2	ACT	InSid	0,0	20	140	0	8200	235	10,0	0	8	1800
	ACT	79		HPbulb	7	0	3150	1	1	0		600
	LOC		76	4	5,8	oiltst		160	90	62,1	1	0,0
IBIS3	ACT	InSid	0,0	20	80	0	8200	235	10,0	1,5	11,0	4000
	ACT	264		HPbulb	13	0	8200	1	1,09	1,5	11	600
	LOC		255	4	8	oiltst	5900	160	90	38,7	1	0,0

4.5.4 Bulkhead Structure

The model has bulkheads that separate each cargo holds. The minimum requirement for strength of the bulkheads according to DNV rule is assumed.

For water tight bulkhead, the pressure is calculated by equation 28. (DNV Pt. 3 Ch 2, Section 8C)

$$P = 10h_b \tag{28}$$

P = 10*7 = 70 KN/m^2

The plate thickness is also obtained by equation 13.

With σ = 160,

t= 8.24 mm

The minimum thickness alternatively has to equal: (DNV Pt. 3 Ch 2, Section 8C)

$$t = 5.0 + k\,L + tk\ (mm) \tag{29}$$

For k =0.03,

t= 8.804 mm

Hence, the minimum plate thickness taken is 9 mm.

The section modulus for the bulkhead is calculated by equation 24. For the pressure obtained in equation 28, and σ = 160, the minimum section modulus is:

Z =188.244 (Cm^3)

4.6. Final Scantling

The final scantling of the midship section obtained by ‘rule check’ of the plates is presented in Fig 4.9.

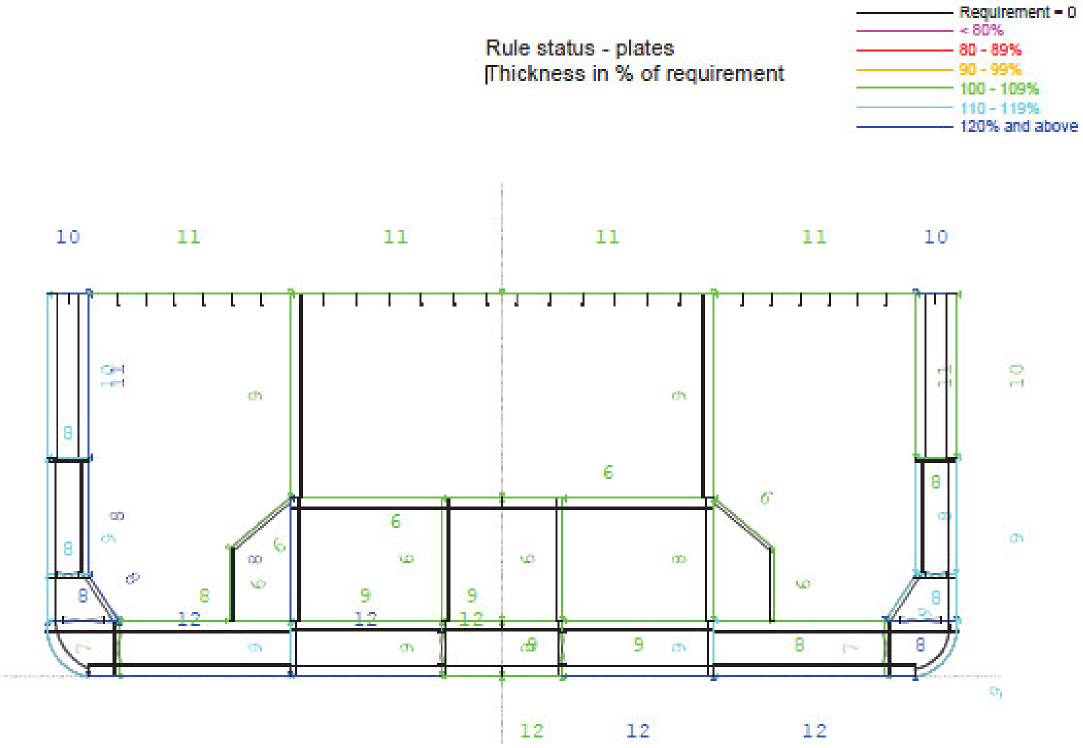


Figure 4.9 Final scantling of plates at midship section

From Fig 4.9, all plates have passed the rule requirement test as they are not painted with red or pink that describe the plate doesn’t suffice the requirement. The Figure also shows that the maximum thickness of the plate is 12 mm situated at the bottom of the vessel. The strength deck of the vessel is where the next thick plate is. In the mid ship section the minimum thickness of plate used is 6 mm.

Due to the nature of the software, the plates and stiffeners are pictured in separate figure to conform to the rule check. The longitudinal stiffeners with the final scantling that passed the requirement is presented in Fig 4.10 for the midship section.

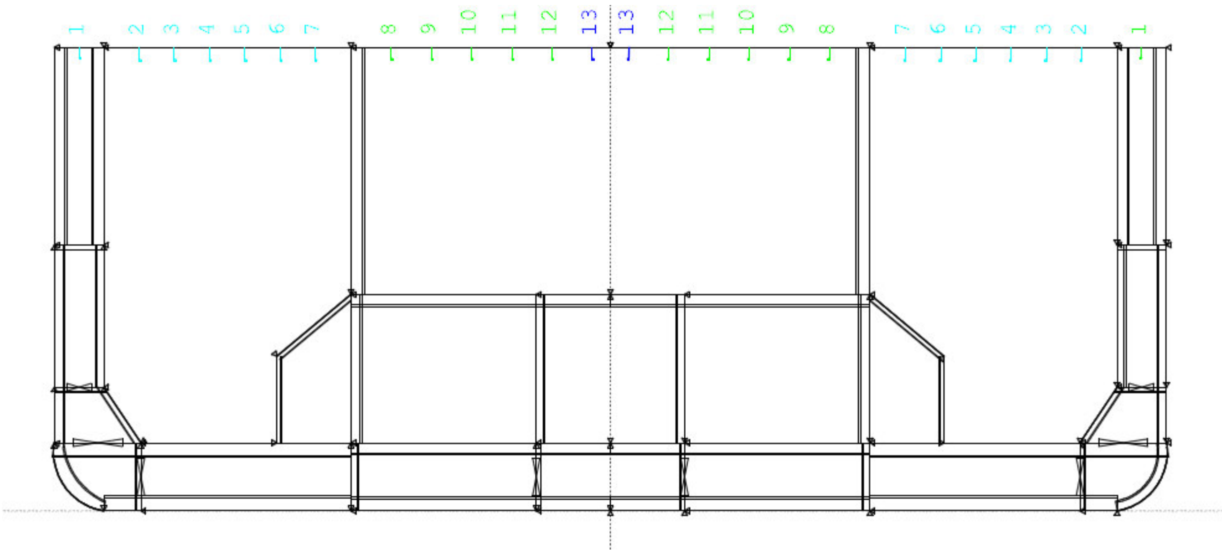


Figure 4.10 Final scantling of longitudinal stiffeners at midship section

The color of the stiffeners show, their dimension and material satisfies the requirements set by the rules.

The final scantling according to DNV for plates in cross-section at frame number 22 is given in Fig 4.11.

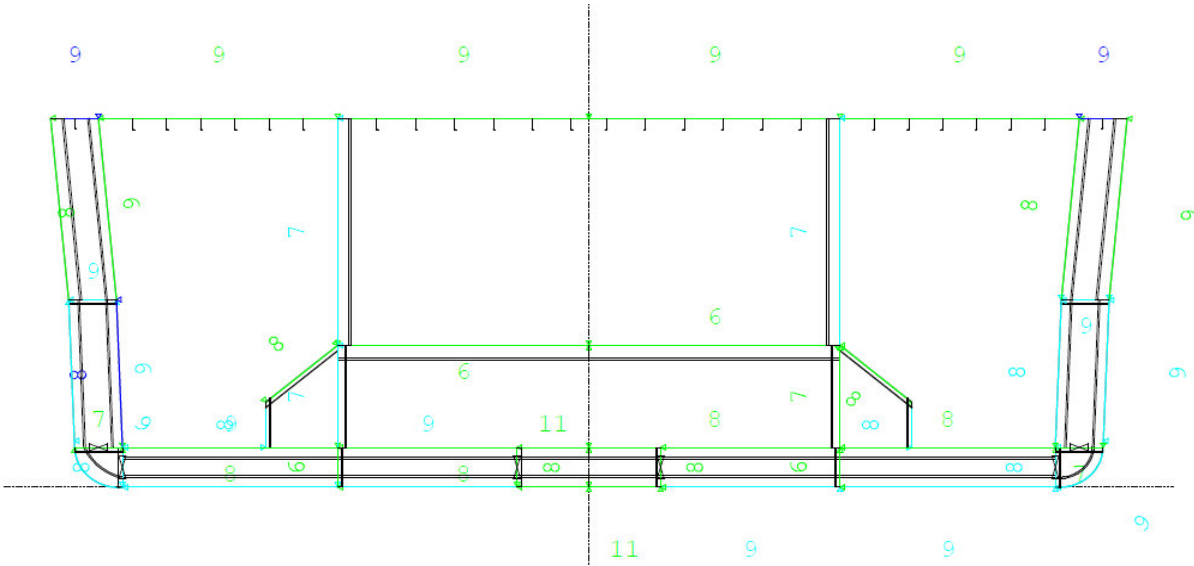


Figure 4.11 Final scantling of plates at cross-section frame 22

From Fig 4.11, the thickest plate used is at the bottom of the deck with 11 mm. like the midship section, the minimum thickness used here also amount to 6 mm. The minimum thickness is at the slant plate covering the shaft vault adjourning bulkhead located at a distance of 4.20 m from the centerline. The colours displayed show are all blue and green indicating that the thicknesses conforms to the requirements of the DNV rule.

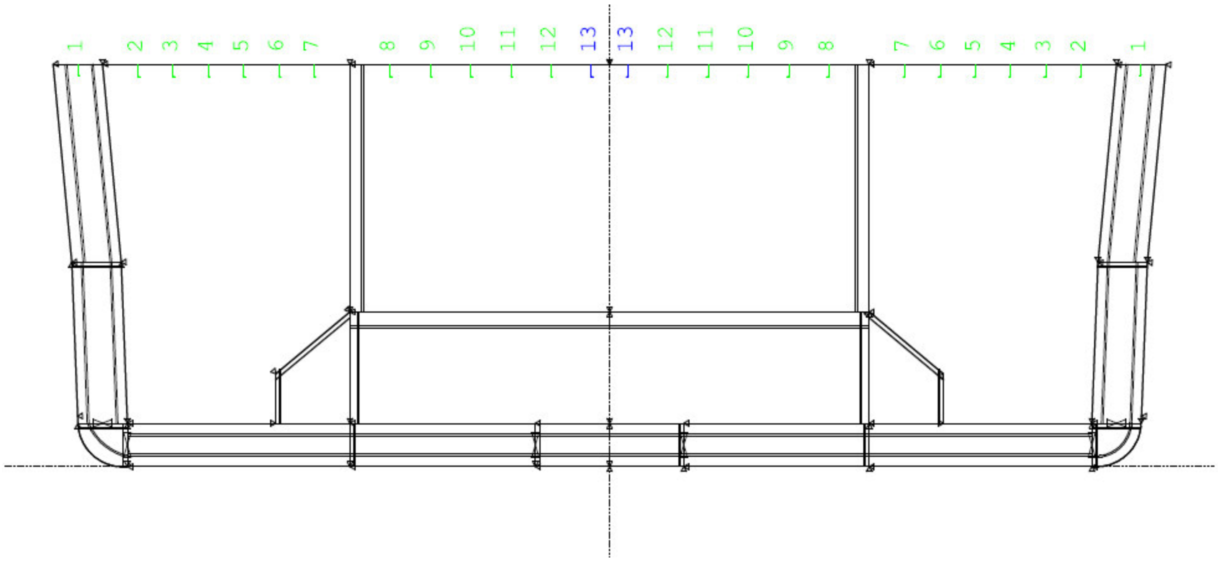


Figure 4.12 Final scantling of longitudinal stiffeners at cross-section at frame 22

From Fig 4.12, like the previous cases, the final scantlings of the longitudinal stiffeners at deck of cross-section frame 22 are all green and blue. Therefore it could be deemed as adequate for it satisfies the requirements.

At this point both cross-section frames (64 and 22) have fulfilled the longitudinal requirements for plates and stiffeners,

The rule check highlights the requirement of the section selected for only plates and longitudinal stiffeners. The rule check for transversal stiffeners is found from the printed result. Stiffener with a star near it in the printed result is the one that doesn't fulfil the requirement. Hence more adjustment in the dimension had to be made until the star mark is vanished. The result of the transversal stiffeners obtained by rule check is attached in the appendix. It showed that all the stiffeners have passed the requirements for both frames.

4.7. Buckling check

After carrying out the requirement tests, it is also imperative that the scantling well assumes and satisfies the requirements for buckling. If the scantling is not checked for buckling, it may result to sudden failure due to compressive loading. The stiffeners for buckling check are portrayed in Table 4.15.

Table 4.15 Buckling strength of the transversal stiffeners at mid ship section

Stiff. No	ACT1 ACT2	Elem.type Long.span mm	Type	h tw mm	bf tf mm	S tp mm	σtp σts N/mm2	Nstif f bf/h act	tkw h/tw act mm	Tkf bff/tf act mm	Tkp s/tp act mm	Y z mm
	Applied Stresses		P KN/m2	σx,1 N/mm2	σx,1 N/mm2	σy N/mm2	τ	Bf/h req	h/tw req	bff/tf req	s/tp req	OK?
	Ultimate Capacity			σx,1UC N/mm2	σx,2UC N/mm2	σy, UC N/mm2	τUC		Crit.limit state	ηuc	ηmax	OK?
	Buckling Strength			σx,1BS N/mm2	σx,2BS N/mm2	σy, BS N/mm2	τBS			ηBS	Elastic buckling	OK?
1	ACT1	S3	20	180	35	400	235	1	1.5	1.5	1	8600
	ACT2	2400	HP bulb	10	17.8	10	235		19.3	1.6	44.4	7500
	Applied Stress			65	65	0	0		90.0	15.0	200.0	yes
	Ultimate capacity			209	209	0	0		4	0.31	0.90	yes
	Buckling strength			209	209	0	0			0.31	no	Yes
2	ACT1	S3	20	220	41	570	235	6	2.0	2.0	1.0	7630
	ACT2	2400	Hpbulb	10	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stress			65	65	0	0		90.0	15.0	200.0	Yes
	Ultimate capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling strength			195	195	0	0			0.33	No	
5	ACT1	S3	20	220	41	570	235	6	2.0	2.0	1.0	5920
	ACT2	2400	Hpbulb	10	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stress			65	65	0	0		90.0	15.0	200.0	Yes
	Ultimate capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling strength			195	195	0	0			0.33	No	
6	ACT1	S3	20	220	41	570	235	6	2.0	2.0	1.0	5350
	ACT2	2400	Hpbulb	10	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stress			65	65	0	0		90.0	15.0	200.0	Yes
	Ultimate capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling strength			195	195	0	0			0.33	No	

From Table 4.15, the stiffeners do not fail under elastic buckling. In addition, the applied stress and the ultimate capacity live up to the requirement.

The Buckling of plates according to DNV rule, for a plate thickness of 12 mm located at bottom center, for instance, is presented in Fig 4.13.

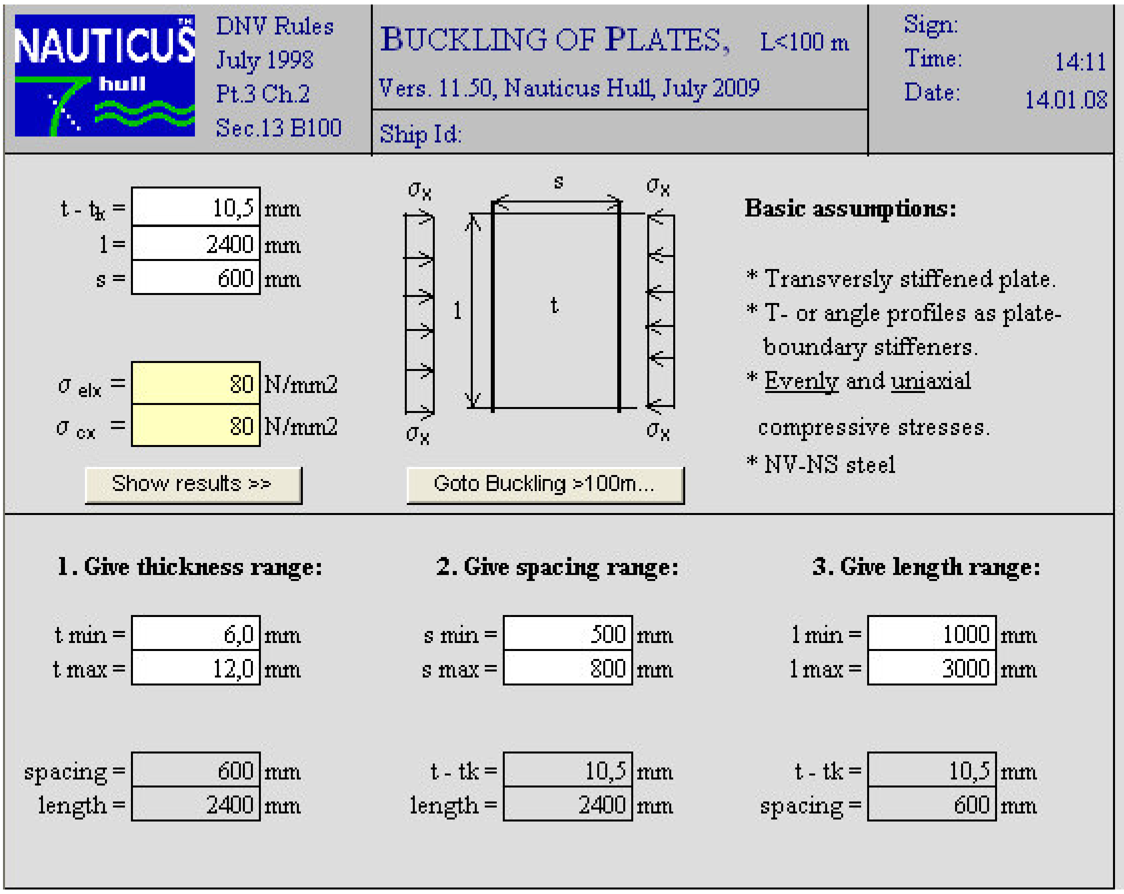


Figure 4.13 Buckling strength of plate

From Fig 4:13, a plate of a mild steel material of stress 235 MPa, thickness of 12 mm with corrosion allowance t_k of 1.5 mm is used and the buckling stress maximum allowable is 80 Mpa. For a different plate thicknesses ranging from 6 mm to 12 mm and spacing of 600 mm, the buckling stress graph is found and depicted in Fig 4.14.

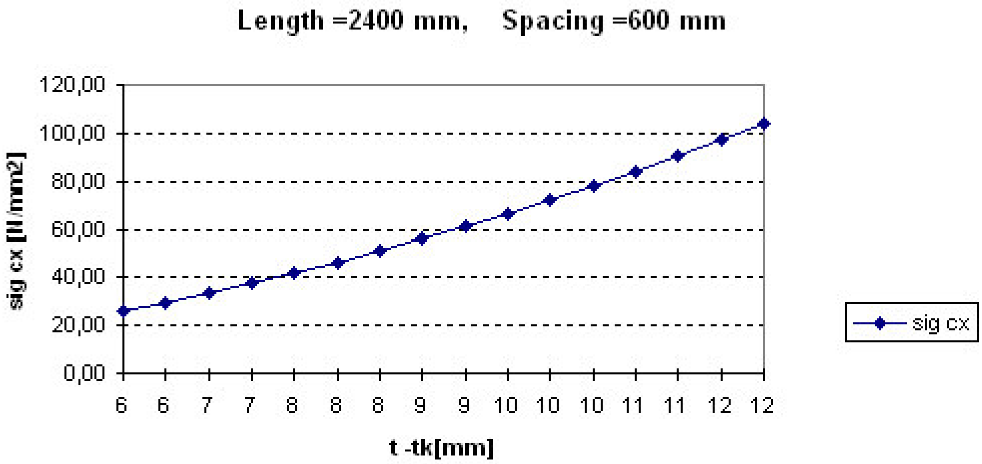


Figure 4.14 Buckling strength of plate for varying thickness

From Fig 4.14, it could be deduced that the buckling strength increases with increasing thickness and keeping the same aspect ratio. The minimum thickness of the plate of the platform supply vessel used in the thesis at the area under consideration is 6mm with a buckling strength a little above 20 Mpa and the maximum is 12 mm with a buckling strength of 80Mpa. Having the same thickness of 12 mm and length of a plate 2400, between web frames, an optimum spacing of the stiffeners could be decided based up on the Fig 4.15.

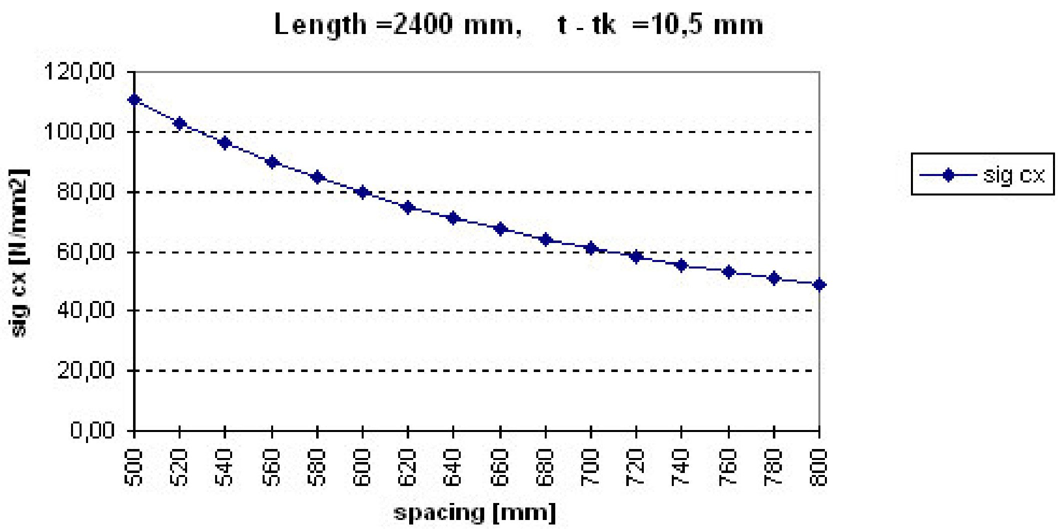


Figure 4.15 Buckling strength of plate for varying spacing between stiffeners

From Fig 4:15, the length and thickness being fixed, the spacing that gives the maximum buckling strength could also be resorted. Alternatively, the spacing and thickness being kept, the way to decide the spacing between web frames for desired level of buckling strength could be obtained from Fig 4:16:

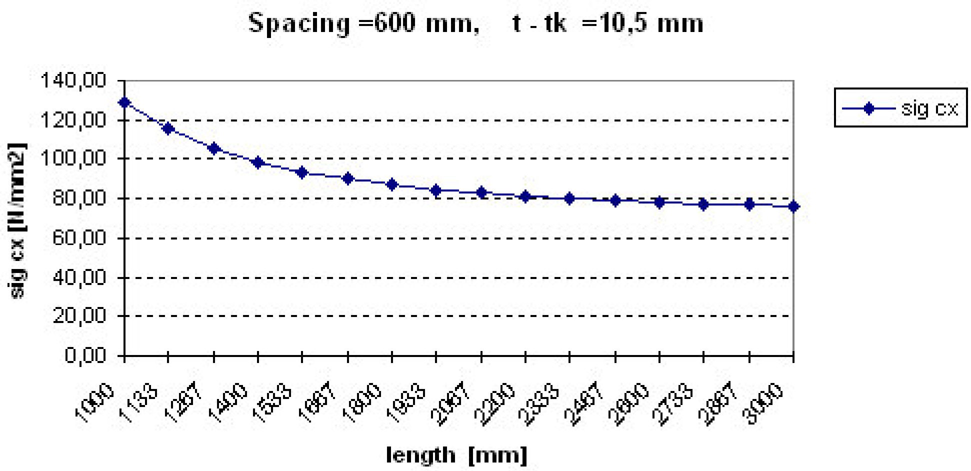


Figure 4.16 Buckling strength of plate for varying spacing between web frames

4.8. Strength of the Structure

The scantling output shows that the longitudinal stress on the points for the hull girder is way below the maximum. Hence, it could be taken as a success on this part. To look in to detail, sample stress for both sagging and hogging case at outer shell in frame 64 is presented in Table 4.16.

Table 4.16 Stress at mid ship section

Position		σx (MPa)	
		Sagging	Hogging
x	y	-55	-55,2
0	0	59,9	-55,2
1200	0	59,9	-55,2
4200	0	59,9	-55,2
7600	0	59,9	-55,2
8200	0	59,9	-55,2
9000	1100	41,9	-38,4
9000	2000	26,6	-24,6
9000	4300	-11,6	10,7
9000	7500	-64,8	59,8

From Table 4.16, the stress in sagging condition is constant around the bottom of the hull where the elements are in tension with a value of 59.9 MPa. It starts to decrease at the bottom half of the hull until it becomes zero at the neutral axis. On the upper part of the hull, above the neutral axis, the stress is compressive one and the maximum value reaches around 64.8 MPa near the top corner of the side and the deck. This upper right corner of the deck where the compressive stress is higher than other parts of the hull is the spot that draws more interest.

The screen shot of pictorial representation of design stress for sagging on the outer shell is given in Fig 4.17.

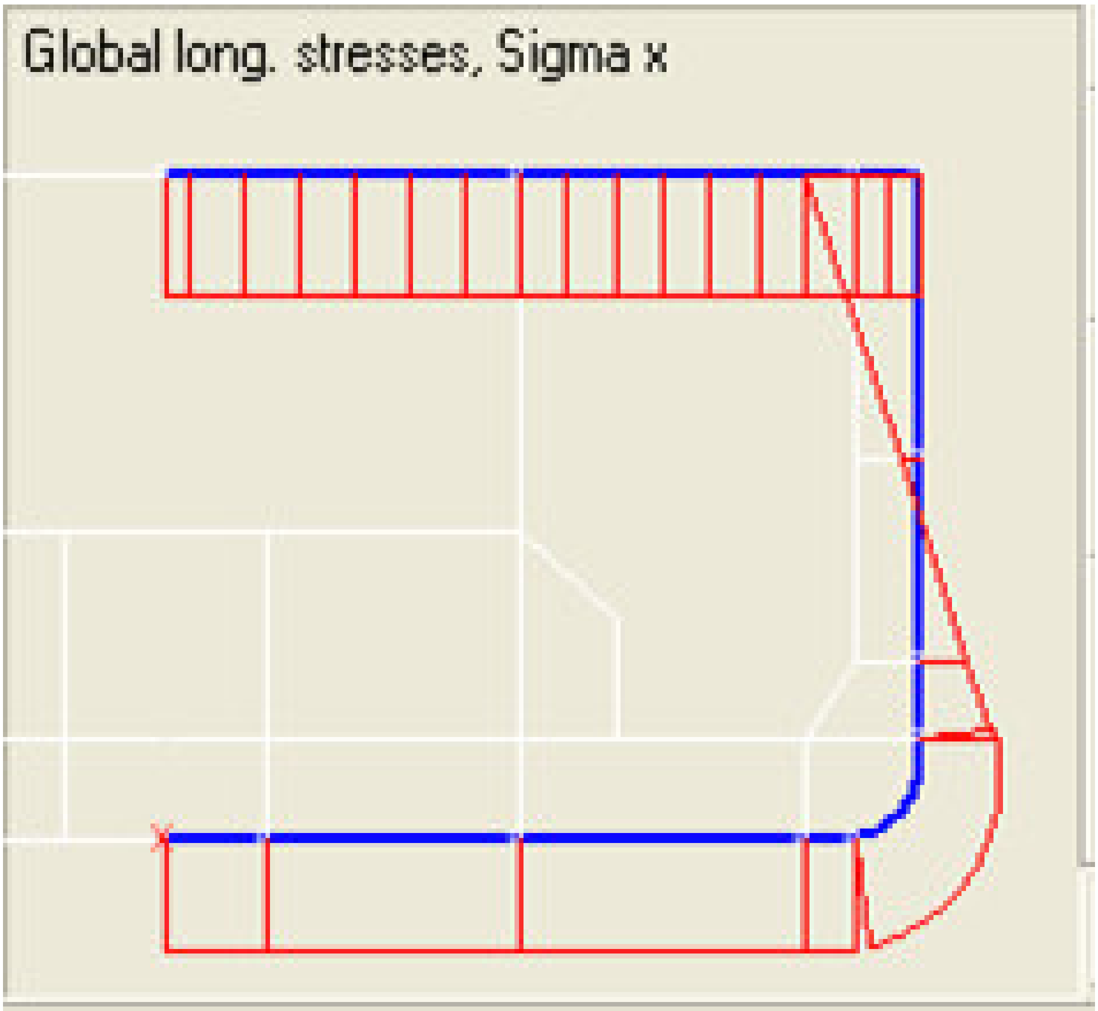


Figure 4.17 Global longitudinal stress in sagging at mid ship section

From Table 4.14, the stress maximum to be encountered in sagging is -64.8Mpa that is compressive one. Obviously, in sagging, the deck part is in compression and the bottom is in tension.

In hogging, the maximum stress was encountered at deck near the corner point of the mid ship section.

The global longitudinal stress for hogging is given in Fig 4.18

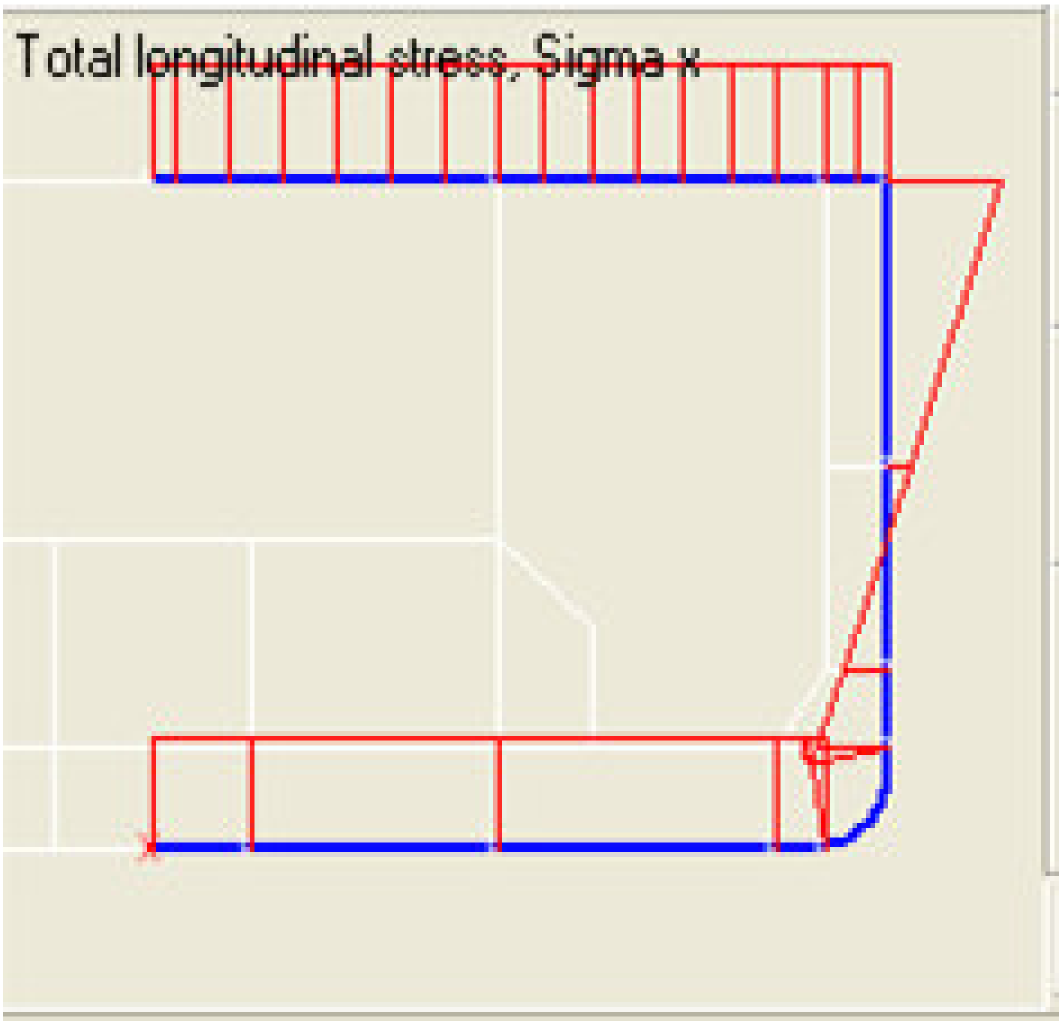


Figure 4.18 Global longitudinal stresses in hogging at midship section

The data presented in Table 4.14 for hogging is presented in Fig 4.18. Both the Figure and the data justify that the global stress in the outer shell is moderate and satisfies the requirement.

In Fig 4.19, the maximum global longitudinal stress in sagging is found out to be 41.2 KN/m^2. It is located at the bottom of the structure.

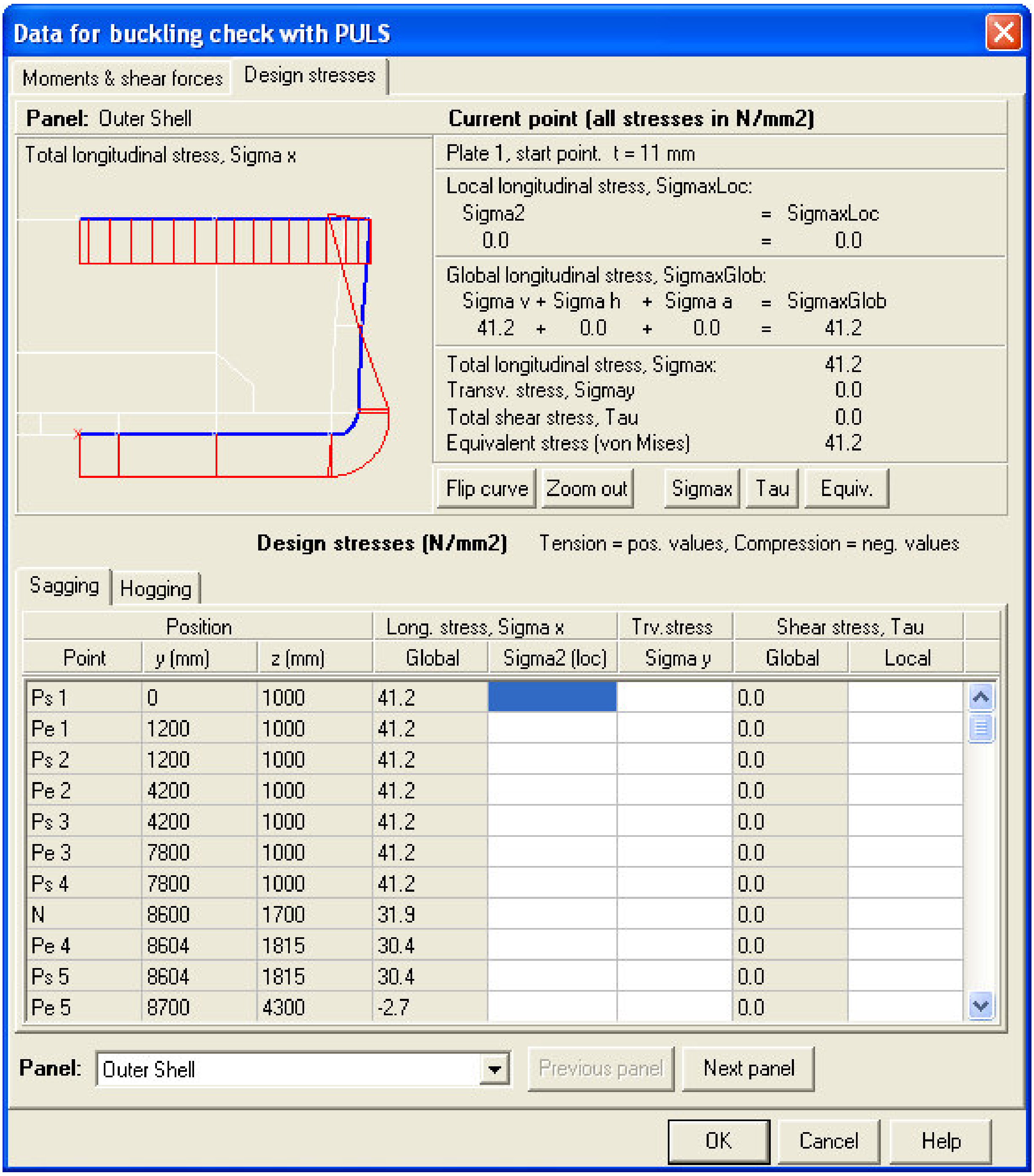
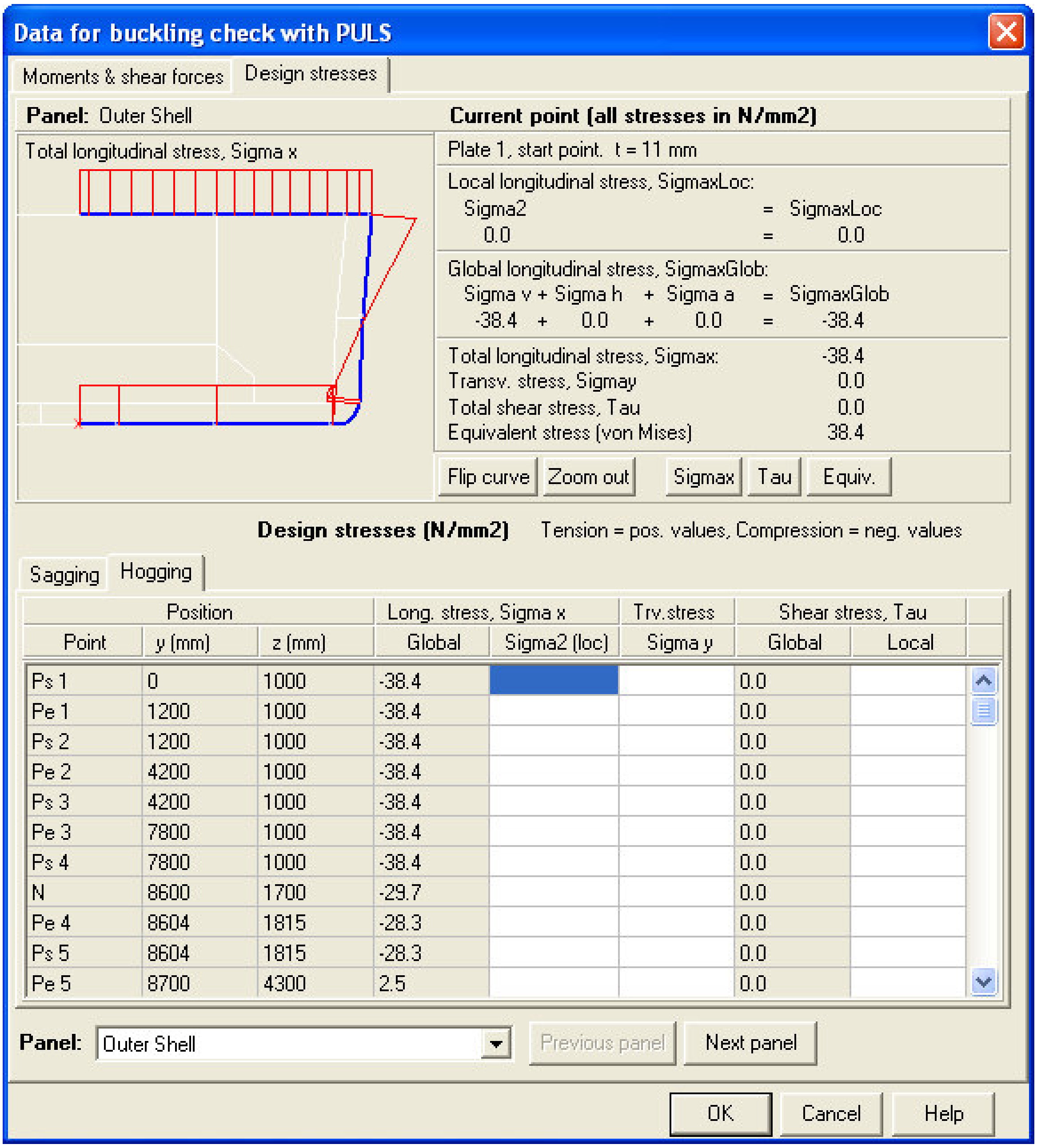


Figure 4.19 Global longitudinal stresses in sagging at cross-section frame 22



Position			Long. stress, Sigma x		Trv.stress	Shear stress, Tau	
Point	y (mm)	z (mm)	Global	Sigma2 (loc)	Sigma y	Global	Local
Ps 1	0	1000	-38.4			0.0	
Pe 1	1200	1000	-38.4			0.0	
Ps 2	1200	1000	-38.4			0.0	
Pe 2	4200	1000	-38.4			0.0	
Ps 3	4200	1000	-38.4			0.0	
Pe 3	7800	1000	-38.4			0.0	
Ps 4	7800	1000	-38.4			0.0	
N	8600	1700	-29.7			0.0	
Pe 4	8604	1815	-28.3			0.0	
Ps 5	8604	1815	-28.3			0.0	
Pe 5	8700	4300	2.5			0.0	

Figure 4.20 Global longitudinal stresses in hogging at cross-section frame 22

From Fig 4.20, the maximum global longitudinal stress on outer shell was found in deck amounting to -38.4 KN/m^2.

The values obtained in sagging and hogging for global longitudinal stress in all plates are in harmony with the requirement. Moreover, it was observed that the global longitudinal stress is less than the local stress at each plate.

4.9. Hull Weight Estimation

Estimation of the mass of the structural elements gives a clue about how heavy the structure is and which members of the structure constitute what percentage of the aggregate. This casts light to which elements the focus should be put on for further optimization of the scantling. The final scantling found by the Nauticus hull is extruded in Rhinoceros and finally exported to Ansys where the total volume and mass of the structure is obtained.

The mass of each group of members constituted in one cargo hold is given in Table 4.17.

Table 4.17 Weight of members of the vessel at midship

	M, t	*M, % Mt*
Longitudinal plates	637.89	62.91
Longitudinal Stiffeners	30.087	2.97
transverse plates (floors and web frames)	38.2122	3.77
Bulkheads plates	6.387	0.63
transversal stiffeners	214.5762	21.16
Bulkhead (frameworks and stringers)	86.816	8.56
Total mass in one cargo hold, *Wt*		1013.968

5. STRENGTH ANALYSIS by FEM

5.1. Introduction

The objective of direct strength analysis is to compare the result obtained with the rule based results of the classification societies. The difficulty associated with the direct approach is that all the criteria and loads must be known. Even though the direct strength analysis may seem an uphill challenge, it helps to substantiate the rule based results.

A finite element method is a numerical method used to solve differential equations constrained by boundary conditions. It is an approximate method that discretizes area of interest in to nodes arranged in a grid form called mesh. The mesh or unit element in the grid behaves physically or structurally to the parent material. In engineering applications, the structure is divided in to elements having a mathematical model and elemental solutions are obtained that aids in solving the overall structure.

For the direct strength assessment, a cargo hold is selected and two more cargo at each side is used. The geometry is modelled in 3D in Rhinoceros Software and exported to Ansys Workbench. A four node shell element, SHELL181, is used for the decks, sides and bulkhead plates. For Longitudinals and transversal stiffeners, beam188 is used. It is a two node element based on Timoshenko beam theory. The effect of shear deformation is included. It is with a notion that a plane element shall not remain plane after deformation. Both the beam and shell elements selected are characterized by six degrees of freedom. With the configuration of shell and beam elements, a quadrilateral dominant mesh is utilized.

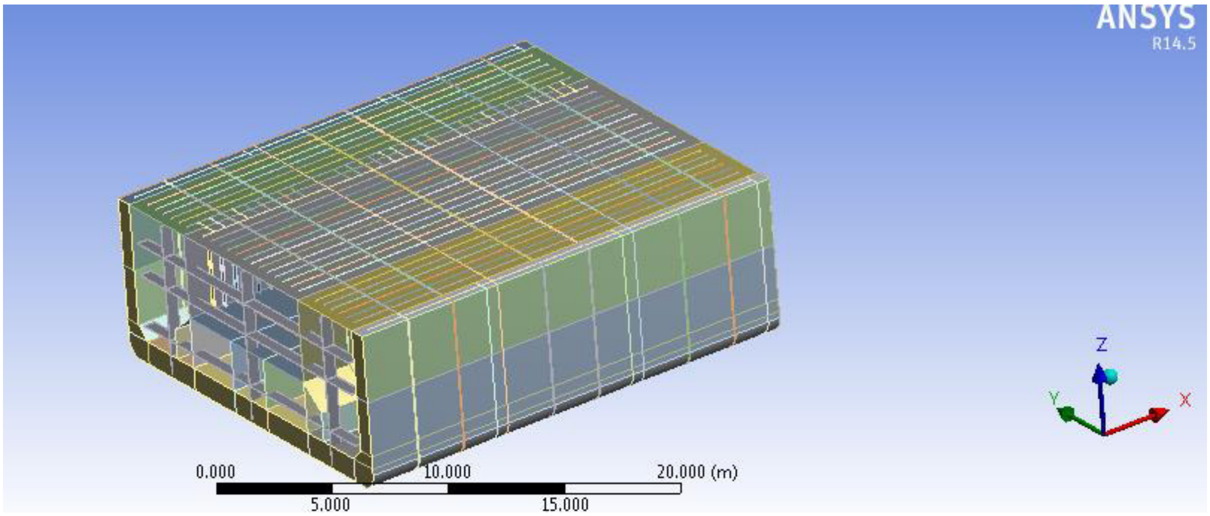


Figure 5.1 Three dimensional view of the model at the midship section

In fig 5.1 Three cargo hold is presented with minimum thickness being 6 mm and maximum one is 12 mm.

- Element size: 233332
- Number of nodes: 883714

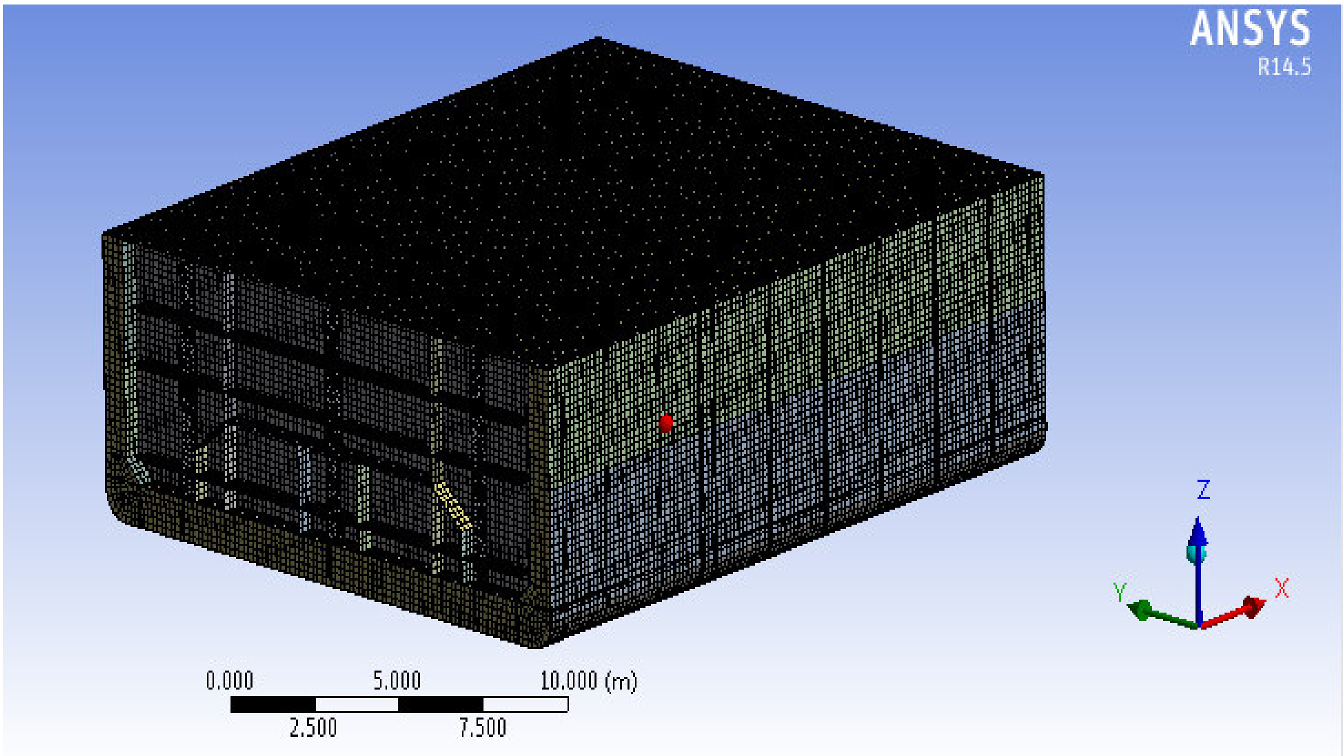


Figure 5.2 Meshed view of model

In Fig 5.3, The mesh for the longitudinal stiffeners is presented.

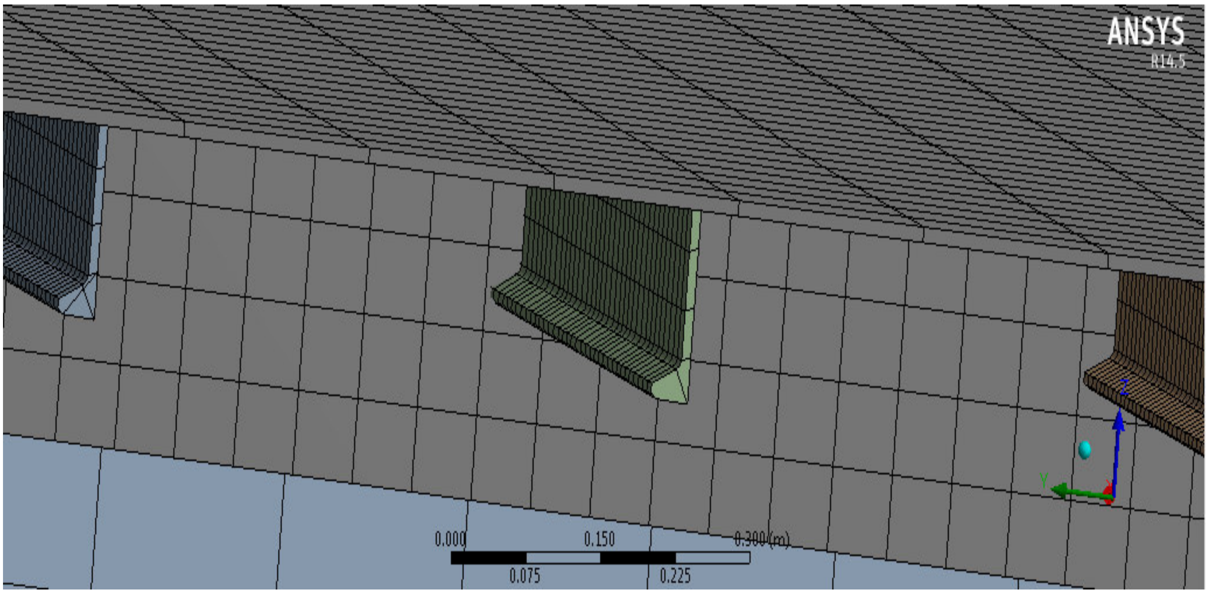


Figure 5.3 Detailed mesh view

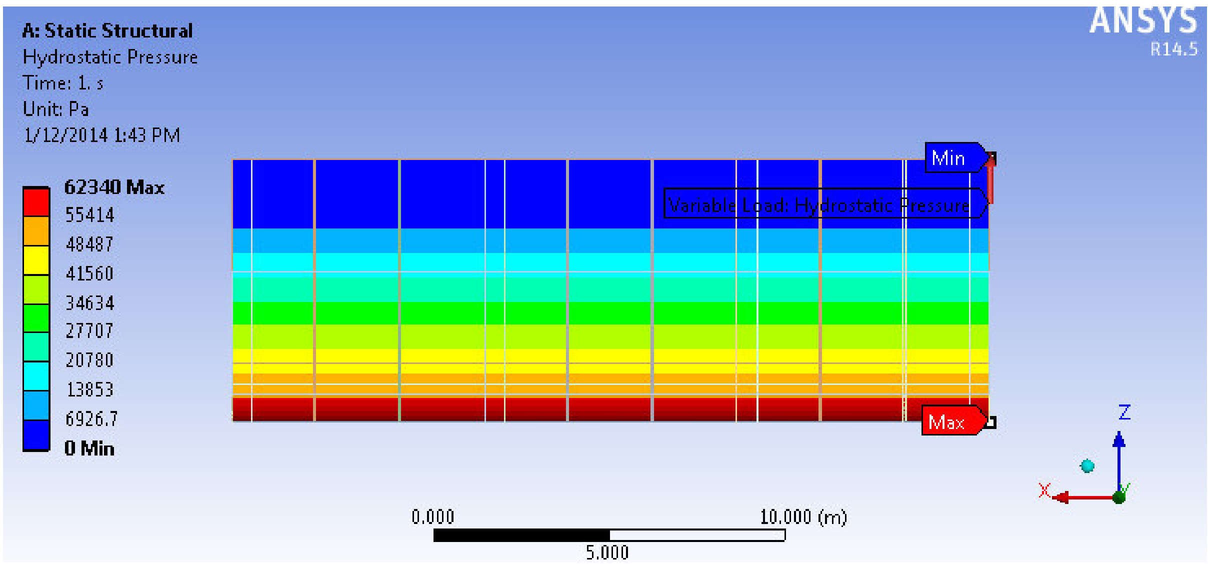


Figure 5.4 Hydrostatic Pressure

5.2. Load Cases

For the FEM analysis still water loading case is applied. The pressure at the deck, due to the deck load is P_d = 78452 Pa.

The hydrostatic pressure at the hull of the vessel is variable, starting at maximum at the bottom, P = 62342.58 to minimum at the free surface.

Pressure at the tween deck, a the longitudinal distance x =6.6 m, transversal distance y = 7.8 m and the height 3.3 m bounding the hold, and the density of amounting to 700 kg/m^3, the pressure is P_{td} = 11128 Pa.

Pressure at the side hold, with a longitudinal distance x =6.6 m, transversal distance y = 3.4 m and the height 5.9 m, and the density of amounting to 750 kg/m^3, the pressure is P = 22072 Pa.

The other load considered is the pressure at tank top, below tween deck, with the longitudinal distance x =6.6 m, transversal distance y = 7.8 m and the height 2.2 m and the density of 750 kg/m^3, the pressure is P = 8000 Pa.

The volume of each cargo holds to calculate the loads are taken up to the filling height of the cargo so as to find realistic loading. To account for all odds of loading conditions, the maximum is used.

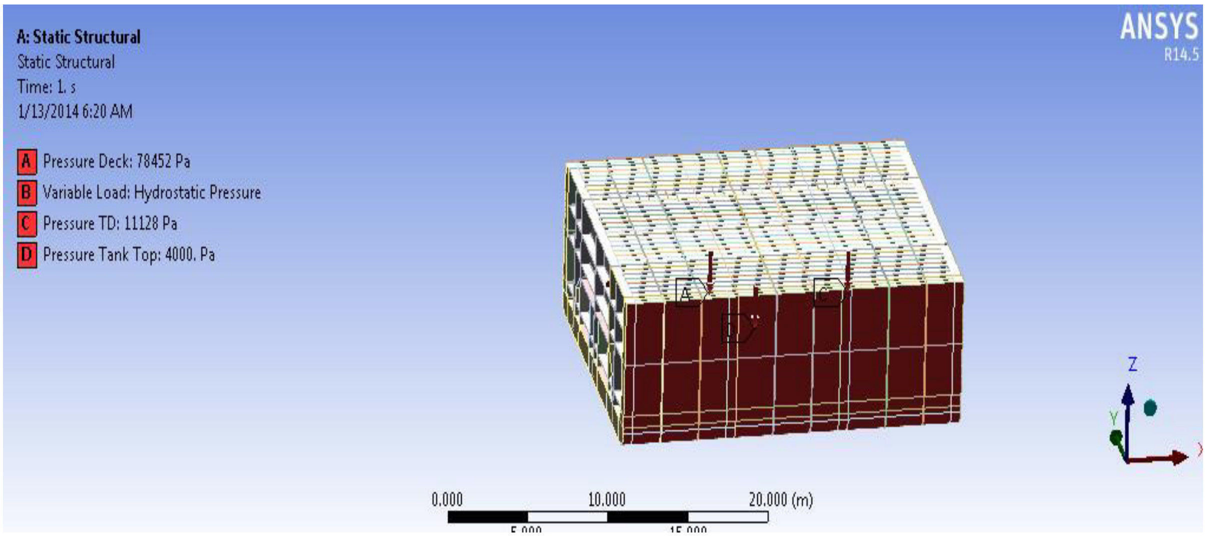


Figure 5.5 Still water loading

5.3. Boundary Condition

The boundary conditions applied to the FE model are dependent on the extent of ship modelled and the load case to be analyzed. Different boundary conditions need to be applied for symmetric and asymmetric load cases. Symmetric boundary conditions for global loads are suitable for the analysis of global loads. These boundary conditions allow the FE model to deflect globally under the action of hull girder vertical shear forces and bending moments. On the other hand, symmetrical boundary conditions for local loads are suitable for calculating stresses resulting from local loads because it removes the effects of hull girder bending from the FE model.

In order to calculate a solution, ANSYS requires data at the boundaries of each sub-task's domain of definition. This means that boundary conditions are required both along external boundaries (boundary sets) and along internal boundaries between subdomains that belong to different sub-tasks. The boundary conditions according to DNV rule are pictured in Fig 5.6.

Location	*Displacement*			*Rotation*		
	δx	δy	δz	θx	θy	θz
Plane A	L	-	-	-	X	X
Plane B	X	-	-	-	X	X
Line C			S			
Point a, b		X				
Point c	F_h					

X	=	Restricted from displacement or rotation
L	=	Linearly dependant of point c
-	=	Free
S	=	Springs
F_V	=	Vertical forces. When vertical forces are applied the model must in addition be restricted from translation in the vertical direction by fixing it in one node.
F_h	=	Counteracting horizontal force

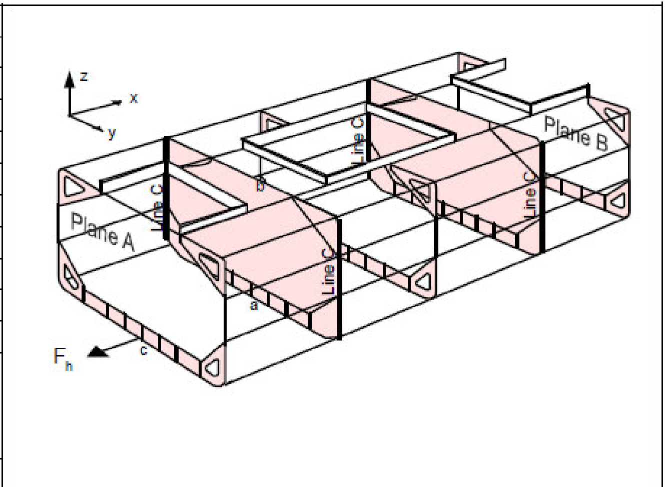


Figure 5.6 Boundary condition for three cargo hold

5.4. Analysis of Local Structures

With the set boundary condition to find out the stress at middle cargo, Ansys was run and an equivalent stress at mid cargo is found out to be 148MPa as shown in Fig 5.7. The stress at one end of the boundary condition looks unsymmetrical for the very fact that one part has bulk head closed part facing outside and on the other hand the bulkhead open part is facing out.

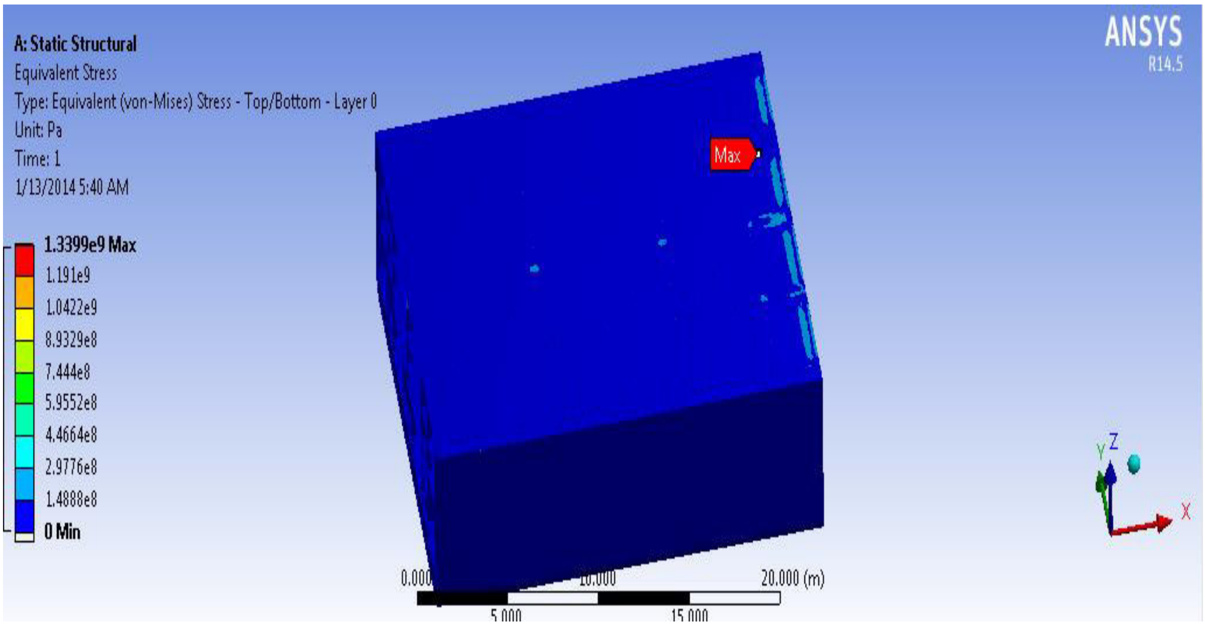


Figure 5.7 Equivalent stress distributions (still water loading)

To decide how safe the structure is, only the maximum stress level that are located in the midship cargo are used. To do so, the probe tool available in Ansys is used to track the spots with the maximum stress level. The maximum normal stress level found at mid cargo area is in fact 157 MPa. This stress value is compared with the stress value of the material used. As the material factor used is 1 in all cases, the safety factor of the stress is 235MPa/157Mpa =1.49. This justifies that the stress level is not severe.

The maximum normal stress is 43.5 MPa at closer to the center of the bulkhead as in Fig 5.8.

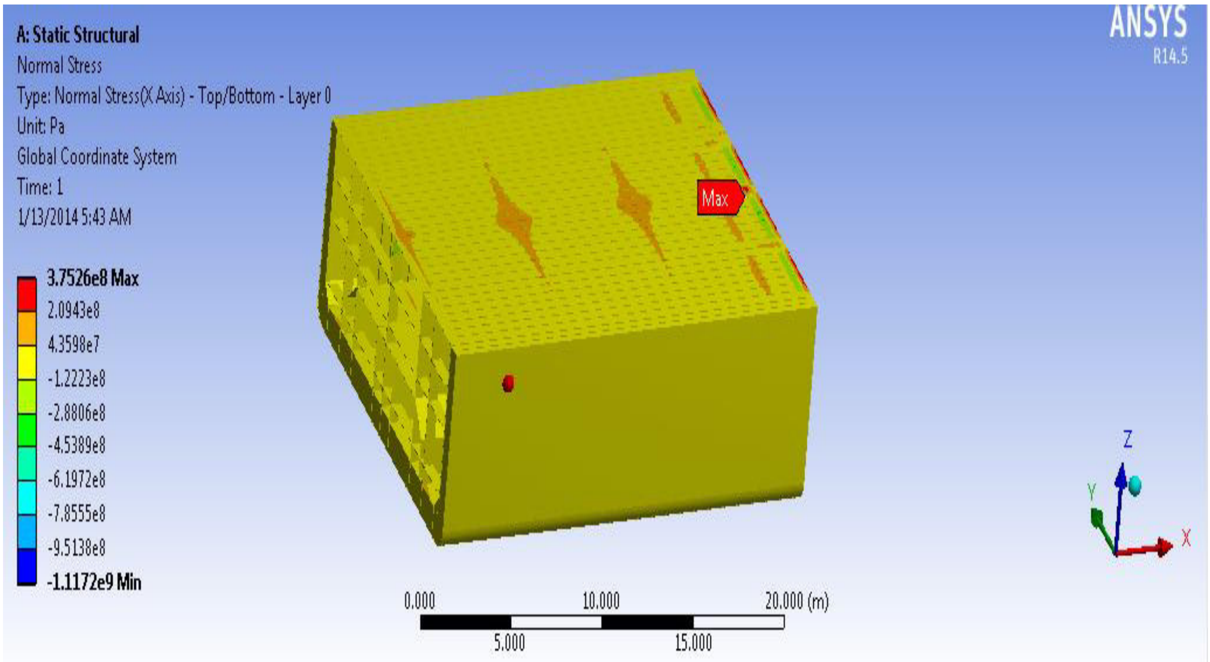


Figure 5.8 Normal stress distribution (still water loading)

The normal stress values obtained satisfies the minimum requirement and it can be said that the structure qualifies.

The maximum shear stress at the mid cargo for the loading case mentioned is144.3 MPa, presented in Fig 5.9.

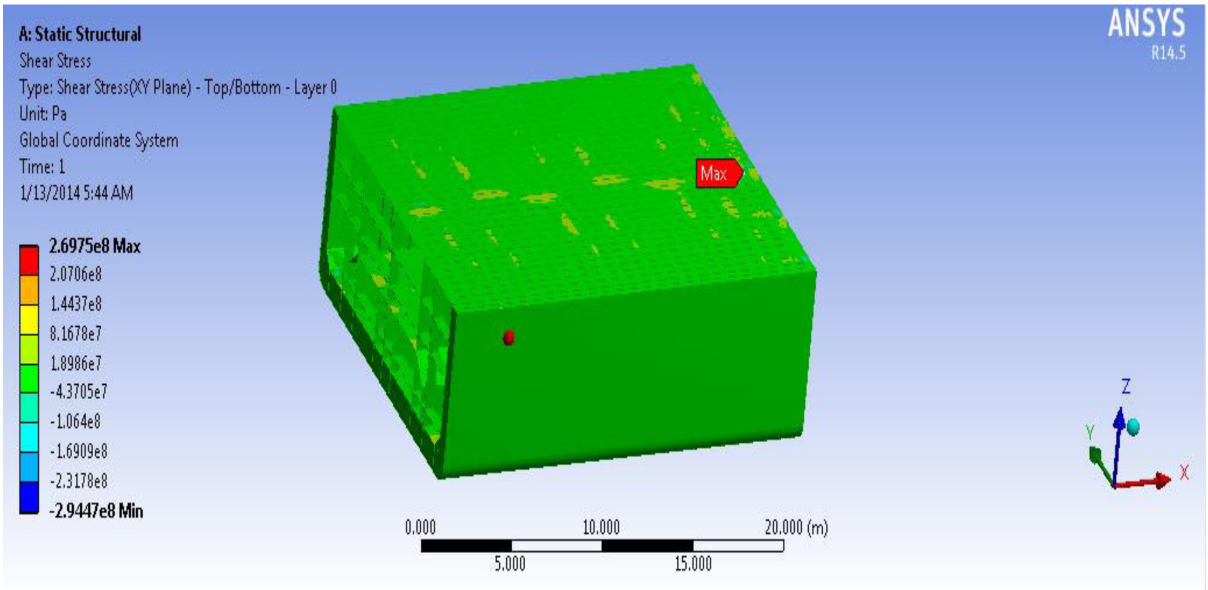


Figure 5.9 Shear stress distribution (still water loading)

From Fig 5.9, the maximum is mentioned to compare the worst stress scenario. Hence, the structure fulfils the minimum requirement.

6. RESULT AND DISCUSSION

The structural design of the platform supply vessel that carries multiple types of cargo is commenced with general arrangement and moves to concept design of the midship area. Having concluded with the concept design, the materials and topology used in the design is meticulously decided. With the existing concept design and materials a DNV, software, Nauticus hull is chosen for its ease and accessibility at the lab. The procedural steps starting from entering the vessel data in to the software to managing the section of the vessel as to where the interest lies to stiffening the section are made. An initial conceptual scantling is made with appropriate type of framing convenient to the vessel. A mixed framing system, length 2.4m and spacing 0.6m, is utilized as longitudinal being on the deck and transversal framing on other places. Furthermore, the compartments existing in the chosen section are structured and loads are assigned to them. Up on the successful completion of the compartmentation and loading, a robust tool, rule check, is used to compare the initial concept structure with the minimum requirement set by DNV rule. A long due of modification steps were necessary to arrive at the structure that qualifies well.

The final values from the scantling result at the midship section shows that, the plating thickness is maximum 12 mm at the center bottom and minimum 6mm at the engine shaft vault.

The largest size of the longitudinal stiffener is found to be *Hp* 200*12 and smaller is 180*10 with a minimum section modulus of Z =57.27 cm^3 with a maximum design bending moment of 60250 KNm.

The maximum shear force at seagoing condition according to the rules is -31465 KN in hogging and 41570 KNm in sagging.

In a nutshell:

- The hull is made of grade A steel with R_{eH} = 235 MPa.
- The height of the double bottom is 1.1 m at the midship.
- Water tight bulkheads are strengthened by vertical frameworks and horizontal stringers.
- The mass the structure in one cargo hold is 1048.54 t, 62 % is dedicated to the longitudinal plates, 13 % is dedicated to the longitudinal stiffeners and remaining percentage is dedicated to the transverse members including the structure of the transverse bulkhead

•

The values obtained above are further substantiated by direct FEM assessment. In the FEM used, the strength of a cargo at the middle of the ship is checked by modelling a three hold cargo according to IACS rule where the mid cargo is the area of interest while the remaining two cargos are used to set the boundary conditions. The equivalent stress is evaluated at the center of the shell elements used in the longitudinal and transverse plating, while axial stress is evaluated at the beam elements. The equivalent stresses are compared with the allowable stress for constituent parts and are less than the later.

The maximum stress value obtained is therefore

7. CONCLUSION

In the course of the design of platform supply vessel in this thesis, a hull structure that supports loads that the ship might encounters in its lifetime. In an attempt to attain the objective, a conceptual model of mid ship section is developed. The constituent parts of the mid ship are then checked individually against the minimum requirement dimensions and section modulus set by DNV.

The software Nauticus hull is used to verify the conformance to the requirement by iteratively changing thicknesses of members until it suffices the need. Moreover, members having adequate thickness may fail to qualify hence; section modulus check is an indispensable tool here. Another cross-section at frame 22 also is used to design the structural members as this section has a narrower size as compared with the mid ship section. . Having all iterative process by modifying the thicknesses, a final scantling that satisfies all the criteria were achieved.

The result obtained by the rules are further examined and corroborated by direct strength assessment using Ansys software on one cargo hold on the mid ship area. Both systems verified that the structural model developed in fact sustains all the loads assumed.

DECLARATION OF AUTHORSHIP

Declaration of Authorship

I declare that this thesis and the work presented in it are my own and have been generated by me as the result of my own original research.

Where I have consulted the published work of others, this is always clearly attributed.

Where I have quoted from the work of others, the source is always given. With the exception of such quotations, this thesis is entirely my own work.

I have acknowledged all main sources of help.

Where the thesis is based on work done by myself jointly with others, I have made clear exactly what was done by others and what I have contributed myself.

This thesis contains no material that has been submitted previously, in whole or in part, for the award of any other academic degree or diploma.

I cede copyright of the thesis in favour of the University of .....

Date: *Signature*

ACKNOWLEDGEMENT

First and foremost I would like to thank God for the guidance and insight in my life. I would like to convey my sincere gratitude to Mr. Jakub Wielgosz and Mr. Slawomir mak, Crist Ship yard for their support and guidance.

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I finally would like to convey my heartfelt appreciation for my family and friends for the love, care and good will that really made me keep on going on the times of difficulty and challenges.

REFERENCE

1. Alderton, P.M. (1995) Sea Transportation – Operation and Economics. Surrey, Thomas Reed Publications, UK.
2. Anthony Beis, 2001. *A Finite Element Analysis of the Nps Autonomous Underwater Vehicle (Auv) Hull Intended To Operate In Deep Waters*. Thesis (Degree) Naval Postgraduate School Monterey, California.
3. Augusto B. Hougaz Carlos A. N. Dias, 2003. FEM Parametric Modeling Applied to the Optimization of a Ship Structure, *Proceedings of the Thirteenth International Offshore and Polar Engineering Conference.* Honolulu, Hawaii, USA.
4. Daniel Tanner, 2008. *The Offshore Industry, its markets and prospects, what are the offshore vessel sector's perspectives for the future?* Diplomarbeit Grades. Hochschule Bremen.
5. Dimitris Servis*,George Voudouris, Manolis Samuelides, Apostolos Papanikolaou, 2004. *Finite element modelling and strength analysis of hold No. 1 of bulk carriers. Department of Naval Architecture and Marine Engineering*, National Technical University of Athens, Athens, Greece.
6. Robin Sebastian Koske Rose, 2011. *Future Characteristics of Offshore Support Vessels.* Thesis (Master). Massachusetts Institute of Technology.
7. Sunil Kumar P G, 2008. *Finite Element Analysis of Warship Structures*, Cochin University of Science and Technology, Kerala, India
8. V. Zanic, J. Andric, M. Stipcevic, M. Grgic, K. Piric, 2009. *Full ship FEM Analysis of Livestock Carrier*, Yard.no.486-487, Technical Reports ULJ 486-487/1÷5, University of Zagreb, Croatia
9. W D Biggs, 1960. *The Brittle Fracture of Steel*. United Kingdom: MacDonald and Evans.
10. Y.V. Satish Kumar, M. Mukhopadhyay, 2000. Finite element analysis of ship structures using a new stiffened plate element *Applied Ocean Research,* 22 (2000), 361-374
11. Det Norske Veritas. 2011. The OSV Market - Today and Tomorrow
12. Farstad Shipping 2012. *The Market for Offshore supply Vessels- Farstad Shipping ASA Annual Report.* Available from:

https://www.farstad.com/en/annualreport2012/the-market-for-offshore-supply-vessels#the-market-for-offshore-supply-vessels/the-market-in-general. [Accessed 10 August 2013]

13. http://en.wikipedia.org/wiki/Oil_platform [Accessed 17 September 2013]

14. http://officerofthewatch.com/2013/09/02/a-brief-history-of-us-offshore-oil-drilling[Accessed 17 September 2013]

15. John Ferentinos, 2013. Global Offshore Oil and Gas Outlook– Infield Systems Gas/Electric Partnership

16. Marketsandmarkets.com, 2013. *Offshore Support Vessel Market - Global Trends &Forecast to 2018.*Available from: http://www.researchandmarkets.com/reports/2588246/offshore_support_vessel_market_global_trends. [Accessed 14 September 2013]

17. Office of Ocean Exploration and Research, 2008. *Types of oil and Gas Structures*. NOAA Ocean Explorer: Expedition to the Deep Slope. National Oceanic and Atmospheric Administration. Available from: http://oceanexplorer.noaa.gov/explorations/06mexico/background/oil/oil.html [Accessed 10 August 2013]

18. Tim Reardon, 2012. *Review of the Guidelines on State aid to maritime transport, UK Chamber of Shipping*.

19. http://www.dnv.com/services/software/products/nauticus/hull/

APPENDICES

NAUTICUS hull

BY DNV

SECTION SCANTLINGS

Hull Section Scantlings according to DNV Rules for ships with L < 100 m

Rule edition .... :	Jan. 2009
Program version .... :	11.50.5860

Ship Identification
Vessel ID: Hoba

ID No .... :	Hoba
Date/Sign .... :	2013-12-07 Student

Cross Section Identification
Frame 64

Midship section? .... :	Yes
Distance from AP .... (m) :	38.400
Date/Sign .... :	

Database: C:\Documents\DNV\Nauticus\Vessels\Hoba\CrossSections\55-73.PW

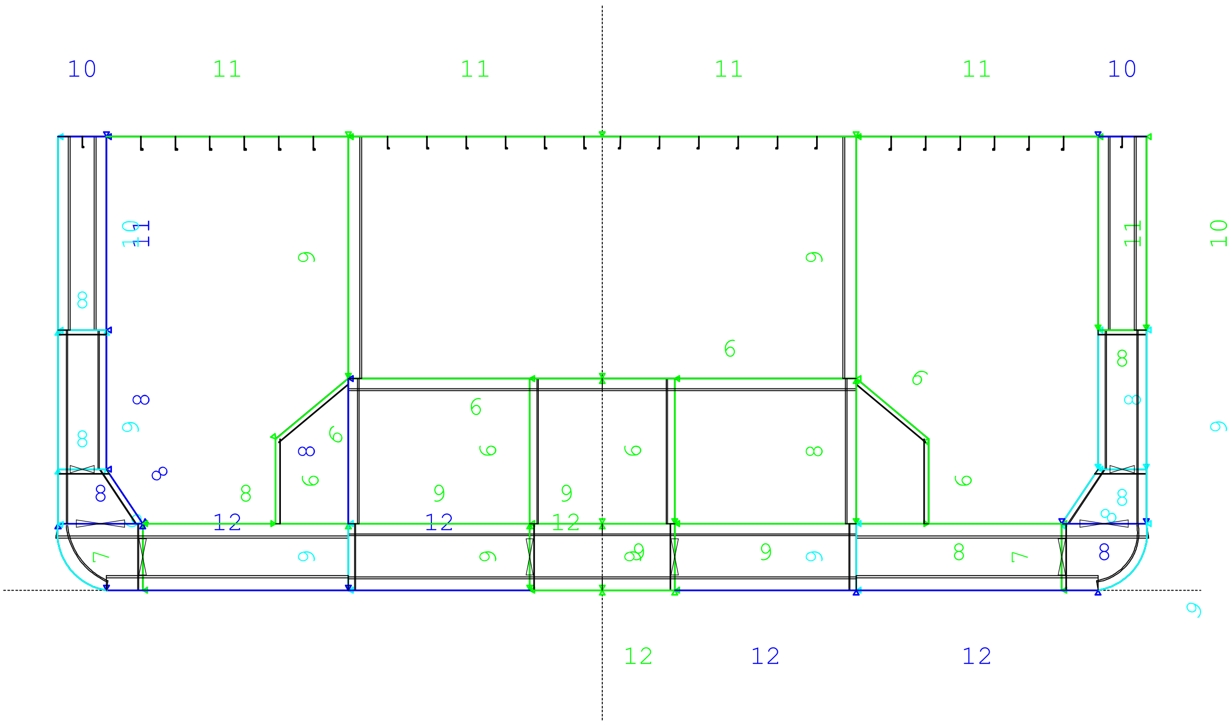
Main Dimensions

Length betw. perpendiculars, Lbp .... (m) :	76.800
Rule length, L .... (m) :	76.800
Breadth moulded, B .... (m) :	18.000
Depth moulded, D .... (m) :	7.500
Draught moulded, T .... (m) :	6.200
Block coefficient, Cb .... :	0.721
Min. design draught at AP .... (m) :	0.000
Min. design draught at FP .... (m) :	0.000
Waterplane area coefficient, Cwp	0.849

Table of Contents

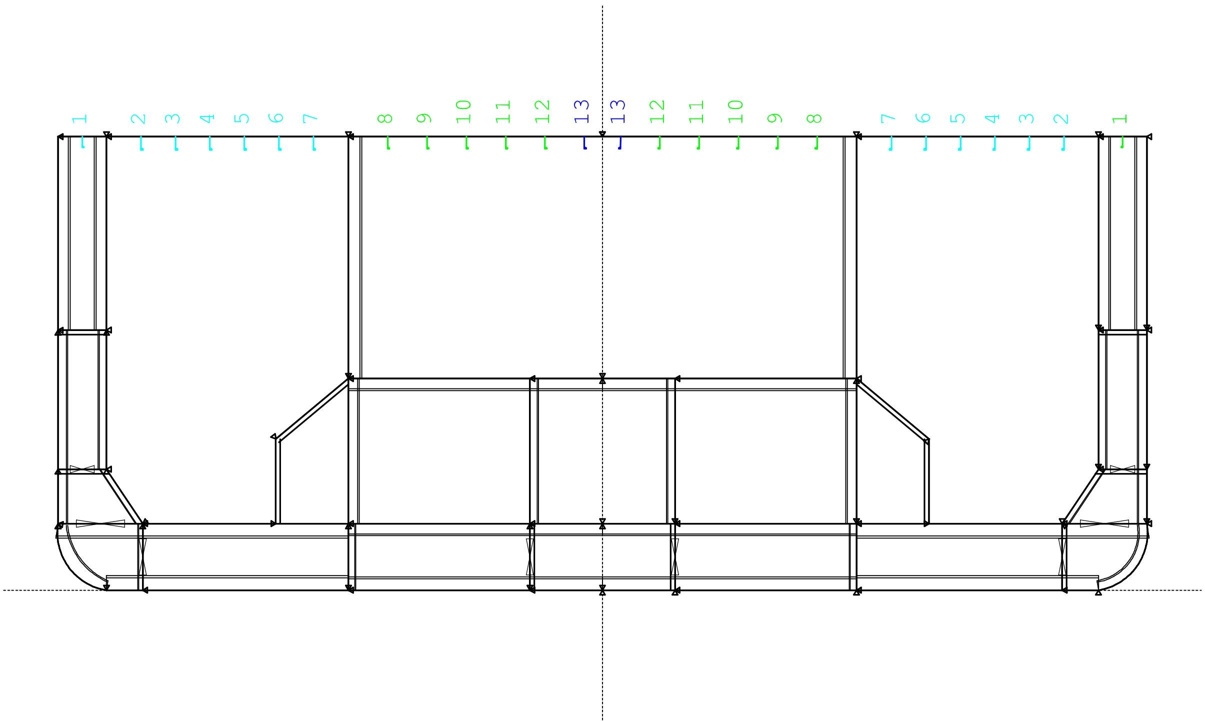
Rule status - plates
Thickness in % of requirement

Requirement = 0
< 80%
80 - 89%
90 - 99%
100 - 109%
110 - 119%
120% and above



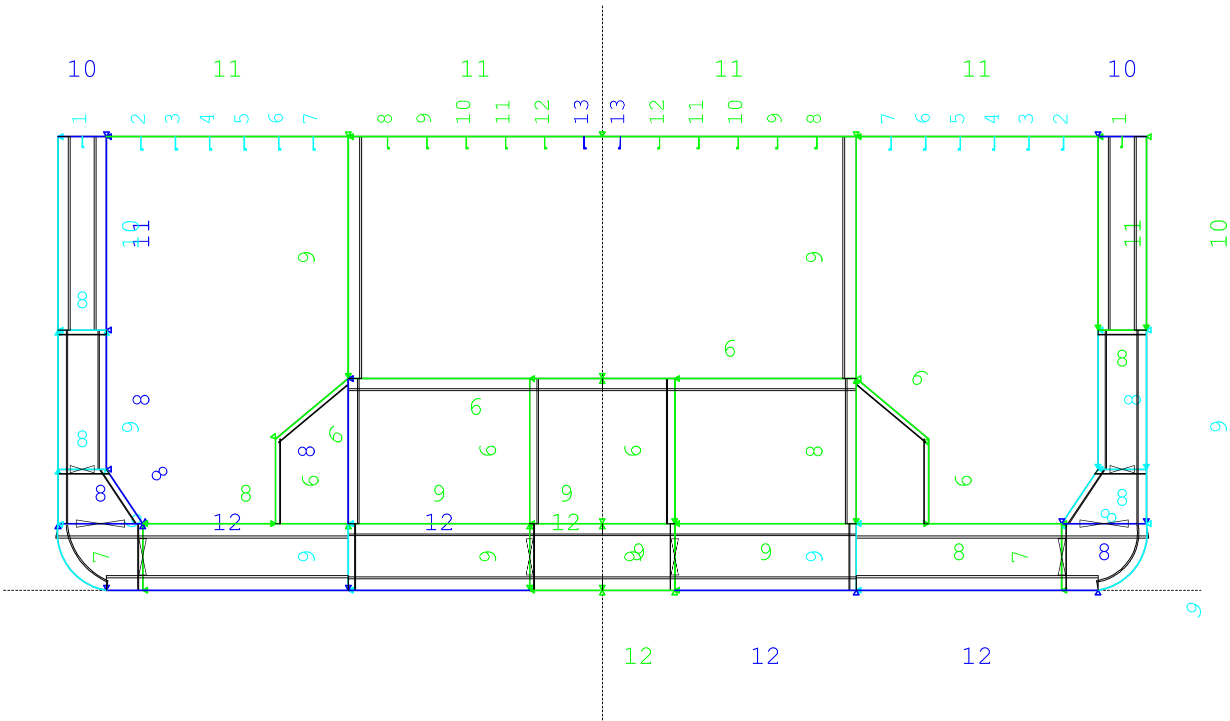
Rule status - stiffeners
% fulfilment of the Rule requirement

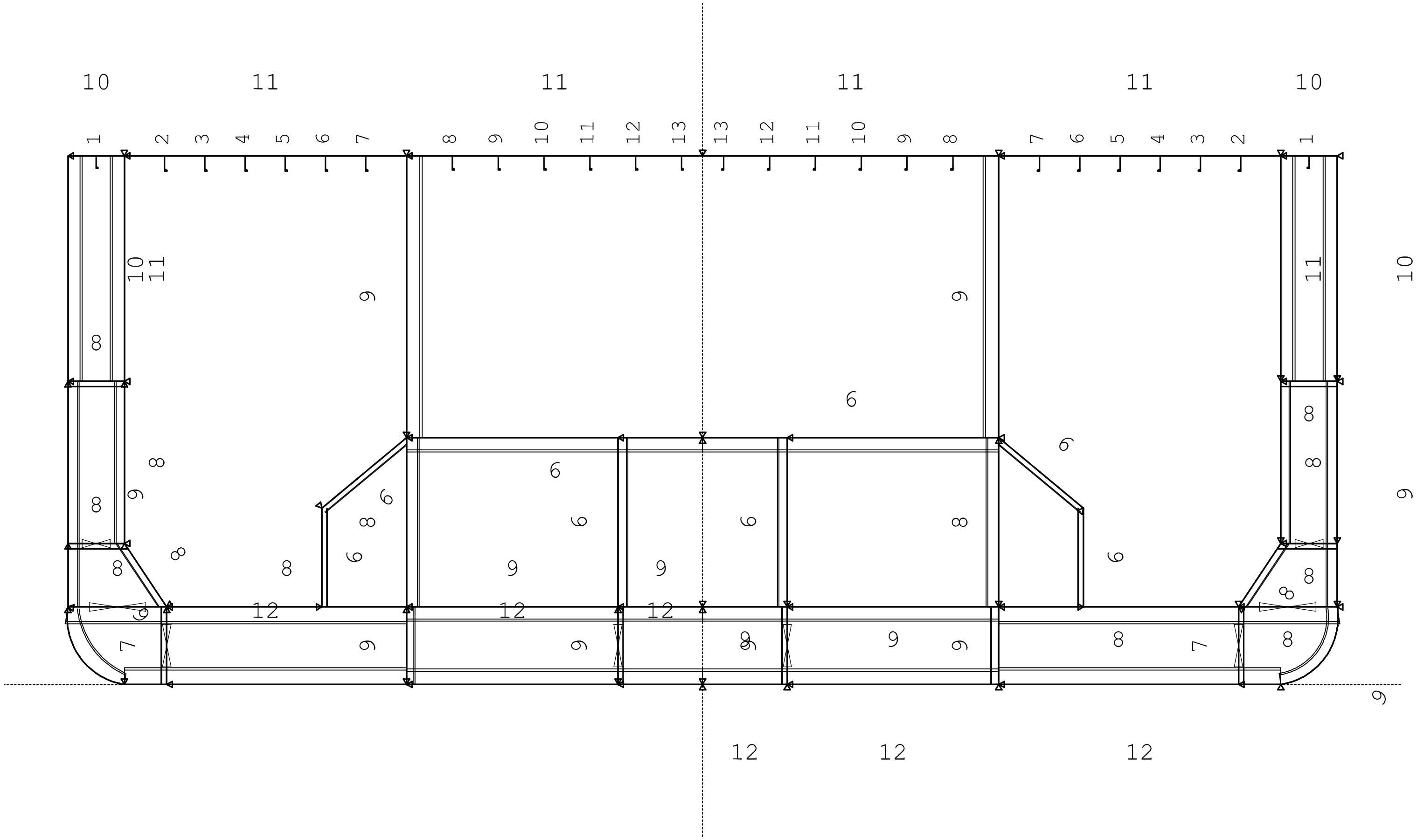
Requirement = 0
< 80%
80 - 89%
90 - 99%
100 - 109%
110 - 119%
120% and above



Rule status - plates and stiffeners
% fulfilment of the Rule requirement

- Requirement = 0
- < 80%
- 80 - 89%
- 90 - 99%
- 100 - 109%
- 110 - 119%
- 120% and above





Nos	Type	Profiles: Dimensions	Steel
Outer Shell			
1	HPbulb	180 x 10	
2 - 7	HPbulb	220 x 10	
8 - 13	HPbulb	200 x 11.5	
Outer Shell SB			
1	HPbulb	180 x 10	
2 - 7	HPbulb	220 x 10	
8 - 13	HPbulb	200 x 11.5	

Nauticus Hull Section Scantlings	Vessel ID: Hoba Date/Sign : 2013-12-07 Student Main dim. : Lpp=76.8 B=18 D=7.5 T=6.2 {m} CB=0.721	Scale: 1:100

1 Rule Reference

DET NORSKE VERITAS' Rules for Classification of Ships, January 2009.
Ships with length less than 100 metres.

STRENGTH ITEM	REFERENCE TO DNV RULES
HULL GIRDER STRENGTH	
- Design bending moments	January 2009 Pt.3 Ch.2 Sec.4 B
- Bending strength	January 2009 Pt.3 Ch.2 Sec.4 C
PLATES	
- Bottom and inner bottom	January 2009 Pt.3 Ch.2 Sec.5 C 100-402
- Side	January 2009 Pt.3 Ch.2 Sec.6 C 100-102
- Deck	January 2009 Pt.3 Ch.2 Sec.7 C 100-202
- Bulkheads	January 2009 Pt.3 Ch.2 Sec.8 C 100-104
- Wheel loaded decks	January 2009 Pt.5 Ch.2 Sec.4 C 200-302
- Container carriers	January 2009 Pt.5 Ch.2 Sec.6 B & C
- IB(+) Notation	January 2009 Pt.3 Ch.2 Sec.6 G 400-402
Buckling strength	
- Trv. stiffened bottom, inner bottom, decks	January 2009 Pt.3 Ch.2 Sec.12 B 100-103
- Otherwise	January 2009 Pt.3 Ch.1 Sec.13 B 100, 201, 205, 206
STIFFENERS	
- Bottom and inner bottom	January 2009 Pt.3 Ch.2 Sec.5 C 600-801, 803, 805
- Side	January 2009 Pt.3 Ch.2 Sec.6 C 300-405
- Main frames	January 2009 Pt.3 Ch.2 Sec.6 C 400-405
- Deck	January 2009 Pt.3 Ch.2 Sec.7 C 300-302
- Bulkheads	January 2009 Pt.3 Ch.2 Sec.8 C 200-402
- Wheel loaded decks	January 2009 Pt.5 Ch.2 Sec.4 C 200-202, 401
- Container carriers	January 2009 Pt.5 Ch.2 Sec.6 B & C
- Buckling strength	January 2009 Pt.3 Ch.1 Sec.13 C 300-402, January 2009 Pt.3 Ch.2 Sec.12 B 101, D 100-102
- Connection area at supp.	January 2009 Pt.3 Ch.2 Sec.11 C 402 (C=1.25)

2 Design Bending Moments

AT ACTUAL POSITION (38.4 m from AP)	SAGGING (kNm)	HOGGING (kNm)
Still water bending moments:		
- Standard values according to Rules, Ms .... :	60250	60250
- Given as input (actual cargo/ballast conditions) .... :	60250	60250
Design still water bending moments, Ms	60250	60250
Design wave bending moments, Mw .... :	100941 (Input)	88465 (Input)
Design wave bending moments, Mw for buckling check .... :	100941 (Input)	88465 (Input)

NOTE: The wave bending moments are given as input.

3 Hull Girder Strength Summary

	ACTUAL	RULE	STATUS (%) (100=Rule)
Cross-sectional area .... (cm2) :	11599		
Height to the neutral axis .... (mm) :	3599		
Moment of inertia .... (m4) :	9.683		
Section modulus, bottom .... (m3) :	2.691	1.115	241.4
Section modulus, deck line (z = 7500 mm).................................. (m3) :	2.482	1.115	222.6

4 Rule status - Plates

Position	Plate No	Thickness (mm) Rule	Thickness (mm) Actual	Loc %	Buc %	Final %	Com-ment
Bottom	1	11.84	12.0	100	141	100	
Bottom	2	9.94	12.0	133	120	120	
Bottom	3	9.02	12.0	150	133	133	
Bottom	4	8.07	9.0	113	0	113	
Side	5	8.07	9.0	113	164	113	
Side	6	9.51	10.0	111	105	105	
Strength deck	7	8.04	10.0	125	222	125	
Strength deck	8	10.45	11.0	105	169	105	
Strength deck	9	10.59	11.0	105	183	105	
Stringer 1700	1	7.20	8.0	114	200	114	
Long. Bulkhead 4200	1	8.04	9.0	120	113	113	
Long. Bulkhead 4200	2	7.36	8.0	107	145	107	
Long. Bulkhead 4200	3	8.58	9.0	150	106	106	
'tween deck 3500	1	5.88	6.0	100	400	100	
'tween deck 3500	2	5.50	6.0	109	171	109	
'tween deck 3500	3	5.95	6.0	109	100	100	
Long. Bulkhead 1200	1	8.54	9.0	120	106	106	
Long. Bulkhead 1200	2	5.77	6.0	100	109	100	
Stringer 4300	1	8.20	8.0	100	229	100	
'tween deck 1100	1	6.55	8.0	145	123	123	
'tween deck 1100	2	8.02	8.0	100	100	100	
'tween deck 1100	3	8.45	9.0	106	106	106	
'tween deck 1100	4	8.30	9.0	106	129	106	
Long. Bulkhead 7500	1	7.04	7.0	117	100	100	
Inner Bottom & Inner Side	1	6.83	8.0	114	145	114	
Inner Bottom & Inner Side	2	6.85	8.0	114	178	114	
Inner Bottom & Inner Side	3	10.57	11.0	157	105	105	
Bottom SB	1	11.84	12.0	100	141	100	
Bottom SB	2	9.94	12.0	133	120	120	
Bottom SB	3	9.02	12.0	150	133	133	
Bottom SB	4	8.07	9.0	113	0	113	
Side SB	5	8.07	9.0	113	164	113	
Side SB	6	8.51	10.0	125	118	118	
Strength deck SB	7	7.04	10.0	143	286	143	
Strength deck SB	8	10.45	11.0	105	169	105	
Strength deck SB	9	10.59	11.0	105	183	105	
Stringer 1700 SB	1	7.20	8.0	114	200	114	
Long. Bulkhead 4200 SB	1	8.04	9.0	120	113	113	
Long. Bulkhead 4200 SB	2	5.77	8.0	133	145	133	
Long. Bulkhead 4200 SB	3	8.58	9.0	150	106	106	
'tween deck 3500 SB	1	5.88	6.0	100	400	100	
'tween deck 3500 SB	2	5.50	6.0	109	171	109	
'tween deck 3500 SB	3	5.95	6.0	109	100	100	
Long. Bulkhead 1200 SB	1	8.54	9.0	120	106	106	
Long. Bulkhead 1200 SB	2	5.77	6.0	100	109	100	
Stringer 4300 SB	1	7.20	8.0	114	320	114	
'tween deck 1100 SB	1	6.55	8.0	145	123	123	
'tween deck 1100 SB	2	8.02	8.0	100	100	100	
'tween deck 1100 SB	3	8.45	9.0	106	106	106	
'tween deck 1100 SB	4	8.30	9.0	106	129	106	
Long. Bulkhead 7500 SB	1	7.04	7.0	117	100	100	
Inner Bottom & Inner Side SB	1	5.77	8.0	133	145	133	
Inner Bottom & Inner Side SB	2	5.77	8.0	133	178	133	
Inner Bottom & Inner Side SB	3	9.07	11.0	183	122	122	

RULE REQUIREMENTS
"Rule" = The Rule thickness, i.e. the highest of the requirement due to lateral pressure, the minimum thickness and the buckling strength requirement.
"Loc %" = the percentwise fulfilment of the local Rule requirement.
"Buc %" = the percentwise fulfilment of the buckling strength requirement.
"Final %" = the percentwise fulfilment of the decisive requirement, i.e. actual thickness/Rule thickness * 100.

Note that in the % columns, the Rule thickness has been rounded to the nearest half mm.

If the requirements are not fulfilled, an asterisk (*) will be printed in the Comment column.

5 Rule status - Stiffeners

Position	Stiff. ID	Prof. type	Dimensions (mm)	Sect. mod. (cm3)		Decisive req.	Loc %	Buc %	Final %	Com-ment
				Rule	Actual					
Strength deck	1	20	180 x 10	166	172	Zr	103	290	103	
Strength deck	2	20	220 x 10	243	288	tminw	119	294	118	
Strength deck	3	20	220 x 10	243	288	tminw	119	294	118	
Strength deck	4	20	220 x 10	243	288	tminw	119	294	118	
Strength deck	5	20	220 x 10	243	288	tminw	119	294	118	
Strength deck	6	20	220 x 10	243	288	tminw	119	294	118	
Strength deck	7	20	220 x 10	245	289	Zr	118	294	118	
Strength deck	8	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck	9	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck	10	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck	11	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck	12	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck	13	20	200 x 11.5	181	243	Zr	134	292	134	
Strength deck SB	1	20	180 x 10	152	172	Zr	113	290	113	
Strength deck SB	2	20	220 x 10	243	288	tminw	119	294	118	
Strength deck SB	3	20	220 x 10	243	288	tminw	119	294	118	
Strength deck SB	4	20	220 x 10	243	288	tminw	119	294	118	
Strength deck SB	5	20	220 x 10	243	288	tminw	119	294	118	
Strength deck SB	6	20	220 x 10	243	288	tminw	119	294	118	
Strength deck SB	7	20	220 x 10	245	289	Zr	118	294	118	
Strength deck SB	8	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck SB	9	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck SB	10	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck SB	11	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck SB	12	20	200 x 11.5	248	248	Zr	100	291	100	
Strength deck SB	13	20	200 x 11.5	181	243	Zr	134	292	134	

DECISIVE REQUIREMENT
The "Decisive req." column indicates the decisive Rule requirement for the stiffener, as follows:
"Zr" = Section modulus (Final % = Actual/Rule * 100)
This requirement also includes the fatigue strength requirement concerning stiffeners in the ship side acc. to DNV Rules Pt.3 Ch.1 Sec.7 E 500.
"lat" = Lateral buckling strength (Final % = usage factor/max usage factor * 100)
"tor" = Torsional buckling strength (Final % = usage factor/max usage factor * 100)
"web" = Web buckling strength (Final % = usage factor/max usage factor * 100)
"bf/tf" = Maximum width/thickness ratio for the flange (% = actual ratio/max ratio * 100)
"tmin" = Minimum thickness (Final % = web thickness/min thickness * 100)

"Loc %" = the percentwise fulfilment of the local Rule requirement (Actual/Rule * 100).
"Buc %" = the percentwise fulfilment of the buckling strength requirement (usage factor/max usage factor * 100).
"Final %" = the percentwise fulfilment of the decisive requirement (lowest value of "Loc %" and "Buc %").

If the requirements are not fulfilled, an asterisk (*) will be printed in the Comments column.

5.1 Profile type overview

Type	Name
20	HPBulb

6 Moments and Shear Forces used in the PULS buckling check

Distance from AP to considered section	(m) :	38.400	
Section modulus, bottom	(m3) :	2.691	
Section modulus, deck line (z = 7500 mm)	(m3) :	2.482	
Height from base line to the neutral axis	(mm) :	3599	
BENDING MOMENTS:			
Still water bending moment, sagging	(kNm) :	60250	(Input)
Still water bending moment, hogging	(kNm) :	60250	(Input)
Wave bending moment, sagging	(kNm) :	100941	(Input)
Wave bending moment, hogging	(kNm) :	88465	(Input)
SHEAR FORCES:			
Still water shear force, sagging	(kN) :	0	(Rules)
Still water shear force, hogging	(kN) :	0	(Rules)
Wave shear force, sagging	(kN) :	0	(Rules)
Wave shear force, hogging	(kN) :	0	(Rules)

NOTE: The wave bending moments are given as input.

7 PULS 2.010 buckling check

Stiff. No	ACT1 ACT2	Elem. type Long. span mm	Type Type	h t_w mm	b_f t_f mm	s t_p mm	σ_{fp} σ_{fs} N/mm²	N_{stiff} Bf/h act	t_{kw} h/t_w act mm	t_{kf} b_{ff}/t_f act mm	t_{kp} s/t_p act mm	y z mm
	Applied Stresses		p kN/m²	$\sigma_{x,1}$ N/mm²	$\sigma_{x,2}$ N/mm²	σ_y N/mm²	τ N/mm²	Bf/h req.	h/t_w req.	b_{ff}/t_f req.	s/t_p req.	OK?
	Ultimate Capacity			$\sigma_{x,1UC}$ N/mm²	$\sigma_{x,2UC}$ N/mm²	$\sigma_{y,UC}$ N/mm²	τ_{UC} N/mm²		Crit.limit state	η_{UC}	η_{max}	OK?
	Buckling Strength			$\sigma_{x,1BS}$ N/mm²	$\sigma_{x,2BS}$ N/mm²	$\sigma_{y,BS}$ N/mm²	τ_{BS} N/mm²			η_{BS}	Elastic buckling	OK?

Outer Shell

Strength deck

1	ACT1	S3	20	180	35.0	400	235	1	1.5	1.5	1.0	8600
	ACT2	2400	HPbulb	10.0	17.8	10.0	235		19.3	1.6	44.4	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			209	209	0	0		4	0.31	0.90	Yes
	Buckling Strength			209	209	0	0			0.31	No	
2	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	7630
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
3	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	7060
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
4	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	6490
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
5	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	5920
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
6	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	5350
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
7	ACT1	S3	20	220	41.0	575	235	6	2.0	2.0	1.0	4780
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.5	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			194	194	0	0		4	0.34	0.90	Yes
	Buckling Strength			194	194	0	0			0.34	No	
8	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	3550
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
9	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	2900
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
10	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	2250
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
11	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	1600
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	

PULS 2.010 buckling check (cont.)

Stiff. No	ACT1 ACT2	Elem. type Long. span mm	Type Type	h t_w mm	b_f t_f mm	s t_p mm	σ_{fp} σ_{fs} N/mm²	N_{stiff} Bf/h act	t_{kw} h/t_w act mm	t_{kf} b_{ff}/t_f act mm	t_{kp} s/t_p act mm	y z mm
	Applied Stresses		p kN/m²	$\sigma_{x,1}$ N/mm²	$\sigma_{x,2}$ N/mm²	σ_y N/mm²	τ N/mm²	Bf/h req.	h/t_w req.	b_{ff}/t_f req.	s/t_p req.	OK?
	Ultimate Capacity			$\sigma_{x,1UC}$ N/mm²	$\sigma_{x,2UC}$ N/mm²	$\sigma_{y,UC}$ N/mm²	τ_{UC} N/mm²		Crit.limit state	η_{UC}	η_{max}	OK?
	Buckling Strength			$\sigma_{x,1BS}$ N/mm²	$\sigma_{x,2BS}$ N/mm²	$\sigma_{y,BS}$ N/mm²	τ_{BS} N/mm²			η_{BS}	Elastic buckling	OK?
12	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	950
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
13	ACT1	S3	20	200	39.5	475	235	6	0.0	0.0	0.0	300
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	43.2	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			208	208	0	0		4	0.31	0.90	Yes
	Buckling Strength			208	208	0	0			0.31	No	

Outer Shell SB

Strength deck SB

Stiff. No												
1	ACT1	S3	20	180	35.0	400	235	1	0.0	0.0	0.0	-8600
	ACT2	2400	HPbulb	10.0	17.8	10.0	235		16.2	1.4	40.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			212	212	0	0		4	0.31	0.90	Yes
	Buckling Strength			212	212	0	0			0.31	No	
2	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	-7630
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
3	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	-7060
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
4	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	-6490
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
5	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	-5920
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
6	ACT1	S3	20	220	41.0	570	235	6	2.0	2.0	1.0	-5350
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.0	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			195	195	0	0		4	0.33	0.90	Yes
	Buckling Strength			195	195	0	0			0.33	No	
7	ACT1	S3	20	220	41.0	575	235	6	2.0	2.0	1.0	-4780
	ACT2	2400	HPbulb	10.0	22.6	11.0	235		24.9	1.6	57.5	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			194	194	0	0		4	0.34	0.90	Yes
	Buckling Strength			194	194	0	0			0.34	No	
8	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	-3550
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
9	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	-2900
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	

PULS 2.010 buckling check (cont.)

Stiff. No	ACT1 ACT2	Elem. type Long. span mm	Type Type	h t_w mm	b_f t_f mm	s t_p mm	σ_{fp} σ_{fs} N/mm^2	N_{stiff} Bf/h act	t_{kw} h/t_w act mm	t_{kf} b_{ff}/t_f act mm	t_{kp} s/t_p act mm	y z mm
	Applied Stresses		p kN/m^2	$\sigma_{x,1}$ N/mm^2	$\sigma_{x,2}$ N/mm^2	σ_y N/mm^2	τ N/mm^2	Bf/h req.	h/t_w req.	b_{ff}/t_f req.	s/t_p req.	OK?
	Ultimate Capacity			$\sigma_{x,1UC}$ N/mm^2	$\sigma_{x,2UC}$ N/mm^2	$\sigma_{y,UC}$ N/mm^2	τ_{UC} N/mm^2		Crit.limit state	η_{UC}	η_{max}	OK?
	Buckling Strength			$\sigma_{x,1BS}$ N/mm^2	$\sigma_{x,2BS}$ N/mm^2	$\sigma_{y,BS}$ N/mm^2	τ_{BS} N/mm^2			η_{BS}	Elastic buckling	OK?
10	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	-2250
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
11	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	-1600
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
12	ACT1	S3	20	200	39.5	650	235	6	0.0	0.0	0.0	-950
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	59.1	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			193	193	0	0		4	0.34	0.90	Yes
	Buckling Strength			193	193	0	0			0.34	No	
13	ACT1	S3	20	200	39.5	475	235	6	0.0	0.0	0.0	-300
	ACT2	2400	HPbulb	11.5	20.2	11.0	235		15.6	1.4	43.2	7500
	Applied Stresses		0.0	65	65	-0	0		90.0	15.0	200.0	Yes
	Ultimate Capacity			208	208	0	0		4	0.31	0.90	Yes
	Buckling Strength			208	208	0	0			0.31	No	