Bachelor's project

NTNU
Norwegian University of Science and Technology

Ola Furustøl
Simen Småriset
Sondre A. Tornli

Preliminary Design of Long-Liner

Bachelor's project in Skipsdesign
Supervisor: Håvard Vollset Lien
May 2020

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Kandidat nr.:	
Emnekode/emnenavn:	
Dato:	

Til Sensor

Dysleksi/lese- og skrivevansker

Vi gjør oppmerksom på at eksamenskandidaten har dokumentert dysleksi/lese- og skrivevansker. Vi ber derfor om at det ikke legges vekt på rettskriving ved sensurering av besvarelsen.

Med vennlig hilsen
Avdeling for studieadministrasjon
NTNU

Til sensor

Vedlegg til eksamensbesvarelse fra studenter med dysleksi

Trondheim, oktober 2017

Studenter med dysleksi

Studenter med dysleksi har utfordringer med lesing og skriving. Noen har større vansker med skriving og staving enn med lesing og noen har størst vansker med lesing. De fleste har vansker med både lesing og skriving/staving.

Studenter med dysleksi har normalt gode generelle evner, men de har vansker med å bearbeide språklydene, slik som å huske hvilke lyder som tilhører bokstavtegnene, lytte ut lyder i ord, trekke lyder sammen til ord, stave ord rett. Dette til tross for at de har hatt tilstrekkelig opplæring. Det er den tekniske siden av leseprosessen de har vansker med, dvs selve avkodingen. De er også ofte usikre på staving, tegnsetting og setningskonstruksjon.

Karakteristiske trekk er lavt lesetempo og feillesing av ord. Dette fører til vansker med å komme gjennom hele pensum, samt fare for misforståelser av eksamensoppgaver. Den skriftlige produksjonen kan være preget av mange rettskrivingsfeil, og ofte også vansker med tegnsetting og setningsformulering. De bruker så mye energi på å prøve å skrive riktigst mulig og de unngår ord de er usikre på, slik at det går ut over evnen til å formidle den kunnskapen de har. Lese- og skriveprosessen er ikke automatisert i tilstrekkelig grad. De bruker derfor lengre tid på å få vist hva de kan.

Forskning og erfaring har vist at med nødvendig tilrettelegging har studenter med dysleksi de samme mulighetene som andre studenter til å få vist sin kompetanse. Tilretteleggingen vil være individuelt tilpasset.

Disse vanskene øker under press så til tross for kompenserende tiltak vil ofte sluttproduktet være preget av lese-skrivevanskene.

Trine Lise Dahl, logoped MNLL, M.A., Cand.ed.

Kilder:
- *Dysleksiprosjektet* (1996), Institutt for spesialpedagogikk, UiO, v/ Helge Strømsø, prosjektleder
- *Dysleksi Norge* (18.08.2016):
http://www.dysleksiforbundet.no/no/rettigheter+rad/dysleksi/Hva+kjennetegner+dysleksi%3F.b7C_wtfM07.ips
- *Statped* (18.08.2016):
http://www.statped.no/fagomrader-og-laringsressurser/sprak-og-tale/dysleksi/dysleksi-og-tilrettelegging-ved-eksamen/

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Nybygg linebåt for Nyliner AS

Nybygget skal erstatte eksisterende fartøy, og skal fiske min. 3500 tonn rundvekt pr. år. Hovedsakelig torsk, hyse, sei, blåkveite, lange og brosme.

Operasjonsområde, sertifikater
Båten skal normalt operere i Nordsjøen, Norskehavet og Barentshavet, men skal innimellom til Newfoundland og Grønland.
Den skal sertifiseres som fiskebåt med norsk flagg, og klasses i DNV-GL.

Hoveddimensjoner

Lengde over alt:	maks 63 m
Bredde:	maks 14 m
Seilingshøyde:	maks 25 m

Tank kapasiteter

Fuel (MDO)	Min. 350 m^3
Ferskvann	Min. 100 m^3 (produksjon på min. 12 t/d)
Urea	Min. 40 m^3

Fremdrift
Skal være diesel mekanisk, men kan være Hybrid?

Propelldiameter	Min. 3 m
Power på propell	Min. 2300 kW
Hjelpemotor	Min. 800 kWe

Vil ha azimuth i forskipet for manøvrering og take-me-home funksjon.
Må ha fokus på fuel økonomi. Ønsker ikke high speed hjelpemotorer. Maks 800 rpm på hovedmotor.
Må ha NOx rensing.

Fabrikk/Last
Ren H/G fabrikk

Innfrysing	Min. 45 tonn/døgn
Kapp/sløy/utblødning	Min. 10 t/time
Lasteromskapasitet	Min. 450 tonn på palle, kanskje mer hvis restråstoff skal

fryses (Pallestørrelse: L=1.07, B=1.07, H=1.1m. 3 paller i høyden)

Ønsker utredning på hva man kan gjøre med restråstoff (hoder og innmat er ca. 40% av rundvekt).

Fiskeredskap

Autoline-anlegg	Min. 70.000 krok (EZ 4000 og 12mm line)

Moonpool
Ønsker utredning på hvordan man kan få til ett snurrevadarrangement om bord (med vakuum pumping).

Innredning
20 enkeltmannslugarer + hospital
Egen kino/spillerom og messe
Bysse skal ha tilhørende rikelig proviantrom (minst 1 fryse og 1 kjøle + tørrprov.).
Skal ha skylleri (egen oppvask i separat avlukke utenfor bysse).

Generelt
Ønsker at det fokuseres på utseende. Skal være en fin båt!
Vil ha så lite vind-areal som mulig. Fokus på sjøegenskaper! Må være god i motsjø!!

Hovedprosjekt

for

Sondre Aarøe Tornli, Ola Furustøl og Simen Småriset

Skipsdesign

Vårsemester 2020

Tittel:

Prosjektering av linebåt

Linefiske er en av de mest skånsomme kommersielle fiskemetodene som brukes idag, og linefisket oppnår også noe bedre kvalitet på fisken enn for eksempel tråling og fiske med snurpenot. Linebåtene har utviklet seg mye de siste 20 årene, og har også økt i størrelse. Basert på en kravspesifikasjon fra en lokal reder skal kandidatene designe og prosjektere en ny linebåt. Det er ønskelig med en båt som er så effektiv som mulig. Bærekraft og miljø står høyt på listen over det som er viktig for rederen, og bør være med i vurderingen i alle deler av oppgaven.

Kandidatene skal i denne oppgaven gjennomføre følgende:

1. Utforske systemene og komponentene i en moderne linebåt.
2. Bygge opp en eller flere GA på konseptnivå, og etablere et vektsestimat
3. Koble sammen systemene og se på samhandling mellom dem.
4. Utforme skroglinjene og utføre beregninger på fremdrift og stabilitet.
5. Revaluere GA og vektsestimat (designspiralen)
6. Modelltesting.
7. Dimensjonere skroget iht. gjeldende regelverk.
8. Vurdering av effektiviseringspotensial (designoptimalisering)
9. Velge endelige løsninger og utforme endelig GA og klassetegninger.
10. Priskalkulasjoner på drift og konstruksjon.

Veileder ved NTNU i Ålesund er Håvard Vollset Lien, og kontaktperson / faglig veileder ved Skipsteknisk AS er Kjetil Nyvoll.

Besvarelsen redigeres som en teknisk rapport, med et sammendrag både på norsk og engelsk, konklusjon, litteraturliste, innholdsfortegnelse etc. Ved utarbeidelsen av teksten skal kandidaten legge vekt på å gjøre teksten oversiktlig og velskrevet. Med henblikk på lesning av besvarelsen er det viktig at de nødvendige henvisninger for korresponderende steder i tekst, tabeller og figurer anføres på begge steder. Ved bedømmelsen legges det stor vekt på at resultatene er grundig bearbeidet, at de oppstilles tabellarisk og/eller grafisk på en oversiktlig måte og diskuteres utførlig.

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Introduction

In this paper we go through our process for making a preliminary design of a long-line fishing vessel. Kjetil Nyvold from Skipsteknisk presented the requirements for the ship and acts as the ship-owner, which means we present our ideas throughout the process. The paper is separated into three main parts; The Exploration, The Design Spiral and The Prestige. In part I, The Exploration, we research earlier vessels on how they typically operate, which systems are included and explore how they may come together or other ways to solve the tasks. For part II, The Design Spiral, the vessel starts to take shape with setting up a general arrangement, lines, weight and structure. Which continues in a spiral until part III, The Prestige, where we make final adjustment to the preliminary design we could present to the owner. Then ending with our evaluations on the final design and our process.

We base our methodology on the design approaches thought to us in ship design III by Gaspar with an inclination towards the Bottom-Up approach. Starting by researching how a long-liner operates to better understand what the shipowner expects, in addition to researching the different systems that will satisfy the requirements. The bottom-up approach allows for new thinking and direction. You are able to explore the design process with a more hands-on feel to it, starting from the essential fishing-systems and then adding the ship systems in order to end up with a functioning ship. Although, we will try not to lean too much in the bottom-up direction, since a top-down approach would be faster in execution and more accurate in predictions and estimations. Essentially, the Bottom-Up approach works best when the main mission is a new kind, where there is not a lot of already existing vessels. We have some vessels we can research and we will use it in predicting the physicality's of the systems, or "blocks" that will form the ship around.

The design is mapping between the forms and functions, hence the different systems will be explored in regards to their specific functions and how they are "usually" formed (Gaspar, 2017). The functions are based on the tasks leading to the main mission of the vessel and are principal defined from that main mission. However, a more rapid approach is to use the typical long-liner's systems to define the corresponding functions. The corresponding forms is the physical aspect that will be manufactured, and we will use earlier designs to estimate this for the individual systems, with small alterations to match the specific functions for our Long-Liner. After this we can start forming solutions on how they might fit together in a general arrangement by using a brainstorming process. Where the goal is to inspire lots of ideas that can be evaluated with guidelines focused on certain aspects of the ships systems, behaviour, and personnel. Moreover, the appropriate ideas will be implemented in the first draft of the GA. In part II of this paper we will show how we proceeded with shaping the vessel through the design spiral based on the different ideas we came up with. The spiral will include aspects like GA, weight estimation and structural design, but the order will be adapted to our specific vessel.

CAD

Computer Aided Design is the use of software to create digital models or drawings. CAD tools are vital for 'one of a kind constructions' like ships, where prototyping is not possible, modelling is used for more accurate physical testing. CAD enables the usage of CAE.

Computer Aided Design tools helps a designer to visualise and communicate ideas both in 2D and 3D and Computer Aided Engineering tools can calculate how the design will behave in a specific environment.

We have taken advantages of several of these tools, both in the design process and engineering process. It reduces the need to create something physical or the need to rely on verbal communication to express an idea. 3D models can also be brought into CAE software or used to create accurate physical models through additive or subtractive manufacturing processes.

Autocad

Autocad is a drawing software we use for creating General Arrangements and class drawing in 2D. In autocad one can apply much information in one drawing as we can draw the ship one to one as well as including details to millimetres prescriptions; the reader only has to zoom in and out on the details they want to review. We can draw repeating elements and copy paste it wherever we need it, saving much time compared to drawing it physically.

Siemens NX

A 3D CAD tool we have applied to our bottom up system-based design procedure. It's hard to describe and imagine a 3D object or arrangement, with NX one can create several small 3D components, and arrange them against each other in a 3D space. This helps us to visualize design and ideas, and maybe open our eyes for solutions that would without this tool be uncomprehensive to us. We used NX to arrange and understand the space in the machinery and see the limitations of the storage room.

We also applied NX in the design of our Moonpool Test models, as the model can be sent to a 3D printer or a CNC machine. The components we designed could also be entered in NX's CAE tools.

CAE

Physical testing of a ships dynamic characterisations requires big models up to 5-10m, and if change of design is needed the test might have to be done again with a new model. This is a time consuming and costly process, but there are ways to predict the ships behaviour before physical testing. Computer Assisted Engineering are the usage of analytic software for checking a component's physical capabilities. Such software varies form FEA (final element analysis) for strength, CFD (computational fluid dynamics) for fluid resistance and MDB (multibody dynamics) for physical translations between parts. With CAE one can use CAD file to simulate a component's physical characteristics. Rapid change of the component can be done in the CAD file, so it is not needed to wait for a prototype to be created. In this paper *Maxsurf, Nauticus hull's section scantling* and *3D-beam* were primarily used

Part I: The Exploration

"The Process of design starts with exploration, but ends with refinement. The best designers carefully move from one to the other, making sure they spend enough time exploring before locking themselves into a design approach

- *Jared Spool*

Jared Spool might not have thought about ship-design specifically when he made this quote, yet it encapsulates a lot of what is in the designing of ships. As designing anything is a constant battle of the knowledge you put in against the solutions you get out on time. And with our limited knowledge about the vessel in question, we instantly knew that we would have to spend time research existing long-line vessels before we can start to design our own vessel. We gather an understanding of the problems and possibilities of the individual systems by exploring how the vessel typically operates, which systems are included and how those systems perform the tasks. Where the forms are compared to the functions, so that we can estimate better how the functions defined by the requirements can take physical form. Which leads into creating new ideas of our own with a brainstorming session, before establishing guidelines from the meetings with the owner as well as the written requirements. And before we move onto *The Design Spiral,* we drag out a few ideas that will be the foundation for our general arrangement draft.

Long-liner Operation Profile

The operation starts ashore, were a long-liner loads up on fuel, equipment, supplies and crew before heading out to the fishing ground. The vessel needs to be prepared for an independent operation lasting at least a month out at sea and are expected to do ten to eleven trips each year per the owner's request. The crew consist of seven fishers, three factory workers, one chief cook, one chief engineer, a Captain, skipper and potentially 6 scientists, resulting in a maximum of 20 crew members. The supplies include the food, water and health care for all crewmember during the entire month of operation. After the ship is loaded it "steams" out to the chosen fishing location at transit speed of 11 knots.

Arriving at the chosen fishing spot, the crew first needs to set the line. They often set the line in an uncompleted loop (or nearly completed) with a buoy and anchor at each end. The line is fastened to the anchor, and possibly other weights along it, in a way that positions the line a given height from the seabed. The desired height may vary depending on the targeted fish and the fishing area. The line and hooks are taken off the arrangement and baited, by a baiting machine, on the way out of the ship. The bait is feed by a fisher into the machine, which cuts it to a pre-determined size before hooking one piece on each hook. When setting the line, the ship operates at approximately 9 knots.

When the line is entirely laid out the vessel moves to the first buoy to start the hauling, completing the loop. A crewmember grabs the buoy in order to lift the anchor and get the line. The line then feed into the hauling machine and the hauling may begin. Long-liners with moonpool hauls the line through it to make for a safer work environment. The line is then washed, inspected, and arranged for storage. The fish is de-hooked before a fisherman have the fish stunned for bleeding and sends it to the factory for processing. Speed is around 1 to a maximum of 2.8 knots, so the fish can be processed by the workers. The fisher also must retrieve fish that falls off to soon and ends up in the moonpool. In the factory the fish is bleed, headed, gutted and eventually frozen and stored. Which will be elaborated further when looking at the factory as a system, *Factory Elaboration*.

A typical day of fishing consists of 4-5 hours of setting and 19 hours of hauling. Overall, the percentage of operation are 50% hauling, setting 20% and transit 30%. Even though the vessel will only be in transit for a fraction of the operation time, it still accounts for 50% of the fuel consumption.

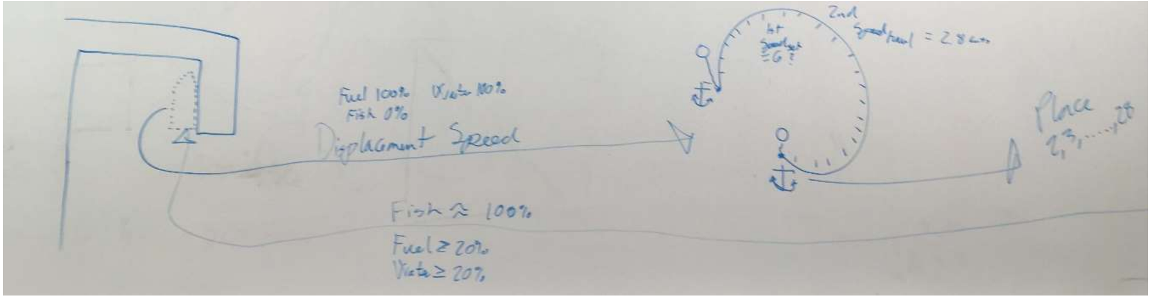


Figure 1 Operation profile

The longline operates in several heavy seas in the north Atlantic. Such as the North Sea, Norwegian sea, Barents Sea and Labrador Sea. North Sea is located in between Great Britain, Norway and Germany. Norwegian sea is located in between Norway and Iceland above North Sea. Barents Sea is located north of northern Russia and northern Norway. Labrador Sea is located between western Greenland and eastern Newfoundland.

After spending about a month at sea while fishing and storing that fish, the time comes for the long-liner to head back to shore. The ship arrives at the shore to unload all the caught fish in order for the fish to go to marked and to be sold. Then the ship loads up again on new fuel and supplies before starting the process all over again on a new trip.

Important Factors for a "Good" Long-liner

From our first meeting with the shipowner, as well as conversations with other ship owners and their crew. The ship in question are the Atlantic and Geir, where we got some pointers towards what usually the decisive factors on the ship design are. Although, it was clear that there were different opinions and prioritisations laying ground for the minor tasks and goals of the two vessels. For our design we focused on singling out the common interest and compare that to what our owner seemed focused on and the requirements he had given. This resulted in some factors that will be considered to be of highest importance throughout the design process.

- Large storage capacity
- Direction Stability - Small Wind areal
- Reliable, PTH system
- Comfort with rolling
- Less workers
- Less Complexity, less work and less part to go wrong
- Environment

Systems
As the basis for a bottom-up design is the necessary systems in order to fulfil the task in question, we take a deep look into the different systems that are typically used in a long-liner. The operation profile is used to determine what kind of systems are needed, as well as what is commonly used. After identifying the systems, a general study of the systems' components helps estimate the overall dimensions and weights. Most systems also require numerous decisions by the designer to narrow down the estimated values.

The Functions
The basis for the included systems is the function they are expected to have. This refers to the tasks that needs to be completed onboard that are necessary in order to complete the main mission of the vessel. In an extreme Bottom-Up approach, would we define the functions out from the main mission. However, as Long-Liners have been around quite a while we can use theirs's systems to define the corresponding functions. This process is a lot more rapid, however it may overlook large innovations. In a true Bottom-Up approach you can end up with something that is not at all close to a traditional Long-Liner, you might end up with something that is not even a ship. Therefore, since it is stated in our task, and the requirement, that we are designing a Long-Line vessel we take the more effective approach.

Factory – Processing
The function of the Factory is to process and freeze the fish ready for storage. The caught fish enters the vessel through the moonpool and are removed from the hooks (either automatically or manually by a fisherman). It falls off the line into a well, is brought up by a lift and are cut at the neck, for it to blead out in a tank with cold running seawater (slush). There can be used multiple tanks in order for the process to be continuous, and the fish will spend approximately 30 minutes bleeding out. When fully out-bled the fish is transported to the heading and gutting process, which can either be done by a machine or a fisherman. Usually smaller fish go through the machine (for instance a Baader 444) and the larger ones are done by hand (Larsen, 2013, s. 29:39). The H/G fish is placed in plate freezers within one hour of coming aboard the ship, where it spends about 4 hours freezing. When completely frozen the plates are brought on pallets to the main storage.

Cold Storage – Maintaining Quality
When the fish has successfully been processed, frozen and packed it is important that it maintains the high quality till it can be sold ashore. Therefore, will the fish be quickly transported into a storage after being frozen, and that stay frozen by Norwegian regulation at -18C.

Moonpool – Safe Hauling
Moonpool is a safer and more efficient alternative to the traditional longline method. In traditional longline fishing waves can knock down fishermen, and without sufficient

pumping and with human errors, like open watertight doors, the ship is at the risk of capsizing. Moonpool eliminates these risks and is proven to be a more effective fishing method:

> "*Teknologien med dragebrønn kom best ut angående tap av fangst. Her ble det registrert tap på torsk og hyse på henholdsvis 0,40 og 0,82 %, mot 1,35 % og 1,55 % for ALH og 2,49 % og 3,03 % ved tradisjonell haling. Resultatene er basert på antall fisk.*" (Rindahl & Larsen, 2009)

Hauling is considered by fishermen to be the most dangerous task during a longline operation. The traditional longline fishing is by hauling the line though a hauling hatch through the side of the ship. A fisherman's task is gaffing the fish onboard and occasionally retrieve unhooked fish from the sea. That puts the fisherman in an exposed position to weather and waves, risking getting knocked down or dragged overboard. Fish hauled through the hatch are also exposed to the weather and crashing waves, which risks unhooking the fish. (Rindahl & Larsen, 2009)(11-12)

Autoline/Hauling – Catching the Fish

A Long-line vessel catches the fish, as the name indicates, with the use of a long line. This line needs to be stored on board, put out on in the sea with bait on the hooks and hauled back up again. In the modern Long-liners is there an Automatic Long-line Hauling (ALH) system in place as this increases the automation of these processes (Larsen, 2013). Automation can decrease the need for manual labour as well as increase efficiency. An ALH system automize the setting and the hauling process of the longline fishing operation.

Machinery – Power Supply

A ship has to function as its own "society" and this includes energy production. So, all the different operations and systems that requires any form of energy onboard must have access to a sufficient energy source. The production of this energy will find place in the machinery, and mainly consists of the main propulsion system and the auxiliary machinery for electrical generation. The electrical generation is for systems like fishing equipment, thrusters, refrigeration, air conditions, pumps, electrical, firefighting, thruster etc. The Machinery also needs to be controlled from somewhere and safety equipment like switchboards.

Connections

After the function of the main systems that will, or at least may, be included in the long-line vessel have been explored and understood as single units. The next step would be to understand how the systems are connected and depend on each other. As this is also an important function of them, especially when they are united on a ship that should also serve as one unit. How one system might rely on being adjacent to another system, at least get an understanding if it is favourable to have them close together or perhaps the opposite is the case. For instance, separating the accommodation from the noise and smell of the factory.

The line needs to be led from the hauling-hatch to the moonpool before it can be connected to the main hauler. The line then continues through a pipe that leads it to the hook separator and the line arrangement. Of course, it is favourable to have as short distance as possible, nevertheless the moonpool and the line arrangement can be positioned relatively independently in the general arrangement. With the line fish enters the vessel through the moonpool and it should be brought quite rapidly to the factory. A lift is used to lift the fish from the hauling well to the factory belts. Therefore, will there have to be factory belts going to the location of the moonpool, although the length of the elevation could possibly vary.

In order to deliver fish even if something went wrong with the lift a crane can be used. This means that the crane needs access to the main storage, which can be accommodated by using the lift shaft. Lowering the lift manually and having a hatch on top of the shaft enables the crane to access the main storage. There should also be easy for loading food, wrapping and bait effectively, and this has to be incorporated in the design. Another aspect that affects the food storage location is having the galley close by, and for serving would it be a significant advantage to have the table is close. The first thought could be to have the galley wall to wall with the dining room, another could be to introduce a dumbwaiter and place the two rooms on top of each other.

There are a few systems that needs refrigeration and they are using the same refrigerant; the vertical plate freezer, the hanging freezer, the fish hotel, the bait storage and the main storage. The refrigerant is transported in pipes so it is possible to have all of them positioned independently, although minimizing the pipes on-board saves weight and lowers the risk of leakage.

The Forms

As a “response” to the desired functions, the different systems will have a corresponding form. This is the physical aspect that will be manufactured, and it is important to gather an idea and estimate this in order to make a sufficient design. We will take a look at the systems one-by-one and study how these are normally formed for other Long-liners. This will serve as the estimates’ backbone and will only feature small alterations to match the specific functions for our Long-Liner.

The Factory

There are a lot of smaller tasks that are done in the factory, which were mentioned in *The Function.* Although there are multiple ways of completing these tasks a “usual” Long-Liner factory would include: bleeding tanks, heading machines, gutting machines, cleaning tanks, freezers, packaging machines and a conveyor belt to move the fish between them. Hence, a factory should be able to accommodate all these different machines and components. In order to get an idea about what exactly this entails, we can study the different machines and make arranging suggestions to establish an estimate. This estimate will include the dimensions of the factory and serve as a system block in the *GA Brainstorming.*

Released from the Hook

The Autoline hauling system (AHL) releases the fish into a well by the moonpool walls. From there, a lift lifts the fish and place it at the start of the factory. Traditionally a fisherman took of the fish with a hook, the well (and ALH) was added in order to reduce the number of fish lost.

Bleeding Tank

The fish need to be bled in order to ensure a high-quality product. The shelf life of the fish would be greatly reduced if blood where to be retained in the fish, since blood is a good nutrition for bacteria. Also, blood in the meat will lead black and yellow spots in the meat (Grete Hansen Aas, personal communication, 23. Jan.2020). 15 to 20 minutes in clean circulating water is a sufficient time to bleed the fish properly (Aðalbjörnsson & Viðarsson, 2017, p. 4).

The fish is stunned, cut at the neck, as soon as it comes aboard so it can start to bleed immediately. For most long-liners the fish is sliced open in the neck by a fisherman. However, there are some automated stunners that are used on larger fish processing factories, those are inexpedient for our vessel (H.P. Holmeset, personal communication, 5. Feb.2020). An example of what may be used in larger fish processing factories is the WMT’s Seafood Innovations Fish stunner:

Figure 2: Fish Stunner: https://www.w-m-t.com/seafood-innovations-fish-stunner/

Our vessel needs the ability to bleed 10 tonnes of fish per hour. As seen in some videos the bleeding process is executed in two large tanks, where the fish has a small occupancy in both to ensure all fish get at least 15 min in clean water.

Resulting in two tanks of 3.5 tonnes when taking into account the extra weight of the gut and head. From that, a deck area can be estimated; 2 metres long and 1.8 metres wide, perhaps some additional length due to the rotation devise in the tanks.

After touring Geir we “measured the tanks to be 4x2.5 metres in the deck area. The same size of tanks where utilized in the cleaning before freezing.

RSW-system

In order to produce cold water can there be used either an RSW (refrigerated sea water) or CSW-system. Difference from CSW (chilled sea water) is that RSW uses mechanical energy to cool down the water. This is a widely used system, since it eliminates the need to carry ice from land. Large tanks are filled up with sea water and this water is circulated in a closed system that cools it down close to freezing point. In our situation this will be mainly used for filling up the bleeding tanks. The bleed off water cannot be recycled and therefore it will be a need for always supplying cold water (Teknotherm Marine, 2020).

RSW system were deemed unnecessary by the shipowner.

Heading

After the fish have been fully bled-out it is time to take off its head. Numerous machines on the market that can do this process. Although they require a fisher to insert/line-up the fish in a certain way to make it work, and here as well are their size restrictions. The large and modern long-liners use a machine to cut the smaller fish and leave the large fishes to be headed and gutted by hand.

Curio’s C-3027 heading machine is an example and can take a broad range of fish from 2.5 kg to 12 kg. This machine is 2.9 metres long, 1.9 metres width, and stretches 2.3 metres in height (Curio Food Machinery Ltd).

Gutting

As the fish continues along the conveyor-belt, it passes the gutting station, as mentioned in *Heading* the larger fishes done by hand. Nevertheless, there are machines for this process as well that work up to a certain size of fish.

Freezing

After the fish have been processed, it needs to be frozen down. The most common type of freezer is a plate freezer due to its fast-freezing rate and easier fish handling. Freezing rate are decreased for a plate freezer due to more transfer of energy due to larger area of contact, compared to for example an air blast freezer. Also, a plate freezer creates smaller blocks resulting in easier handling, faster unloading and loading compared to other freezing methods (W.A. Johnston, 1994).

There are two types of plate freezer vertical and horizontal. Horizontal plate freezers are utilized for higher quality product like fish fillets. Since the quality remains high due to the fish being frozen horizontal and therefore maintaining shape. Vertical plate freezers are used for products with more irregularities like a fish that is only headed and gutted (W.A. Johnston, 1994). Since a vertical plate freezer is loaded by dropping the fish into the compartment versus placing the product in a horizontal freezer. Reducing the loading time drastic for irregular products. Since by design requirements our vessel is only going to process headed and gutted fish in the factory is that the optimal freezer choice.

Thickness of the blocks can range from 50mm to 100mm. A smaller thickness leads to a faster freezing time but is limited by the necessary space to fit fish properly. Smaller block size is utilized for high volume small fish. Since a long-liner fish large fishes and in order to utilize the vertical plate freezer for the ability to utilize plate freezer for largest percentage of processed fish, where a 100mm block thickness selected.

Standard block sizes depend on the pallets, since the fish will be stored on a pallet measuring 1070-1070mm can not the blocks surpass those dimensions. Standard block size for a vertical plate freezer fitting our pallet size is therefore 1060-530mm (Freezertech, 2020). A block of that size weighing approximately 50 kg.

Freezertech states that the traditional time for freezing fish in a vertical plate freezer is a total of 132 minutes for ammonia at -40C. Which consist of 107-minute freezing time, 4 minutes defrost time and 20-minute unloading / reloading time. They say that their freezer can reduce defrost time from 5 to 1.5 minute, resulting 8.7% increase in production. In order to be on the safe side, the traditional time where chosen for the necessary calculations.

Dimensions of a vertical plate freezer have typically a fixed width due to the block length of 1060, freezer is dimensioned at a width of 1500mm. For a block with dimension 520mm height, the maximum top of the freezer height is 810mm. Due to necessary space for equipment is the machinery space a larger height 500mm above that resulting in 1700mm.

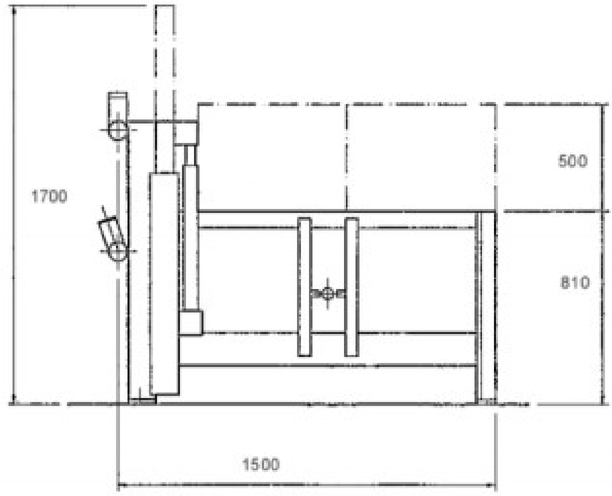


Figure 3 -Dimensions For Vertical Plate Freezer (Teknotherm Marine, 2020)

Since the capacity per area increases with the length due to less necessary space for mechanism. Therefore, largest practical length for a freezer should be chosen. Values for batch sizes for a vertical plate freezer where obtained from the manufacture freezertech.

Table 1 represents the values from batch load and freezer length obtained from *Freezertech*. Area/Unit explains the necessary floor space necessary for a specific size of vertical plate freezer using width dimension obtained from *Teknotherm*. Efficiency where calculated in order to understand the impact of a larger freezer length.

Freezer Size Nr. Stations	Batch Load [kg]	Freezer Length [mm]	Area/Unit [m^2]	Weight/Area [t/m^2]
10	500	1986	2.98	168
16	800	2727	4.09	196
20	1000	3221	4.83	207
26	1300	3962	5.94	219
30	1500	4548	6.82	220
36	1800	5289	7.93	227

Table 1: Relationship of Length of Freezer and Weight/Area Efficiency

Difference for a freezer with a length of 1986mm to one width 5289mm, where a large increase from approximately 168t/m^2 to 227t/m^2. Which is approximately 74% increase in more t/m^2 which shows the significant difference choosing the largest possible length per unit of a vertical plate freezer.

Unloading methods for a vertical plate freezer.

After a fish blocks are finished freezing does it need to be unloaded. That and following processes should be automated since block can weigh as much as 50kg. Automation are therefore crucial as it both

Several different methods for unloading where noticed when exploring different solutions.

From *Optimar* had created two different solutions whether the equipment loading integrated into the vertical plate freezer or not (Optimar AS, 2020). *Geir* utilized the not integrated solution of the robotic arm.

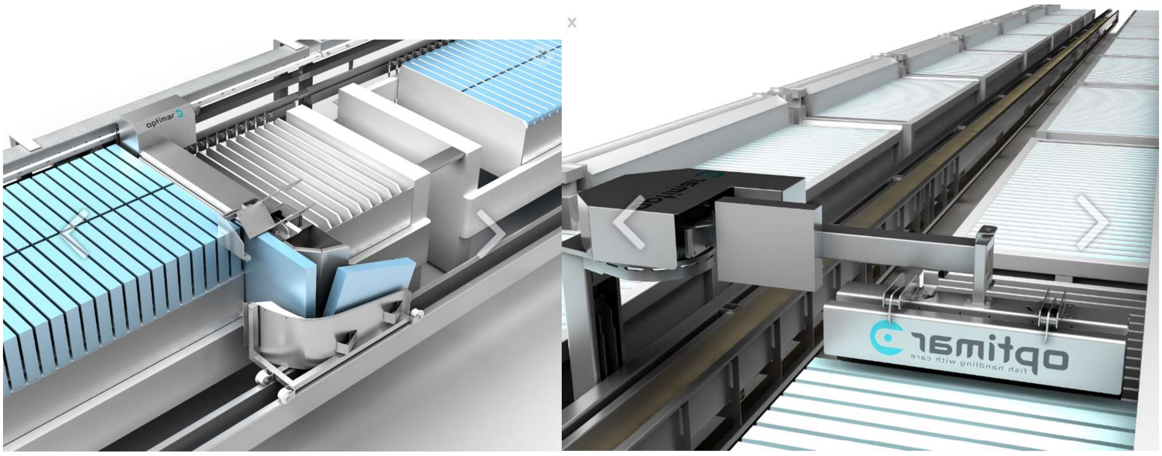


Figure 4 – Vertical Freezer Unloading Optimar Solutions (Optimar AS, 2020)

After visiting *Atlantic* did another solution present. That solution dropped the block on a transport band underneath the vertical plate freezer. Transporting it from underneath the freezer. That solution seemed to utilize more floor area space, than the two *Optimar* solutions above, but increasing the height.

AS, 2020)

For larger fish that does not fit into the vertical plate freezer. A hanging freezer is utilized as that makes the fish straight, giving the best quality possible.

Further the ice blocks will be sealed in plastic or paper, typically done in a machine. Then the wrapped fish blocks go into a freezing hotel. The purpose of the freezing hotel is to store ice blocks until enough of one type to fill a whole pallet. The blocks are automatically stacked on a pallet machine.

Then transported to a lift that brings it down into the cold storage. The lift where estimated to be around 2x3 meters. In order to be on the safe side, depending on whether customer wanted solution (3) or (4) in figure 6, so enough space where allocated.

Figure 6 - (Conveyor Systems Ltd., 2020)

Cargo

For the ship to fulfil the task is it necessary to properly design cargo spaces, both for accessibility and necessary space requirements. Some cargo spaces have been explained in the relevant system, as it is much easier to connect them together. Therefore, section will include the relevant information about the main cargo spaces in the design such as cold storage and tanks.

Cold Storage

Purpose of the cold storage are to store the frozen fish processed from the factory until offloading at dock. Design requirements are the storage needs to hold a minimum capacity of 450 tons. The fish is to be stored on wooden pallets with dimensions (L=1.07m, W=1.07m and H=0.1m). The fish stored on top has a total height of 1m making the total height of a loaded pallet to be 1.1m. From the design requirements should any stack of pallets not be more than three. In order for the refrigeration system to function is it necessary with a clearance of 0.15 meters from the roof to the pallets. That information was given from our supervisor (Håvard, personal communication).

Functionality

As when the frozen fish arrives from the factory, or during the unloading personnel the fish must be transported. That requires a forklift as a pallet weighs approximately 1 ton. Necessary space must be allocated to allow a lift operating in between the cargo and the lift. Space is required to operate the forklift by its turning radius and height. That is a factor from the size of the forklift by the lifting requirements.

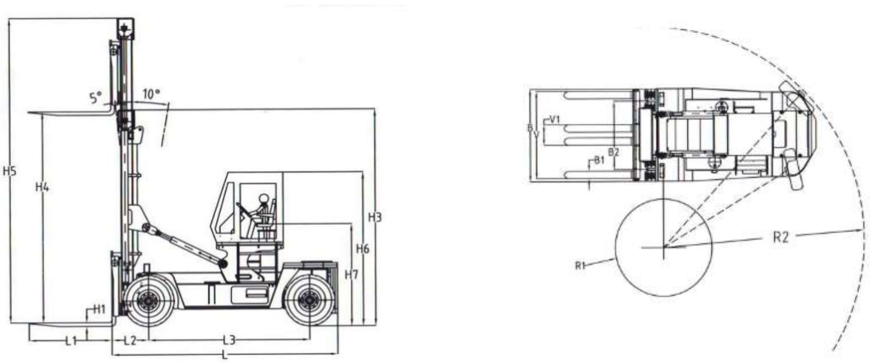


Figure 7 Forklift Radius

Efficiency

To keep a storage at -18 degrees consumes a significant portion of the total power consumption for the vessel. Therefore, designing the storage room to reduce the power consumption are both beneficial for the environment and operational costs. A large part of the heat loss can occur through heat transfer through the wall to the surroundings. Heat transfer can be explained by equation under.

$$Q = \frac{U * A * (Temp\_Out - Temp\_in) * 24}{1000}$$

Q = KWH/day

U = U-Value of wall $W/m^2 * K$

Temp out = ambient temperature

Temp in = temperature of the storage room

A = Surface Area

Temp inside and outside of the storage are factors that one as a designer has a small amount of influence on. Factors one can influence is the heat transfer coefficient, and surface area. Surface area is the total area of the walls of the storage linking it to the surroundings. The heat transfer coefficient is a factor on how much heat can be transferred through the material. Steel has a high heat transfer coefficient therefore are cold storages designed by having an insulator separating the cold storage walls from the steel. Such material can be polyutherane which are most used for industrial cold storages. A problem with polyutherane when operating at sea is that the material burns fast and easy. Therefore, another material called rockwool are typically utilized since it will not burn, even though it has a higher heat transfer coefficient.

Also, any product entering a cold storage with a higher temperature will require extra heat in order to cool down the product. Therefore, is it optimal to have a low temperature on all fish entering the factor. Also limiting the loss of heat due to opening from the lift and crew doors will reduce the heat loss from the environment. Other factors consist of the internal load such as heat from workers, forklift, lights and other equipment.

Refrigeration System

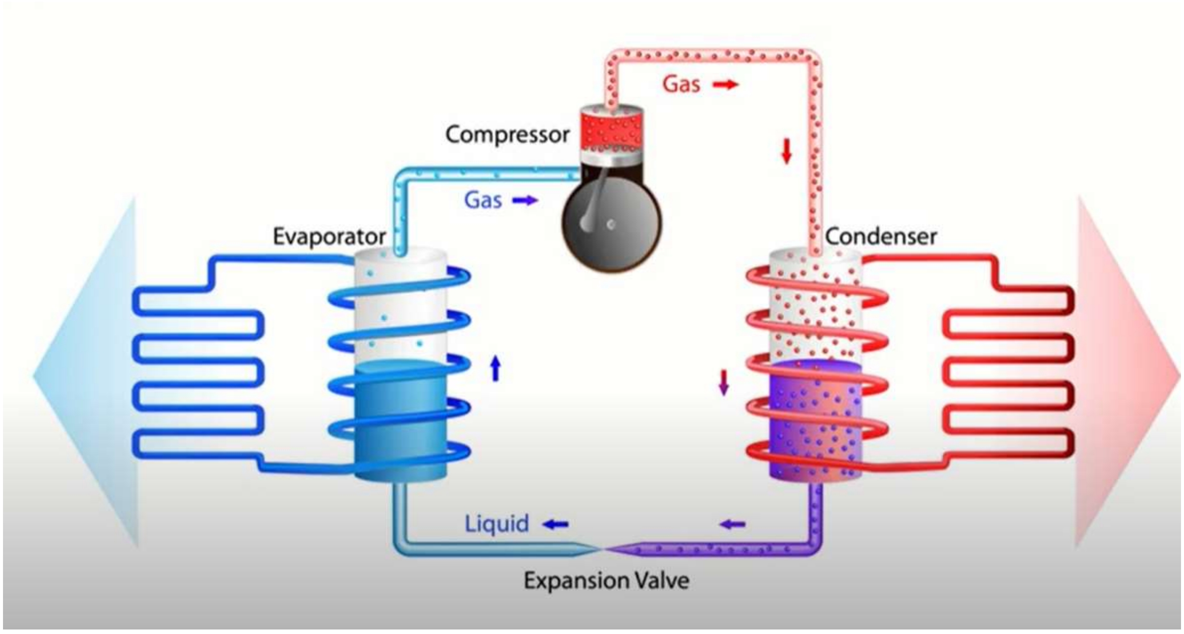
A vital system for many different types of vessels since it extracts heat cools cargo, provision and other items or components that need refrigeration. There are four main parts of refrigeration system. Compressor, condenser, expansion valve and the evaporator.

Compressor works by raising the pressure of the system raising temperature of the refrigerant, that a gas with a constant volume if the temperature increases must the pressure also increase due to particles moving faster and vice-versa (Helmenstine, 2019). Also works by pumping the refrigerant around the refrigeration system.

Basic function of a condenser is to either circulate sea water or air to sub-cool the refrigerant, for cooling the system by the evaporator.

Same as with the compressor it releases pressure, decreasing the temperature by decreasing the pressure. The lower pressure causes the evaporator to be more effective.

Cools the air by transferring heat from the cold storage for example. Cooling down the air, that is inside the system. Evaporator are explained by the Clausius statement of the second law of thermodynamics, which states that the heat flow spontaneously from the hot to cold and not cold to hot.



https://www.youtube.com/watch?v=dfLFt2X1uh4 0:49

Refrigerants

"MARPOL, Annex VI (Regulations for the Prevention of Air Pollution from Ships. Regulation 12 – the use of ozone depletion substances (ODS) in marine applications). New installations containing CFC or Halon are not permitted on ships constructed on or after 19 May 2005"

When it comes to a refrigerant system it has apparent consequences what type of refrigerants one utilizes both for environment and operational costs. Since ozone depleting refrigerants are banned, and many non-natural refrigerants have high GWP (Global warming potential). Are Natural refrigerants good choice since it has zero ODP (ozone depletion potential) and near zero GWP (Global warming potential). Most common of the natural refrigerants in applications are R744 (carbon dioxide) and R717 (ammonia) or hydrocarbons such as R290 (Propane), R600a (Isobutane) and R1270 (propylene) (CAREL, 2020). These will be evaluated against each other to find the most effective refrigerant, and which of them that would function best for refrigeration system at sea.

Natural Refrigerants

Carbon Dioxide (R744) is not toxic and non-flammable. It is a cheap substance with a high efficiency and are therefore cheap for heat extraction purposes. The greatest challenge with carbon dioxide is the high pressure necessary in order to become a liquid. That requires the equipment to become more expensive and harder to maintain, which makes the equipment higher cost and repair more costly (Linde, 2020). The benefit of the high pressure is that it has a very low temperature and can be evaporated down to about -50 degrees, which significantly increases the efficiency of the system (Verpe, Spring 2018). Carbon Dioxide is a highly toxic substance even with small concentration as seen in figure 8. It is not detectable as it produces no smell. Therefore, proper detection system is required

Table 3 Carbon dioxide potential health effects

Concentration of CO_2 (%)	Time	Adverse effects
17–30	0–60 s	Loss of controlled activity, unconsciousness, death
>10–15	1–3 min	Dizziness, drowsiness, muscle twitching, unconsciousness
7–10	1.5–60 min	Headache, increased heart rate, shortness of breath, dizziness, sweating rapid breathing
7.5	5 min	Significant performance decrement
6	Several hours	Tremors
6	<16 min	Headache, dyspnea
6	1–2 min	Hearing and visual disturbances
4–5	A few minutes	Headache, dizziness, increased blood pressure, uncomfortable dyspnea
4–5	4 h	Drop in body temperature (1 °C)
3	1 h	Mild headache, sweating, dyspnea at rest
2	Several hours	Headache, dyspnea upon mild exertion

Figure 8 - CO2 Health Effects (Bhatkar, 2013)

Hydrocarbons are safe to work with, non-toxic as well as very cost-efficient, because of its energy efficiency. On the other hand, hydrocarbons are highly flammable and requires more extensive design to maintain effective safety (Langde, Ali, Shahid, & Sultan, 2014). Fire and explosion danger area are a significant risk for vessel operating far from shore. Therefore, have not found any fishing vessel utilizing that natural refrigerant

Ammonia benefits have a high efficiency. Also, the substance is cheap and non-flammable (CAREL, 2020). The substance is toxic and hygroscopic. Hygroscopic means that the substance will bind with water molecules and therefore will damage moist parts of the body like moist skin, eyes, and throat. A fishing vessel have a high degree of moistness in the factory and a leakage might cause severe damage to crew members. Therefore, is it a requirement that the refrigeration system operates in a separate room. Even though it is acutely toxic at even low levels, it is easily detectable as it produces strong odour at very low levels of saturation. Therefore, deaths are rare from exposure to ammonia (Walter S. Kessler).

"Maintenance demands can be high and even in normal operation they generally require more frequent routine tasks than fluorocarbon plants. One such example is the regular draining of oil from the evaporator, which is vitally important for the safe operation of the system. This task may be laborious and frustrating, but it introduces the risk of gas leakage if not carried out correctly" (NORTH, 2016)

Non-Natural Refrigerants

It is possible to use a non-natural refrigerant with zero ODP such as HFCs (hydrofluorocarbons). HFCs is efficient, reliable, high safety and cheap, but the major problem is high GWP level. Therefore, will it contribute to global warming, which is not ideal for an environmentally friendly fishing boat. HFO (Hydrofluro-Olefins) might be the future since it has the same benefits as HFCs. The major problem with HFO is the high price of the refrigerant. (MarketWatch, 2019). Figure 9 represent the environmental factors for different refrigerants.

Type	Refrigerants	ODP	GWP (100 years)	Atmospheric lifetime (years)
CFCs	CFC-11	1	4,680	45
	CFC-12	1	10,720	100
HCFCs	HCFC-22	0.05	1,780	12
	HCFC-141b	0.11	630	9.3
	HCFC142b	0.065	2,000	17.9
HFCs	HFC-32	0	650	4.9
	HFC-125	0	2,500	29
	HFC-134a	0	1,320	14
	HFC-407C [HFC32/125/134a (23/25/52 wt %)]	0	1,674	29
	HFC-410A (HFC-32/125 (50/50 wt %))	0	1,997	29
Natural refrigerants	Carbon dioxide (R-744)	0	1	0
	Ammonia (R-717)	0	0	0
	Isobutene (HC-600a)	0	3	0
	Propane (HC-290)	0	3	0
New	HFO-1234yf	0	4	0

Source: montreal protocol science assessment of ozone depletion 2002

R-407C and R-410A are blend refrigerants of R-134a, R-32 and R-125. Highest value among the components (i.e. R-125) is considered for the atmospheric lifetime

Figure 9 - Environmental Impact (Bhatkar, 2013)

Discussing type of Refrigerant

Carbon dioxide versus Ammonia where chosen to be analysed since they had a strong enough safety profile, low price and environmentally friendly. Ammonia is a common refrigerant used on fishing vessels. Carbon dioxide present a distinctive advantage versus CO2 since the evaporator can operate down to -50 degrees. For ammonia the evaporator is only suitable to work under -35 degrees. The lower operation temperature can reduce the freezing time of fish.

Espen Halvorsen Verpe did a master thesis exploring if the reduced freezing time utilizing CO2 against ammonia in a vertical plate freezer were beneficial. His numerical freezing models were validated by a physical test in a vertical plate freezer. Source of error is for irregular fish with less contact ratio. Since model versus the physical test had a difference of 3% were the results determined to be realistic to utilize in this analysis.

The analysis resulted in a higher coefficient of performance (COP) of 11% for ammonia against CO2. For a vertical plate freezer system utilizing at -50C resulted to be 70% more expensive than the same system operating at -30C, but when freezing at -50C where the production capacity increased 66%. Freezing cost where estimated to be 90-190kr per ton of fish. In order to analyse how significant, the higher cost of freezing was a calculation for the cost increase for a trip for utilizing CO2 for our vessel. In order to understand the impact on the difference in price also expected earnings from the fish where calculated to calculate the ratio for maximum decreased earnings.

$$CostIncrease = (C_t * 70\%) * W_F \approx 60000kr/trip \text{ or } 2000kr/day$$

$$Earnings = P_F * W_F = 144{,}00000kr/trip \text{ or } 480000kr/day$$

$$Ratio = \frac{60000}{14400000} \cdot 100 = 0.416\%$$

Ct = is the cost per ton of fish here were 190kr/ton used in order showcase most realistic increase in cost. Since freezing whole fish causes a lower contact resulting in higher freezing cost.

WF = Weight of fish frozen on a trip which by design requirements are 450 tons.

PF = the price of the fish from sale which are approximately 32000kr/ton (Verpe, Spring 2018).

Earnings or increase in cost per day where calculated from a standard length for a trip which is 30 days.

Decrease in the profit margin resulted to 0.416% which are small, but can be a more significant difference since a long-liner already operates at a smaller profit margin than a trawler. For ships where the freezing time are a bottleneck in the function such as trawlers, due to high variability in catch. CO2 utilized in those ship can therefore reduce the operational cost by decreasing freezing time. Also reducing the amount of time before the fish is frozen can increase the quality. Since a long-liner's factory have a steady flow of fish is the benefit of freezing time not significant compared to the increase in price. Another benefit from utilizing CO2 achieve is possible reduce the amount of vertical plate freezers reducing the factory space.

Further since CO2 can increase operational cost both from more maintenance and initial investment into equipment. Will that lead to a decrease in the profit margin of the vessel.

Conclusion on Refrigerant type

Disadvantages and benefits determine that ammonia is the best choice for this long-liner. Since for a long-liner is it concluded that extra space gained is are far less significant than the decrease in profit margin both from operational, maintenance and initial investment. It needs to be noted that one should utilize one refrigerant for the whole system included freezer storage. Therefore, is this analysis correctness limited since it only analysed the effect for utilizing it on the vertical plate freezer. An added increase in operational costs are likely to occur from the cold storage as well.

Earlier it was explained why it is so important to get the point of view of the owner after explaining benefits versus the disadvantages for a correct conclusion. Ship owner explained that he had no experience utilizing CO_2, but the added complexity and difficulty of maintenance were the most crucial factor for him. Therefore, ammonia is the correct refrigerant to utilize.

Moonpool

Moonpool is a hole through the bottom of a ship, allowing access to the water surface within the ship; typically used for drilling ships, pipelaying ships and ships carrying submarine vessels and instruments. Weather, waves and transiting can cause pumping and sloshing inside the moonpool, which makes for unreliable and unsafe working conditions.

A central hauling pool (dragerbrønn) is a moonpool specially designed for longline fishing. It is a tube going through the ship with an opening towards the sea and an opening inside the ship. The cross-section at the top of the tube is increased to such extent that changes of the surface level in the pool are low enough to be considered a safe working condition. Additionally, the increased top section reduces the speed of the water in the tube, which lowers the risk of losing the fish. (Enerhaug, 2004)

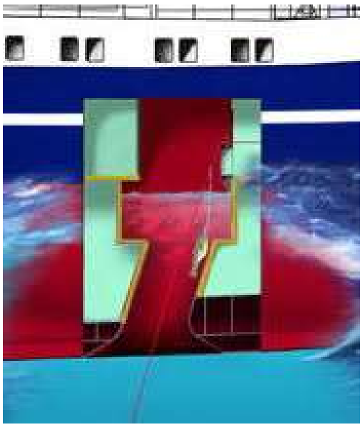


Figure 10 Moonpool as illustrated in (Enerhaug, 2004)

Hauling through a moonpool decreases the rate of lost fish, as stated by [Rindahl and Larsen](), because the fish is released as soon it leaves the water. As stated in the *[Factor-processing]()*; the fish falls into a well after being released from the hook. If the fish gets unhooked before the well, the fisherman can easily gaff it from the pool. Having this process inside, shields the fish from being unhooked by waves crashing and wind, in addition to making it easier for the fisherman to gaff the "lost" fish.

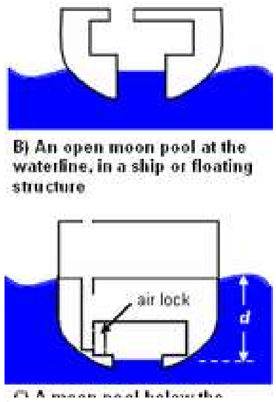


Figure 11 Two different design concerning pressure. The image is a cropped version of https://upload.wikimedia.org/wikipedia/commons/3/34/Moon_pool_diagrams.PNG

There are many ways to design the moonpool. The most common design is by allowing the sea pressure being equalized by the atmospheric pressure, making the water level in the moonpool and outside the same. By creating an air and pressure tight room, one can equalize the pressure inside the moonpool area with a certain depth, having the moonpool surface lower than the actual sea surface. This is however a dangerous environment for humans, as the pressures is high and changes as waves affect the internal surface.

With the right tube profile, the tube can be self-cleansing, although there is little research done on the hydrostatic effects in a moonpool. The first “Dragerbrønner” was designed with an elliptical tube, this tube had a circulation or change of water within it, replacing old dirty water with clean water. To save of space, newer boats are installed with circular tubes. The circular tube did not have the same effect, but the water surface proved to be much calmer. Sivert O. Sæther investigated in his master thesis (Sæther, 2019) the effect of tube profile on the circulation of water inside the tube. He said the results where hard to read and needed further testing, “however, the current results show that the elliptical and rectangular moonpool shapes provide a better behaviour of cleaning out dirty water.”

Due to the nature of the fishing line the tube needs to be angled. It is normal in longline fishing to drive towards and over the line; this way it appears the line is hauled from the back. To avoid the line from grinding the opening of the tube, the tube is at an angle. This lowers the chance for fish to get knocked off hooks and the line and hull do not wear off quickly. (Canada Patentnr. 2307650, 2007)

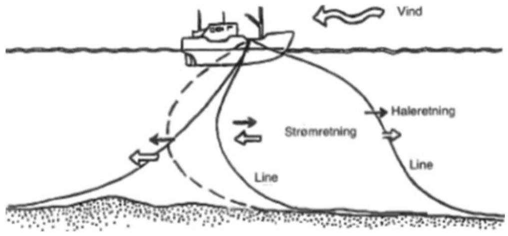


Figure 12 Line allignment form (Karlsen, 1997)

The difference and transition between the narrow profile and the wide profile damp the change of the surface level in the pool. The volume flow in the moonpool is constant, due to waters incompressibility, increasing the profile will therefore slow down the surface’s vertical speed. The patent (Canada Patentnr. 2307650, 2007) suggests that the turbulence that occurs in the transition between the two sections converts some energy to undetectable heat, and therefore has a damping effect of the flow through the tube.

Hallgeir Holmeset claims in an interview with NorskFisk.no that position of the moonpool is vital to its success. He points out that ‘Antarctic III’, a vessel with moonpool build in 1995, had its pool forward in the ship, causing splashes and foam due to the great pitch motions. He adds that a rectangular profile and plate damping measures could contribute to said effects. (Holmeset, 2018) To avoid splashing and foam, the moonpool should be placed nearby LCF, where there are pitch motions are negatable. Since LCF moves for different load conditions, an approximation is ok.

Moonpool Test

We want to test the effect different tube profiles and transitions between pool and tube has on damping the vertical surface speed. We wonder if the profile had anything to do with the damping effect as suggested in Sivert. O.S. master thesis (Sæther, 2019)concerning switching to a circular shape, or if it got something to do with the area of the profile, limiting the flow of water through the tube. We will therefore test moonpools with an elliptical profile, a circular profile and another elliptical profile with the same area as the circular profile.

The fist model we made in 3D where with 6 pools. Two different transitions, one with a sharp edge and one with a slope. Per transitions we added three different tubes, one oval, one circular and another oval with the same area as the circular tube. In the model the pool was 200 mm wide. Before it got made it was decided the differences in the model was too small to be able to detect any difference in the waves.

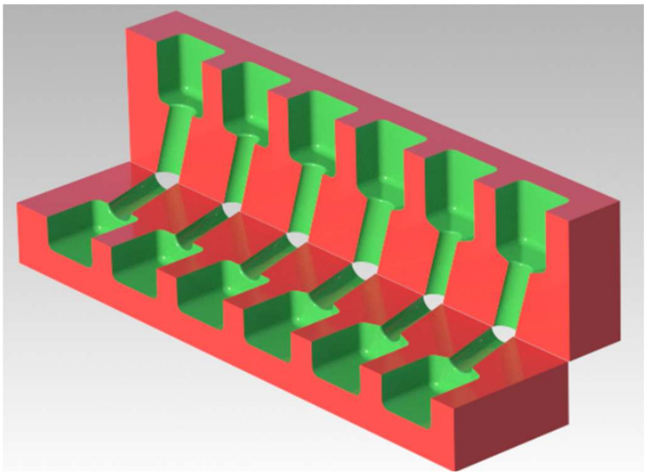


Figure 13 Moonpool model test with six designs

In the second model we only made two designs, with a much more noticeable difference between the two transitions. One with a sharp edge and one with a slope. For the test to be as similar as possible the pools where made so the volumes in the pool where the same, and the pool section had the same area. So, the damping effect we would see would not be due to any difference in the volume. We decided also to model the pool in a hull so the test would resemble to the actual condition of a moonpool in a ship. This would more accurately resemble the flow direction of water under the tube.

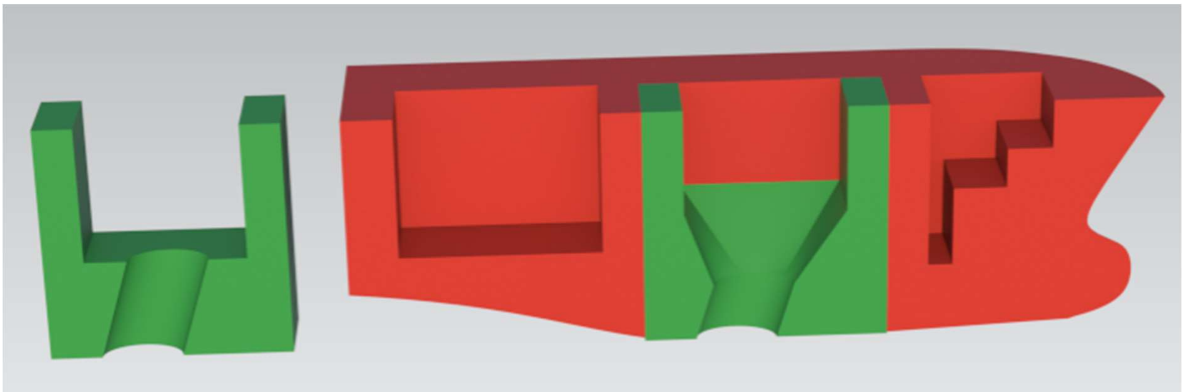


Figure 14 Final moonpool model

To resemble the reality as much as possible, we scale the hull so that the water depth in the moonpool resembles 1 meter in reality, and the hull have a B/T relation of 2,55. We have chopped up the middle part of the hull, so the we can fit as much weights as possible to submerge the model at required depth.

We also want to test the effect of an asymmetrical pool. Meaning installing the pool to the side of the ship while the tube still is cantered according to the ship. An asymmetrical pool could give more flexibility in the ships floor design. We suspect this may have negative affect on the ships yaw motion.

We plan to use the same moonpool for the test, just putting a block to simulate an asymmetrical pool. The test will be visual, to see changes in the model's yaw motions.

Now, there is still some technicalities to left to figure out, however, due to the recent pandemic the test was suspended. Technicalities that remain include what sensors to use, where to mound the sensors, how to keep the vessel still without disrupting the result of the test.

The result we wanted to see, was if there are a significant difference in the damping effect of a 'Cone' transition compared to a 'square' transition. We expect the 'Square to have a better damping effect due to the turbulence created in the transition, but we are not sure to what extent.

Already we know there are some sources of error. The model is incredibly small, and due to the size, the hydrodynamic properties will not mimic the effect on a large scale moonpool. In ship model testing, one puts on tape, to induce turbulence for a more realistic result, we could do the same on our models, but we are not sure if this will delegitimize the result.

Due to the model's size and the height of the waves created by in the tank, we would need incredibly accurate sensors to detect any difference. Our model is made of foam, which have a different roughness than steel.

Moonpool Calculations

There are not many guidelines concerning the central hauling pool, so we choose the dimensions based on eyeballing the pools in *Atlantic* and *Geir* and the dimensions given in Siver.O. Sætheres master thesis. However, we chose to do some calculations to review the moonpools properties.

The main function of the moonpool is the profile change reducing the speed of the water surface. Since water is an incompressible the volume flow is constant, meaning we can calculate the speed reduction of the profile change.

$$Q_t = Q_p$$

$$v_t A_t = v_p A_p$$

$$v_t \frac{A_t}{A_p} = v_p$$

$$v_t \frac{4.71m^2}{16m^2} = v_p$$

$$v_t 0.29 = v_p$$

We see that the surface speed in the pool is reduced to 30 percent of the speed in the tube. And if we assume that the vertical speed of the water inside the tube is the same as the surface outside, we have reduced the wave height by the same amount, as distance is the integral of velocity. Without any proper guidelines, we assumed this is ok.

The Bereau Veritas S.A. has a guideline for moonpool assessment. We chose to use their formula for moonpool resonant pumping to see if there is need for further work. (Buerau Veritas, 2016)

We have a moonpool with changing profiles; it consists of two constant profiles; therefore, we modified the formula a little.

$$M_{eq} = \rho * A(0) * \left\{ \int_0^{<4.5} \frac{A(h)}{A(0)} dz + \int_{4.5}^{5.5} \frac{A(h)}{A(h)} dz + \frac{A(h)}{A(0)} * K * \sqrt{A(0)} \right\}$$

Der A(0)=4.71m^2

A(h)=16m^2

Ro=1.025 t/m^2

$$M_{eq} = \rho * 4.71m^2 * \left\{ \int_0^{<4.5} \frac{16m^2}{4.71m^2} dz + \int_{4.5}^{5.5} \frac{16m^2}{16m^2} dz + \frac{16m^2}{4.71m^2} * 0.476 * \sqrt{4.71m^2} \right\}$$

$$T_m = 2 * \pi * \sqrt{\frac{M_{eq}}{\rho * A(0) * g}}$$

$$T_m = 2 * \pi * \sqrt{\frac{M_{eq}}{\rho * 4.71m^2 * g}} = 9.3s$$

3 Typical hydrodynamic phenomenon

3.1 General

3.1.1 The moonpool may induce the following hydrodynamic phenomenon, having potential undesireable effects to be considered in the design of the vessel.

3.2 Resonant pumping mode

3.2.1 Definition

Pumping mode is generated by vertical oscillation of the water column within the moonpool. When waves period enters in resonance with moonpool natural period, the water column may be "pumped" above the upper part of the moonpool with undesireable effects on personnel and equipments installed on deck.

3.2.2 Risk assessment

As a rule, the risk of resonant pumping mode is to be considered when:

$$0,6 < \frac{T_m}{T_w} < 1,3$$

where:

T_m : Natural period of the moonpool, in s

T_w : Wave peak period, in s.

The natural period of a constant cross-section moonpool can be evaluated as follow:

$$T_m = 2\pi\sqrt{\frac{h + K\sqrt{A}}{g}}$$

where:

h : Water column height in the moonpool, in m

A : Cross-sectional area of the moonpool, in m²

g : Gravity acceleration, in m/s²

K : Factor depending on the moonpool cross-sectional shape, to be taken equal to:

- 0,479 for a circle
- 0,460 for a rectangle (b/l = 0,5)
- 0,473 for a square

obtained by linear interpolation for rectangle shape with b/l values between 1 and 0,5.

l : Moonpool dimension parallel to the considered direction, in m

b : Moonpool dimension perpendicular to the considered direction, in m.

Figure 15 Guideline for moonpool assessment 1/2 (Buerau Veritas, 2016)

We then need to compare it to wave periods to see if we will risk resonant pumping.

$$0.6 < \frac{T_m}{T_w} < 1.3$$

$$\frac{0.6}{9.3} < \frac{1}{T_m} < \frac{1.3}{9.3}$$

$$0.065 < \frac{1}{T_m} < 0.146$$

We will have risk of pumping for wave periods between 15.4s and 6.8s. These values should be checked against wave scatter diagram for the oceans the vessel operates in to review the possibility for pumping. If we had a risk of pumping, altering the tube profile would give the greatest change.

We are not sure if the tube angle will illegitimatize the results of the calculation, so we would need assistance from an expert to review our result.

The natural period of a varying cross-section moonpool can be evaluated as follows:

$$T_m = 2\pi\sqrt{\frac{M_{eq}}{\rho A(0) g}}$$

where:

ρ : Sea water density, taken equal to 1,025 t/m³

A(z) : Cross section area at z level, in m²

A(h) : Cross section area at water level, in m²

A(0) : Cross section area at moonpool bottom level, in m²

M_{eq} : Equivalent mass, in t, to be taken equal to:

$$M_{eq} = \rho A(0)\left\{\int_0^h \frac{A(h)}{A(z)}dz + \frac{A(h)}{A(0)}K\sqrt{A(0)}\right\}$$

Figure 16 Guideline for moonpool assessment 2/2 (Buerau Veritas, 2016)

Moonpool Summary

We chose an elliptical profile for the cleansing. Our dimensions where based on eyeballing the moonpools in 'Geir' and 'Atlantic'; tube profile radius of 1.5m*1m; top section a rectangular profile with sides of 4m. A tube angle of 17° based on the moonpool patent (Canada Patentnr. 2307650, 2006)

Autoline system

An auto line system is an automated version of the traditionally longline fishing method. It is mostly used for large scale fisheries with high capacity. The line has evenly spaced hooks attached to it, and the hooks are hanged on steel rails called magazines during transportation and other non-fishing operations.

When setting the line, an anchor drops dragging the line subsea. A "line setter" maintains the tension of the line through an "auto baiter" which apply bait on hooks. The "auto baiter" cuts evenly sized pieces of bait and as the hooks are dragged through it hooks the piece as it triggers a new cutting and baiting process.

After the line is set, it is time to begin the hauling process where the line runs the components as it was laid out in *Connections*. Some haulers are equipped with hook cleaners, if not this has to be done by a crewmember. After going through the hook separator, the hooks are hanged on rails, where a worker is check for faults and sends the hooks into magazines, where its stored for a new setting process. Line retrievers are used to hold tension in lines while releasing the load on the haulers. (Mustad Autoline, 2020)

Storage

Storage of the hooks is done in Mag-packs. The hooks are hanged on rails with the lines hanging from the hooks. The rails are cut in magazines at a certain length, arranged in a carousel. As one magazine is either emptied or filled, a new magazine is readied. It is not clear how many hooks a magazine can carry, but from visits at *Geir* and *Atlantic,* we assumed that a mag pack area of 80m^2 would accommodate about 70000 hooks; when we asked the owners for dimensions, we received some drawings, and gathered that we needed an area of 65m^2.

The Mag-pack comes in sets with pre-determined magazine lengths from 3-5m. The depth of the carousel can be costumed to reach the hook capacity. A formula where n= number of Mag-packs and L is taken from a list of predetermined magazine lengths; one can find the depth needed.

$$\frac{65m^2}{L * n} = depth$$

Setting

Setting the hooks are done by releasing the line as new magazines is prepared. The hooks go through an auto baiter, which puts bait every hook. The auto baiter is triggered by each hook going through, as a crewmember fills up with bait. Based on visits and Maustad Autolines' catalogue, we estimated a required area of 4m^2 including space to manoeuvre around the machine.

Receiving

To receive the line, fishermen needs access through the side of the ship to pick up the buoy. A hauler drags the anchor to the hatch, then the line has to be led into the moonpool as stated in *Connections*. The line is connected to a rail arrangement dragging the line from the side hatch up and trough the moonpool. Although the rail will cause a discontinuity it the shipside, the effect of the hydrostatic elements of the ship are neglectable. The hauler takes up as circular space with the diameter of 2m

Hauling

When the lines are retrieved a hauler hauls the lines with fish. The lines go between two cylindrical bars too narrow for fish to pass, which unhooks the fish. Then hooks goes through a hook cleaner before the line is dragged through a tube to a hook separator. The hook separator aligns the hooks on rails going to the Mag-pack. The hook separator is a slim structure occupying an area of 500mm*1300mm but needs additional space for crewmembers to move around.

Arrangement

Most of the components require little space, and its arrangement are adaptable. Hauler needs to be at the moonpool, so the line can be hauled in the tube centre. The auto baiter needs to be at the stern where the lines are released. These are the only fixed components of the auto line system. The hook separator and Mag-pack are more flexible concerning its positions, however arranging these components in a tangled arrangement will cause much complexity to the floor arrangement in general, might reduce efficiency and add work/maintenance.

Machinery

The machinery often fills up the rest of the room aft of the storage as it entails a lot of different components and additional space for maintenance is also needed. The main part of this system is the main engine and propulsion unit, so this is the first thing we investigate. There are numerous different alternatives and combinations, so we will try to include as many as possible. However, our focus will be on the usual solutions.

When it comes to rules and regulation do fishing vessels only required to follow the regulations outlined in SOLAS by international maritime organization (IMO). SOLAS were large and extensive. Though fishing vessels are not obliged to follow the amendments outlined in MARPOL, can be smart to utilize, due to the important topics regarding the environment. Their amendments are as outlined below.

- Annex I (3)
  Prevention of pollution by oil
- Annex II (3)
  Control of pollution by noxious liquid substances
- Annex III (1)
  Prevention of pollution by harmful substances in packaged form
- Annex IV (1)
  Prevention of pollution by sewage from ships
- Annex V (2)
  Prevention of pollution by garbage from ships
- Annex VI (2)
  Prevention of air pollution from ships

Propellers

A propeller converts the energy from the machinery into propulsive power. There exist several types of propellers with each of their own advantages, and our job is to determine which one them suits this vessel. Following the mission based on the owner's requirements can a main propeller be determined. Furthermore, should we also assess a retractable azimuth thruster in front, as the shipowner has stated in the requirements that he wants. That should work as a side thruster while retracted, increasing the manoeuvrability at low speeds, and as a Safe Return to Port (SRTP) unit when deployed. Under are some of the relevant propellers' advantages outlined

Fixed Pitch Propeller

FPP is the most prevalent used propeller today. It is commonly cast as one mono-block, but can also consist of separately mounted blades. The propeller is typically uniquely designed for each ship according to the ship's wake.

The benefits of having an FFP are the low cavitation and high efficiency around the design speed, if designed correctly. If designed poorly on the other hand, these "benefits" can become disadvantages.

Controllable Pitch Propellers

CPP is a propeller that can change the pitch using hydraulic pressure. Since the pitch determines the load on the propeller, changing the pitch gives it a broader range of efficiency than an FPP. It also gives the skipper more manoeuvrability as the power in one given direction can be more rapidly changed. In addition, can the blades be easily replaced if needed, because the blades are bolted on individually (Nerland, Marine Hydrodynamics - Propulsion Part 4 of 4, 2017).

The drawbacks of a CPP is the complexity of the mechanism. Making them more expensive as well as a higher chance of error in the mechanism and higher repair costs. Since it is not primarily designed for a given design condition and has a more massive hub than FPP, it less efficient in that condition. Hub sizes are 146% to a maximum of 200% larger, which gives it a maximum of 5% open water efficiency loss. (Nerland, Marine Hydrodynamics - Propulsion Part 4 of 4, 2017).

Ducted Propellers

A propeller can be encased in a cylindrical cone for two different purposes: either for increasing the fluid flow or because of restricted room. Increasing fluid flow increases the efficiency at low speed and high force operation, because the duct helps accelerates the water to ease the load on the propeller and generates a separate thrust.

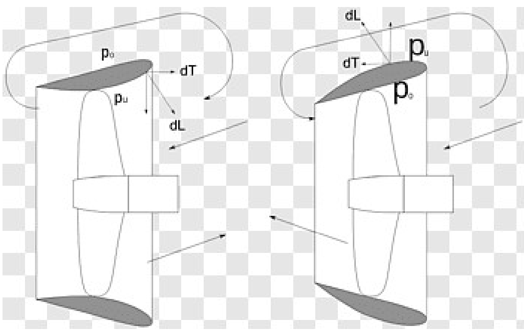


Figure 17:Two different profiles for a ducted propeller

In trawlers, it is shown that the duct can increase the thrust with 30% in 5-knot speed. Though in and above 12-15 knots it is shown that open water propeller has higher efficiency, due to increased drag. (Nerland, Marine Hydrodynamics - Propulsion Part 4 of 4, 2017)

Azimuth and Azipod Propeller

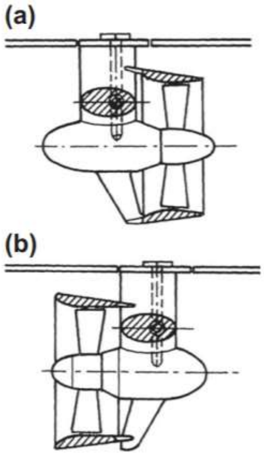


Figure 18: Azimuth thruster types (a) pushing, (b) Pulling unit, (Carlton, 2012)

Azimuth defined as a rotating pod with propeller driven by mechanical transmission (Z-drive). While an azipod has an electric motor and a short shaft enclosed in a pod. They can be used as both main propulsor and/or thruster units. They can be a free propeller or a duct propeller, for better thrust at slow fluid flow. This can be configured in two typical ways: a pushing unit in front of the housing or a pulling unit at the back as seen in figure 18. They can also have either have tandem or contra rotating propellers for added efficiency.

Azimuth has the benefit that it can rotate and therefore produce forces in any direction, which gives it great manoeuvrability at lower speeds. However, it can give weaker directional stability at higher speed (Nerland, Marine Hydrodynamics - Propulsion Part 4 of 4, 2017). It also consists of a streamlined body and rudder, which makes a separate rudder not needed. The directional stability can be improved by making a more streamlined body. For a ducted azimuth the pusher in a test produced 10-20db more noise compared to a puller. A pusher had more efficiency than the puller, but the cyclic variation in cavitation was higher. (Carlton, 2012)

Propeller Type	Benefits	Disadvantages
Fixed pitch propeller (FPP)	High efficiency at one speed	Less effective at several speeds. Cannot change blades
Controllable Pitch propeller (CPP)	Broader range efficiency	Increased hub (146-200%) Higher Maintenance and Cost
Ducted Propellers	Higher thrust efficiency in lower speed	Lower efficiency than an open propeller at (12-15kt) Higher Cost
Azimuth / Azipod	Higher Directional Stability at lower speeds	Weaker directional stability at higher speeds

Table 2: Propeller Types

Main-Machinery System

Choosing the optimal main-machinery system is a complex task, as many elements need to be considered. The primary factors influencing the choice should be equipment cost, reliability, maintenance, efficiency, and adaptability.

In a line-fishing vessel, the shipowner frequently is the operator of the vessel. Therefore, could a shipowner have preferences for a typical system based on earlier experience. The shipowner, in this case, has experience with a diesel-mechanic layout, but he is interested in seeing the possibility of using a hybrid solution. He has told us that the vessel will presumably be operating at:

60% of the time at from 1-3 knops, during hauling.

20% of the time at steady 9 knops, during setting.

20% of the time at transit speed, typically 11 knops.

During a typical day it will operate throughout all these conditions, and at the ratio mentioned in *Operation Profile*. Then it is our job as designer to consider and evaluate the different main machinery layouts that best satisfy the shipowners wishes.

The initial cost is not the only considerations for choosing the optimum layout. Other factors are: maintenance cost, size, weight, necessary manning levels, availability of spare components, and operating cost for instance 20-25 years into the operation. The diesel engines are a central part of the layout as it mainly determines the size of the machinery space required. One must give enough space for both the installation and maintenance of the engines, involves giving sufficient headroom and adjacent spaces. (Molland, The Maritime Engineering Reference Book, 2008)

Diesel Engine Type

The shipowner requested a main engine that is has less than 800 rpm. Not a high-speed diesel engine. So, two remaining options slow-speed or medium-speed engine where analysed.

Slow speed is defined as engines running in between 70 to 200rpm (Æsøy & Langset , 2007). The main concept in the cycle is that the fuel self-ignited, leading to less cycles called 2-stroke. Slow speed engines have a higher thermal efficiency in large engine, by they can burn lower quality fuel due they have more space for the combustion. They also have less parts resulting in more reliability and less maintenance (Kyrtatos). The negatives with are because they have a higher unit cost, since they have less development than medium speed engines (Molland, The Maritime Engineering Reference Book, 2008). Also burning a lower quality fuel can create more NOx gasses.

Medium Speed are defined as engines running from 300 to 900 rpm (Æsøy & Langset , 2007). Difference from 2-strokes is that it needs an extra two strokes since it manually ignites the fuel. Therefore, they offer a higher power to weight or volume ratio than a slow-speed engine. Medium-speed engine can faster change speed and direction. They have also very good NOx values. General rule is that the longer stroke cost more per KW of output. Though since a higher stroke ratio can eliminate more NOx emissions, exists a good balance (Molland, The Maritime Engineering Reference Book, 2008). They tend to go to a higher stroke. Since they have more moving parts comes the disadvantages from increased fuel consumption, increased engine noise, reliability and maintenance. Also, since they need to run on higher quality fuel does that lead to higher fuel costs.

Mechanic Propulsion

Mechanic Propulsion is when the main propeller-powered directly from one or more diesel engines. While auxiliary loads are powered from a separate system, as seen in figure 19.

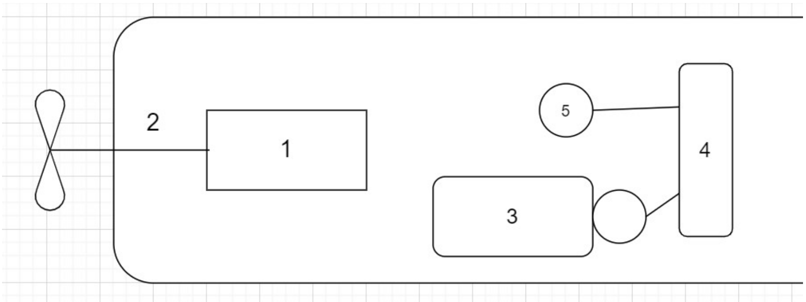


Figure 19: Mechanic Propulsion

The system functions as follows. (1) The mechanical propulsion engine delivers all the power to the (2), propeller either directly or through a reduction gear. If it is a slow speed engine, then the engine can power the propeller directly. If a medium speed engine powers the ship, then the rpm must be reduced by a reduction gear. All the electricity for (5) auxiliary loads and azimuth is powered by a (3) generator separately. (4) A transformer converts the electricity to correct frequency and distributes it where it is necessary.

The benefits of mechanical transmission come from a small number of parts, resulting in higher fuel efficiency, less initial cost, and reduced maintenance cost (Geertsma, Negenborn, Visser, & Hopman, 2017). Higher fuel efficiency comes from the small amount of energy conversion, only through the axel, reduction gear, and propeller. Although a diesel engine is only efficient at 80-100% of the design speed, running at a lower result significantly lower efficiency. The plot for specific fuel consumption clearly shows the quadradic increase of power compared to load on the engine rpm.

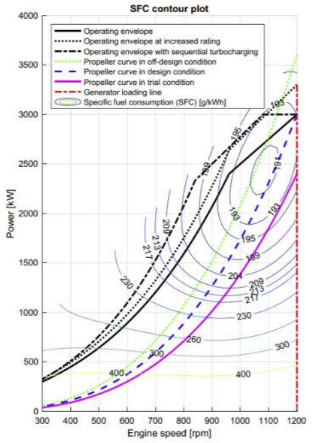


Figure 20 SFC contour plot (Geertsma, Negenborn, Visser, & Hopman, 2017)

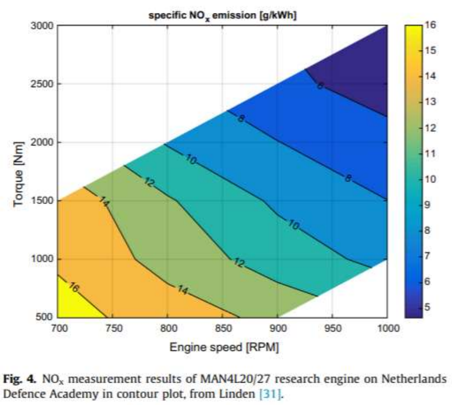


Figure 21 Specific NOx emissions (Geertsma, Negenborn, Visser, & Hopman, 2017)

It is possible to increase the range of efficiency by utilizing two engines or a father-son layout (Molland, The Maritime Engineering Reference Book, 2008). A father-son design consists of one large and one small propulsor engine, which increases the productivity at more speed, by combinations of those two motors at their optimal performance. Those engines need to be of the same type with different number of cylinders. (Vilmar Æsøy, personal communication, 17 January 2020)

Electric Transmission

A fully diesel-electric system is when both main propulsion power and auxiliary loads are generated from generators, often diesel powered. Although the source of the energy to the generator could potentially be something else.

The biggest advantage with a fully electric system is that electricity generates instant torque which gives the captain great manoeuvrability. An electric engine will also run more efficient at a lower speed compared to mechanic-transmission. The disadvantage appears when running the engine at high speed since this will have a reduced efficiency compared to mechanical transmission due to the increased number of energy transfers. In addition, can instant torque become a disadvantage at sea in that it is shown that can increase cavitation (Geertsma, Negenborn, Visser, & Hopman, 2017), which might cause destruction of the material of the propeller and hull.

Hybrid

Hybrid can consist of both electrical and mechanical propulsion, but also possibility to generate electricity from mechanic propulsion. Therefore, a hybrid system with the correct design can combine their advantage by complementing each other's weaknesses. That is done by designing a correct power management system (PMS) (Geertsma, Negenborn, Visser, & Hopman, 2017). The disadvantage is that both the initial cost and maintenance can be greatly increased, since the complexity increases significantly as seen in figure 22 .

Due to high complexity and no previous knowledge professor Vilmar Æsøy were questioned to present us how such a system can be built up. He drafted up a system and explained basic function of each component. Figure 22 represents the draft obtained.

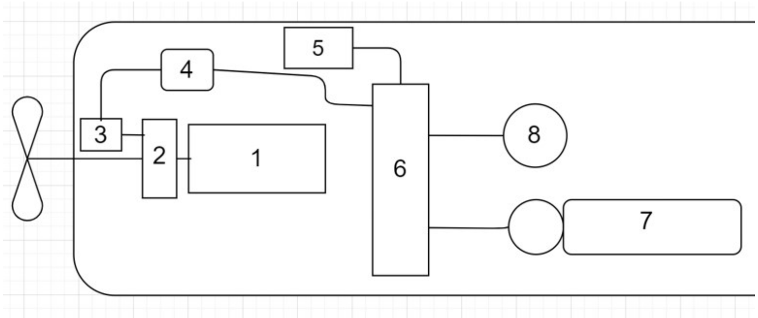


Figure 22: Hybrid

(1) Diesel engine rotates the axel connected to the (2) reduction gear. Reduction gear is both connected further to the propeller and (3) a shaft generator. The shaft generator can either generate power from the main engine or deliver power from (5) a battery or (7) a generator. (4) A variable frequency drive is needed to convert the electricity to the correct frequency. (6) Transformer that control the power distribution, for example to the (8) auxiliary systems. Transformer also converts the electricity to a correct frequency between the systems. To be noted that the shaft generator can be connected directly in between the main engine if there is a direct connection from propulsor engine to the propeller. (Vilmar Æsøy, personal communication, 17 January 2020)

- *Shaft Generator*

  Shaft generator are the fundamental component for a hybrid system, since it connects the mechanical components with the electrical system, and vice versa. To generate electricity from the mechanical components or utilize the electrical power, is it either mounted on the propeller shaft, or connected through a reduction gear.

  Some shaft generators can assist the main engine, taking power from batteries or generators and turning the main axel. That can be beneficial when operating in heavy seas, as small bursts of power can assist the engine, allowing it to have a more stable load and therefore increasing the durability. Also, can give the captain more ability to achieve a higher speed than the propulsor engine can achieve, by inserting extra electrical power, which are called a “Booster” function. Function of consuming electrical power from the ship are called power take out (PTO). This power function can also reduce impact on the environment by implementing a battery, since this battery can be charged at shore, and then generating thrust from PTO.

Other shaft generators can function by creating electricity from the axel or reduction gear, to deliver power to the ship, Shaft generator can stand for the whole electricity production. (Prousalidis, Hatzilau, Michalopoulos, Pavlou, & Muthumuni, 2005) concluded that can be more cost efficient than generating power through the generator if the main engine is a low speed diesel engine. Connecting a generator to the main shaft can cause issues if the axel rotates at different rates. To achieve a constant frequency, the axel must rotate at a constant speed. Without a way to transfer energy is it impossible to have the axel rotating at the same rate, at a lower speed. The energy can be transferred to another axel connected to the shaft generator, which is achieved by a two-speed gear. The lower speed should not be designed for less than 80% of the design speed (Hauland, 2016).

Shaft generator can increase safety by utilizing electrical power for creating thrust, when a mechanical component does not function, or vice versa. That function of a hybrid system is called power take home (PTH).

It is to be noted that shaft generators can include both PTO and PTI. At a higher price.

Determining Most Applicable Machinery System

To determine if hybrid machinery system is a better solution than a mechanical system, is it beneficial to conclude which system are most optimal for each deciding factor.

Best System Regarding	
Equipment Cost	Mechanical
Operability	Hybrid
Reliability	Mechanical
Maintenance	Mechanical
Efficiency	Hybrid
Environmental	Hybrid
Safety	Hybrid
Space	Mechanical

Table 3: Determining Most Applicable Machinery System

Mechanical machinery propulsion system involves less components than a hybrid. Also, hybrid system complex components which are more expensive.

A hybrid system would give better operability, due to the speed flexibility. For example, with a small burst of energy with function PTO, the shipowner can maintain the same speed at high seas. As a designer one can give more design speeds, due to multiple functions.

Mechanical where determined to be best system for maintenance. Due to fact that less complexity both lead to faster repairs, and understanding what component is malfunctioning. The reliability decision is based on that reliability can be increased by decreasing complexity of maintenance. At the same time maintenance on a hybrid system can occur less, due to reduced stress on the main engine, decreasing need for amounts of maintenance.

Hybrid system where deemed the most efficient and most environmentally friendly. Since utilizing a mechanical machinery system would be very inefficient at hauling, due to the speed being lower 70% of design speed transit. Also, since the vessel vary speed many times a day, at least 3 times. Also, since the space is not wide enough, and length limited by the moonpool, is it probable that a father son layout is a not a practical solution to the problem of efficiency and operability. Increasing efficiency has a clear benefit on the environment as well.

At the same time being possible to utilize a battery to generate thrust would be better for the environment. Clearly running the main engine at optimal speed during setting, is an added benefit as reduced efficiency are generated.

Selective Catalytic Reduction (SCR)

SCR is used to reduce NOx emissions from a diesel engine. This is a necessary technology to reach the NOx emission regulations. It can normally reduce the NOx gasses by 90% (Wärtsilä - Selective Catalytic Reduction (SCR), 2020). The exhaust goes through a diesel particulate filter to take out the big particles. Further it goes through the selective catalytic redactor which a Catalytic reduction liquid which consist of urea and water mixture is mixed into the exhaust. Then the NOx is reduced to near zero levels, and then released out through the exhaust pipe.

Batteries

The battery system is divided in two under systems. Where one is the modules, containing the battery cells, sensors and battery managing systems. Batteries can release gases and in case of abuse or failure they can combust and they are sensitive to the thermal condition. The other system is therefore fire protection, ventilation, thermal control.

Example on a battery pack with capacity of 2400 kWh I 2000 mm high 8600mm long and 1200mm wide weighing 23,3 tonne

A smaller battery with capacity at 125 kWh, height of 2241 mm, width of 865 mm, depth of 738mm or a horizontal arrangement, height of 1260 width 1730 and depth 738.

Waste Management

From our visit on the Atlantic we saw that trash-compactor and/or incinerator, all in all takes up about 3x3 metres.

- *Incinerator*
  Since a long-liner operates for a long time does it require limiting the amount of waste stored. A way to remove organic material is by burning it for hand which is done in a shipboard incinerator (Wärtsilä, 2020). The shipboard incinerator should be placed close to where the waste is generated. In this long-liner that is on the accommodation deck.

- *Sewage Treatment*
  70 litres black water and 120 litres grey water is accumulated per person a day (Molland, The Maritime Engineering Reference Book, 2008). Physical-chemical sewage treatment utilizes chemical to process the sewage into more

compact to be stored in a tank (Wartsilla - sewage treatment, waste treatment, 2020). That tank needs to be emptied every 14 days, since this vessel operates 30 days at a time is this sewage treatment not possible. Electrocatalytic oxidation breaks down the sewage into small particles using electrolytic cells which oxidizes the sewage. (Wartsilla - sewage treatment, waste treatment, 2020)

Auxiliary Machinery

Some other components that might be placed in the machinery could be: garbage compactor (or an incinerator), sewage treatment, oil/water separator and an oil heater

Accommodation

There are famously a lot of exchanging of personnel working on long-liners. This is assumed to be the result of the hard work that goes into the fishing and processing. Therefore, will there be a focus on making the work more tolerable in addition to making the accommodations better.

Cabins/Staterooms

For our hardworking staff, there needs to be a space each of them can have on their own. A space for them to sleep, regroup and recharge their battery. They will be working long days, and the work they will be doing is a hard so at the end of the day they truly deserve some good rest. The cabins will be there to offer them sleep and relaxation. Filled with a bed, seating, a head and a window that is letting the lethargic sun breath in. There are some rules and regulations concerning the staterooms of fishing vessels, this must be followed and we intend to do so. Nevertheless, we aspire to put our intentions to creating a comfortable environment for the workers. As this will allude potential workers to work on this specific ship instead of a similar vessel with inferior comfort levels. In order to create this type of comfort, we must not limit ourselves to what is acceptable (from the rules), and rather ask ourselves what would the crew want?

Our vessel will end up being over 45 metres so there needs to be a separate stateroom for each of the crewmembers (Lovdata, 2018, s. 5 (Kap 11.)). This is also a part of the demands form the shipowner: 20 single-cabins for the six officers, eight crew and potentially six scientists.

The cabins should not feel cramped, there needs to be enough area for a person to comfortably move around. Inside the cabin there should be space for a single-bed, closet (lockable), seating for two with a table and a small head(on-suite, bathroom) (Lovdata, 2018).

Shaping the Room

When it comes to designing the cabins there are a lot to take into consideration. For a long-liner the accommodation is a secondary system, and therefore it has to make do with what is left to a larger degree than higher priority systems. It is common to place the cabins along the side of the vessel; this allows for windows that fill the cabin with light. However, in order to place larger quantities of cabins in a small area there is a danger that they will become quite narrow. This is especially a problem for larger vessel because you want to make use of the full beam. Nevertheless, when the beam becomes long enough you might be able to will the centre with other rooms. Although there would then be necessary to have two corridors.

Figure 23: Single Cabin with longitudinal bed

Our vessel can have a maximum beam of 14 metres, leaving probably 12 metres of inside beam to work with. As an assumption to get us started, however the beam of the ship will to a larger degree be determined by the storage space, since this is of a higher priority.

- *Bed*
  The beds will be at least 1.980 metre times 0.9 metre and there needs to be 0.78 metres from the edge of the bed to the wall (Lovdata, 2018). Furthermore, it is considered to be more comfortable to have the beds placed in the lunitidal direction due to how the ship behaves in waves.

Galley

The meals served to the crew will be made in the onboard galley. The capacity needs to be enough to serve up-to 20 persons, which is the capacity the ship must accommodate. The galley needs to have supply-rooms, at least one for frozen food, one for cooled and one for dry. Vessels longer than 24 metres needs to have a separate room for galley. Working in the galley there will be the chef himself and a "helper" or kitchen porter.

The galley of the Atlantic where about 3x5 metres so that will be our first estimate of the area needed. The vessel will be around a month out at sea at the time and needs to have sufficient rooms for food storing. The storage rooms occupied about 8x5 metres, although all these values are based on eye-measuring. After getting some estimation formulas, we calculated that we need 120 m^3 with food storage all together. 40% of dry food, 25% of cooled food and 35% of frozen food.

Dining

Food is served to the crew in an eating location, where there must be at least a table area of 0.6x0.4 metres per person and should have the ability to seat all the people onboard. Although for the most time there will some vacancies, they'll be sleeping on shifts and some will always be working for example driving the vessel.

Corridor

The corridor must be at least 0.9 metres. Although we might start with 1050 as way of achieving a roomy feeling, in addition to having some wiggle room with the width of the walls. If there comes an extra need for strength in the inside walls, we will be able to increase the plate and stiffeners a bit without having to rearrange the GA.

Lounges

Some comfortable seating places is useful to have. Letting some of the crew get together after dinner for example, maybe to watch a football match or just to converse about other things than the job. Since the dining is set-up to take all members at once the favourable thing would be to have multiple lounges that seat 4-6 people. This opens the possibility for different style of lounges, some more open and some closed off. There are some rules that if smoking is to be allowed, it should only be in specified locations and should not affect they who do not smoke in any way.

Cinema/Gaming room

As specified in the demands from the owner there should be a room for watching movies or gaming etc. Being there to increase the comfort onboard and achieving a high living standard.

This could possibly be done in a way that simulate a cinema, with rows of chairs, a screen, a projector and a "banging" stereo. This was done in the Atlantic, however there is also the possibility of taking another approach. Perhaps making it more homely, with a big couch and coffee-table.

Figure 24: Cinema/ Gaming Room

Danish Seine

In the requirements set by our owner he asks for us to outline how a Danish Seine-arrangement could be implemented and what effects it would have. This arrangement should use vacuum-sucking to get the fish aboard.

Danish Seine in Videos

This type of fishing differs quite a lot from long-line fishing, as Danish Seine uses a net to capture the fish. This net is hanging out from the stern of the vessel, as the vessel moves slowly through the water. The net is filled up with fish to a certain point, which is predetermined. Where the Danish Seine net reaches its desired capacity the "extra" fish is able to swim out of the net. The capacity of the net is determined pre-fishing by placing a closing-device on the net at the desired volume. When the net is filled and the ship starts to haul it up the closer-device automatically releases due to a pre-determined pressure release (Havforskning, 2018). This allows better space for the fish in addition to lowering the risk of losing fish at the surface.

The restriction on the amount of fish caught in the Danish Seine enables us to control how much fish we get each time. Thereby avoiding the risk of filling the net up to a degree that is unmanageable, keeping the safety abord the vessel as well as assuring the quality of the fish (Lorentzen, 2018).

Demersal Seine Fishing

This method consists of two ropes and one net, where one rope is dropped down with an anchor and a buoy. The vessel starts move forward setting out the net and second rope before circling around back to the first rope, starting the collection phase. The ropes should have reached the seabed before the vessel starts to move, and when it starts to move the speed is about 1-2 knots. Circling back to the first rope, both ropes get closer together and the net starts to close in. The second rope gets picked up and the vessel continues to slowly move forwards until the ropes get to a certain distance apart. At this point the rope drums starts to haul in the ropes in order to close the wings of the net faster in the closing phase before the fish can be brought aboard (Madsen , Aarsæther, Herrmann, Hansen , & Larsen, s. 2).

VS Trawler

The Danish Seine nets are quite similar to the trawler nets; however, the wings are most often longer for the Danish Seine (FOA, u.d.). The wings are the side-pieces of the net that are not the main bag, they are used to close in more fish when the ship is in the circling phase.

Making Space

Danish Seine fishing differs from long-line fishing in several ways and for our vessel to accommodate the Danish Seine-arrangement, there has to be reserved significant spaces. We would have to evaluate if the changes should be done in the design or after production, based on need. Firstly, the arrangement needs two drums in order to release and hoist the net and ropes. The ropes will be pulled in from the stern of the vessel and the drums should therefore have a clear path to the stern. Another thing that takes up significant space is the vacuum pump for sucking the fish onboard.

Net drums

These are the storage of the Danish Seine-net onboard, there could be multiples of them if that is wanted. Which it most often is, perhaps one becomes damages or tangled. So, having one or more backup nets ensures operation goes on even after you encounter some problems. The Atlantic had space for three nets.

Researching Similar Vessels

Since there exist very limited number of long-liner are given such large dimension and technologically adapt. Since in a many vessel, consist hundreds of different main elements and components. Also understanding how those system where connected where needed as none of us working had ever been inside a long liner. For instance, how the line was brought from the hatch to the moonpool.

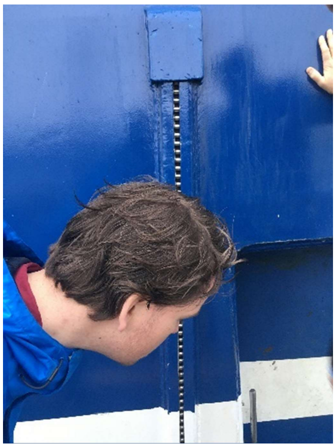


Figure 25: Line Guider and an engineering student who is examining this technology thoroughly

Visual Approximation of Main Components

The research combined with visits to *Geir* and *Atlantic* turned out to be successfully for understanding the main systems and components involved in the design. The major problem where estimating the size. Since most manufactures do not reveal the dimension on many components publicly, especially in the factory. To get those dimensions a trial of contacting firms was made, to mostly no response. Even though the dimensions for the main components in the machinery where publicly available information. Is a large part and space for the machinery from small components in the auxiliary system. Typically dimension for those are assumed from scaling up from earlier designs. Same problem regard other system as refrigeration system, ventilation and other systems involving small component, for example piping.

To create an idea for system, for example how the factory should be outlined. Some rough approximations needed to be made for many components. Since these problems were known before the visit to *Geir* and *Atlantic*. Did that benefit from the understanding of need to be taking rough visual approximations, during the visits. These are the foundations for estimating size of many systems in this vessel.

GA – Brainstorming

After we have established an understanding of the different systems included in a long-line fishing vessel and how they are dependent on each other, we can start forming solutions on how they might fit together in a general arrangement. In order not to disclose potential ideas and come-up with fresh ones, we try to take a step away from the commonality by brainstorm different GAs. To start the brainstorming process, we write down a few “titles” that represent what different visions we might want to purse in the design. Furthermore, these titles are elaborated with a few keywords before we draw potential solutions that could accommodate these keywords. The goal is not to come up with a perfect, or even functional, GA, it is rather to inspire ideas that can be implemented in the first draft of the GA. Thereby, using the potential that a bottom-up approach can give, and only after all of them were draw we wrote down the key point on each drawing. The points were then colour coded with green for positive outcomes, red for negative outcomes and some were left blank due to ambiguity.

The Cowboy

- Laid back, yet things still get done efficiently.
- Manual labour, solution oriented
- Nothing is difficult, nothing is surprising.
- Don’t worry about consequences, rely on experience.
- Nothing too fancy.
- Danish Seine

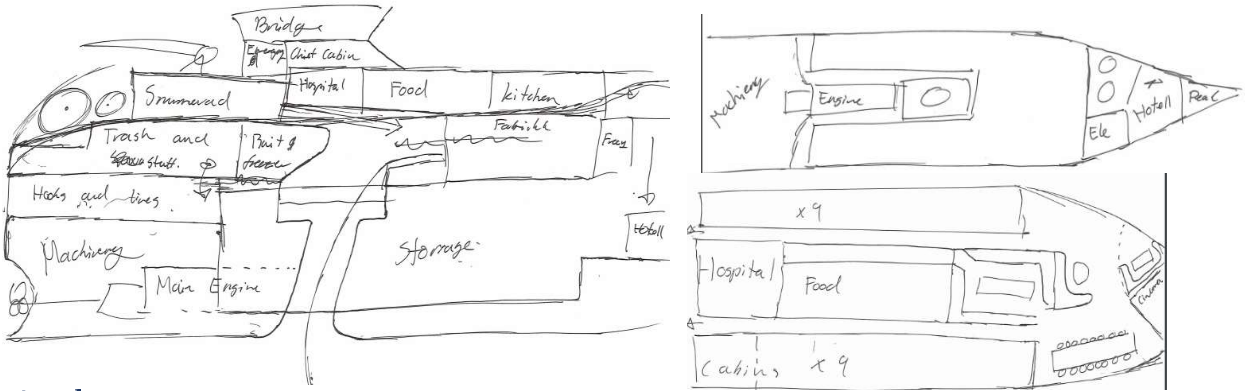


Figure 26: Cowboy I

Cowboy I

- Traditional
- Large fish capacity
- Easier handle of bait
- Needs more space for bait loading
- Large and bulky, much wind area
- Comfortable accommodation
- Larger wall area in storage -> more isolation
- Cramped machinery needs higher cb

Cowboy II

- No innovation
- Bait storage close to and on same deck as auto baiter.
- Takes away approx. 70m^3 of either factory or line space.
- Using space besides moonpool for AUX flattens machinery and moves weight forward and down.
- Storage further forward, effected by lines

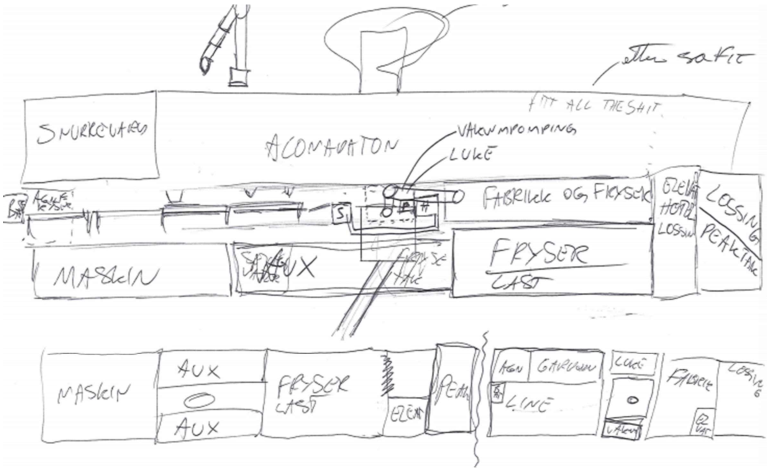


Figure 27: Cowboy II

Fish Quality

- Slow and steady wins the race
- Efficient factory
- Selling point
- 100% fish condition.
- No bad fish
- Top of the line equipment.
- Design from factory point of view.
- Straight factory? No gay.

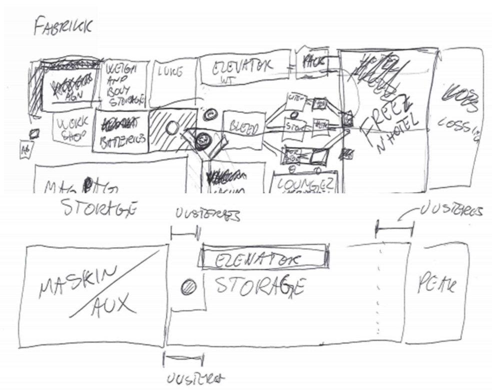


Figure 28: Quality I

Quality I

- Straight line factory split in small/large/too large.
- Freezer in front; weight forward but also increased beam forward (flare?)
- Pack on way to lift.
- Lift at centre of the ship.
- Factory might be too long. If so, send freezer to the side.

This also have Danish Seine possibility but is not required.

Quality II

- Needs two lifts down to fish hotel.
- Storage further forward, increased trim
- Quite low profile, good considering wind.
- Man putting fish in freezer needs to move from one side to the other.
- Needs double machinery for putting fish to freezer.
- Easy movement of bait to lines
- Moving storage forward reduces the capacity
- Using the length of the vessel.
- Complicated to adapt for Danish Seine due to placement of dining/kitchen

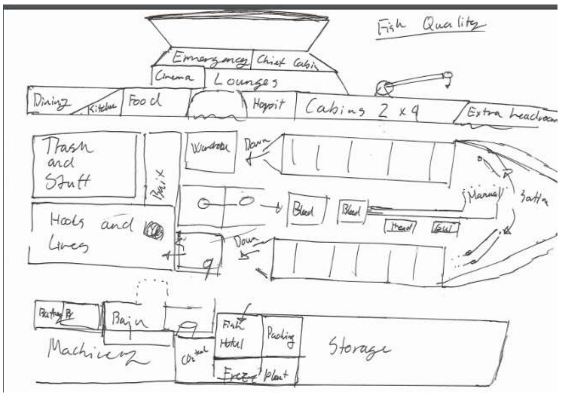


Figure 29: Quality II

Quality III

- Cabins all around the stern.
- Lower forecastle
- Even worse for Danish Seine than Quality II
- What to use the space behind the hospital for?

Rogue Sea Conditions

- Difficult weather in the seas in question
- Focus on ship handling
- Balance
- Tough and strong
- Top-Down ish, start with hull and insert system
- Think about the lines, so no dairy
- Small wind area
- Exceptional in head sea

RSC I

- Storage around moonpool, more isolation
- Bait far away from baiter
- Wind area in bow
- Almost Axe-Bow deign, cutting through waves.
- Perhaps too low power and speed in operation profile
- Much volume lost in bow area
- Battery laid out flat, worse for ventilation?
- Factory, freezers, hotel and packing on same deck and has a clear flow direction.

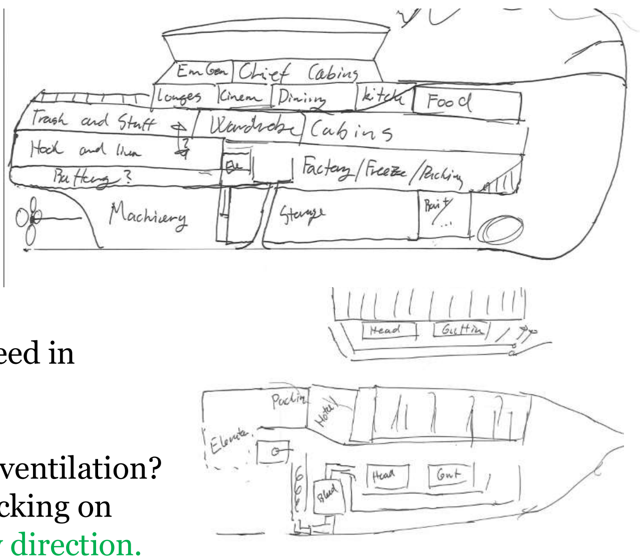


Figure 30: RSC I

RSC II

- X-bow and x-stern for rough conditions aft.
- Large wind area
- Cramped storage and machinery space
- Does not look like a ship

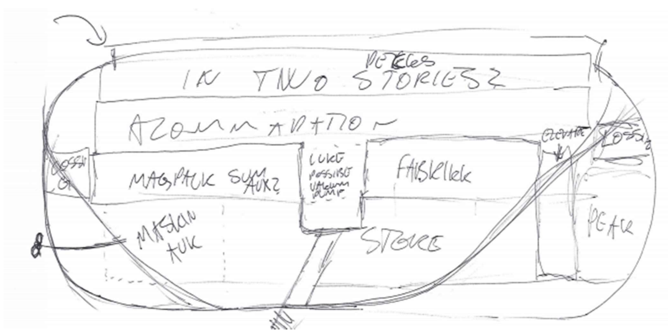


Figure 31: RSC II

Speed 3: The Unloading

- Short time in port
  - Owner does not like to spend time in port
- Focus on storage
- Specialized unloading equipment?
- Easy access to storage

Speed I

- Weight in storage lifted about 3.5 metres, possibly more, negative effect on stability
- Factory under waterline? Is that possible or even legal?
- Moves volume upwards, so possibly lower Cb in bow?

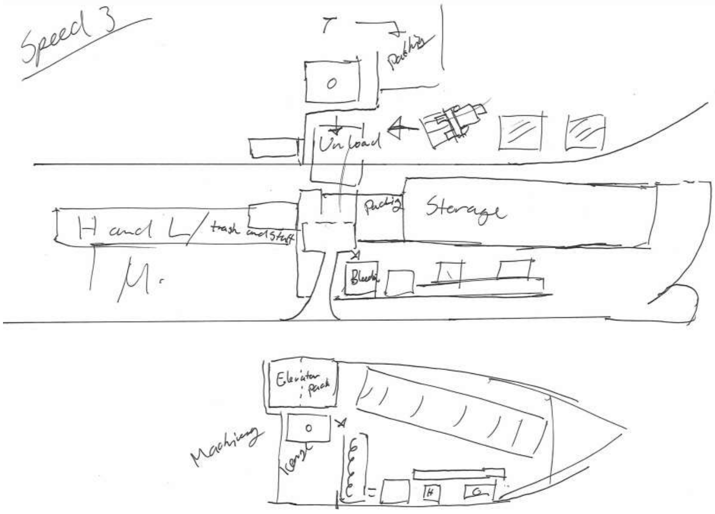


Figure 32: Speed I

Speed II

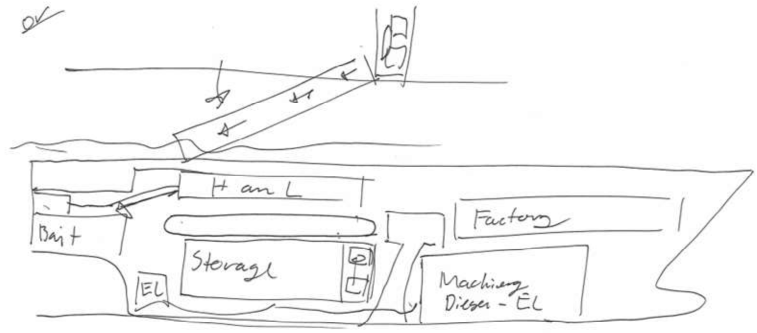


Figure 33: Speed II

- Complicated ramp
- Must have diesel electric, which is less efficient than hybrid.
- Lower storage length and breadth
- Higher storage capacity /unsafe for loading.
  - Can be forward trim, quite negative for propulsion.

Speed III

- Storage on main deck
- Ramp directly into storage
- Challenging stability
- Fish directly into the factory
- Less awkward walls for storage
- Workers need to work beneath waterline. Claus is that you?

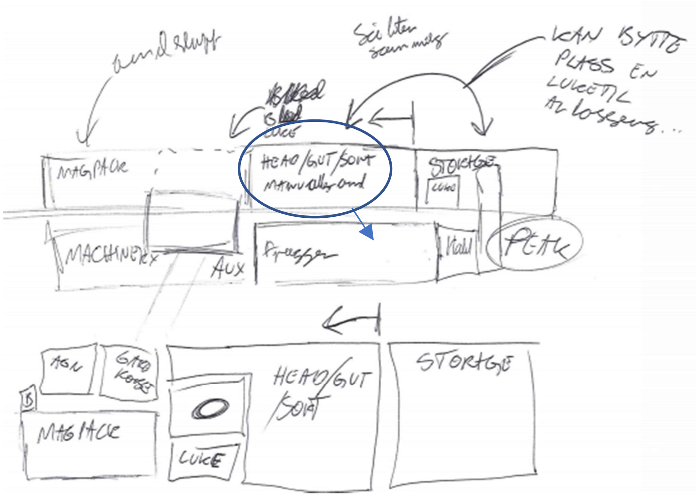


Figure 34: Speed III

Speed IV

- Unloading with ramps
- Ramps take a lot of space, a lot
- Unknown if it is faster than lift

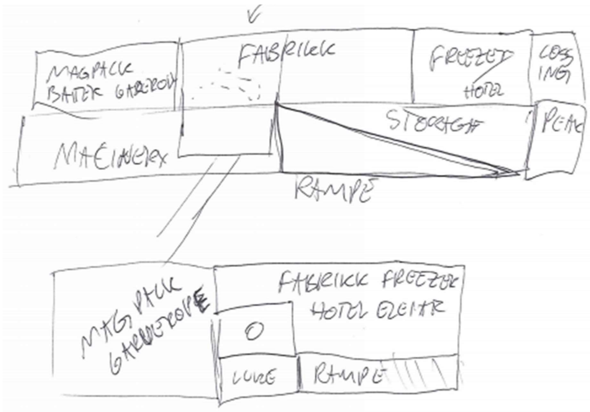


Figure 35: Speed IV

Possibility

Can be stored in the floor

- Higher machinery cost, more complexity

Speed V

- Paternoster lift/ski lift solution/carousel
- Larger than lift, smaller storage less money.
- Shorter waiting time for new pallet delivery
- More complexity, mtenance installation, price.

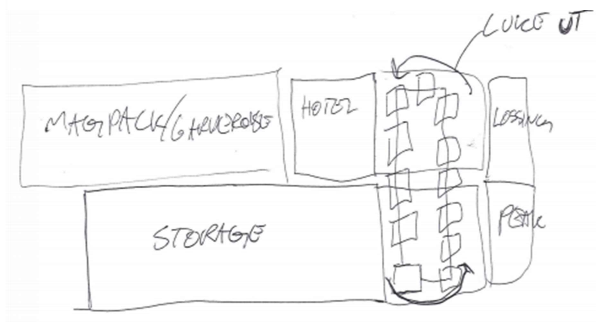


Figure 36: Speed V

Safety

- Try to keep people alive
- Minimize dangers
    - Throw line off-board
    - Grab line again for hauling
    - Handling the hooks on-bord
    - Squeezed between storage
    - Stuck in factory machinery
- High freeboard
- Closed environment
- Good stability/comfort.
- Emergency protocol
- Placement of hospital
- High onboard comfort with cinema/ gaming room

Safety I

- Setting done through moonpool
- Getting fish from hotel and packing to storage.
- Ramp takes a lot of space, 12 degrees ramp.
- Far from bait freezer to autoline

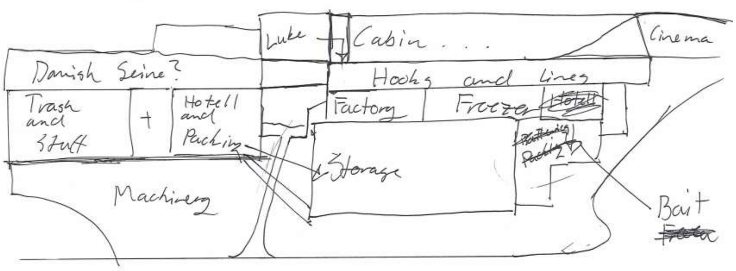


Figure 37: Safety I

Safety II

- Moonpool section surrounded by watertight bulkheads. Potential flooding reduced to moonpool area, and water can flow into the pool.
- Megapacks arranged along the width of the ship
- Hauling through the moonpool? Safer? What about the propeller?
- Need new bleeding tanks

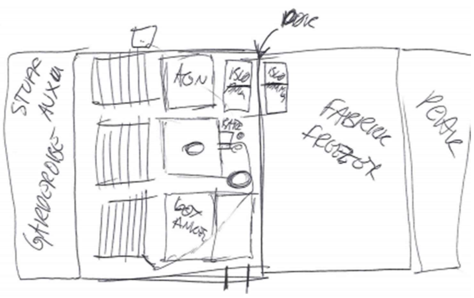


Figure 38: Safety II

Better Than Trawler

- Inspired Fishermen
- Relying on knowledge
- Quality of fish/product
- Less resources
- Adaptability
- Danish Seine
- Good looking vessel

BTT I

- Tall amidships
- Where is lift? Portside of moonpool?
- Factory higher up, due to pack and hotel being on own deck.
- When leaving shore there will be high trim due to much weight in stern
- Stepped deck ads complexity in the construction process
- Nice curves

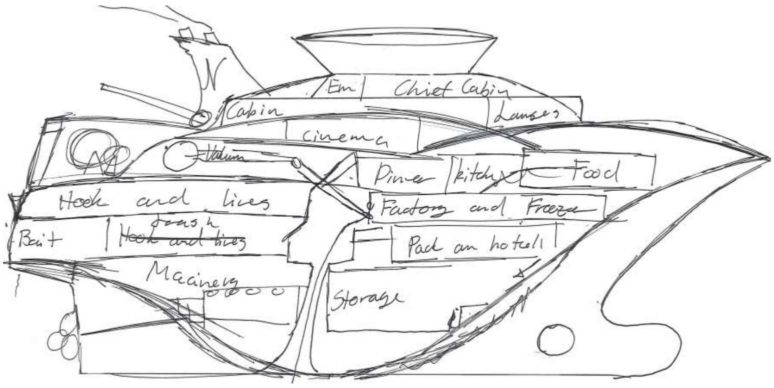


Figure 39: BTT I

BTT II

- BEAUTIFUL BOAT MH-HM
- Consumption tanks and storage in front to reduce trim when not carrying cargo
- Flares makes for awkward storage and machinery arrangements
- Larger roll, more uncomfortable
- Danish Seine arrangement

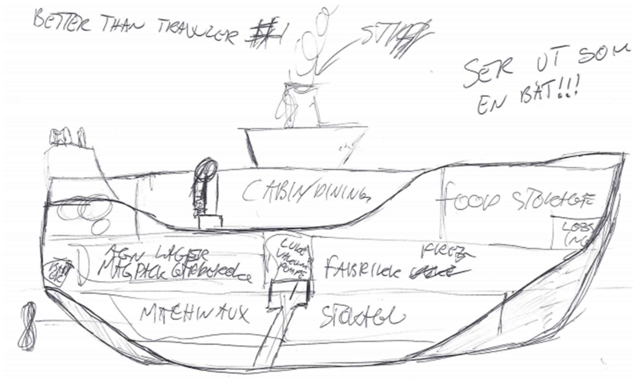


Figure 40: BTT II

Environmentally Sustainable

- Hybrid propulsion
- Using as much as possible of the fish.
- Sewage treatment
- Garbage treatment
- Energy efficient

ES I

- Tall
- Much movement in food storage, when waves.
- Moving AUX down and forward on each side of the moonpool, using the natural curves of a fishing vessel.
- Battery in front of moonpool, then tanks.
- Difficult to get to battery
- Good weight distribution?

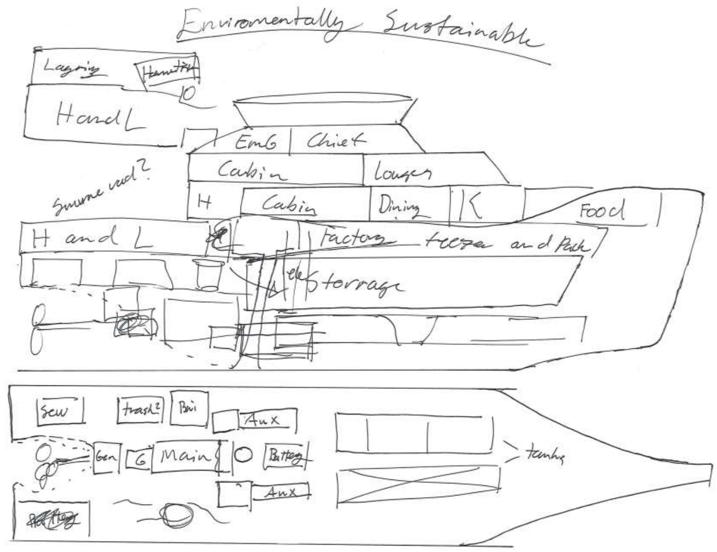


Figure 41: ES I

Machinery solutions?

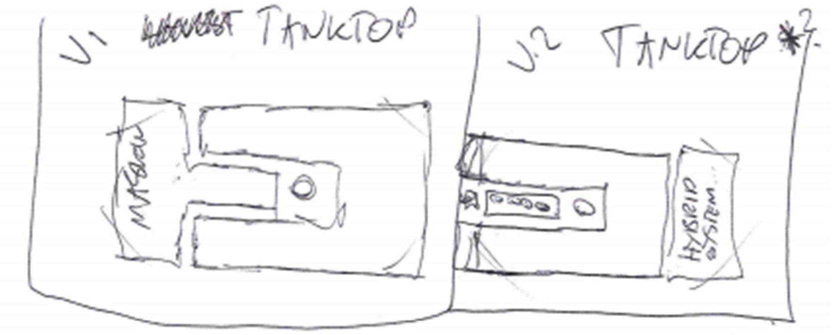


Figure 42: Machinery Layout?

- More centred (longitude) storage
- Awkward machinery arrangement
- Possible in need of two machinery compartments or place the components in height. Increasing the height of the centre of gravity. Way more machinery, two nox cleaners, two pipes, two noice redactors.
- Storage Hight under moonpool is reduced.
- Avoid storage along flares

Let's Make a Design

After the fresh ideas generated by the brainstorming is it necessary to bring it all back to earth. To evaluate the pros, cons and relevance of each idea, singling out the ones that can be implemented into the design. To do this, designer have to establish some guidelines to evaluate them by and determine what defines a good design. To create a good guideline, is it important to include the crew's viewpoints. As they have experience in the pros and cons of different aspects of design. The crew will not give clear answer to guidelines as they might have limited theoretical knowledge. Thereby applying the technical knowledge from the designer to create guidelines for the design. The guidelines focus on certain aspects of the ships systems, behaviour, and personnel. The features are solutions that correspond to the guidelines.

Understanding Waste with Lean

To understand how to create good guidelines, one must understand what causes inefficiency. In this section inefficiency are stated as a waste process. The eight wasted are explained below.

1. Overproduction
2. Inventory
3. Waiting
4. Motion
5. Transportation
6. Rework
7. Over Processing
8. Lack of Creativity

In the following section these eight different waste types are explained according to lean theory. Though since waste theory of lean where created by Toyota Factory which is a land-based factory. For example, a factory processing fishes at sea, have significant differences. Therefore, a decision was made to include some examples from fishing vessel next to the definitions. To get a comprehension on how these wasted function in a long liner.

Overproduction

Producing more products/components than necessary clearly illustrates wrong priorities. Overproduction can both be inside the cycle, where one process produces more than the next needs, or it can be at the end, meaning that you produce more than the customer wants. There is a danger that the things that are overproduced will "never" be demanded by the customer, meaning the business wasted a lot of capital on raw material and production costs for a product that will not sell. Predicting what the customer wants in the future is a dangerous game to play, as new technology arises, and popularities swing the demand changes rapidly with the times.

Overproduction is considered the worst of the eight wasted because it leads to other wastes. When there has been produced to many units, there comes a need to store all this units. Leading to excess inventory (NEHP, 2018).

In a longline overproduction occur mainly inside the factory. As rarely caught more fish than the storage capacity allows. During catch there is a necessary limit to how much fish that can be storage, as normally the freezers are the bottleneck. Freezers are bottlenecks due to the significant time it takes to freeze the fish as explained earlier. Not freezing fish are not an option as the quality demands rapid freezing time.

Overproduction can occur from building a factory for processing raw material of the fish as well. Since as explained earlier there are some clear cultural differences in which part of the fish it is acceptable to consume.

Inventory

Stacking up all the products and components that are not needed is a waste. Firstly, it is taking up space and space is not free. That space could be used in many ways that would be more useful than storing “unwanted” units. Furthermore, in those large stacks of products and components there might be some unknown problems hiding. As the products stacked up have not been in use there might be some fault in the manufacturing without anybody noticing, and the factory continues to produce without fixing the problem as well as storing broken products and/or components.

Depending on the size of the fish it varies how many fish must be headed and gutted by hand. With a large quantity of large fishes, and/or needed rework after the machine, the fish might pile up before the manually cutting.

Waiting

Letting people or machines be idle, while waiting for a task is a waste that can be a result of many different things. If the work is not divided evenly across the production one process might take more time than the processes after it, thereby will they need to wait on that one process. Dividing up the work properly will eliminate this and make full use of the people and machines. Another way waiting will arise is by delayed decisions from management or delayed information, it is therefore crucial to have good flow and structure of information and decisions.

Since the factory is on a ship and about 30 percent of the time goes to traveling to the and in between fishing locations, the fishermen will have some waiting in periods. As well as the setting of the line has become more automated there is not the same need for personnel at this stage as the fishing stage.

Motion

Without good planning of the layout of the factory time and resources will go to waste due to excessive movements. Visualizing movements inside the factory and highlighting what movements are value-adding and which are not is a good start to remove this waste. By structuring the workplan and placing the equipment in throughout location less time goes to waste for non-value adding activities.

There is quite a limited area onboard fishing vessel so the placement of the different equipment is thoroughly thought out to be as effective as possible, and as tasks becomes more automated by machines there moving to getting the necessary equipment are at a minimum.

Transportation

A produced item needs to get form the manufacturer to the consumer, this waste encapsulates movement of items, equipment and workers. During transportation the items may suffer damage and defects.

An unnecessary transport in the factory process could be elevating the fish to the factory. If the fish entered at the same level as the factory, some transportation could be eliminated.

Rework

Rework is a task you must repeat due to incorrect execution or a task required to correct a failed task. This process is an unnecessary waste since one must put in extra work on an item and does not add value to the item.

Examples on rework wastes in the longline process are "to large", "manually cut", "attach hooks" and "retrieve fish"

Over Processing

Over processing is when more work is put into an item than required. One does not have to put in more functions and details than the costumer wants or pays you for. Over processing is also be having capability to improve an item more than it requires, like unutilized tools and machines.

An example of this relating to the longline fishing ship, is the costumers asking for a headed and gutted fish, but you fillet it as well.

A potential over processing waste is "sorted", but this might be a required task for some costumers and adds worth to the item.

Lack of Creativity

Skilled workers doing unskilled task is a wasted resource. Workers might have ideas on how to improve a process, they can have a greater understanding on what takes time in a process and what's tiering. By not listening to workers one might miss out on valuable improvements. Simple and repetitive task does not suit humans well. If humans do unnecessary lifting or work with bad posture, will this leave this human with tiredness, boredom or even injuries. Therefore, it should be switched with robots. Robots can perform very well on tasks it is programmed to do, but not to improvise. Hence, a worst-case scenario the process will be stopped if something unforeseen happens. Making humans necessary for some processes.

Guidelines

To spawn new ideas and solutions, we have established some possible guidelines of which we want to improve our vessel with. The guidelines are chosen based on the requirement spec we received from the shipowner and conversations we had with owners and crew on *Geir* and *Atlantic*. The most important attributes they seemed to favour are;

1. Simplicity
2. Sea Attributes
3. Low Number of Workers
4. Storage Capacity
5. Shore Time
6. Comfort and Luxury
7. Safety
8. Wind Area

Out of these guidelines we looked for new design features, and evaluated the solutions created in GA brainstorming. In addition to comparing our solutions against typical long-line general arrangements, based on *Atlantic* and *Geir*.

We approached this task by solely looking for ways to improve upon the guidelines. This way we find solutions that we would not find if we limited our imagination to the general expectations of a good design. The result is often ridiculous and unrealistic solutions, but even these can spawn new ideas and alter the way we review a problem. With this approach we increase the design space we work within, allowing more possibilities for our design.

Simplicity

Sometimes advanced solutions can be elaborate and inefficient. Choosing solutions which are simple might be more efficient and cheaper.

One Lift Centre

With only one lift will reduce cost of installing and maintenance, as well freeing more space forward. Centring the lift means the unloading hatch will be installed at the middle of the ship where the ship side is a straight plane, which is an easier and possible cheaper solution than installing the hatch where the hull side is flanging and curving.

However, there is little space in the ships centre due to the moonpool, and factory somehow needs to circle back to midships, lowering the prioritization of a good factory flow.

Factory Flow
A good factory flow will simplify the work needed for the crewmembers. Reducing the distance between workstations will increase efficiency and allow the crew to observe all parts of the factory. A spacious factory arrangement will allow the crew to quickly move between the different posts.

Factory Below Deck
By setting the factory on the tank top, the fish do not need to be elevated from the pool to the factory but sent directly to the factory after its unhooked. This slightly reduce the time for the fishes spend over water before its frozen. There is will be no need to install a lift.

However, this will result in having the storage being placed on a high deck, ultimately resulting in great stability problems, and since the workers must work in tight spaces below deck, it can seem claustrophobic

No hauling lift?
In order to simplify the factory, could an arrangement of the factory and hauling space that would neglect the need to elevate the fish be explored.

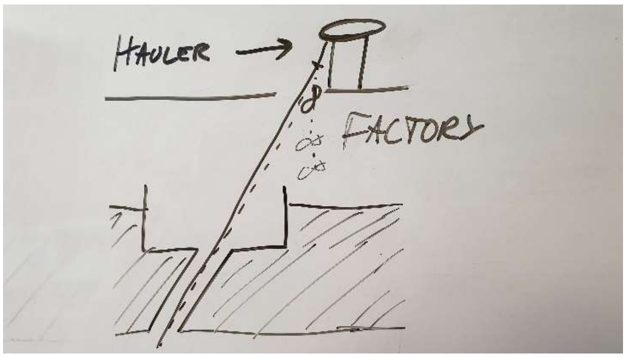


Figure 43 Hauler on deck above moonpool

Raising the hauler up, would mean the fish could be unhooked at the same level as the factory deck, effectively eliminating the need for elevating the fish from moonpool. This solution's problem can be that the fish must travel longer before it is released, thereby increasing the chance to losing it. Although it should be noted that the moonpool "damping-pool" has the advantage of working as a well of its own, so the fishermen have a good chance to retrieve the lost fish before it can swim out of the moonpool. However, this will mean that the fish will travel further over water and are therefore in risk of detachment. This will decrease the catch efficiency.

We were also toying with the idea to have a fully-automated section of the factory one deck down. With the use of a stunning machine and automated bleeding pools the first part of the factory could be done one deck down and then an lift connected with the bleeding-pool elevates the fish to the deck above where the rest of the process, with operators, happens. This would however require space from the storage and add more complexity than simplicity.

Sea attributions

Good seakeeping capabilities are important to avoid interruptions in the operation. Minimizing roll and pitch movements makes for a more reliable, safer workplace, possible to work during rough conditions and comfortable living situations. Controlling the trim can also increase the ships manoeuvrability and make sure the vessel operates reliably.

Consumption tanks and storage forward

Longline vessels usually have storage forward, which causes change in trim when loaded compared to unloaded. As the ship leaves the shore there is little weight at the front of the ship, but at the end of an operation it is a lot. This change of trim can be reduced by putting weight which is consumed in such position that reduce the trim. Such weights are bait storage, food storage, fuel etc.

Cooling

The traditional weight distributing tends to put heavy systems aft, which contributes to trim when storage is loaded or unloaded. Cooling system are heavy, and its positions would preferably be close to the freezers and freezing storage. By positioning the freezers forward, it could be advantageous to put the cooling tanks forward as well, causing the weight to be distributed evenly across the ship.

Machinery Distribution

Machinery is a system with many heavy components. Dispersing the components on the lowest area of the ship can be advantageous for the ship's stability.

This solution is very area consuming, and might take space from the storage space. In order to maintain the same storage capacity, the storage would have to be risen. Which increases the centre of gravity and thereby undoes the reason for dispersing the machinery components. Hence, this solution seems to be ineffective for our vessel.

The main engine may reach height of 5-6m meaning there will be a lot of space lost above the machinery components. With a mezzanine deck, one could utilize the space over the components with more lighter ship systems.

Axe / X-stern/-bow

Convectional ship hull has increased volume in upper deck forward, to increase the bouncy when entering a wave. The increased buoyancy however causes rapid pitch movements making the vessel difficult to work in. X bows do not have this increase volume, so the buoyancy increases more slowly, making the pitching movement smother.

Axe bow consist of a long vertical stem, which stays submerged during pitching to avoid any slamming. With a linear volume change due to pitch motion will cause a smooth change of vertical velocity.

Although these bow forms cause a smooth change of roll, it probably increases the roll motion. With low change in volume and a bow without flanges one is also in danger of receiving green water on deck.

Centre storage

The storage is a room with variable weights causing trim when positioned off centre of buoyancy. To minimize the trim, one can put the variable weight closer to the centre of buoyancy.

Lower Number of Workers

To limit the amount of people needed on board is a priority for some shipowners. Crewmembers needs pay, food, water and accommodation. A solution which reduces the amount of people needed, but remain its efficiency is appealing for a shipowner. Less accommodation means the ship can be smaller, less space is required for food and water, but the manual labour might have to be replaced with advanced machines and automated system.

Smaller accommodation

A small cabin and common area can be cleaned by the crew. If the crew easily can clean these areas, there is no need for any workers mainly cleaning the areas.

However, a small and compact accommodation might feel claustrophobic for the crew, and the decrease of luxury and comfort might cause the long liner to be less attractive for new crewmembers

Autonomous factory

Automizing the factory will result in fewer workers needed. Machines and robots are perfect for use in repetitive and tiering work, as humans can get bored and tired. However, robots need maintaining, they require a complicated arrangement and expensive to buy and install. Some task in the ship needs human adaptability and decision making in case something unforeseen happens.

Storage capacity

The storage capacity decides how much fish the ship is able bring to shore. A shipowner would like to be able to carry as much as possible home in after a good operation.

There are many challenges regarding the space required for storage. The pallet the fish is stored in need to be stored on a flat surface. Flanges limits the area where the pallets can be stored, but shelves can be used to store in the volume above. Ideally the storage is an open room to allow more storage, less isolation area and easier for the forklift to move about. This would however result in massive beams, so columns may be used to carry the load, but these occupies some of the area used for storage.

Storage by vertical hull side

A way to avoid lost volume to flanges, is to place the storage where the ship side is vertical. This is usually in a central longitudinal position elevated form the bottom. This would allow more efficient use of the storage volume, but it will require the other ship system to be able to be installed by flanges. It will also challenge the ship system layout as well as the ship stability.

Fish Quality

A selling point for longline fishing is fish quality. A design based on fish quality is focused on a factory arrangement where the time the fish spends out of the water until it is frozen is minimal.

Factory flow

A good factory flow needs short time between the posts. We could apply lean and digital factory to review and visualize what takes time, detect bottle necks and find ways to improve the flow. Prioritizing the factory outline to reduce the process time, can be advantageous for the fish quality

Safety

Moonpool has contributed greatly for the safety of longline fisheries. A safe work environment is a great selling point for worker to join the ship, it can be advantageous to find ways to improve the safety.

Workstations close to LCF.

Many of the machines the crew operates involves sharp cutting tools, and some work posts involve operating knives and checking hooks. Makin sure the environment is reliable and as motions less as possible may reduce the risk of any injuries.

To build the factory close to the LCF is however a challenge concerning space. In midships we have the moonpool, taking much space in centreline of the ship. The factory must be fitted awkwardly in tight spaces probably decreasing it efficiency, as well being claustrophobic for the workers.

Bait storage up and conveyor belt

To carry bait to the auto baiter is hard and tiering work. The bait blocks can weigh up to 50 kg and can cause back injuries and if not handles properly, they can slip and squeeze crew members. By having the bait storage on the same deck as the auto baiter, the workers don't have to carry the bait vertically, and by either putting the bait storage closer to the baiter or possibly using a conveyor belt, the crew don't have to carry the blocks as much.

The bait storage is required to carry around ??tonnes, which is a significant weight in terms of stability. For stability it would be advantageous to place this storage as low in the ship as possible; having the bait storage on main deck would require counteracts in form of hull shape and/or ballast.

Separate moonpool

Even though moonpool is a safer alternative to traditional hauling, there is still a chance of water pumping over the pool side and onto the deck. By separating the moonpool and hatch room from the rest of the ship, with a bulkhead, one limits the areas the water can flow.

Wind Area

Wind have great effect on a vessel's seakeeping capabilities. Longline fishing is more efficient when the line is hauled trough the centre of the moonpool tube, and wind can cause unwanted drift which requires unwanted manoeuvring for the captain. The wind can also cause the vessel to roll, which cause a danger to crewmember working with sharp tools and around heavy machinery.

Use length of the ship (lover the wind area)

To lower the hight of wind area will reduce the moment the wind is working on the vessel. By using the whole length of the ship for accommodation one will reduce the need of several decks

Shore time

Some shipowners might prioritize solutions which reduce time the vessel spends ashore. The less time spend ashore, the quicker the vessel can leave for another operation. During the time ashore the ship needs to unload the fish, and fill the food storage, bait storage and consumption tanks.

Storage up

For quicker unloading, one can put the storage on a higher deck. The time spent on elevating the fish from the bottom is eliminated. However, the high storage poses a challenge concerning stability and the system arrangement of the ship. We would have to need a great breadth and hull shape that support the high KG, greatly affecting the ships resistance.

Lift

Traditionally a lift is used to elevate the cargo to an unloading hatch. A lift requires little space but can only lift a few pallets at once.

Ramps

Ramps would eliminate the need for a lift; a ramp requires less maintenance and one eliminate the risk of not being able to unload due to a lift failure. However, it requires a lot more space than a lift both in storage and the deck above and it's not clear if a ramp will reduce the unloading time.

Paternoster Lift

A paternoster lift is an elevating system where several compartments are lifted and lowered in a continuous loop. It is primarily used for transporting people, but the idea could be transferred to cargo. This way one can elevate cargo continuously and at the same time lower bait. This

Figure 44 Paternoster lift for vertical transportation of people https://i.ytimg.com/vi/KoCQ6tq5wJE/maxresdefault.jpg

solution however has many challenges; it requires more space than an ordinary lift; it would be a complex design with many moving parts meaning high maintenance; to our knowledge this technology does not exist for cargo and has to be developed; the logistics of loading and unloading is complicated.

Comfort/Luxury

There is a competition for manual labour in offshore and fishing industries. A high standard of comfort/luxury may be a deciding factor for crewmembers to stay at one ship.

Spacious accommodation

From Lovdata we gathered some limits on how small the cabins may be, however we know that we should be well over these limits in order to have a sufficient comfort level. A common area with lounges and activities for the crew might also help the luxury and the camaraderie. A well looking interior may also increase a sense of wellbeing.

However, luxury is more costly for the ship owner. The increased space in accommodation will ultimately result in a greater wind area.

Other solutions

There many design features that does not fulfil a design guideline. But to increase the design space, all possible solutions should be reviewed and considered.

Transverse arrangement of Mag-pack

The Mag-packs are adaptable in the way the area can be arranged. The magazines come in certain lengths and the depths can be customized giving the Mag-pack area much flexibility to be fit where need be. However, the length of the magazines is preferably as long as possible to limit the amount of work needed to connect and release the lines. The Mag-packs can be arranged transverse to allow more flexibility in the design. However, this will require bends in the hook rails which might cause the lines to be tangled. A longitudinal arrangement will allow the hooks to go in a straight line, allowing a more efficient workflow.

Moonpool with incline

The moonpool requires much space, and its position is not adaptable, usually the space of the storage and machinery. Being able shape the moonpool as a funnel will allow more flexibility in the design of those rooms. However, the incline might decrease the damping effect of the moonpool.

Moonpool with an offset

Moonpool is a component of the ship with little flexibility in terms of position. Searching for ways to adapt its shape might be beneficial for the deck arrangement. The moonpools of today have the tube opening at the centre of the tube; allowing the pool to be offset from the centreline gives flexibility of the deck arrangement.

However, the imbalance might affect the ships sea handling capabilities, by creating rotation in the moonpool sloshing motion.

Designing an Efficient Work Deck

The factory and hauling deck have many workstations and components, increasing the difficulty visualising the whole system and flow of work. Therefore, a visual process utilizing process mapping where utilized. This makes it easier to understand how all the components are connected. As well as finding and eliminating redundant tasks. As well as clearly defining waste processes either minimizing the waste or eliminating redundant tasks. Crucial step as now the design is being outlined. Boxes, colours, and lines one can see what type of task it is, how important it is and what order of tasks are done in.

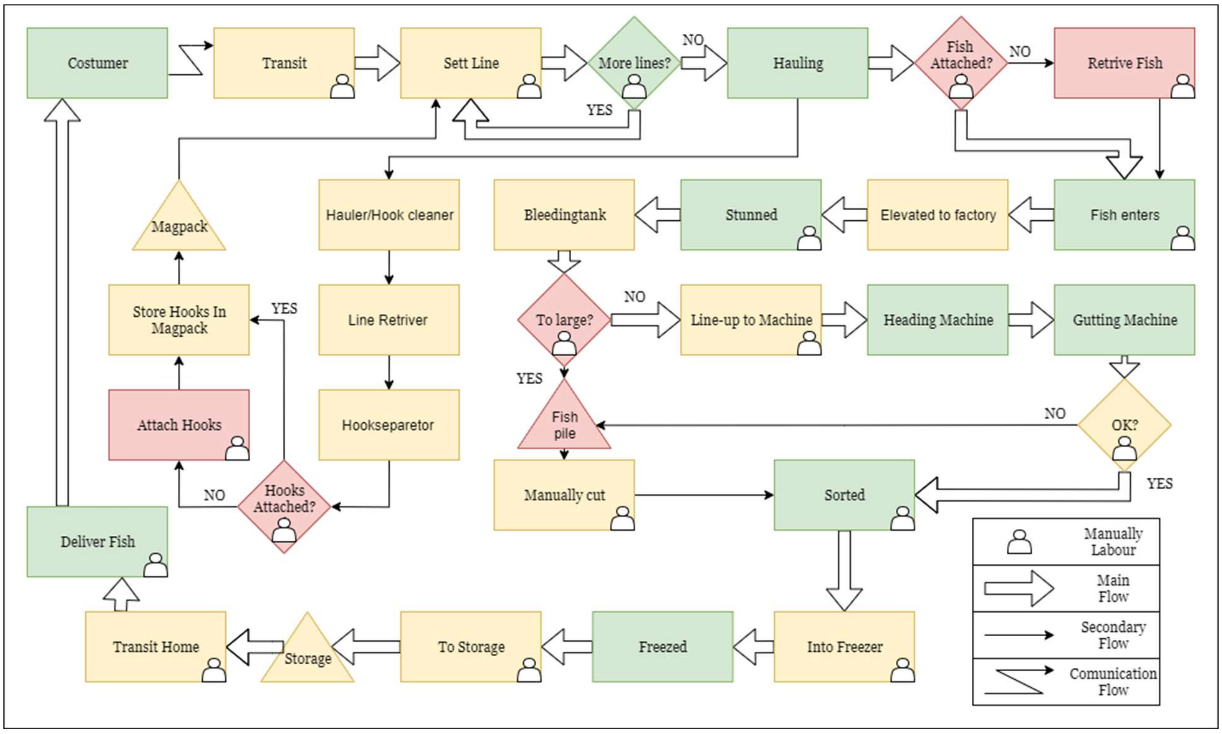


Figure 1: Process Map; current situation

Red

These are unnecessary task which are highly manual and therefore are very likely to increase waste. Of the importance of minimizing waste should this be considered as a priority

Retrieve Fish

When the fish leaves the water surface, it has a risk of getting unhooked before having boarded the ship. A fisherman then needs to retrieve the fish. It is not much technology one can utilize to eliminate this problem. Should be designed so the fisher has the least wasteful method retrieving the fish. Same person is responsible to cut the fish. So, both areas should be designed to be close together.

To Large

This task uses a worker to determine if a fish is too large to go through the heading, gutting machines, and vertical plate freezer. At the current technology heading and gutting machines are not able to be automated for large fish. The fish needs to be manually inserted into a “hanging freezer”. Making the process manually intensive. As both the factory worker need to carry the fish in between the workstations. Shortening the distance between these areas will therefore decrease the waste.

Attach hooks

When the line is sent for storage, it must go through a hook separator. The hook separator setts the hooks on a rail leading to the Mag-Pack. If the hooks are not attached, a crewmember needs to hang the hook on the rail. Reacting and completing rapid for the hook separator to not stop functioning. Therefore, this process should be tried to be visually present for the worker and clear and short path.

Yellow

They are less labour intensive, but very simple repetitive tasks therefore creating waste. Also, some are considered to be bottlenecks in the factory. One should aim to minimize these tasks by automating them. If not since the workload is smaller compared to red, should these be accessed fast.

Bleeding

The fish needs to bleed for 15 to 20 minutes (Aðalbjörnsson & Viðarsson, 2017), however due to how the bleeding tank is setup to take large quantities at the same time some fish spend about 30 minutes in the tank. Furthermore, this leads to batches of fish and unregular flow in the process.

Line-up to Machine

The heading machine needs the fish to be placed in a certain way for it to work. A person must manually do this task, can a robot do it instead? With current technology is that not possible. As one need to handle the fish delicately to not destroy the meat, which are difficult for a robot. To these kinds of task should the worker be able to walk fast in between such workstations. Also visualizing where work needs to be done.

Baiting hooks, into freezer, line-up and stunned

Baiting hooks during setting of the line, requires a worker to feed a baiting machine. This is a very simple and repetitive task which requires little creativity by the worker. Other task that suffers from this waste in this process are “into freezer”, “line-up to machine” and “stunned”

Green

Tasks can be very automated, and generally is not very labour intensive. These tasks therefore produce great value in the factory.

Sorted

The customer demand that the pallets of fish he receives are sorted by different kinds of fish, therefore is this a task we valuate as value added activity. Which also can be automated.

Maximize Storage space

Storage space is considered the "first" priority since the fish is the payload, and with a higher capacity there can be fished more fish on each trip. Some solutions we are evaluating to maximize this space is; letting the storage evolve further aft and thereby going around the moonpool, having one central elevator and using the gained space from the elevator removed for extra storage, or move the storage room upwards to get away from the flare since this is ineffective volume.

If we implement the two first points, they are somewhat conflicting. Letting the storage further aft and having the lift centred means that the storage room is cut off. It is important to note that the storage room would be cut of a bit due to the moonpool so perhaps placing the lift at one side of the moonpool and the storage on the other side could be a favourable. Here it would be critical to get the bait storage close to the lift or at least approachable from the main storage.

Going for one lift centred

The lift will be just aft of the moonpool at the port side of the vessel. Connected to the main storage through a path besides the moonpool. While the bait storage is positioned just aft of the lift and over the machinery, yet still aiming to keep it as low as possible. It is important that the bait is easily "reached from H and L.

The advantage by having one single lift, instead of one large for fish pallets and one smaller for bait storage loading, is that the volume gained by eliminating one lift. This volume can then be used for other compartments, either increase storage, increase room in machinery, lowering the profile or a bit of all.

Sea-Handling Attributes

There's a lot of factors and elements that goes into how a vessel handles at sea. Stability, comfort, directional stability and how the vessel handles waves all play a part in what we would consider sea-handling attributes. Most of them are results of the hydrostatics of the hull and weight positions onboard. A common problem that becomes apparent before we have come so far as to calculate the weights is that the storage will be at least mostly in front of LCF and therefore will we encounter a large difference in trim from empty to full. We considered if we could move the storage a bit further aft, yet then it would have to "go-around" the moonpool. This would mean there would be a larger wall area and thereby the need for more isolation. When "around" the moonpool the total beam of the isolation in the walls would be double that of the normal storage and therefore we lose potential storage volume.

Machinery System of Choice

Since the vessel has three different main operational speeds, with a significant difference in speed, would a hybrid solution with two speed gear be favourable, it adds more flexibility, reduced fuel consumption, and reduces the necessary maintenance. During transit, it would operate using the higher rpm, and all the power goes to the propeller. Setting would use the lower RPM speed, and the rest of the energy from the main engine to generate electricity for the vessel through the shaft generator PTO. During high seas, the PTO function will assist with power bursts, to assist the main engine. Since we want added flexibility from a controlled pitch propeller will frequency vary greatly. To get this under control, a variable frequency drive (VFD) will correct the frequency. During hauling would it use electricity from the auxiliary generator (PTI). Since we utilize a hybrid system, which can generate power if the main generator fails, a decision where made with our supervisor to only have one generator. A battery would be added, which can be charged ashore, which can come from renewable electricity. The battery will only be small and contain 600kwh same as Geir and Atlantic, as a bigger requires a lot more space.

Accommodation

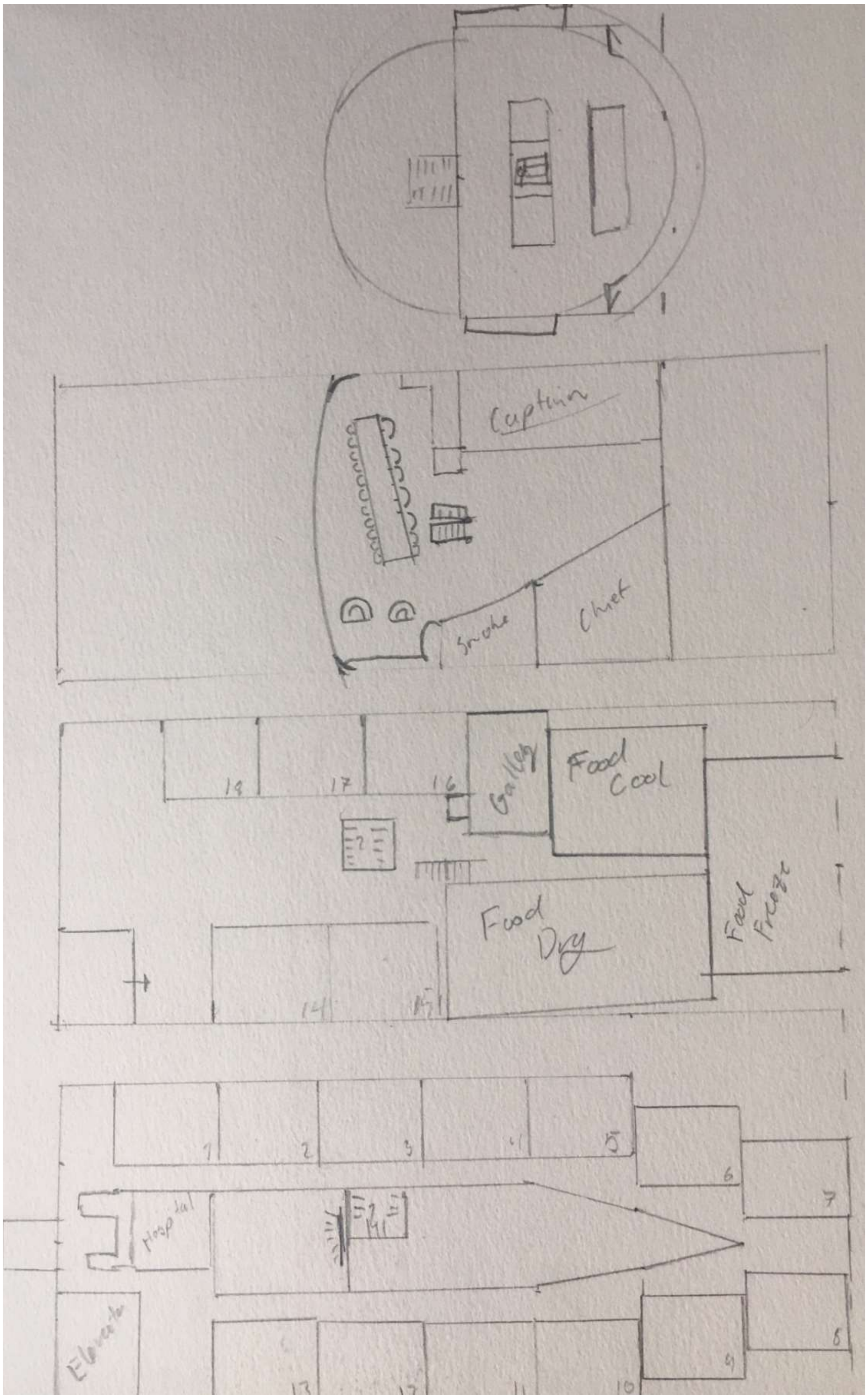


Figure 45: Draft of Accommodation

Deck 4

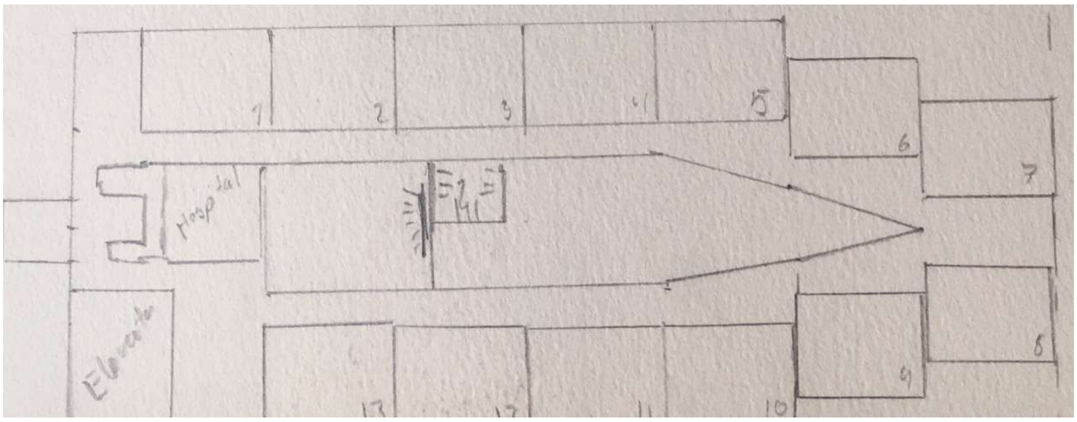


Figure 46: Draft of Layout Deck 4

The accommodation is by no means the main priority of the vessel. Nevertheless, it should not be forgotten, so what it in the end means is that we use wherever there is some space after the core systems have been positioned. And due to the massive weight of the core systems they are placed in the bottom of the ship. This leaves room for the accommodation to be placed on the top. In this first GA draft, we chose to make the first accommodation deck to be mainly cabins. The cabins have been designed to be 4.1x3.2 metres, this is because we wanted to offer longitudinal placed berths. In addition, this both makes the cabin feel roomier than narrow cabins and it leaves space in the centre for cinema and hospital.

Figure 47: Standard Cabin as used in Draft

In this draft the whole centre section has not been filled as we consider this space either for some extra lounges or making some of the cabins larger if that would be desired by the ship-owner. Larger cabins and fewer lounges could neglect the need for a regular cleaner.

Deck 5

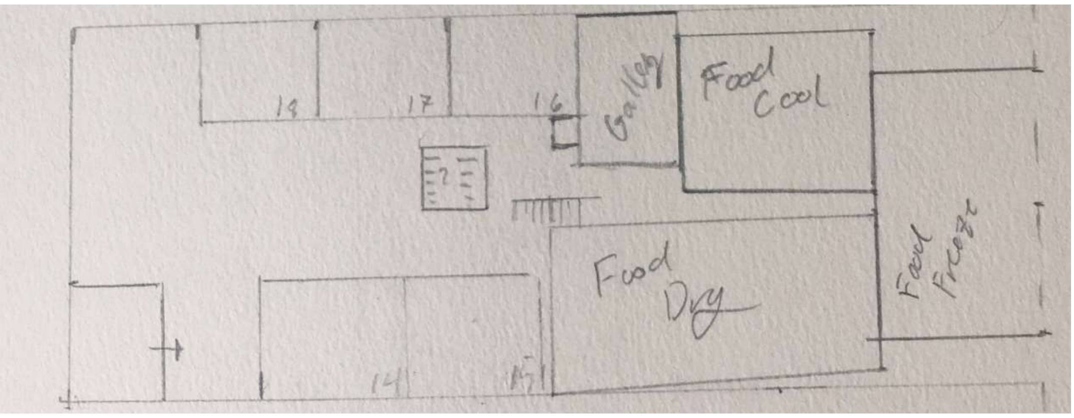


Figure 48: Draft of Layout Deck 5

The next deck features the remaining cabins (excluding the captain's and chief's quarters) and the food storage with galley. The food storage is positioned quite forward in this design, somewhat to get more weight further forward and somewhat to fit with the placement of the diner. The weight of the food storage might not be a lot in the grand scheme of things yet it is still a weight so it should be considered as one.

Deck 6

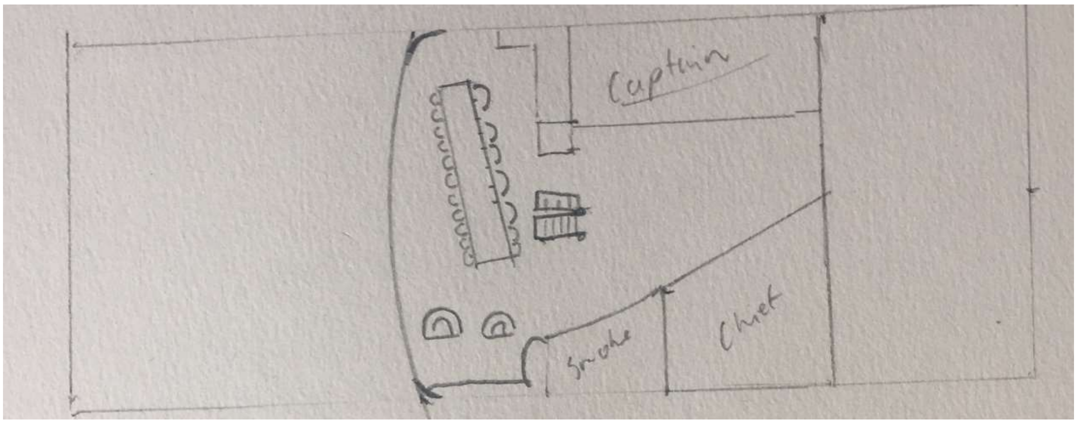


Figure 49: Draft of Layout Deck 6

Now we look at the diner, placed one deck above the kitchen. This means we must have a dumbwaiter that can bring the food up. The advantage of this placement is that the diner can have a lot of windows that brings inn a lot of natural light in addition to giving a great view. The view will be directed aft wards due the forecastle in the bow and to be more protected.

This deck also includes cabins for the captain and for the chief in addition to the hardware for the bridge controls. These have not been mapped out exactly, since we do not know yet how large they will be. We assume that the true area we eventually use on this deck is not restricted too much, and therefore have we not spent as much time on the rooms that it shall contain.

Deck 7

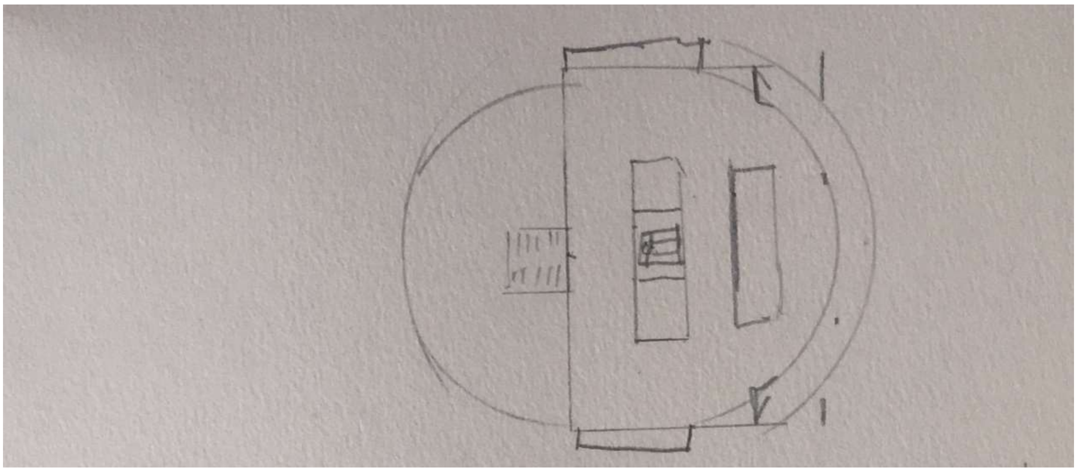


Figure 50: Draft of Layout Deck 7

This is the deck of the bridge and is the highest deck on our vessel. The bridge is placed quite forward in contrast to the other vessels we have looked on. The decision-making factors behind this include, the shape we want on our bow, the fact that we are trying to use only one lift placed amidships.

We are exploring the possibility of using a bow shape resembling a straight stem or possibly a water piercing bow. This in order to move volume forward, softer slopes and making use of the full length. Since the ship usually heads into the sea with the bow first so there is already a need for high a high profile over the waterline. Since we are trying to keep a low profile, we put more of the superstructure forward where we already have a high profile.

Placing the lift amidships means that we could consider only one crane as well placing that one close both the lift shaft (and/or storage hatch) and potential Danish seine. For this to work the superstructure will have to be forward.

Access

When it comes to loading, food and packaging is stored in the bow and cannot be reached by the main crane. An additional crane can be placed in the bow, yet there also has to be a hatch and shaft that goes down to the storage. The food is stored on deck 5, the same as the forecastle, so just below the hatch. The packaging on the other hand will be stored on deck 4 and 3, and there would therefore be an open shaft down to these deck for easy access.

Storing the bait on the main deck and in a central position gives advantages in terms of access when loading and consumption of bait. Placing the storage nearby the hatch and/or the lift makes for quicker loading of bait. Having the storage on the same deck as the auto baiter, eases the workload for the crewmembers.

Part II: The Design Spiral

For the next part of this paper we will show how we proceeded with the actual shaping of the vessel. Where we take the knowledge, ideas and prioritizations established in Part I and put them into a specific design and working them over and over through the design spiral. Since there is quite a lot of systems that each take up significant volume, and our prioritization of the storage capacity and how we aim to maximize the available space for it, we start by drawing up some drafts of a general arrangement. Testing a bit of the different ideas we came up with and show them to the owner in order to get feedback from his point of view. Then we will establish a quick weight estimate to get some essential values that will be used, together with the GA, to form the hull. With both the hydrostatics from the hull and the weight estimate the stability will be calculated. After this we would perhaps need to go back a bit and make alterations to improve the stability and improving the weight estimate of the hull. Furthermore, should the structural design be implemented as the design spiral continues until it reaches a convergence within the time frame available.

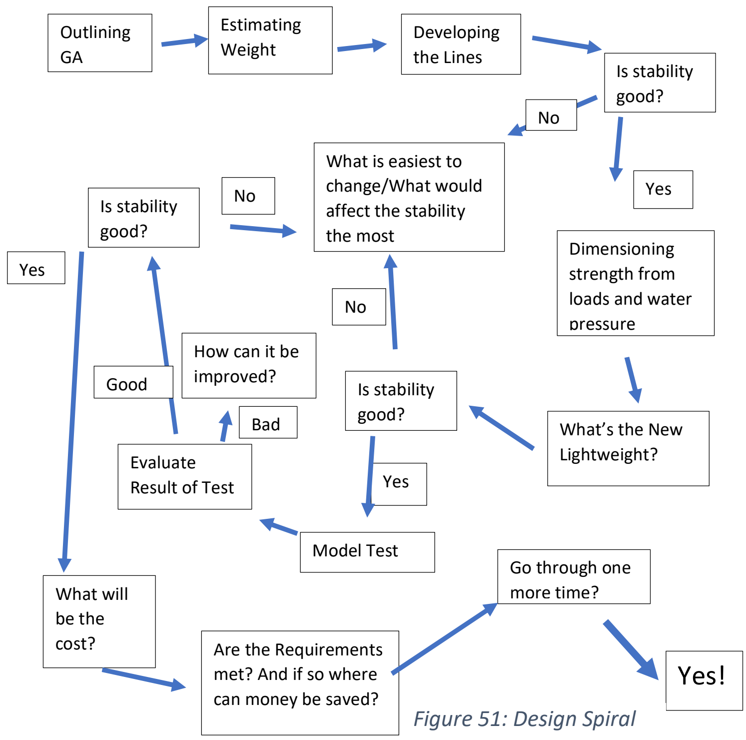


Figure 51: Design Spiral

We chose to utilize the dimensions given in the requirements to fit as much cargo as possible. Therefore, we presumed with a length of 63m. The required beam was 14m, our pallets would require a breadth of 1.1m, we assumed that we needed additional 0.7 on each hull side to fit beams and isolation, therefore we calculated our beam to 1.1m*11+0.7m*2=13.5m.

First Drafts of GA

After our vigorous GA brainstorming, we came up with a lot of different ideas and solutions, then we explored some of the problems with the solutions. These problems called for their own solutions and in order to decide which we should spend more time on than the others we had to decide on some priorities. All this information where to come together in one first draft of a GA that we could present to our customer. Nevertheless, after developing one main GA we quickly put together some different alternatives, including Danish Seine, that we could keep up our sleeves if they were called for.

Main Draft GA 101

Our ship should first and foremost be a long-liner, as requested by our customer. So, for our main “First Draft” GA this would be our most pressing concern. Thoughts about Danish Seine where cast aside for this version and we started with our ideas from the brainstorming. This ship focuses on making the storage capacity largest possible, and making the ship most cost-efficient by making the workplace the most efficient possible.

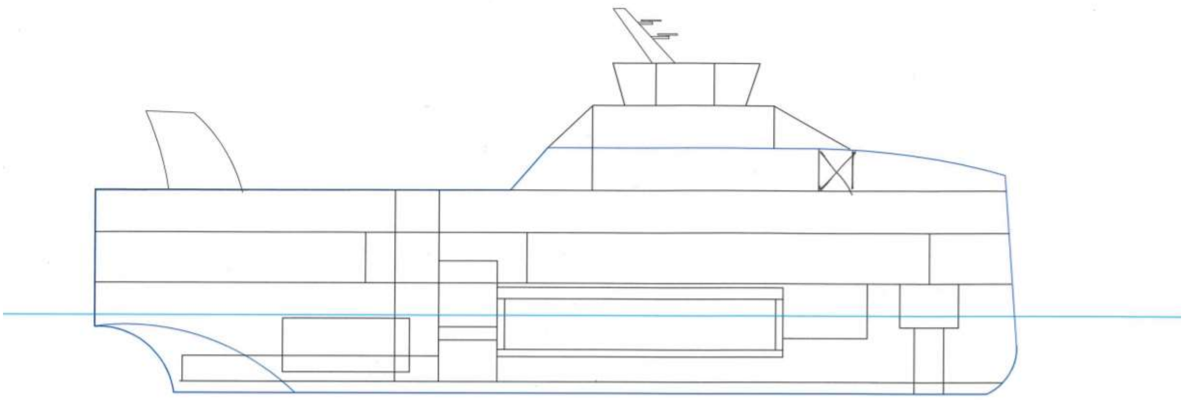


Figure 52:Profile GA 101

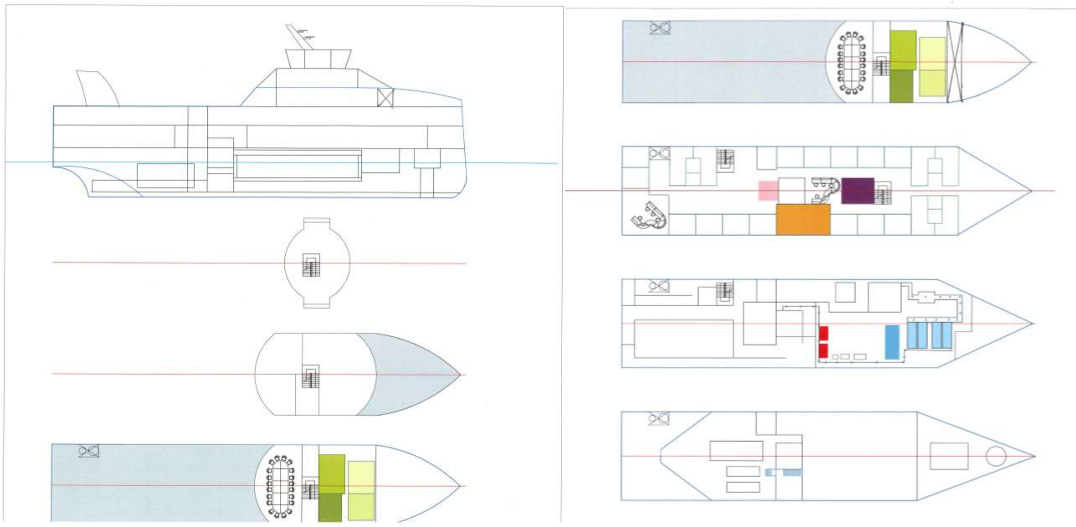


Figure 53: GA 101 Plan

One Elevator Amidships

As stated in the brainstorming the first focus would be to maximize the potential storage space. One of our ideas for doing so was to eliminate one of the elevators, this is incorporate into the design early on. To not take away storage space where the beam is at a maximum, we chose to put the elevator on one side of the moonpool. However, this seemed to possibly be a bit tight and the forklift would need some space as well. Therefore, ended we up with putting the elevator slightly aft of the moonpool on the portside. This takes away a bit of the space from the machinery, and in order to make up for this the cooling generators would have to be moved away (Using a Top-Down approach based on the machinery of Geir). There is always a problem with the trim on these vessels when the storage is empty so in order to get weight forward and to get the cooling generators close to both the storage and freezers, we ended up putting them in the bow. By placing them in front of the main storage they could still be in a sperate room as required, although on smaller vessels the owner might fear that there would be too much movement in waves.

Elevator where estimated to be around 2x3 meters. In order to be on the safe side, so enough space where allocated.

It should be noted that this position of the lift also aided in making a clear flow in the factory, where the fish starts at starboard, moves forward and comes back to lift at port.

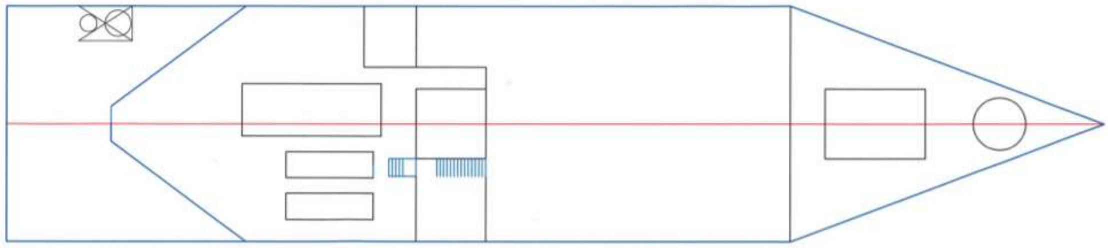


Figure 54: GA 101 Deck 1

Clear Flow Factory

For the factory we had already done some designs on how the machines could be positioned. To make the work place efficient it was important for us that the workspace was placed so that easy movement between work areas was possible. Also, that the factory workers could see where their work was required. Hence, we developed some of a factory drawing and made an outline with some estimated sizes. The arrows represent the fish flow through the factory. The plate freezers where placed at the front. A manual freezer for large fish would be placed in the middle area close to the bleeding tanks (red), this would make it possible to manually process the fish next to the freezer. The freezer hotel and pallet-machine were placed on the same deck as space freed since no machinery needed for fileting.

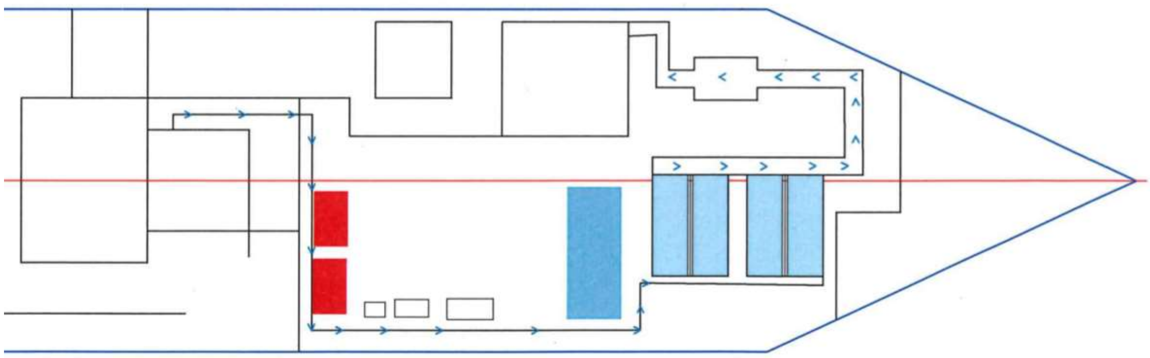


Figure 55: GA 101 Factory Layout

Weight Movement

Due to the uneven load in a fishing vessel especially from factory was it necessary to add a transverse bulk tank. From calculations was the critical load when the fish hotel, freezing and palletizing was loaded maximum. From calculations needed the aft bulk tank be 25 m^3 in order to equalize that when it was placed between 2-5 meters transverse. Calculations can be seen under. H/G is removed as that process will cancel out the maximum transverse moment.

Factory Fish				Tanks		
					AftBulk1	Aftbulk2
	Weight	Trans Arm	Trans*m	%	98%	98%
H/G	0	3	0	Tot-Vol	12.50	12.50
Freezers	12.8	-0.75	-9.6	Volume	12.25	12.25
Fish Hotel	15	-4.025	-60.375	Weight	12.56	12.56
Palletizing	1.03041	-4.75	-4.89445	Trans Arm	2.75	4.25
Lift	2.06082	-4.75	-9.7889	Trans*m	34.53	53.36
			-84.6583	87.89		

Table 4: Aft Bulk Heeling Tanks

Bait Storage

Here we wanted to place the bait storage so the bait could move out from the storage on a pallet without the need to lift. That would reduce the wear and tear of the employees. So, the bait storage was placed on the same deck as the hooks and lines. This also gives us more space to work with for the machinery. The consequence is that it makes the gravity point higher and therefore reduces the stability. Therefore, this needs to be revised during stability calculations. The best placement would be behind the moonpool since this would be next to the elevator and reduce the trim in departure as the weight is move amidships. On the other side, placing the bait storage this high raises the cumulative centre of gravity. Thereby, giving us worse stability, although at this point, we choose to try-out this position and keep it in the back of our mind that this might accumulate problems later in the process. Hence, if they do, we already know a solution.

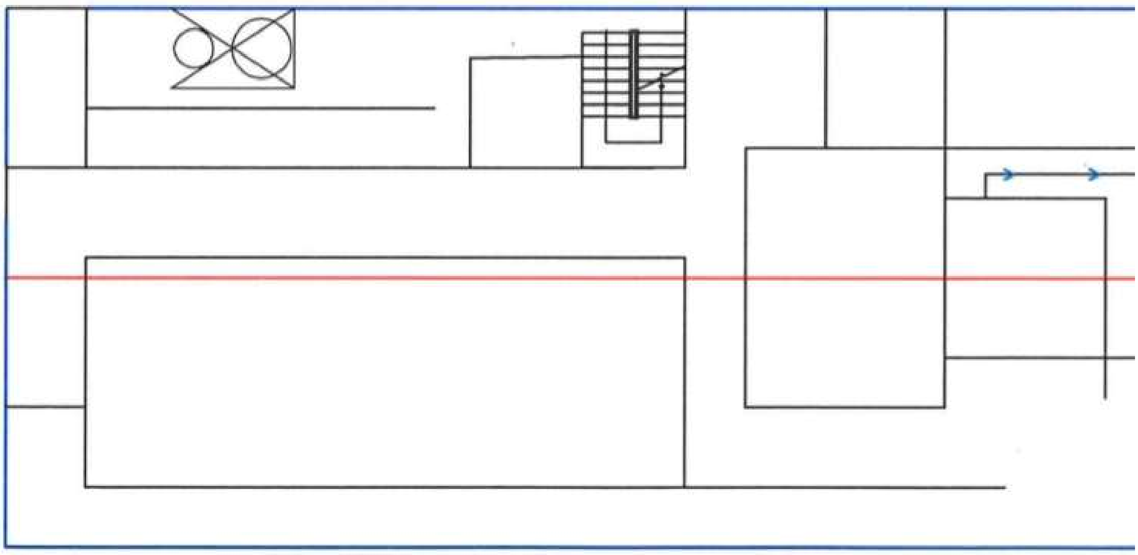


Figure 56: GA 101 Bait Storage and Mag-pack

Low Profile (21.5m)

The oceans the vessel operate in tend to have strong winds. The wind forces the vessel off course, and with more force, more energy is needed from the azimuth at the front to keep the vessel on course. Therefore, is it important to minimize this area to have greater directional stability. Having this in mind we tried to reduce the height of the vessel by spreading out the accommodation on the whole length. Nevertheless, the bridge needs to be somewhat elevated in order to give a better overview of the vessel. In addition to this the bow should be have some height to it since this is where the vessel "cut" through the ocean. This is also aided by the idea that we want to have the flare quite high up to make a smoother bow. Henceforth, we positioned the bridge in the bow with a deck under it for the hardware. And below this the galley and dining are enclosed by the hull-lines on the sides, forecastle with rope handling and chains for anchor in the front and a panorama view out aft.

Machinery

The chosen propulsion version was the hybrid solution since it gives greater flexibility and in principal lower consumption for a vessel that operates at various speeds. The problem with this is that it requires more space than other layouts, and we already have "taken" away some of the space from the machinery with the lift position. However, as we already mention the refrigeration system are placed in the bow for that very reason. So, without going into all the components of the machinery, we estimate that it should be enough space for a hybrid solution if we use a "mesanin" deck. At least at this point, while we will look at this issue as we continue the design process.

Accommodation

Deck 4 includes all the cabins, 14 "normal" cabins and six officers' cabins that are a bit bigger. In addition to a cinema, two lounges and a hospital (3x3 metres). In order to keep a low profile, the whole length of the vessel should be used for most deck possible. Moreover, the cabins are the rooms that windows would be most essential for. Hence, to make use of the "window-area" of the deck the other rooms are placed in the centreline. This restricts the available width for the cabins, except for some that we will use for the six officer-cabins.

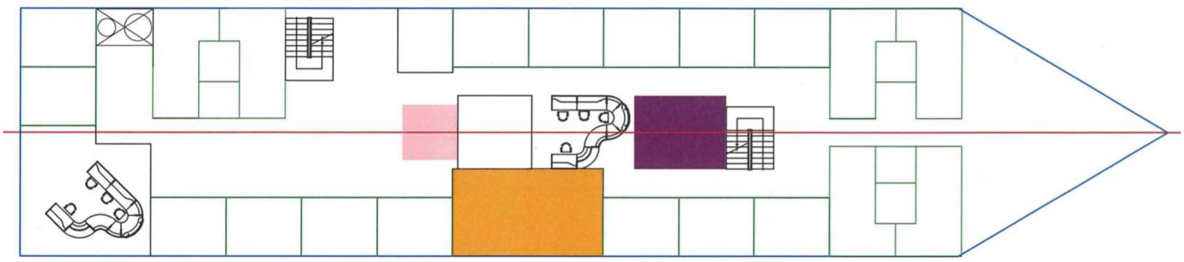


Figure 57: GA 101 accommodation

Standard cabin, version 105

Figure 58: Standard cabin, version 105

For the standard cabins we estimated that there would 3.2 meters available in width, the limited width means we have to use the length to make a comfortable space. So, when they were drawn the width was kept as a fixed number while the length was a result of including all that we wanted to include in the cabin. Taking into account what was mentioned in the brainstorming and forming a room to accommodate them. Resulting in a length of about 4.1 metre.

Figure 59: Standard cabin, version 105 Top View

Officers' cabin, version 107

Figure 60: Officers' cabin, version 107

With the officer-cabins we had a lot more of the width of the ship to work with, and we try to take the full advantage of that fact. While still keeping a good feeling of space, not slim the cabin down just because we can. Rather we focused making the cabin significantly larger, and placed the bathrooms so that we can make them almost square.

Figure 61: Officers' cabin, version 107 Top View

Cinema

Figure 62: Cinema

We decided to firstly make a “cinema” with a homely feel to it, so instead of rows of separate seating we went for a big couch and a low table. Furthermore, there is still a large screen and surround sound.

Deck 5

Consist of a large dining room and kitchen and food shown as green. The idea was to place the food forward to reduce the trim in departure. Infront of the food storage the antiroll tank was placed, to utilize the total breadth of the ship. Infront of the antiroll tank would be used for anchor handling equipment.

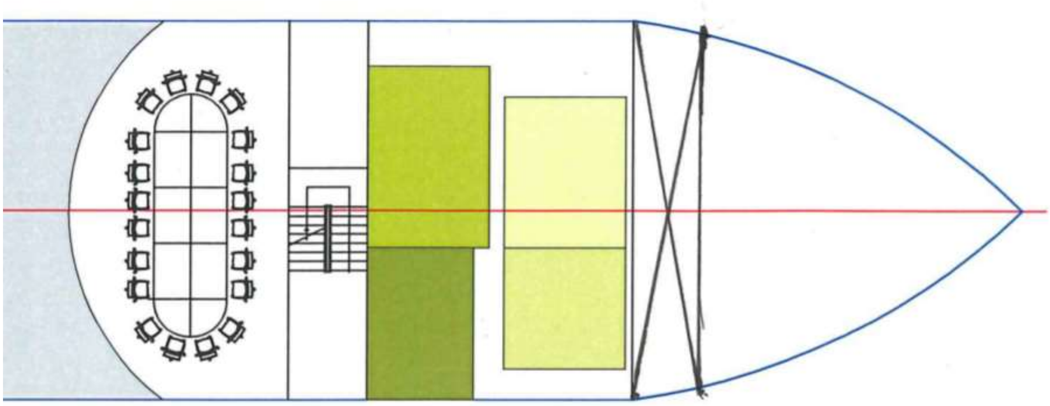


Figure 63: GA 101 Deck 5

Superstructure

Underneath the bridge this space would be used for a lounge and electrical system. The sized for the electrical system is unknown and therefore we did not draw this into the system. Above this the bridge would be placed.

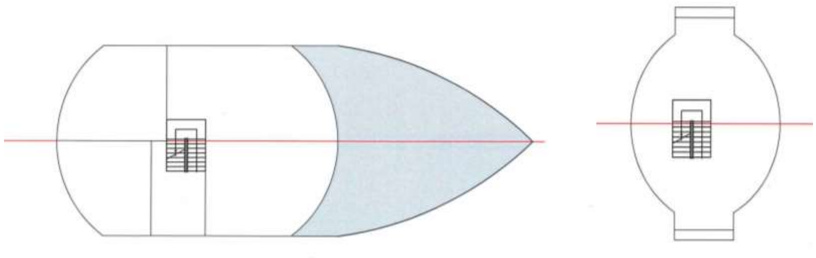


Figure 64: GA 101 Superstructure

GA 102

Our task stated that we should outline the ways that the implementation of a Danish seine arrangement would affect the vessel. Without going into the possible added necessity for added strength, there are some other factors that became apparent when we tried adding it to our GA 101. It should be positioned aft, and the only place that could fit this would be above the deck 4 (cabins) if the vacuum pump would be placed above this as well, the fish would have to be transported through the accommodations in order to reach the factory. Another “intuitively obvious” point is the added weight at this height would badly affect our stability. So perhaps there would be a need to get this arrangement down to deck 4 and move the cabins accordingly, which is what we did I in the 102 GA.

This would use the concepts from GA 101 with same factory, hauling layout and elevator. The main difference is due to the some of the space on the first deck of accommodation is used for Danish seine equipment. This takes up space, therefore it was necessary to place the officer cabins at a deck above. This increases the wind areal for the vessel, so is the negative from having also having ability to use Danish seine. The Danish seine would make the trim larger, due to 30-40 tons extra placed aft. This could possibly be avoided by moving some weight forward, although most of the weight that can be pushed towards the bow already has been. So, the next step would be to change the lines to accommodate the new LCG with a fitting LCB. At this point we have not jet designed the lines so this conflict will have to wait.

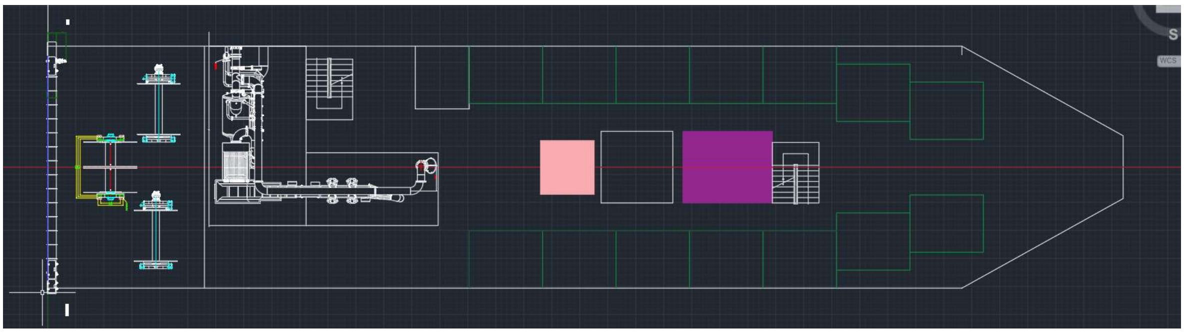


Figure 65 - GA 102 Accommodation Deck 4

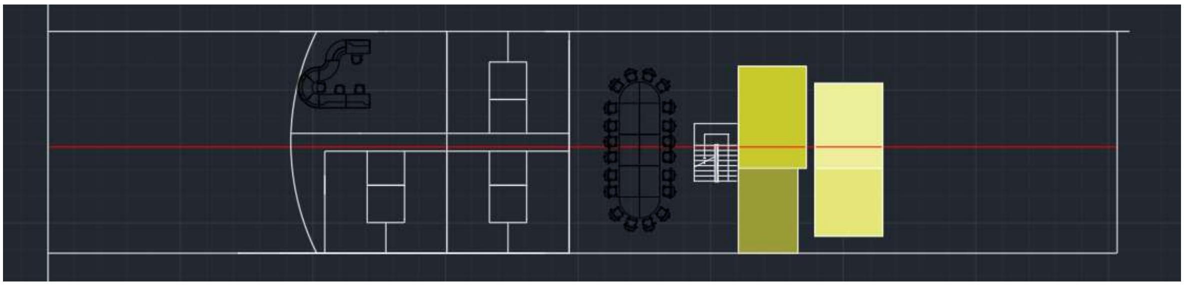


Figure 66 - GA 102 Deck 5

GA 103

If the Danish seine arrangement was to be implemented on the vessel, would we encounter a lot of complications. Not only considering strength and stability, but also the arrangement and changes that would have to be made to get the fish into the factory. For our GA draft 103 we incorporated the Danish seine from the very start and, both considering the space needed for the gear and how to get the fish efficiently to the factory. Which leads us to the main change, a rearranged factory deck.

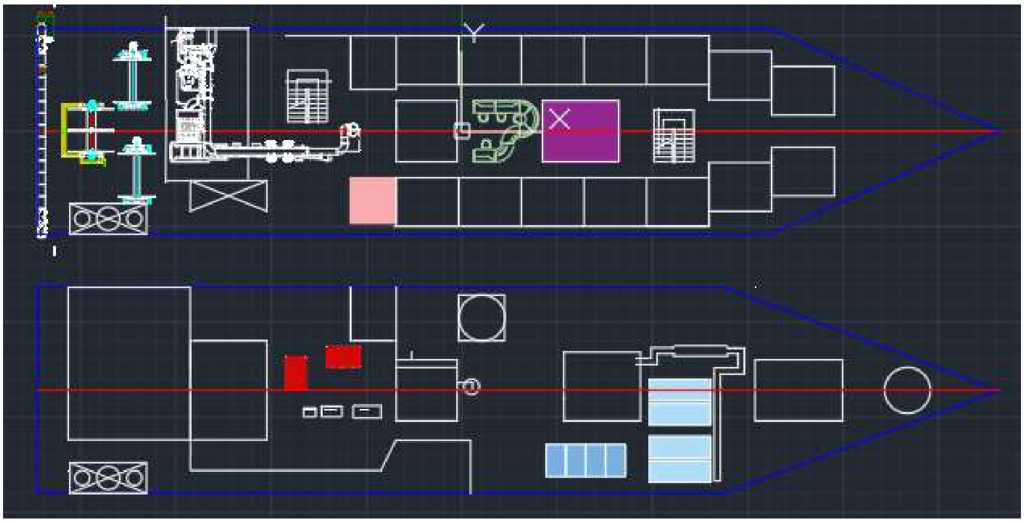


Figure 67: GA 103

Factory ala Danish Seine

In order to get the fish swiftly from the vacuum pump to the factory the factory has been pulled back a bit. Letting the fish enter out of the aft side of the moonpool elevator and starting the factory there, lets the fish caught by the Danish seine be pumped into the ship by the vacuum pump on the deck above then transported straight down to the start of the "normal" factory. This allows from almost seamless transitions from using the long-line to using the seine. There is little added transportation and the factory works the same way. Our version of the factory still places the heaviest equipment towards the bow. This "frees up the space in the centre in front of the moonpool, which could for instance now be used for wardrobes, storage or electrical cabinets for the factory machinery.

The Trim

Although this outlays the factory eases the flow and works both for long-line fishing as well as Danish seine, it places more of the weight further aft. The bait storage is pushed back in addition to the added weight from the Danish seine arrangement with the vacuum pump. This could result in a drastic change in the LCG position, and thereby also in the trim if the hull is not appropriately changed. Perhaps the main disadvantage is that the changing load, the storage, now has a centre of gravity further away from the lightship LCG. Making the effect it has on the trim while being filled grater, hence greater differences between the conditions.

Allocation of Accommodation

Adding the Danish seine arrangement to the Accommodation deck means we have to move some cabins up to the next deck if the cabin sizes are to remain. This will lead to increased height of the vessels profile, although there are numerous ways of allocation the accommodation volume. For the 103 GA, the officers' cabins where raised to the next deck as a separate compartment. The volume of the cabins as they are now is not enough to fill out the whole deck, so by keeping them separate from dining and galley allows us to place them "quite freely", by putting them as aft as possible without disturbing the Danish seine maintains the "balance of design". Keeping the profile area spread out over the length of the vessel lets the vessel have a more balanced visual representation. And due to the fact that the shipowner wants a good-looking vessel this could be an important aspect to maintain. We also wandered if the fact that we could round of the edges of both "compartments could lead to a lower drag force from the wind. Although it would not seem that we could do this to an extent that actually would make a difference, furthermore the wind can attack from all directions so it would be difficult to make useful calculations.

This could be used as an argument to put the two compartments together, followed up by the building costs. Then we could incorporate the deck from GA 102

GA 104

Using the same factory and Danish seine setup as 103, however changing the accommodation to add a new deck for the officer's cabins. This would mean the total height of the vessel would increase by one deck, and more of the weight would be pushed forward. This could hopefully counter the added stern weight a bit.

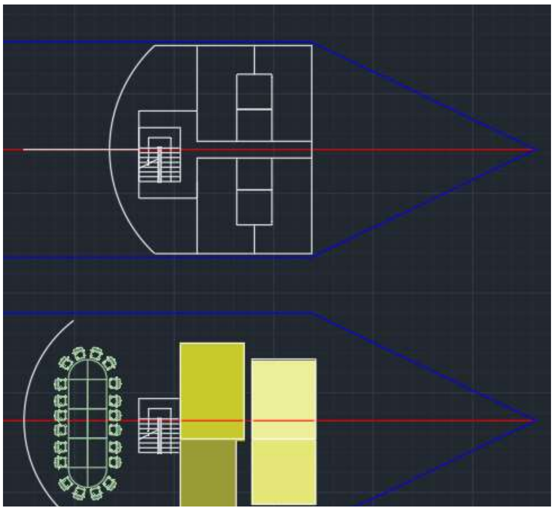


Figure 69: GA 104

Meating the Owner

The shipowner seemed quite pleased with the thoughts and evaluations we had incorporated into the design of our first draft GA (101). The concept of putting one centre elevator to give space to large storage in addition to letting us start storing from the bow to amidships was much received. Therefore, is this a concept we will pursue further and include in our future designs. Furthermore, we take this as a central idea of what our vessel should represent: *Simplify and earn more money!*

Factory Flow

The focus on keeping a good overview of the factory with clear flow and suitable space was met by the owner and stated as an essential aspect. That should be kept as we further develop the design. Although the factory should have allocated space for the electrical cabinets for the machines, which we not had included in the design. The placement of the fish hotel was accepted. On the other hand, he commented on the pelleting machine. It seemed to be unnecessarily far away from the elevator. The position of this machine has continuously been moving since we were not sure how close it could be to the elevator. So, this comment from the owner will be further explored and could potentially lead to better space in the factory. This added space could, for example, be for the electrical cabinets.

Exhaust: Placement and Size?

The exhaust stack where placed almost as an afterthought in the 101 design and the owner reacted on the size of it. It did not seem large enough so we would have to calculate an estimate on how large it would need to be. In addition, was the placement discussed. Due to how we want to shape the stern (underwater) would the placement of the stack be better suited to be before the stern lines are “dragged” that much up.

Tanks and Vessel Depth

When it comes to the position of the tanks (fuel, freshwater and urea) we had firstly added some volume under the storage, not using the double-bottom tanks in order to have a clan design. Although, the shipowner told us that the DB tanks would be used for this. Therefore, can we eliminate the added tanks in the GA allowing us to reduce the depth (or height for that matter) of the vessel.

Accommodation

The cabins where appreciated, they seemed comfortable without taking too much of the space onboard. Both considering the standard cabins and the officers cabins made a positive impression on the owner. However, the placement of the officer’s cabins where a bit disputed, the Shipper and Chief would be reluctant to have cabins so far away from the LCF. This we aid to solve by switching up the placement for the cabins and consequently the cinema, lounge and hospital.

Furthermore, it could be added that the owner seemed to want to have some lounges on the deck under the bridge, possibly facing forward. This will be brought on into the design and assessed according to how much space would be left when the systems to the bridge and HVAC are placed. The area around the window should be prioritised for the lounges and the midsections can be used for the other systems.

Position of Officers cabins

As he mentioned that the skipper and chief regularly demand to have cabins close to the LCF, because of the waves. In the 101 GA all the officer's cabins where placed on "the edges" of the ship, four of them in the bow and two behind the elevator. We had not considered that this would be unfavourable, and since the officers has an important saying the cabins will be repositioned.

Firstly, the four cabins in the bow are being placed just in front of the moonpool, just enough for a corridor between. Then the other "standard" cabins are being pushed forward. It should also be noted that the cinema had to follow as well due to the available space.

There are still two officers' cabins that needs a placement, the two aft officer cabins could be placed on the aft side of the moonpool. However, this would force the position of the MOB-boat as well as the hospital aft. Another alternative could be to place two cabins on the deck 6 around the HVAC and Bridge systems. Since we now got some dimensions on these systems, we are able to make use of the space around them.

Figure 70: Officer Cabins in Deck 6

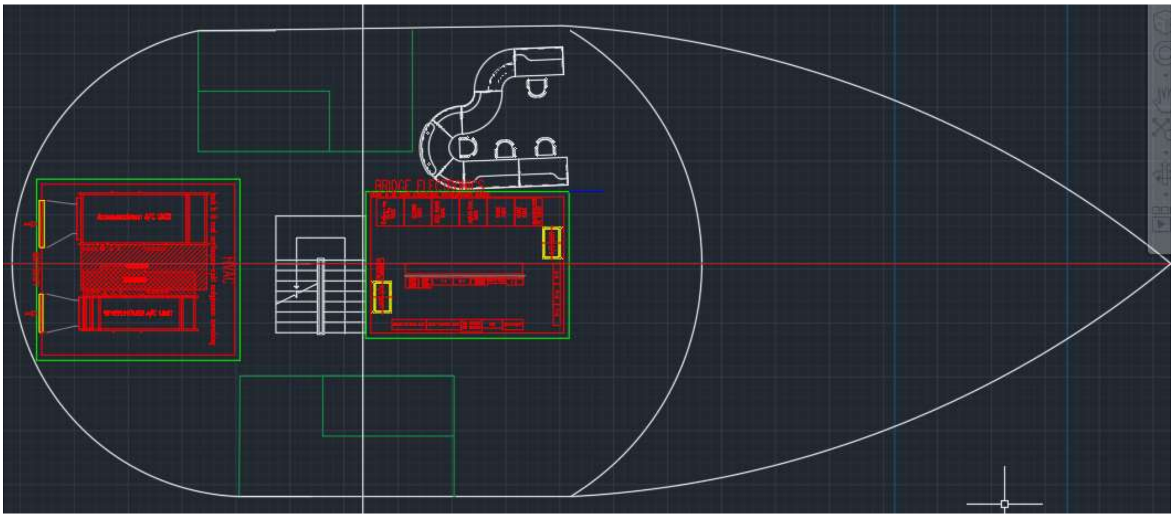


Figure 71: Lounge and 2 Officer Cabins

Lounges

The owner notified that it seemed like the potential window area had not been made full use of. There is a limit to how much outwards area there are on a ship, and positioning the systems that do not benefit from having windows may come off as wasteful. The food storage is an example of this, as it is taking the full beam of the ship at the moment. Yet this is not the main concern for us, as our first intent is to further make use of the area on deck 6. The two officers' cabins took up some, nevertheless we can still fit some seating forward to make a lounge. This lounge will have a panorama view out over the bow although the total area of the lounge may not be impressive. This is due to the fact we are still trying to keep down the wind area.

Bait

The owner agreed on the assessment of our placement of the bait storage, that keeping it on the same deck as the baiter would make life easier for the workers, especially if it is possible to use a "pallet-trolley". And keeping it close to the moonpool due to the effect on trim, although since it is positioned quite high up, it could perhaps have a to great effect on the stability. This will become apparent when we draw the lines, if it is a problem. So, we keep in mind that it still can be repositioned if extra stability is needed, yet for now it will stay put. Danish Seine Factory Behind Moonpool. Placing the factory behind the moonpool if one would have a Danish Seine arrangement, to make the factory flow better adapted for this would not be possible since the line-arrangement sideways is not possible, we got told by the owner. It was impractical to place the line arrangement this way. Therefore, the space the factory would take up behind the moonpool would lead to it not being possible placing the some of the factory behind the moonpool.

Bow Hatch

For loading the wrapping for packaging and food supplies we discussed having a hatch in the bow in front of the food storage. In order to be self-sufficient there should be a crane close to this hatch as well. Since this hatch will be quite far away from the other crane, and there is a superstructure in the way, we will need another crane on the bow.

Refrigeration System

Supervisor commented that since the cylindrical coolers bow additional wear and tear, occur due to the large pitch motion. Although the client accepted it hesitantly due to the size of the vessel.

Continuing GA 101

Even though the owner was overall satisfied with this GA as a starting point, he had quite a few comments and we aim to implement these in the design. The design of our GA will not be a constant, it will be developed throughout the entire design process. By using the design spiral, we continually go back to previous aspects to improve them in light of the new "founding's". We will also let the feedback from the owner and the discussion we have among ourselves, influence the design.

Weight estimation

Since the hull-lines will be highly dependent on the position of LCF, LCB and KG, it is important to establish an estimate for the weight of the vessel and its placement early on in the process. The total weight (or displacement) of a ship is divided up as deadweight and lightweight.

Deadweight

Deadweight is defined as the difference between the total displacement to the lightweight of a vessel at any given draught. For our vessel this consist of cargo, stores, provisions, tanks, ballast tanks, stability tanks and crew.

The cargo capacity for this long-liner was estimated to have a capacity of 550 tons. Stores include the weight of bait, packaging and pallets. Provisions include such as food storage. Tanks consist of MDO, fresh water and UREA.

Storages

- Estimated to be a possible capacity of 500 tons, although 450 is the requirement set by the owner.
- Bait storage of 50 tons
- Food storage of 10 tons
- Packaging and pallet storage of 10 tons

Tanks

- MDO minimum 350 cubic meters, weight 350 tons
- Fresh Water minimum 100 cubic meters, weight 100 tons
- Urea minimum 40 cubic meters, weight 52.8 tons

Shown underneath is the tank arrangement that will be used for stability calculations. Although, it will be changed a bit as we go through the design spiral it is mostly just increased in the detail level. The figures here is what we ended up with at the end, nevertheless it resembles quite well how it started as well. Some alterations where made to adjust the volume when the lines changed, to keep the same volume as the requirement.

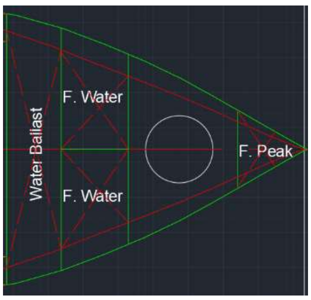


Figure 72:Tank Arrangement Bow over DB

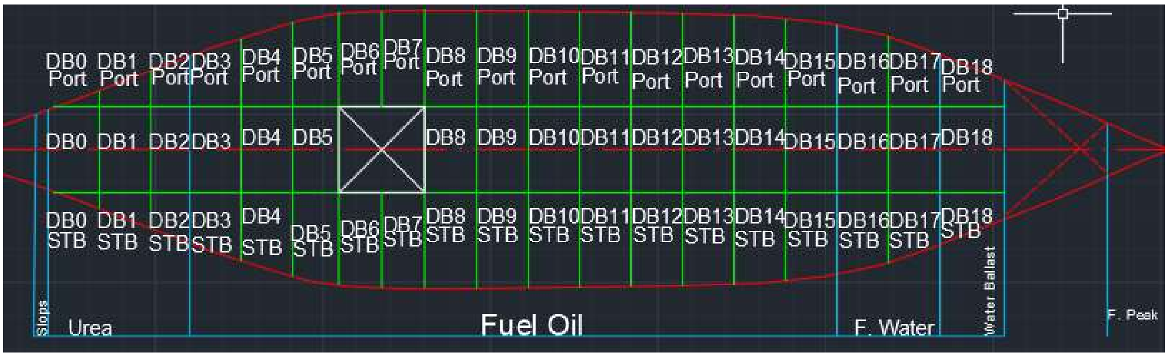


Figure 73: Tank Arrangement DB

Lightweight

Since this is a bottom up design, with no statistic on similar vessel. Estimating the lightweight out of parametric formulas generated from other type of vessels, would not give us an accurate representation. For us therefore it is done by gathering the weight and placement of both all major known components as well as estimating the hull weight.

The lightweight is divided into these categories.

1) Hull steel weight
2) Superstructure
3) Accommodation outfitting
4) System outfitting (pipes, electrical cables and etc…)
5) Machinery
6) Main equipment (winches, cranes, lifeboat and etc…)
7) Construction margin
8) Uncertainty margin
9) Future grow margin

Estimating steel weight

This is a major factor in the lightweight. This consist of all the plates, stiffeners and girders in the ship. This will be calculated from the area of the plates and then multiplied with a factor to approximate the weight of the stiffeners and girders (Håvard Vollset Lien, Personal Communication). Here was placement of the plates important to accurately measure the LCG and VCG.

Plate type	Factor	Thickness [mm]
Double Bottom	2.5	14
Side plates under freeboard	1.5	10
Side plates above freeboard	1.5	8
Side plates superstructure	1.5	8
Strength-deck	1.5	12
Other Deck-plates	1.5	7
Watertight bulkheads	1.4	8
Other bulkheads	1.3	6
Super Structure Bulkhead	1.3	6

Figure 74: Plates

Side plates, deck areas were measured by taking the Maxsurf hull into NX. In NX the area and centre of the plates where established as well as the centre of the side plates. That is a rough estimation and therefore will be regularly updated as the structure design becomes more and more detailed. All bulkheads where measured with weight, LCG and VCG.

Hull Lightweight					
	Weight	LCG	VCG	LCG*m	VCG*m
Decks/double	399.79	29.51	4.59	11798.57	1834.79
Side	334.37	32.68	5.90	10927.15	1972.77
Moonpool	92.47	25.50	7.70	2357.91	712.00
Factory/haul Bulkheads	43.85	25.01	7.79	1096.75	341.45
Accommodation Bulkheads	61.62	28.47	11.40	1754.26	702.50
Superstructure Deck5	103.05	45.24	15.56	4662.07	1603.19
Superstructure Deck 6	32.44	40.44	20.29	1311.67	658.05
Watertight bulkheads	15.40	28.31	3.31	435.99	50.92
Cylindrical Cooler room	7.97	52.75	4.38	420.63	34.89
SUM	1090.96	31.87	7.25	34765.00	7910.56

Table 5: Hull Lightweight

We fear that this rough estimation of the steel weight is lower than what it would be in reality. Therefore, the added factor of 20% was implemented as a safety factor. The resulting VCG have a height of 7.25 m, but if this would impose a problem the superstructure can be designed in aluminium to ensure lower centre of gravity. Theoretically that could save 60%, however, since it has a lower e-module and melting point will it need a higher thickness than steel. So, practically it would save 45-50% of the total weight (Håvard Vollset Lien, Personal Communication), which means a lower centre of gravity half a meter. Decision was made to use steel for the first weight estimation as this could be a safety factor for later stability. To ensure the vessel always have sufficient stability.

Calculating the Inner Walls of Deck 4 and 6

The standard cabins are designed with the head at on end, so the resulting gravity point of the walls would defer towards the bathroom. However, there are the same number of cabins with the head in the front as there are with the head towards the stern. The different layout will cancel each other out, so for calculating the LCG we can assume that for each cabin the LCG is placed in the middle of them. Another thing we have to take into account is that the wall between two cabins is only one wall and we should be careful not to count them twice. The same goes for the ship side, which were calculated in the exterior calculations. So, for our spreadsheet we define each cabin as having only two main walls and the two walls for the head with the LCG in the middle, 2 metres forward of the first wall (seen from AP). For the combination we take groups of cabins together and put in their positions, and we end with a total length of the walls and their resulting LCG.

It should be noted that the estimate for the standard cabins only goes up if we define the officer's cabins as having all three main walls. This seemed like the simplest way to calculate them accurately, so we have defined them accordingly.

Some of the same thoughts went into the calculation of the two officer cabins at deck 6, where they are placed in opposite direction so the placement of the head walls cancels out each other.

Accommodation outfitting

To calculate the weight of outfitting for accommodation a factor for where used 0.15 (t/m^2) of floor area. This includes weight from concrete and tiles also needs to be calculate for washing room, wardrobes, kitchen and toilets. Does not include extra weight from tiles and concrete in food storage (Håvard Vollset Lien, Personal Communication). The value was used to calculate the weight of outfitting for accommodation areas including the bridge for equipment and electrical area, also were used for control room for machinery.

System outfitting

This is hard systems to measure accurately and requires knowledge-based numbers. We got some knowledge-based numbers from (Håvard Vollset Lien, Personal Communication).

- Electrical wiring including lights is 12 tons/1000GT.
- Pipe system is 50 tons/1000GT, since our vessel require high level of pipes for refrigeration and using exhaust to warm up the ship.
- Ventilation is 2t/1000GT where the weight is centred in ventilation system.

Machinery

Machinery consist of roughly the equipment underneath.

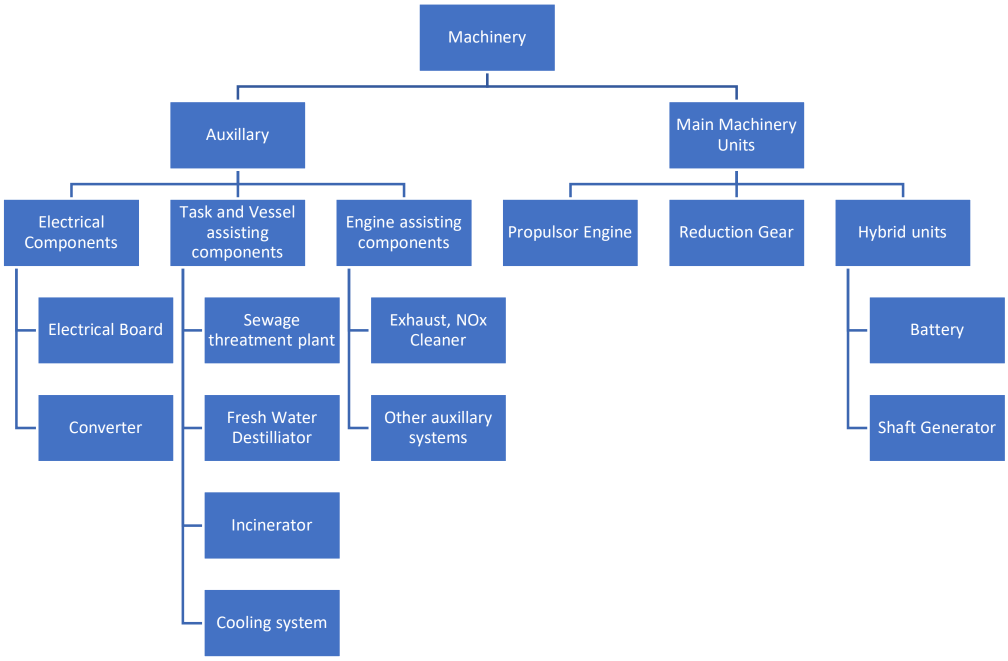


Figure 75: Machinery Main Components

Main Machinery

Weights from manufacture was found for Propulsor engine, auxiliary engine and two-speed reduction gear. Shaft generator was informed by our supervisor since we could not find any product info, and battery was informed by Holmseth.

- Propulsor Engine: Yanmar 6N330-SW,
  Total dry weight with marine gear 43385kg
- Auxiliary Engine: Yanmar 6EY22LW,
  Dry weight engine 11000kg, gen. set 18100kg
- Two Speed Reduction Gear, Wartsila SCV80/2-P54
  Weight 14000kg
- Retractable Azimuth Thruster 16500kg from Kongsberg
- Shaft Generator 5000kg
- Battery 3000kg

Auxiliary

- Wartsila Serck Como single-stage Destilliator: 620 kg +10%kg
- ACO Clarimar MF Sewage Treatment Plant, Clarimar MF-3: 3660kg
- Incinerator: 3000 kg
- Electrical Board: 6000kg
- Exhaust and NOx-reduction: 2200kg
- Cooling compressors: 12000kg

- Other Auxiliary Systems: 200 tons;
  preliminary formula pg. 176-177 (H. Schneekluth & V. Bertram, 1998)

Sewage treatment, and distillatory was found from product catalogues. Wartsilla SSD says that during operation it has 10% added weight. Incinerator, electrical Board, exhaust and NOx reduction, cooling compressor. Other auxiliary system weight was calculated from ((H. Schneekluth & V. Bertram, 1998), but at a later stage were found that that value was a high overestimate. Therefore, our supervisor gave us an estimate of 30 tons at a later stage. Normally other auxiliary system is found by using earlier vessels and scaling up.

Electrical Board is with also transformer, the transformer might vary widely in weight from manufacturer to manufacturer, the estimate above were given by our supervisor.

Main Equipment

Main equipment on this vessel consist of two deck cranes, elevator, MOB-boat, elevator, factory equipment, hauling equipment and azimuth.

Deck Cranes

In this ship it needs to cranes. Since the weight and dimensions are not available from suppliers the weight of these two cranes taken from (H. Schneekluth & V. Bertram, 1998, p. 171). Both cranes should have sufficient max load 10 tons. The crane in the front only needs max working radius of 15 m and crane at the back need to reach from deck to aft, which is approximately 20m.

- Front crane 18 tonnes 17.25
- Back Crane 22 tonnes 14.35

The book also mentions the centre of weight for a crane is 25-30% of height of the crane.

Lift

Elevator weight consist of the machinery and weight due to extra steel structure. The extra steel structure necessary was roughly calculated to be 3 tons, while our supervisor gave us the value for the elevator machinery at 1 ton. For a total of 4 tons. The centre of weight was measured to be approximately at 40% of the height, due to the added weight from machinery at the bottom.

MOB-BOAT

The weight of a MOB-boat including the crane was estimated by our supervisor to be 3.5 tonnage. Due to the weight mostly consisting of a crane, was the centre of gravity estimated to be 30% above deck height.

Factory Equipment

Consist of many machineries and therefore was an impossible for us to determine the factor for. Therefore, we were given an estimate from *Skipteknisk* of 50 tonnage, were most of the weight is located at the vertical plate freezers. Therefore, the LCG was determined to be forward regarding the placement of the vertical plate freezers. The

VCG was assumed to be approximately 1 meter above the deck, cause the centre of gravity for the equipment is located towards the bottom of the deck.

Hauling Equipment

The hauling equipment where already estimated. Same as the factory VCG where estimated to be 1 meter above the deck.

	Weight
Baiter	0.38
Hauler	1.1
Separator	0.3
Mag-pack	6.5
Total	8.28

Table 6: Hauling Equipment

Results of lightweight

Lightweight					
Lightweight	Weight	LCG	VCG	LCG*m	VCG*m
Hull	1309.16	31.87	7.25	41718	9492.676
Pipes and etc	123.20	31.5	5.5	3880.8	677.6
Machinery	321.28	19.32	1.82	6205.785	584.22
Outfitting	175.84	37.65648	13.92635	6621.456	2448.788
Factory Equipment	50	44.75	7.5	2237.5	375
Hauling	8.28	13.19	7.50	109.205	62.1
Elevator	20.00	22	8.5	440	170
Azimuth	15	57	4.5	855	67.5
Coolers	12.00	50	4.76	750	71.4
Crane aft	22.00	20.5	13.85	451	304.7
Crane fwd	18	52	16.75	936	301.5
Total	2077.76	30.90101	7.005387	64204.75	14555.48

Table 7: Results of Lightweight

The results from the total lightweight represented to be a not so different compared to the similar vessels. Therefore, confident that this weight estimate, will help us get started. One surprising element though is the LCG being behind midship in lightweight. Probably because of the high weight from the machinery since this was calculated to be a very significant component. Hard to know if the correct machinery is correct estimated, as not much information to compare it too.

Curves and Lines

After establishing both a GA and a rough weight estimation, we start constructing the shape of the hull that will surround it. The necessity to evaluate if the placement of the weights in the arrangement can give enough stability for the vessel means we have to check it according to the hull shape. The hull shape and lines are the basis for the ship's hydrostatic values. For instance, KM (metacentric height), which is compared with the vertical centre of gravity (VCG) to determine the vessel's initial stability (GM). Furthermore, the LCB and LCF plays a part in the trim of the vessel, which we also need to establish. So, after establishing a GA that is works with the given space, we have to determine how the weights of the different parts effect it. And if it is still functional, most likely it will have to undergo some changes. Nevertheless, this is all part of the design spiral.

In order to "test" the designed hull we will create a 3D model of the hull in Maxsurf, where we defined the zero point vertically at the baseline, bottom part of the hull. The longitudinal zero point is defined as the most aft point of the hull at DWL, in this paper this is referred to as the Aft Perpendicular (AP). The Forward Perpendicular (FP) is defined as the most forward point of the hull at DWL.

Ship Motions

A ship can be regarded as a solid body rotating freely in 6 degrees. Three degrees are regarded as linear motions, such as surge which is an acceleration longitudinal. Sway are acceleration, sideways along y-axis on figure 76. Yaw are the upwards linear acceleration. Last three degrees of motion are rotational, such as roll, pitch and yaw.

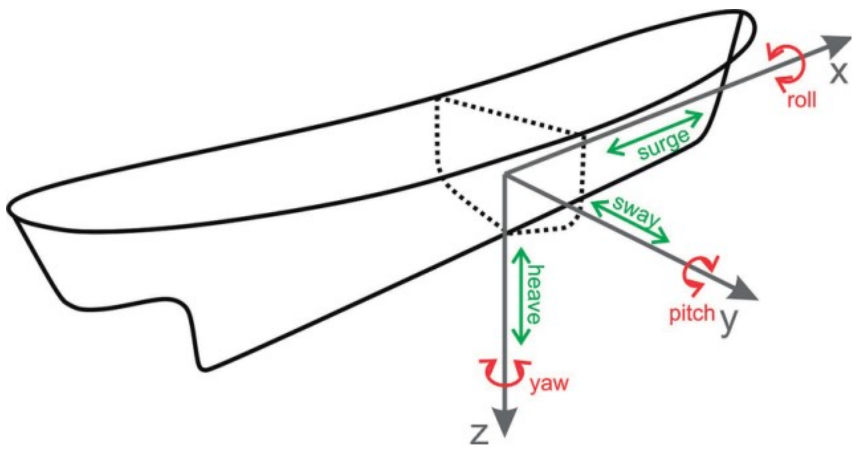


Figure 76 - Ship Motion; 6 Degrees of Freedom (DOF)
https://www.worldmaritimeaffairs.com/ship-motion-6-degrees-of-freedom-dof/

Understanding how these rotational motions are affected by the hull is important. Due to Archimedes law, a larger ratio of increase in the bow regarding create a shorter period. That is crucial to know cause such forces as roll and pitch have a tangential force, which correspond with the acceleration due to Newtons second law. If the period is halved, the tangential velocity is doubled by moving both upwards and downwards. If the velocity is doubled then the acceleration is quadruplet. So even though one manages to limit the distance of the motion by half. The acceleration can be quadruplet, which can cause the ship to be highly uncomfortable. Same regard also linear forces.

Parametric Rolling

When a wave moves along the hull, the area of waterplane changes. Since transverse BM are defined as equation under.

$$BM = Moment\ of\ inertia\ of\ waterplane/volume\ desplacement$$

Changing of the waterplane changes the moment of inertia, causing the righting lever (GM) to vary. Which can cause the ship to start rolling (wartsila - parametric-rolling, 2020). Since the roll is dependent on time, if the variations synchronous is, can cause the ship to roll more and more each wave cycle which is called parametric roll.

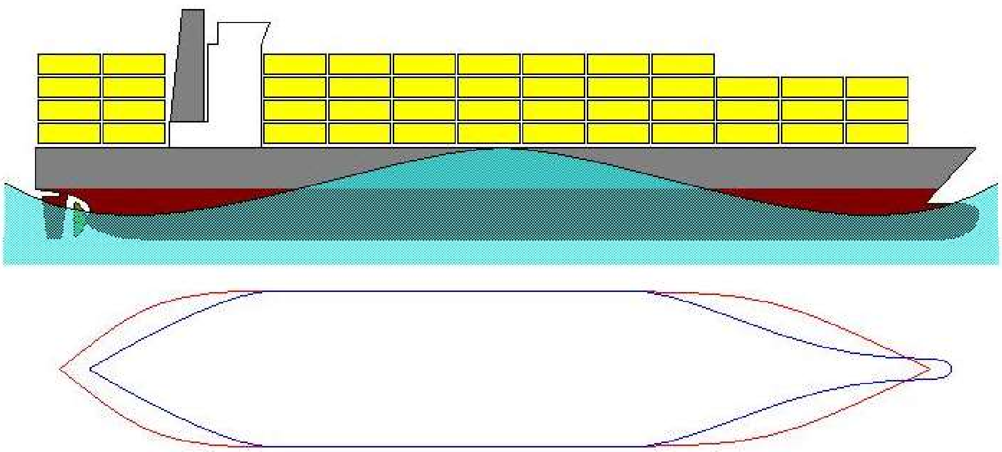


Figure 77 - Parametric Rolling
https://www.marineinsight.com/marine-safety/what-is-parametric-rolling-in-container-ships/

It is important to take into account parametric rolling when designing a vessel. As the parametric rolling can cause the ship to roll at extreme degrees. Causing discomfort for crew, damage to cargo, damage to the hull structure or even sinking the ship.

Natural Roll Period

When a ship is rolls, every hull has each own time before the roll is righted. This is called the natural rolling period. Which can be significant since if it is at resonance, can the roll be increased more and more. Example if one pushes at a swing with small push, causing the force to increase more and more.

Anti-roll tank

Since a fishing vessel needs to reduce the high roll effect of the ship due to high seas operation. To achieve roll damping of a vessel a tank is fitted in order to use the free water surface effect since water starts moving as soon as the angle of ship is increased, utilizing the free surface effect. Due to the limited space on the side of the fishing vessel will this be a simple rectangle filled with a certain percentage of water. In order to slow the water effectively down the rectangle space is fitted with 3 slosh dampers with 60% opening typically in order to slow the water down. That is to get the roll damping effect of the water to one side as seen in figure 78. To stabilize the ship optimum, the tank should span over the entire breadth of the ship as the water can exert maximum force. The tank should be able to exert the maximum stabilizing effect at the maximum possible roll. The tank should also be placed as high as possible as that will create a greater roll damping effect due to it being as far away from metacentre as possible.

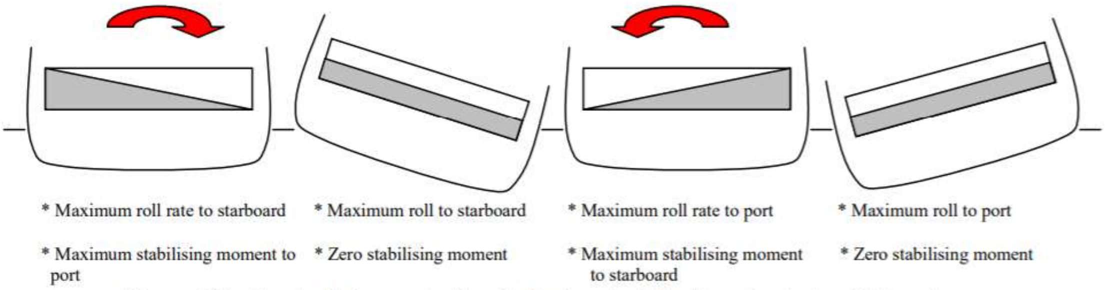


Figure 4 Surface tank demonstrating the basic principle of passive tank roll damping.

Figure 78 - Basic Princieple of Passive rolll damping tanks (Winden, 2009)

Since a fishing vessel vary greatly in weight, will also the metacentre vary greatly from different load-conditions. During heavy cargo or full tanks, a high metacentre will affect the ship by having larger correcting force, this can affect the work-effectivity in the vessel. During these conditions, the tank can also work by being filled completely to lower the metacentre height.

If one should face low stability it is a requirement that the tank should be emptied fast in order increase the stability. Depending on what kind of substance the tank consists of where one can empty. Either the tank can have ballast water, but then the tank needs to be properly made anti corrosive beforehand, as else the water would need treatment. Optimum choice was to make this tank of reserve fuel oil, to use the volume efficient. To include reserve fuel oil a tank below would need to ready to be emptied into, as one cannot empty fuel oil straight into the ocean. At a later stage were utilized as a ballast, as enough fuel was included in the double bottom tanks. Also, since fuel tanks are locating far from the machinery, adding unnecessary space for piping.

Since a ship have a specific natural roll period, which means the time it takes for the ship time for correcting. If even small waves have a natural frequency equal to this can the rolling of the ship greatly increase to create large roll.

Bilge Keel

Bilge keel lowers the maximum roll at resonance, by creating a counteracting force to the roll (Samolescu & Radu, 2002). The bilge should not extend farther than the width or depth of the vessel. As being wider or deeper at a point can harm the vessel.

Hull Resistance Theory

Chapter 3 (Molland, Turnock, & Hudson, Ship Resistance and Propulsion Practical Estimation of Ship Propulsive Power, 2011) explains that the total resistance for a hull can be summed up by three main groups.

1. Viscous Resistance
2. Viscous pressure Resistance
3. Wave Resistance

Firstly, the viscous forces work by bounding a layer of water to the hull, which affects the water outwards in a specific distance called the boundary layer. The forces result in tangential shear forces that make it necessary to create a counteracting force to move at a constant speed. Friction resistance acts on the entire wetted surface of the hull. Frictional forces are increased with a higher hull roughness. Also, such appendixes as bilge keel will increase the frictional resistance. Due to Bernoulli is the velocity around a hull not constant since the pressure varies due to 3d effects of flow around the hull. That is defined as the form factor, which is a factor multiplied by the frictional resistance to obtain the total viscous resistance.

The boundary layers cause typically a turbulent flow. Viscous pressure resistance is a pressure resistance due to viscous effects on fluid flow. Which normally occurs due to separation in the stern. The degree of separation determines the amount of resistance. Since normally around the stern of a hull, there is a low degree of separation, are the viscous pressure resistance low. Therefore, this is typically included in form factor (Øyvind Gjerde Kamsvåg, Lecture Notes). Where viscous pressure resistance can be a more significant factor are in the appendixes of the ship, for example, stern transom and side thrusters.

When the hull is moving through the water the pressure differences creates waves, due to the same effect to fluid flow around a hull as viscous pressure resistance. A larger difference in pressure will create larger waves. The total energy necessary to create waves, breaking waves and spray cause an added resistance. Total wave resistance is a sum of a series of different waves created a long the hull. The hull creates bow, stern, shoulder, and symmetrical waves.

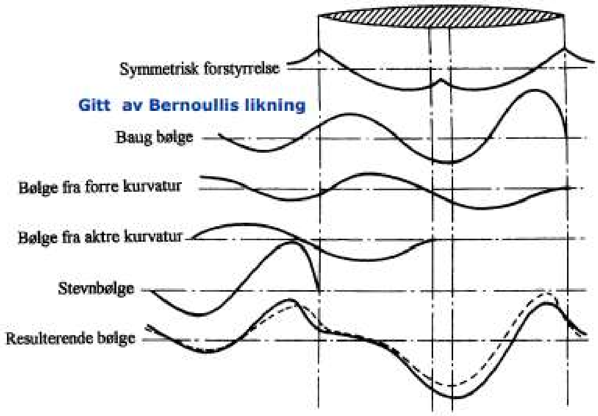


Figure 79: Wave Picture Around a Hull
(Øyvind Gjerde Kamsvåg, Lecture Notes)

A propeller accelerates the water behind does a propeller cause the hull to have more resistance. The added effect is that increase in fluid flow around the stern can cause more separation. Therefore, a propeller can both increase frictional resistance and wave resistance. That added resistance is called thrust deduction.

Hull Effect on Propeller

The flow to the propeller near a hull affects its performance. That is due to the flow around the hull in the stern section is slower than the ships speed affecting the propeller negatively as it must consume more energy. The flow is slower due to several factors combining into the total wake fraction such as potential wake, wave wake and frictional wake (Molland, Turnock, & Hudson, Ship Resistance and Propulsion Practical Estimation of Ship Propulsive Power, 2011) Chapter 8.

Potential wake is not affected by the friction in the liquid, it is purely affected by how water flows around an object. In the stern section of the vessel the flow combines after going around the object increasing the pressure and by Bernoulli's principle reducing the velocity of the flow. That is happens typically where the propeller is situated. Frictional wake is the reduction of the flow caused by the friction from the hull. Which is due to the water sticking to surfaces causing the flow to move the same speed as the hull. In hydrodynamics, that is referred to as being the same speed, called the no-slip condition. A section of the water outwards from the hull is affected called a boundary slowing the water down due to friction. Wave wake is induced by the orbital motion of waves created by the difference in pressure from midsection to the stern section.

Frictional wake is the largest factor, nevertheless potential wake is also a significant factor. For a single screw vessel, the wave wake is a smallest factor. Total wake typically is between 0.2 to 0.4 depending on the hull (Molland, Turnock, & Hudson, Ship Resistance and Propulsion Practical Estimation of Ship Propulsive Power, 2011).

Wake of the vessel can be measured in two different ways, either measuring the wake without a propeller called nominal wake or with a propeller called effective wake.

The Incentive or Initiative

The owner specified that the ship should be beautiful, however, there are numerous ways for a ship to be beautiful so this could not work as a basis for the design of the lines. Nevertheless, it would work as an incentive to present a thoughtful design that follows a clear vision of what the ship expresses. We wanted the vessel to step out of the herd and speak to something new in the long-line industry. Where new ideas and functions complement each-other in a single unit. It is important to us that the ship is not simply multiple parts added together, it is on the other hand a fellowship of systems that compromises to work together. To showcase this all the way through the design of the hull we aim to let lines follow the whole length of the ship, curving into each-other rather that cut each-other off by angles.

A Bow from the Past with a Modern Flare for the Arts

First and foremost, the bow of the ship is to reduce the resistance by having a smooth water flow. In order to reduce wave making resistance, but a good bow for a long-liner is also dependent on several factors listed below:

- Low resistance in calm water in several speeds, to reduce the fuel consumption.
- Low resistance at several draft levels.
- Minimum added resistance in waves, as this vessel will operate during high seas. That is an important factor since the operating oceans tend to have a high sea.
- Minimal pitch motion and acceleration, to be comfortable for the crew.
- Minimize the level of spray and green water. Spray water is only disadvantage that it can knock a crew member of deck and reduce the visibility. The importance is to reduce the green water since green water can damage equipment and structure of the ship.
- Minimize jerky motions both in pitch and surge

To achieve all these factors are impossible, since making a slimmer bow decreases resistance, but increases risk for green water immersion. Some points are counterintuitive, as one might think shortening pitch motion leads to a more comfortable vessel, but that might increase the acceleration. Therefore, a compromise between the factors will be necessary to achieve the optimum bow.

Most long-liners favour the traditional bow with a bulbous due to several factors. The benefits of having a traditional bow is that the flare above greatly minimize the chance for green water on deck. Also, the flare deflects spray greatly. Disadvantage is that the bow produces large jerky forces in heavy seas. Large waves can rob all the forward speed, especially at lower speed. That can greatly increase the resistance in high seas. Typically, a bulbous bow is optimum for calm sea state.

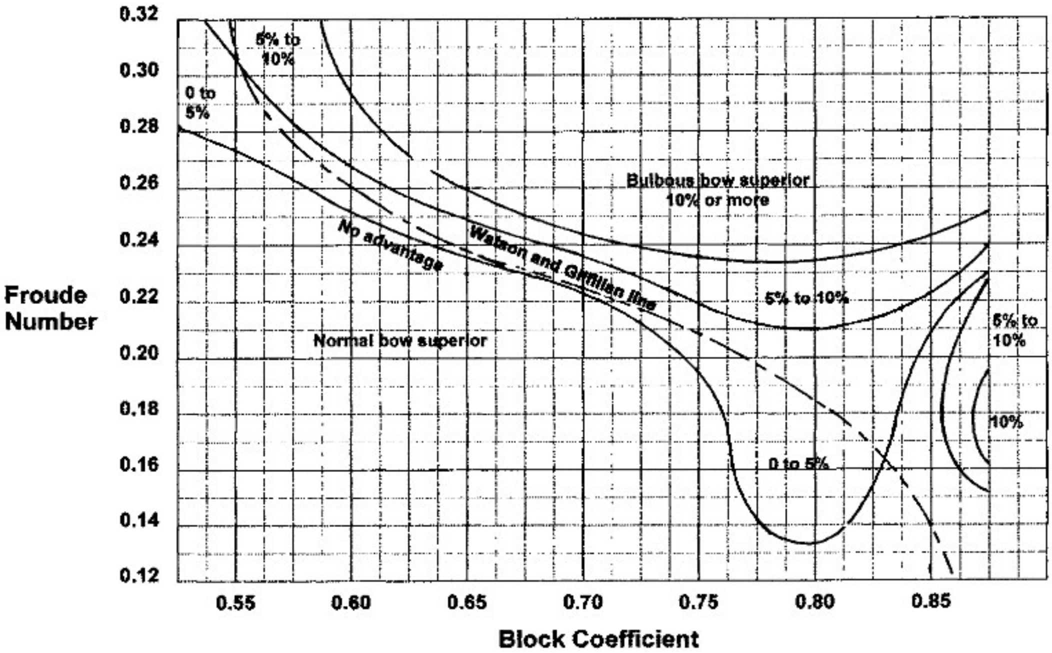


Figure 80- The Combination of Froude number and block coefficient at which bulbous bow are likely to be advantageous (Watson D. G., 1998)

Froude number is a dimensionless value of the ratio of the inertia forces and the gravity of fluid particles. That gives a value for wave making resistance for a vessel.

$$F_n = \frac{u}{\sqrt{g \cdot LWL}}$$

For our vessel is the Froude number 0.227 at 11 knops, 0.186 at 9 knops, 0.0457 at 2.3 knops. Since a bulbous bow is typically efficient at higher Froude number over 0.20 depending on the Cb as seen in figure 80, will it be inefficient during hauling and setting which is approximately 70-80% of the time. Also, since the vessels forward trim vary highly will it only work as lengthening of waterline during low bow trim conditions and a bulbous bow only have optimum efficiency at design speed. All those factors reduce the bulbous bow to low efficiency for a long-liner. A bulbous bow will be near inefficient at 70-90% of the time. Designing without a bulbous bow propose a risk since wave resistance can increase significant if block coefficient is above 0.75 as seen figure 80.

Since the comfortability is an important factor as large jerky pitch motions hinders the work in rough seas. The idea is to increase comfortability on behalf of added resistance in transit in calm water. Idea is to add volume at the bow of the ship, to reduce the pitch motion and at the same time work as lengthening of the waterline. Also, to get a gradual pitch motion to remove the large jerky motion but having more volume in front might cause more surge. That will be explored by having a straight stem with gradually increasing volume over the waterline. Hopefully, this would alleviate some forces acting on the refrigeration system in front.

Although it does have some drawbacks, it is expensive and difficult to produce in addition to not having flare to push the water to the sides. The latter one is something we can work through because we would like the bow to spread out with a more normal flare at a certain height. This will not have an effect with low waves that will not be a problem anyways, however, when the waves rise higher, they will meet the flare and washed to the sides.

Stern

The stern might not always get the right attention, it is hidden behind the vessels instead of being a showpiece like the bow. However, it is a grinder, it does a lot of work and must be deigned with that in mind. Obviously, it has to be raised up from the base line to make room for the propeller, but how it will be raised up is there now good answer to. The lower degree of rise gives better flow, yet it has to be raised up "quick" enough propeller clearance and lead the water to the surface. You also have to take into account the flow to the propeller, yet there must still be enough volume for the machinery.

8.2.4 Sterns

Sterns have to be considered in relation to the following roles:

(i) the accommodation of the propeller(s) with good clearances that will avoid propeller excited vibration problems;
(ii) the provision of good flow to the rudder(s) to ensure both good steering and good course stability;
(iii) the termination of the ships waterlines in a way that minimises separation and therefore resistance;
(iv) the termination of the ships structure in a way that provides the required supports for the propeller(s) and rudder(s) plus the necessary space for steering gear, stern mooring and towage equipment etc. and is economical to construct.

Figure 81 How to Design Good Stern Lines
(Watson D. G., 1998)

Intuitively rounding of the sides of the stern lowers the resistance as it contributes to less separation. So, it will be implemented in the early stages of the design and assessed to what degree it shall be used further out in the process, depending of what other situations allow.

Immediate Concern

The plan was to form the hull to accommodate both the internal space needed for the GA and the weight estimation. From the weight estimation we got a value for LCG which we could match the LCB with the hydrostatics from the hull, which was not a problem. The problem came when we focused on getting a KM higher than the VCG, which we really struggled with. The importance of these values will become clear when we evaluate the stability. Nevertheless, we could already see that this would become a problem so it was made a quick switch to aluminium for the superstructure before moving into stability. This lowered the VCG to 6.7, but this will be further explored in Stability.

20	BMt	3,405	m
21	BML	67,270	m
22	GMt corrected	-5,905	m
23	GML	57,960	m
24	KMt	6,595	m
25	KML	70,460	m

Figure 82: KMt generated thorugh Maxsurf

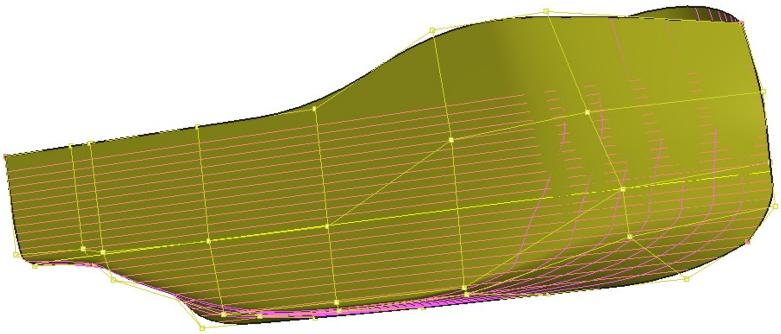


Figure 83: First Maxsurf Modell

Stability

For our vessel, is stability the first aspect we would want to "test" with the line design. Some of the systems that carry large weight are placed quite high up and can have drastic effect on the Stability. We have kept the tanks and main storage low, as intuitive decisions to maintain good stability. However, the bait storage is on a higher deck then most other long-liners and some of the superstructure are as well. Therefore, are there quite a few uncertainties in how stable the vessel will be.

It is generally known that a vessel with a higher length to beam ratio runs more efficiently through the water. However, narrow ships have lower stability which can lead to problems if the VCG does not allow it. There is always an aim to keep the resistance of a vessel to a minimum, yet enough beam to give the appropriate volume onboard and stability. Till this point we have kept us under 13.5 metres in the beam, which is half a metre lower than the requirement. The VCG seems to be quite high so we are concerned that a 13.5 metre beam will not give enough stability, yet we proceed to try.

Due to the fact that most of the weights where elevated in order to make room for large enough tanks, the VCG crept up to about 6.7 metres above baseline. The hydrostatics for the model gave us an KM for 6.595 metres, meaning we would have a negative GM. Nevertheless, before we start to increase the total beam of the vessel, we start to change the lines a bit to maximize the stability. The stability will increase if the waterplane are increases or the displacement decreases, and since we cannot change the latter at this stage, we aim to change the former. We have already increased the volume in the bow to make volume for the tanks, so in order to increase the stability we "move" that volume up from the baseline to the waterline. This gave a bit extra, but nothing to significant. So, we move on the look at the stern of the vessel, where we had previously sloped the curves as this is known to give less resistance. However, since the resistance would increase more if we would have to increase the beam, we are willing to "sacrifice" the curves for better stability. Keeping the full beam further aft and not lifting it up from the water resulted in a massive increase in the KM. Including some small alterations all over the ship, we were able to increase the KM to 6.8 metres above baseline.

Load Conditions

From DNV there are a number of load conditions that have to satisfy a given GM demand. All the relevant conditions must be must pass in order to achieve the classification. For fishing vessel this consists of (DNV GL, 2019, p. 33):

- Light ship
- Departure (100% tanks (fuel, urea and fresh water), bait and supplies, with no catch)
- Departure from fishing ground (100% catch with 30% in tanks, bait and supplies)
- Arrival (100% catch and 10% in tanks, bait and supplies)
- Arrival 20% (20% catch and 10% tanks, bait and supplies)

There is also a condition given in the rules with maximum catch on deck and 50% fuel etc, although it is noted that this should only be considered if it is “consistent with fishing method” (DNV GL, 2019, p. 33). For a long liner it is not part of the operation, since the catch is brought on one by one and not like for instance a trawler where the whole catch is lifted up in a “group”. Therefore, is this neglected for our vessel.

The light ship condition only requires a GM more than zero, for the operational conditions the requirement is 0.35 metre (DNV GL, 2019, p. 32).

Checking Tanks

After making these alterations to increase the stability of the vessel we had to go back and check if there is still enough volume in the tanks and other compartments from the GA. There were some wiggle room before the alterations, and we therefore knew we could decrease the volume a bit (approximately 40 m^3 in the tanks), nevertheless we would have to check that we did not overdo it. In order to do this, check we put the new hull into Maxsurf Stability with the load cases and tank arrangement from the previous design. Then we ran the software and received some values for the tank volumes as well as for the stability in each load case. The volume had decreased a bit, but not enough to go below the requirement. When it comes to the stability, the alterations made a positive impact and the GM came out positive for all the given load cases, as long as the free surface effect in the tanks where ignored. For this tank arrangement divided we the tanks quite roughly and unrealistic. The double bottom was split at each ten-metre interval in order to use the them to even the trim. Nevertheless, the remaining tanks where not split in any way and these as well the double bottom would in reality be divided into smaller compartments. In addition to that they would also feature baffles, holed-out plates to reduce sloshing. All in all, this means that the free water surface effect calculated by Maxsurf with the values we have given will be higher than the actual values. Therefore, do we choose to neglect them for our first estimations.

Trim

As established in the operation profile of a long-liner, will the vessel experience a couple of different load conditions. Departure; where tanks, bait and food are filled, Arrival; where storage is hopefully at a maximum while the others are low, and all that is in-between. The different condition will affect the trim of the vessel in different ways, the storage is placed in front of the moonpool and therefore will contribute to a forward trim (trim by the Bow). This is something we would want to avoid; the ship handles better and runs more efficiently when the trim is level. We can accept a bit of stern trim at some conditions, but forward trim should be eliminated. However, we know there will be some trim when the position of the large weight changes. So, to start we focus on keeping a level trim when we have filled up the storage and finishing the trip.

Arrival

The vessel has caught enough fish and are heading ashore, meaning it is filled with the deadweight that will be sold for money. This is most likely the highest displacement of the vessel, since this is the main purpose of the ship. The captain would not like to add ballast to a ship in this condition, since that would mean added weight without any added value. Therefore, are we starting with this condition to create a hull that benefits this load condition in an extent that there will be no need for added ballast.

The weight was now further improved, since the hull shape existed, as earlier the hull areas, were just an estimate. The measurement was done in Siemens NX. These in addition to the LCF, which we know should be close to the moonpool/hauler, are used as hydrostatics aims for the 3D model. If the LCB can match the position of the LCG we know that the trim will be even for this condition, and then we can check the other conditions.

The LCG are calculated to be about a metre in front of midship (z-position), the VCG about 6.7 metres from baseline and the displacement 2800 tonnes. This are our aim values for the first attempt at the lines, that will be inside of the requirements: 63 metres long and 14 metres wide.

Problem with Tank Volume

After having made a design that would satisfy the values calculated for the Arrival load case we ran into some problems. Firstly, when we filled out the volume, we had planned to use for the tanks we quickly realized that it could not satisfy the required tank capacity. We had planned in the GA to mainly use the double bottom and the forward peak as tanks; however, we also used the minimum value for the height of the double bottom and we had a quite narrow design in the bow. As a result, the tank capacity where significantly lower than what it should be.

To rectify this there where made some changes, firstly the double bottom where increased from 760mm to 1 metre. Then we added a 500m high tank underneath the main storage, this also contributed to a height increase in the machinery which would allow us to smoothen the lines in the stern. We also added a tank under the cooling compressors, to all in all add up to almost 500 m^3 of tank volume (assuming 80 percent filling in DB and Peak, while 90 percent of the rest).

First Circulation of the Spiral

New Draught

As we continued to assess the different load conditions, it appears that the draught never matches the assumed 5.5 metres. Departing at 4.97 metres and arriving at 5.197 metres indicates that the designed waterline should be around that area. A problem with having a to high design waterline could be that the transom will be lifted to high and decreasing the stability and wrongly assessing the hydrodynamic properties.

Therefore, did we lower the design draught to 5.2 in order to make it comparable to the load conditions we had. This meant lowering the stern to an appropriate height, in order to make sure the transom stays in the water for most of the operation time. The bow on the other hand has quite straight lines in the vertical direction and is therefore affected it a such low degree that it makes minimal difference. These straight lines came from the vision we incorporated in the design from stage one. Hence, the bow is unaltered in this process and will remain identical to the previous, only difference being the height of the design waterline.

After making this new design for this new design draught the model is entered into Maxsurf once more. Here it was checked again that the stability, trim and draught all are acceptable for this new design. Since we did very few changes there where not a lot of surprises, and the design passed the given stages.

Vive la Resitance

After the design of the hull gives the hydrostatic properties to comfort the stability and trim as well as making room for the volume given in the GA, it is time for the design to face the next stage. Even though the hydrodynamic properties to keep a low resistance is so imbedded in our brain stem that a lot of intuitive choices went into the hull design to maintain a low resistance, the actual resistance values are still unknown. The resistance of the hull is always kept in mind while designing it, so keeping the angle of incident as low as possible in addition to a smoothly raise the transom are all aspects that are expected before even making the first draft of the GA. We know that the volume in the bow and the stern are massively limited by the lines of the hull of other vessel and there is no way around it. That is why we kept the compartments in these spaces at a minimum, however all of the judgments on the space available are purely intuitive, or educated guesses.

When it comes to the space that we have available, we have now true values before we start designing the lines. The fact that most of these spaces where used for tanks explains why they were wrongly assessed in account to volume. They were guessed due to the fact that they were second priority and we know that we could make changes after we had got some values for the volume. After the first design of the hull, there were made changes to both the hull and the GA to accommodate the necessary tanks.

After this there were made some changes on the hull to achieve the necessary stability, then some changes to fit the new draught. While doing all these alterations to accommodate the different aspects of the design is it easy to forget about the resistance. Because the fact is, we do not have any specific good way to estimate the resistance. So, for all the intuitive choices and educated guesses we put into the design of the hull, few of them make it all the way. In order to continuously keep this in mind, did we regularly insert the “new” design into Maxsurf Resistance. There we assessed the empirical calculations and the wave picture generate to get a feeling where we were on the spectre.

Then with a design that satisfied all the other needs we inserted it once more and assessed it based on the previous designs. And although we encountered a setback in the early process, which is natural, we were able to always have the resistance in the back of our minds so, the final design ended up better than the first.

Although it should be noted that we took an overall assessment of the last design to make it smooth before condoning the tests. All in all, we are satisfied at this stage that the ship seems to be able to hold 11 knots without breaking 400 kw, and at the required power (2400 kw) surpass 15 knots and almost hit 16. Which is quite far from the requirement/expectations from the owner, hence we assume that this is poor estimate from the software due to empirical calculations that do not fit our design. We will come back to this while continuing the design spiral, however at this stage we asses that the importance of effective lines does not indulge further treatment of the design of lines.

New Weight Estimation

By going along the design spiral there will always come up new changes that makes us go back and recheck the different aspects of the design. In addition to gathering new information over time in order to improve our estimates, for instance the weight estimate. As we have moved forwards in the design process, we gather some new intel on the side, by finding new sources or some common estimates from our supervisors. This means we are constantly updating our weight estimate, and thereby we also need to update the stability conditions. Some changes are small and does not cause major changes, nevertheless many small changes together or larger changes can cause changes that need to be dealt with. A dramatic update on the estimate for the auxiliary components in the engine room, along with some other updates, resulted in a significant lower displacement in addition to a higher VCG. Thereby, did our existing lines not suffice with the new loads, and the stability requirements were not met.

All in all, this required some revision of the line design, lowering the displacement and increasing the KM. firstly we lower the draught to about 5 metres in order to lower the displacement without taking away from the inside volume (GA). Although this proves not to be enough so we also focus on slandering the hull under the waterline. While this is done under the waterline, the waterline itself is enlarged, with focus on maintaining the beam further aft and forwards to enlarge the BM and thereby the KM. These two alterations are somewhat conflicting, increasing the water area while minimizing the volume leads to increased angle radius of the bilge and/or introducing angle of the keel.

Furthermore, it should not be forgotten that this vessel will operate on different draughts which all needs to be stable, and if the waterplane area is too low for smaller draughts the stability for these conditions becomes impaired.

Induced Trim

Both the transom and bow are widened in the waterline, although due to the “requested” position of LCF and the danger of having a jerky surge motion in waves, we use the beam to a larger extent at the stern. Yet, the volume needs to be reduced in the stern to maintain the optimal LCB, resulting in a “flat and high” transom. Then stability becomes a problem with lower draughts. Then we have some options, we can lower the transom deeper into the water which compromises the LCB, and we are risking forward trim. Another option could be to slender in the transom further to force the LCB forward and inducing an aft trim. This might seem counterintuitive; however, this aft trim means that at lower draughts where we do not have so much weight to “move around” as ballast, the vessel gets a natural trim aft-wards. If the trim gets to a large enough extent the transom is “lowered” into the sea the waterplane area is increased. This helps for both the Light ship and the Arrival 20% condition and we can achieve enough stability. We opted to try the second option and with the help of the available “ballast” we achieved stability for the low draught conditions, without resulting in trim for the 100% catch conditions. And we maintain required trim for all conditions.

A negative effect of this trim is that the waterlines while trimming are quite awkward. Since the transom is designed to be level at design waterline with no trim, it angles down at the stern like a stern wedge when there is aft trim.

“Case Keel”

After we visited *Geir* in a dry dock and talked to our supervisor we became aware of the trend of filling a case with packed steel and mounting it under the keel of the vessel to achieve greater stability. Both as a weight lowering GM and increasing Initial Stability as well as increasing the directional stability due to the shape. Although our supervisor informed us that this is a possibility and widely used, we would want to avoid adding additional weight to our vessel. So, while we move forward, we keep this option in the back of our mind as a backup if the search for stability reveals to compromise the other factors too much.

Introducing Strength

At this point we have established sufficient estimations of the loads as well as general arrangement that can work as a foundation for the structural design. In addition, would we need to get the dimensions of the beams, stiffeners and plates in order to better asses the available space in the GA. Hence, we start the structural design process for this long-liner and create a first structure as a starting point. Then based on the strength results of that structure it will be revised and adapted. So that the dimensions can be implemented into the other sections of the design process, in order to increase the detailing and propel the design.

The ship strength affects several ship parameters, especially weight and utilizable deck volume, but also building cost. The hull is usually a shell structure consisting of plates, stiffeners and girders, where plates takes the initial load and transfer them over to stiffeners, which ultimately transfer the load to girders. The whole hull itself can be viewed as a massive girder, referred to as the hull girder, which experience global stresses form various loads and sea moments.

Loads

A ship experiences a variety of different loads each affecting different elements of the ship structure. Can be categorized into two main categories longitudinal loads and local loads.

Local loads

There are many forcers affecting the ship, from payload to forces making sure the ship floats (Yasuhisa , Yu, Masaki, & Tetsuo, 2009).

1. **Gravitational loads**
   Weights affected by gravity. This include structural weight, cargo, ballast, system weights etc. These are relatively constant, although cargo, ballast and fuel will change over time.
2. **Hydrodynamic and hydrostatic forces**
   The water pressure keeping the vessel floating. This load is constant at still water, but during waves the pressure can increase and reach higher points of the ship side than the design waterline.
3. **Dynamic loads**
   As waves affect the water pressure, it also affects the ships motion. Pitch, roll, surge and heave motions can make the static loads dynamic. A structure that almost collapses by a constant gravitational load, will collapse as the load increases by one of these motions.
4. **Impact loads**
   When the hull enters the water during a wave, the impact between the shell and the water creates a load. Sloshing in tanks will have the same effect.

Shell structure
Almost all the loads are taken up the hull and deck plating. The plates are what makes the ship watertight and allows the deck area to be used. However, only using plates, would be little effective, therefore one adds stiffeners. Stiffeners support the plates and reduce the plate thickens while remaining the same strength. In long structures like ships, the stiffeners will require support as well in the form of girders. This way one can imagine the shell structure being some sort of an open rectangular honeycomb. Plates, stiffeners, longitudinal and transverse girders are the main elements in the structural arrangement.

It is important to know how the loads are distributed and transferred in the structure, to know how one can improve the strength of the structure.

Plates
When reviewing plate strength, one first need to know that the load is transferred to the to the nearest supporting member. In a rectangular shape the load will be transferred to the longest sides. Plate strength can be calculated thus.

Reviewing a plate experiencing fluid pressure of other evenly distributed loads. Imagine a thin strip of the plate, supported by stiffeners at the ends.

$$M = \frac{qs^2}{c}$$

$$q = Pb$$

$$Z = \frac{bt^2}{6}$$

$$\sigma_{act} = \frac{M}{Z} = \frac{6qs^2}{cbt^2} = \frac{6Pbs^2}{cbt^2} = \frac{6Ps^2}{ct^2}$$

$$\sigma_{act} < \sigma_{crit}$$

As we can see, plate thickness and stiffener spacing is the main contributors to plate strength. Increasing the thickness and reducing the spacing will reduce the plate stress.

Stiffener
We draw the same assumptions as we did for the plates.

$$M = \frac{ql^2}{c} \; M = \frac{qx^2}{c}$$

$$q = Ps \;\; q = Pl$$

$$\sigma = \frac{Psl^2}{cZ} \sigma = \frac{Plx^2}{cZ}$$

$$\sigma_{act} < \sigma_{crit}$$

Here we can see several factors contributing to the stress in a stiffener. By increasing the section modulus, one reduces the stress, while increasing the girder spacing which supports the stiffener, one increases the stress, note that the spacing is in the power of two which makes the change critical. One can reduce the load on the stiffener by decreasing the stiffener spacing, and thereby decrease the stiffener stress.

Girders

The stiffener method could work on a singular girder with the same assumptions. However, the approach should be to review the girders as a system. The girders main purpose it to take the load from the stiffeners creating many point loads, but its more typical to gather them as a distributed load over each girder.

Longitudinal Loads

Imagine a ship as a girder, with structural weight, ballast, cargo pulling it down and the buoyancy holding the girder up. Even though the centre of gravity and buoyancy align, the individual loads are spread out unevenly creating bending moments. There are two different types of bending moments; Still-water moments and wave-moments each creating two additional moments; sagging and hogging.

Still Water Moments

In still water moments we imagine a ship where the buoyancy dispersion is constant, but the gravitational loads deviates. A case where there are great gravitational loads at each end of the ship will cause a hogging moment where the strength deck is in tensile stress and the bottom is compressed.

Figure 84 Stillwater hog https://present5.com/presentacii-2/20171208/7896-chapter_9_longitudinal_hull_strength.ppt/7896-chapter_9_longitudinal_hull_strength_35.jpg

A case where the gravitational loads are centre midships, the buoyancy at each end of the hull will not be "supported" and thus creating a sagging moment, where the strength deck is in compression and the bottom experience tensile stress.

Figure 85 Stillwater Sag (fig.42 link)

These cases would be examples of when the ship fills up or empties compartments with various load cases like a storage, ballast- or fuel tank. However, as is the nature of the ocean and the weather, the water really stays still.

Wave Moments

Changes in wave movements are more frequent than changes in still water moments. The gravitational loads stay relatively constant, while the waves change the location of the buoyancy forces.

Imagine a wave about the length of the ship; when the waves amplitude passes midship, we have excessive buoyancy in the middle causing the ship to hog, and when the amplitudes passes the perpendiculars, we have excessive buoyancy on the far ends of the ship causing it to sag.

Figure 86 wave-moment sag (fig.42 link)

Figure 87 wave-moment hog (fig 42 link)

Global Strength

The hull girder takes up the stress of these bending moments. The hull's shell structure acts like a girder and have its own neutral axis and section modulus. And just like one would with an actual girder, one checks the stresses against the allowable stresses, and uses those stresses to check for plate buckling.

It is important to know that the structural elements in contributing to global strength also experience local loads, and therefore be reviewed against a lower allowable stress since the capacity can't be utilized twice.

Torsional load and strength

When the ship is going through waves at an angle, like in the OSA example in figure 88, the deviation in buoyancy forces create torsion in the vessel. One could more accurately determine the dynamic loads by calculating the ship-motion with strip method, to calculate the dynamic forces applied for the vessel over a longer period of time (Yasuhisa , Yu, Masaki, & Tetsuo, 2009). That is a complex process and therefore the dynamic forces will be estimated from DNV-rules.

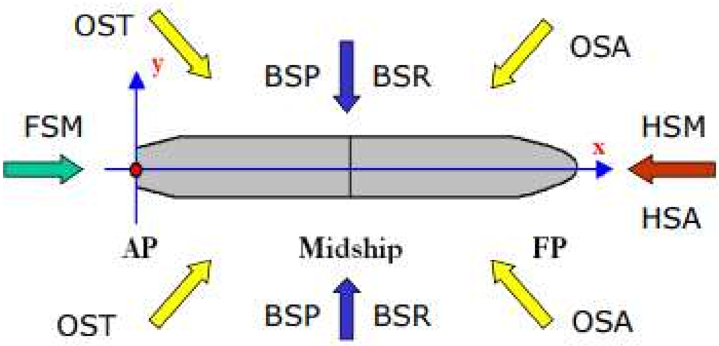


Figure 88 Wave load cases from DNV

Dynamic local loads

The ship will be affected by wave loads. Wave loads are categorized into slamming and green water. Therefore, a benefit from our bow design could be reduced bow slamming forces for the vessel. The vessel will still experience slamming in the aft of the ship.

Failure mode

Although there are many reasons for a structure to fail, collapse or deform, the three most common is yielding, buckling and fatigue.

Yielding

Material strength is in some cases measured by the load needed to deform a material. A stress-strain curve shows where deformation in a material changes from being elastic to being plastic. When the material experience elastic deformation as a result of an applied load, the material will resume it original state as the load is removed. At a certain point, the elastic behaviour changes to plastic, meaning the deformation is permanent. This point is called the yield point.

When using yielding as a failure mode, one usually reviews the material while it is in its elastic state and make sure the material does not exceed the yield point.

The tensile forces usually come in the form of bending moments working on girder profiles. They are calculated by determine the profiles section modulus, which is used to divide the bending moment of a section. A girder dimension is to be determined by the stress created by the greatest bending moment of the girder.

Buckling

When a girder, column or plates experience compression loads, structural engineer must review the elements buckling strength. Buckling is instability in an element; the element might fail before the yield load is reach due to the element deflects in a direction. Bucklin must therefore be considered separately form yielding.

There are two types of buckling to consider in a structural arrangement: plate buckling and column/girder buckling.

Plate buckling

$$\sigma_{Euler} = C\frac{\pi^2 E}{12(1-v^2)}\left(\frac{t}{s}\right)^2$$

Euler's equation for buckling can be viewed in three parts (Arne Jan Sollied, compendium, p.31). The C is the plates buckling coefficient. It is determined by the direction of loads on the plates and deviation of loads. The E stands for the materials Young's modulus and the v is the materials Poisson's ratio, which makes $\frac{\pi^2 E}{12(1-v^2)}$ constant for each material. The t is the plate thickness and *s* are the stiffener spacing, meaning $\left(\frac{t}{s}\right)^2$ represent the plates slimness.

To determine if the bulking strength is sufficient one determine the critical buckling strength.

$$\sigma_{Euler} \geq \frac{\sigma_{Yield}}{2} \quad \rightarrow \quad \sigma_{Critical} = \sigma_{Yield}\left(1 - \frac{\sigma_{Yield}}{4\sigma_{Euler}}\right)$$

$$\sigma_{Euler} \leq \frac{\sigma_{Yield}}{2} \quad \rightarrow \quad \sigma_{Critical} = \sigma_{Euler}$$

Then compare the critical buckling strength to the longitudinal ship tension. If the longitudinal ship tension exceeds the critical buckling strength, then measures must be made to increase the plates buckling strength.

Viewing Euler's equation for buckling one have three elements one can change to improve buckling strength. The buckling coefficient can be modified my changing the stiffener arrangement. A transverse stiffened topology will result in a buckling coefficient between 1 to 4, defined by the relation between the stiffener and girder. A great difference will result in a low coefficient and when the spacing is equal the coefficient will be 4. A longitudinal stiffened topology will result in a coefficient of 4 no matter the spacing relation.

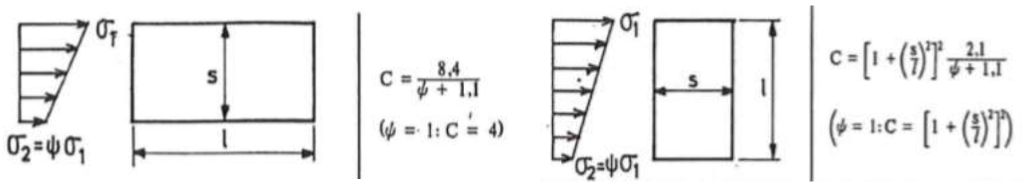


Figure 89 Buckling coefficient for pressure on the long side of the plate (Arne Jan Sollied, compendium, p.31)

Another measure can be altering the plate slimness. One can improve the buckling strength by increasing the plate thickness and reducing the stiffener spacing. This is a spectre of weight and construction cost. Increasing the plate thickest of a deck, ship side or bottom will result in added weight. Reducing the stiffener spacing will require more stiffeners to be added, resulting in increased building cost.

Column Buckling

There are two ways to review a column buckling strength. For a column that experience a uniform compression stresses over the whole profile, the following formula can be used.

$$P_{cr} = \frac{\pi^2 EI}{l^2}$$

P_{cr}=critical force [N]

I=Area of inertia to the column profile [m^4]

l=elastic length [m]

The elastic length is determined by the boundary condition. If the Column is fixed for rotation in one or both the elastic length shortens by a factor determined by the boundary condition. (Arne Jan Sollied, compendium, p.30)

The previous formula can be altered like this.

$$\frac{P_{cr}}{A} = \frac{\pi^2 EI}{l^2} \frac{1}{A} \rightarrow \ \sigma_{cr} = \pi^2 E \left(\frac{k}{l}\right)^2 \ [where\ k^2 = \frac{I}{A}]$$

This formula is great to use as one can compare the critical buckling stress to the material's yield point. One dimension the girder after whichever is the lowest. (Yasuhisa , Yu, Masaki, & Tetsuo, 2009)

Fatigue

Even though we say metals can exhibit elastic behaviour, this is not always the case on a microscopic level. Small loads that could never challenge the yield point of the material can still cause deformation in metallic crystals. If the load is applied repeatedly, the deformation can converge to greater cracks, resulting in a structural element losing its initial strength. (Falck-Ytter, 2014)

The two factors that causes fatigue are loads and frequency. A heavy load with a low frequency might be as damaging as a small load with a high frequency. Vibration from the engine, propeller, etc. can make static loads dynamic, causing seemingly insignificant deformations to take place. However, this happen at a high frequency which will eventually lead to a material failure. (Yasuhisa , Yu, Masaki, & Tetsuo, 2009)

The Structural Design Procedure

The SDP is an approach form Arne-Jan Sollied's compendium.

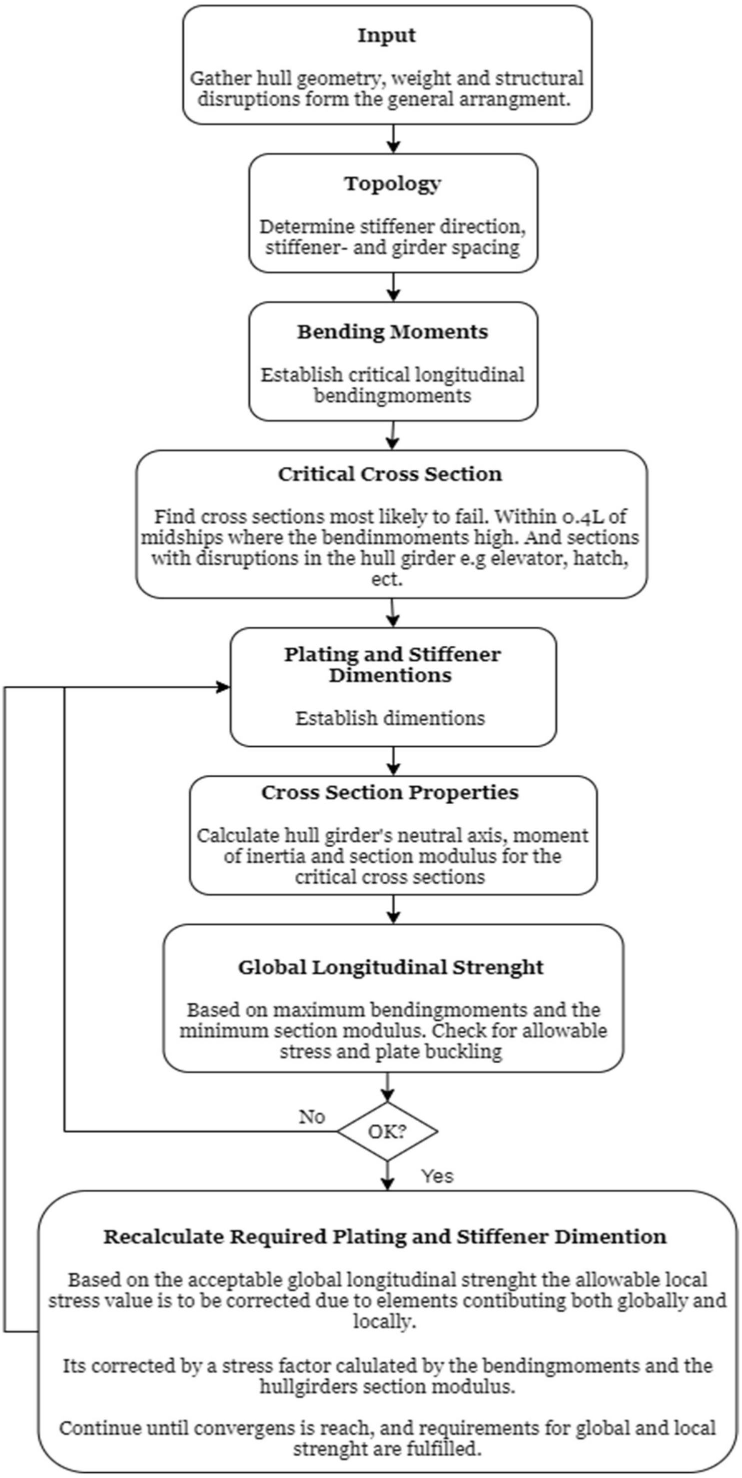


Figure 90: SDP form Arne-Jan Sollied's Lecture notes

The SDP only descript global and local strength on longitudinal members. After this one proceeds to review strength of the transverse members before one analysing structural detail like manholes covers and connections.

CAE

Computer aided engineering is the use of software to calculate physical effects on a virtual model. CAE is a repeating 3-part process, which starts with preparing a model for analysing. During the preparation, the engineers creates the geometry and the physical properties of the model, applies environmental loads and constrains. The second part is solving, where the software uses mathematical formulas based on fundamental physics to calculate the model's response. Finally, the engineer evaluates the results, evaluate if changes are needed restart the process if necessary (Bahman & Iannuzzo, 2018).

CAE are used to allow structural analysis to be performed quickly and perhaps more accurately.

CAE Tools

DNV has provided a set of CEA tools for rule check analysis called Nauticus hull. In Nautucus Hull, one enters the main dimensions and data for a ship, like LBD, Cb, speed ect. And which rules you want the tool to check against. Nauticus then determine what rules applies to the ship.

Section scantling

Section scantling is a rule check CEA tool that calculates both global and local strength. It is done by drawing a cross section of the ship with plates and stiffeners; girders are not analysed but longitudinal girders are added to contribute in the global strength.

The ship geometry may be drawn based on GA or extracted form a 3D model. One applies deck loads, as section scantlings adds the required hydrostatic loads automatically.

Initially one enters transverse- girder and stiffener spacing, so the program can calculate the local strength.

After analysing, section scantlings gather the result in a report for an engineer to evaluate. One gets plating- and stiffener requirement based on local and global strength compared with the actual dimensions based on requirements form DNV. Plating also receive requirement against buckling.

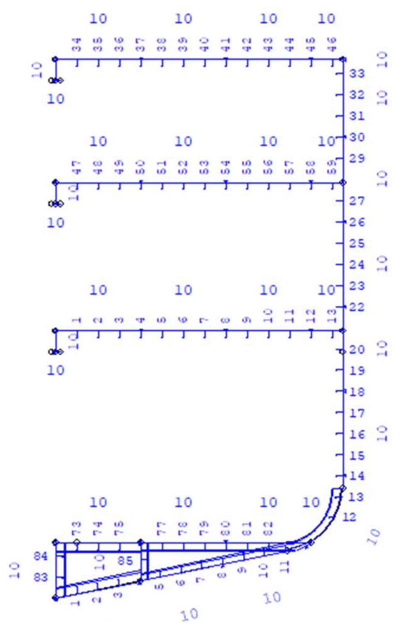


Figure 91 Example of a section scantling model

The software also gives you the section modulus of the cross section, along with neutral axis and moment of inertia.

3D-Beam

As the name suggest 3D-Beam analyses the 3D arrangement of a girder system. A structural engineer models the structure as accurately as possible within the programs calculating capabilities.

There are 4 steps one must complete to analyse a girder structure in 3D-Beam

1. Draw model with nodes and beams
2. Applying loads
3. Boundary conditions
4. Cross section profiles

The model is built up by nodes with beams connecting them (1), where one can apply evenly distributed loads on the beam or point loads on nodes (2). The model must be fixed in all directions, one does this by fixing nodes, but the engineer has to review what boundary condition is appropriate for each node, as one can fix the node against movement in x, y and z direction as well as against rotation around the x, y and z axis (3).

A cross section profile is chosen for each beam. Usually the double bottom girder is considered as one I-beam, with flange width of the girder spacing. In 3D-Beam it is possible to add T-beams, but most beams are welded to some plating, effectively contributing to the beam's strength. Therefore, most girders will have a I-Profile to represent the deck and outer shell plating (4).

One can also make the beam ends rigid, effectively decreasing the elastic length. This should be done where beams connect and technically shear webs.

Once analysed an engineer can evaluate the stresses each beam experience, the forces, bending moments and sheer forces affecting the beam as well as beam deflection.

Stresses

The *Stresses* view shows the maximum values (negative or positive) of axial stresses, shear stresses, bending stresses and normal stresses along the beams.

	Beam No.	Sig-Nx [N/mm	Tau-Qy [N/mm	Tau-Qz [N/mm	Tau-Mx [N/mm	Sig-My [N/mm	Sig-Mz [N/mm	Min Sig-Ny [N	Max Sig-Ny [N	Min Sig-Nz [N	Max Sig-Nz [N/mm2]
1	1	1	-0	-1	-2	7	4	-6	8	-4	5
2	2	0	0	0	0	0	0	0	0	0	0
3	3	0	0	0	0	0	0	0	0	0	0
4	4	0	0	0	0	0	0	0	0	0	0
5	5	-0	-0	1	-1	7	2	-7	7	-3	2
6	6	-1	-0	1	1	7	4	-8	6	-5	3
7	7	1	-0	-1	-0	-1	2	-0	1	-1	2
8	8	0	-0	-1	1	7	4	-7	7	-4	5

Output | Beams | Nodes | Profiles | Beam loads | Node loads | Beam responses | Stresses | Node responses | Response plot | Notes

Figure 4.6-11 Output window - Stresses

Following stress components are calculated:

Principal stresses:

Sig-Nx: Axial stress (N_x/A_x)

Tau-Qy: Max shear stress in local y-direction (Q_y/A_y), in the neutral axis.

Tau-Qz: Max shear stress in local z-direction (Q_z/A_z), in the neutral axis.

Tau-Mx: Torsional stress (M_x/W_x)

Sig-My: Maximum bending stress about local y-axis (M_y/W_y)

Sig-Mz: Maximum bending stress about local z-axis (M_z/W_z)

Stress combinations:

Min/max Sig-Ny: Min/max of the normal stress in local xz-plane, max of (Sig-Nx ± Sig-My)

Min/max Sig-Nz: Min/max of the normal stress in local xy-plane, max of (Sig-Nx ± Sig-Mz)

Figure 92: Stress Definitions

FEM Structural design spiral

The main difference between CAE- and manual structural analysis, is that the calculations are done by the computer and “by hand” respectively. In CAE and engineer still have to create the static problems, evaluate the result and determine what measures to be made. This make the structural design procedure, using a CAE tool a lot like the spiral in the STP mention earlier.

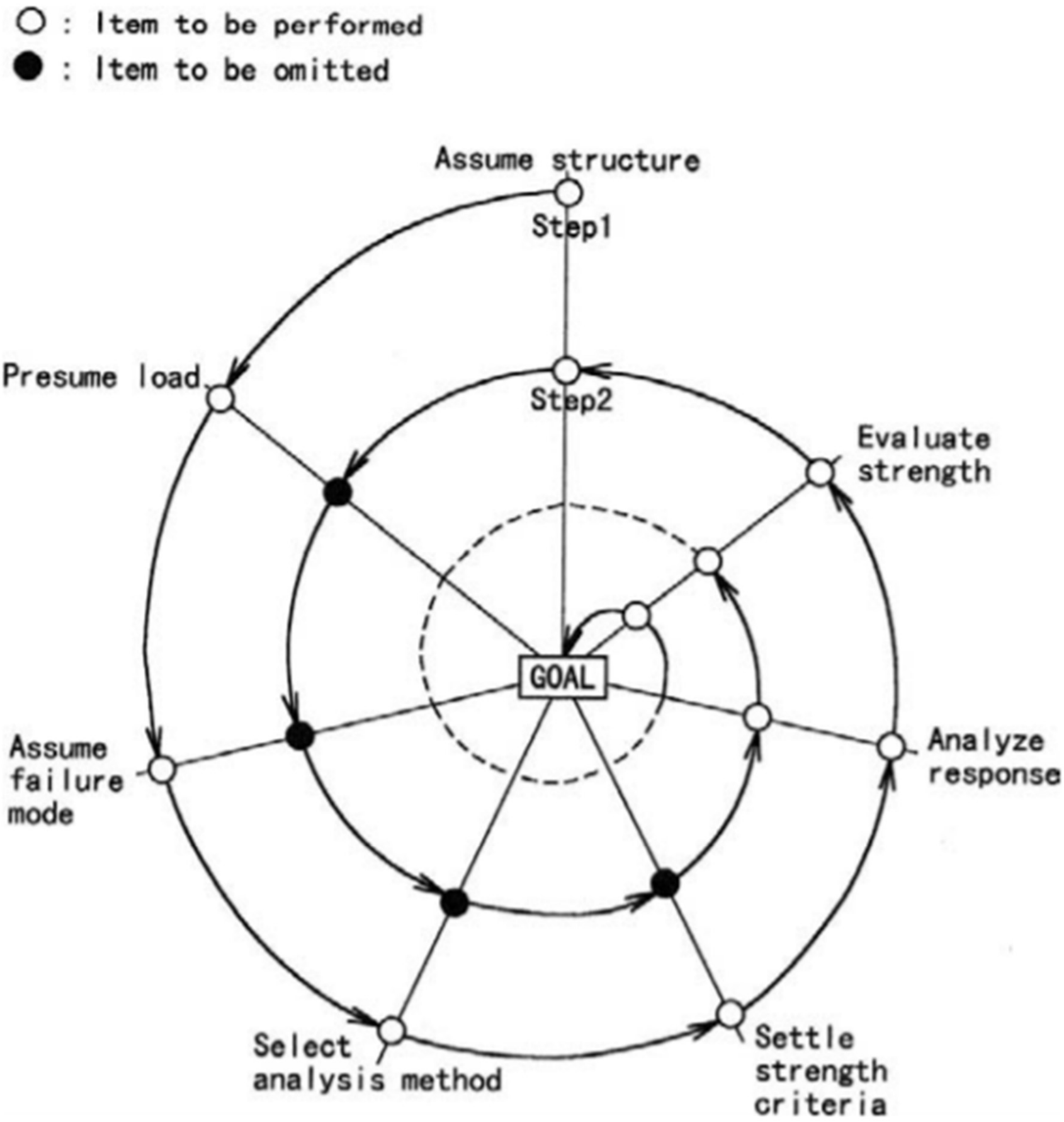


Figure 93 FEM design spiral (Yasuhisa , Yu, Masaki, & Tetsuo, 2009)

Choosing Topology

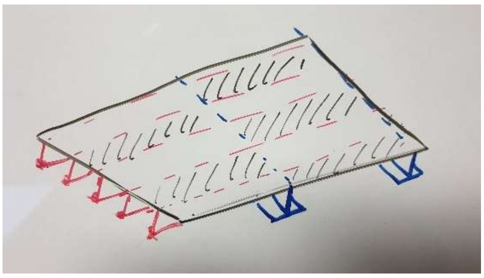


Figure 94 Stiffener and Girders creating rectangular plates

There are three facets that are chosen constringing the topology, distance between girders and stiffeners and orientation. The girders and stiffeners are set 90 degrees of each other, the girders have a longer spacing than the stiffeners, figuratively cutting the decks and ship sides in rectangular plates.

Transverse stiffening is advantageous where the hull has intricate shapes and curves. In these cases, transverse stiffening is easier and cheaper to build. However, the vessels long ship bending moments can reach an amount where the transverse arrangement will cause the plates to buckle in the long ship tensions. This can be calculated in Euler's equation for buckling.

Choosing Profile

Area of inertia determine the strength of stiffeners and girders. A high area of inertia can be obtained by making a profile with high webs and flanges far away from the profile centre. However, this might result in a higher structure, which can cause problems for the ships stability, and a high web is more prone to buckling. One can lower the web height but is then required to increase the area of the flanges to obtain the same area of inertia. The formula concerning area of inertia tells us that an area further away from the profile centre is more effective than an area closer to the centre, meaning the flange area has to increase exponentially compared to the area loss of the decreased web height, ultimately resulting in a profile with a greater area.

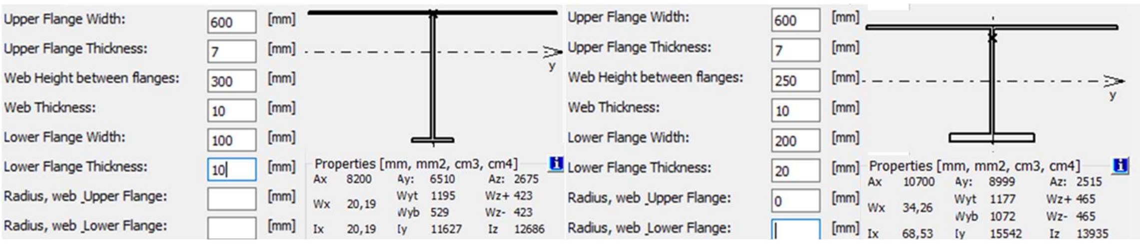


Figure 95 Comparing a high web vs a low web. Same section of modulus but different profile area.

A lesser area of the profile further away from the profiles neutral axis may obtain the same area of inertia by having much greater area close to the neutral axis.

The dilemma stands as follows, a compact girder and beam profile reduces the wall and deck thickness needed, but require a greater profile area, resulting in a heavy structure. Girders and stiffeners with a long step might require less profile area but they require more space from the surroundings.

In the 'Design of Ship Hull Structures' the authors suggest that an optimum flange to web area is 1.5 Aw/Af for a balanced girder with a neutral axis in the middle girder, 2 for a girder with neutral axis closer to one of the flanges, and 3 for a girder with neutral axis on the flanges. (Yasuhisa , Yu, Masaki, & Tetsuo, 2009)

Structural design of The Long Liner

We performed a series of CAD and hand calculations to dimension and evaluate the hull structure. Using DNV software “Nauticus Hull” we could automatically preform rule check analysis and extract values for hand calculations. To be able to use Nauticus hull we had to enter main dimensions and other values to be able to perform the necessary rule check analysis.

Our Approach

Due to our experience, available workforce, and time we chose to mainly use yielding and buckling as failure modes to review; longitudinal, local and transverse strength excluding torsional strength; some details like cut-outs and connections.

We are mostly trained in 3D-Beam and Section Scantlings, so these are the CEA tools we will use, as well as preforming some hand calculations. The training we received in section scantling require us to use the DNV jan 2012 rule set to extract results.

Since the SDP where a bit incomplete, we decided to create our own approach to the structural design incorporated with the CEA tools we used. With the SDP as template we made some changes.

Our plan is to find a general structural arrangement, by reviewing a single cross section that represent most of the ship. Then we check is the sections with disruptions, add elements for it to perform satisfiable and then move on to details.

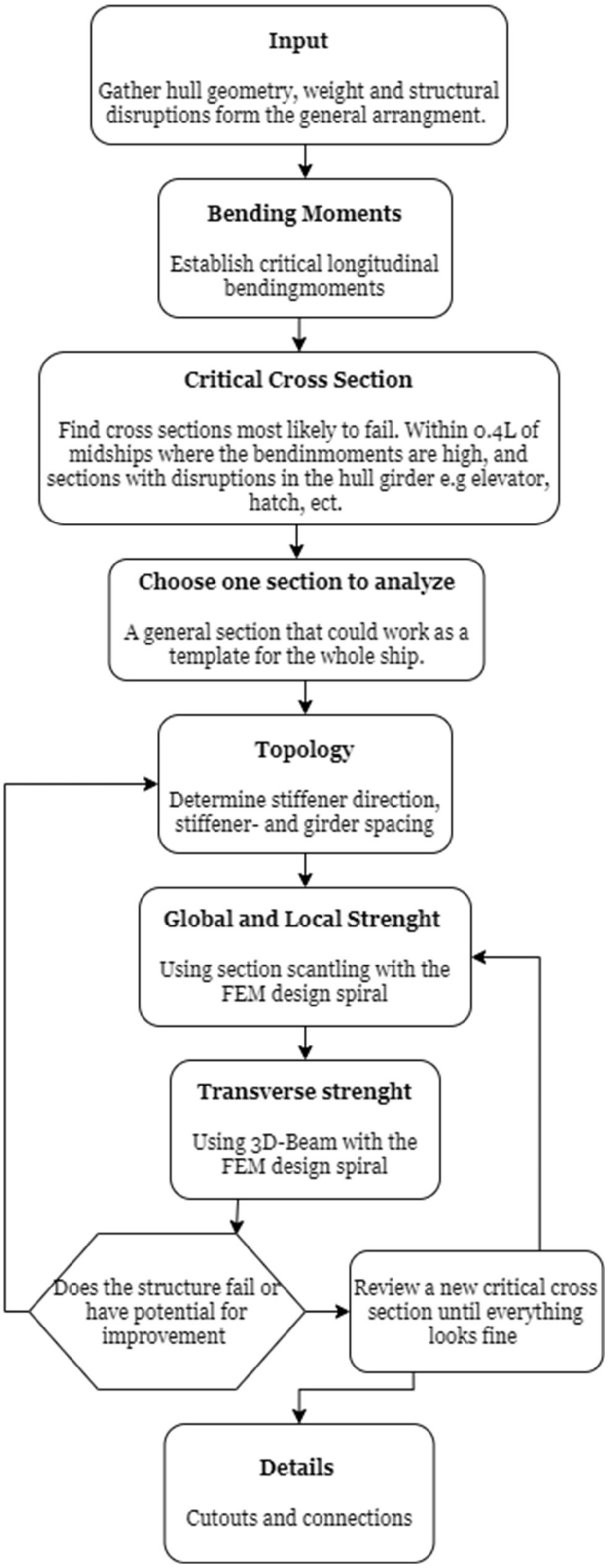


Figure 96 Our approach to the SDP

Pre analysis

Before we can start analysing the structure, we must establish loads, geometry, and critical sections.

Form Maxsurf we can gather actual bending moments for our model. However, these are much lower than the rules bending moments. We have to choose the highest of these two, so we use the rules.

There are a many different hydrostatic and gravitational load cases for different scenarios to review. However, we have chosen in our analysis to just review the normal operation at sea load cases. In DNVGL-RU-SHIP Pt.3 Ch.4 sec.7 2.1.2 we find the loading conditions for each scenario.

External shell- receives static sea pressure at given draught as well as wave pressure. We chose to review a single hydrodynamic lode case due to pressing time, and we chose an HMS load case.

Exposed deck- Strength deck receives green water pressure, mob boat deck, Deck around moonpool and hatches?

Internal Exposed Deck- The internal deck receives the design load with a minimum of $2.5kN/m^2$ added by a dynamic load which is the design load timed by vertical acceleration divided by the gravitational acceleration. Since we could not retrieve our vessels pitch motions, we chose a load of $5kN/m^2$. This a load commonly used by greater ship, but we thought it would be a conservative measure.

Despite having to use DNV 2012 rules for section scantling, we double check some dimensions with DNV 2015 rules for ships under 100 meters, and uses these for 3D-beam.

- Rules for transverse strength was gathered from Pt.3 Ch.2 Sec.3 B301.
- Rules for floors and longitudinal girders was gateherd from Pt.3 Ch.2 Sec.5 C401
- Rules for bottom longitudinal girders was gathered from Pt.3 Ch.2 Sec.5 C601

The geometry of the ship can be taken out by the Maxsurf models, and the distribution and placement of loads are gathered from the GA and weight estimation.

In the GA we detect critical sections.

1. Moonpool section has disruptions in the centreline up till accommodation deck.
2. Superstructure has both hatches and stairwells disrupting the structure, as well as being made in aluminium.
3. Storage receives the highest bending moments and will for us work as a template for the general structural arrangement.
4. The stern does not receive the support from water pressure the rest of the ship receive, and we want to check is there are some failures occurring.

First Structure

Our initial assumed structure was a longitudinal stiffened shell with a transverse girder spacing of 2 meters so the spacing would be consistent over the moonpool; the stiffener spacing was chosen to be 500 mm. Each deck would have a centre longitudinal girder. This was based on earlier projects we had done and theory we were familiar with. Since we had limited time to perform the analysis, we chose a conservative load case through the analysis, with design loads, as described in '[pre analysis]', on each deck.

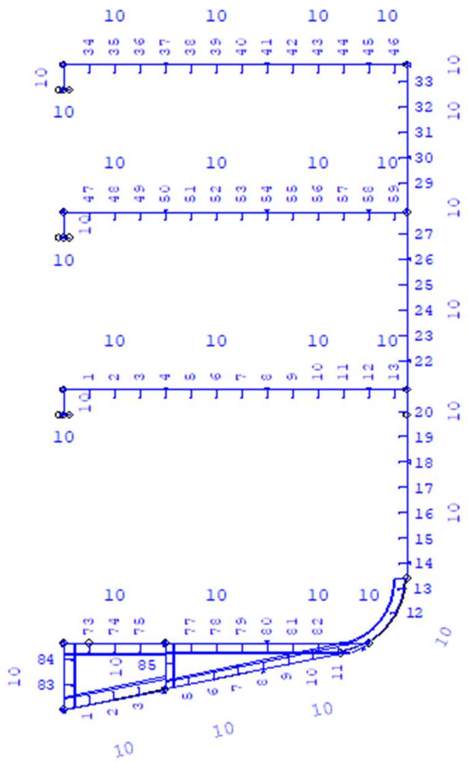


Figure 97 Sections scantling of the first structure

Longitudinal and Local Strength

With "Sections Scantling" we calculated longitudinal strength and local strength. Section scantling can calculate bulking strength and local strength by entering the girder spacing and transverse stiffener spacing.

Inner Bottom				
	1	ACT	10.0	std
		LOC		6.89
		BUC		2.20
	2	ACT	10.0	std
		LOC		6.89
		BUC		2.20
	3	ACT	10.0	std
		LOC		6.89
		BUC		3.00
	4	ACT	10.0	std
		LOC		0.00
		BUC		To be spe
'tween deck 7600				
	1	ACT	10.0	std
		LOC		5.50
		BUC		0.32
	2	ACT	10.0	std
		LOC		5.50
		BUC		0.32
	3	ACT	10.0	std
		LOC		5.50
		BUC		0.32
	4	ACT	10.0	std
		LOC		5.50
		BUC		0.32
'tween deck 10500				
	1	ACT	10.0	std
		LOC		5.50
		BUC		1.79
	2	ACT	10.0	std
		LOC		5.50
		BUC		1.79
	3	ACT	10.0	std
		LOC		5.50
		BUC		1.79
	4	ACT	10.0	std
		LOC		5.50
		BUC		1.79
'tween deck 13000				
	1	ACT	10.0	std
		LOC		5.50
		BUC		2.44
	2	ACT	10.0	std
		LOC		5.50
		BUC		2.44
	3	ACT	10.0	std
		LOC		5.50
		BUC		2.44
	4	ACT	10.0	std
		LOC		5.50
		BUC		2.44

Outer Shell				
Bottom				
	1	ACT	10.0	std
		LOC		10.15
		BUC		2.21
	2	ACT	10.0	std
		LOC		7.52
		BUC		2.35
	3	ACT	10.0	std
		LOC		7.52
		BUC		2.29
Bilge				
	4	ACT	10.0	std
		LOC		7.52
		BUC		1.80
Side				
	5	ACT	10.0	std
		LOC		7.52
		BUC		1.67
	6	ACT	10.0	std
		LOC		7.52
		BUC		1.05
	7	ACT	10.0	std
		LOC		7.52
		BUC		1.45
	8	ACT	10.0	std
		LOC		7.52
		BUC		2.18
	9	ACT	10.0	std
		LOC		6.89
		BUC		2.33
	10	ACT	10.0	std
		LOC		0.00
		BUC		To be spe

Figure 98 Plate result of the first structure from section scantling

First we choose the most critical sections of the hull girder; The storage, with the heaviest internal load and located midships where the hog and sag moments where the greatest; The moonpool section, with discontinuity in both the hull girder and transverse girders; The accommodation deck, experiencing the highest moments, but where added mainly to check local strength due to the superstructure being aluminium.

When drawing the sections in sections scantling we used lines from Maxsurf to manually plot the ship bottom side and bottom. Later, we realized it was possible for "Nauticus hull" to read the Maxsurf files and automatically plot sections in "sections scantlings" making the analyzations more accurate. However, we kept the original manually plotted sections due to the lines constantly being revised and changed.

From the result we gather the dimensions necessary to fulfil the requirements, and it is clear that we have excessive strength in all parts of the structure. The stiffeners range from 700% to 160% excessive section modulus; the deck and hull side and bottom where calculated to be from 5.5 mm up to 7.52 mm with the only exception being the bottommost plate calculated to 10.25 mm.

After changing the thickness and stiffener profile to the calculated dimensions we reviewed the hull girder's section modulus.

	BOTTOM	DECK	ABOVE DECK	SIDE
Material strength group .... :	NV-NS	NV-NS	NV-NS	NV-NS
Yield point of material .... (N/mm2) :	235	235	235	235
Material factor, f1 .... :	1.00	1.00	1.00	1.00
Section modulus ratio, Za/Zr .... :	7.476	5.825		
Based on:				
Za	2.708	2.110		
Zr	0.362	0.362		

Figure 99 Hull girder section modulus from section scantlings

	BOTTOM	DECK
Minimum section modulus, Zo .... (m3) :	0.51257	0.51257
Section modulus requirement based on design bending moments (kNm):		
- Sagging (still w = 23697, wave = 39702) .... (m3) :	0.36228	0.36228
- Hogging (still w = 23697, wave = 33018) .... (m3) :	0.32408	0.32408
Rule section modulus (m3):	0.51257	0.51257

Figure 100 The rule minimum hull girder section modulus form section scantlings

The actual section modulus is 2.7 m^3 for bottom and 2.1m^3 deck calculated from section scantlings, being within the rules of 0.513m^3 by quite some margin.

Even though section scantlings calculated the buckling strength we wanted to review by what margin the plates where accepted. Dividing the longitudinal bending moments with the hull girder section modulus gathered in section scantling we calculated; actual bending tension in the hull girder where calculated to be 14.7N/mm^2 in bottom and 18.8N/mm^2 in deck; Euler's buckling strength is calculated using c=4 due to the pressure being applied on the shorth length of the plate, material is 235MPa steel and the new deck plate thickness 5.5mm. The Euler's buckling strength where calculated to be 90N/mm^2 which is less than the half the yield strength of 235N/mm^2, requiring us to use Euler's Buckling strength as the critical buckling strength. Similarly, to the section scantling results the buckling strength is proven to be excessive by quite some margin.

Transverse Strength

To calculate the strength of the transverse elements, we used 3D-Beam. We chose to check the storage section of the ship, due to the double bottom experience near to max sea pressure and height of the double bottom web having a difference of 40 cm. In 3D-Beam the double bottom is drawn as one girder with flanges representing the deck plates and ship bottom. We chose the x axis to be the longitudinal axis of the ship, y as transverse, and z as vertical.

The girder profile was chosen based on the space we made available with webs of 500mm; flanges with 10mm x 250mm based on theory of an optimum intermediate beam. The load from the stiffeners where applied as distributed loads on the transverse girders.

Since the ship has greater depth in the aft, raked keel, the sea pressure increases in the aft. Therefore, the sea pressure must be calculated for each girder.

For 3D-Beam to compute, the model needed to be fixed in all directions. The forward end of the model is connected to a bulkhead, which is stiffer relative to the rest of the structure. Therefore, the nodes forward in the model will be fixed in the z and y. direction. The ship is fixed on z directions top corners of each transverse girder section. There is no load applied in the x direction, however the computer manages to create an insignificant force in a x direction, therefore a centre girder was fixed in x direction without it altering the result in any way.

Since the bulkhead is located forward or the ship there is not an equal load front and aft of the bulkhead, therefore the beams are prone to rotation around the y-axis. The back end of the model is located in the middle of the ship, which means the load on each side of the moonpool is roughly the same, causing the nodes in the aft end to be fixed against rotation around the y axis.

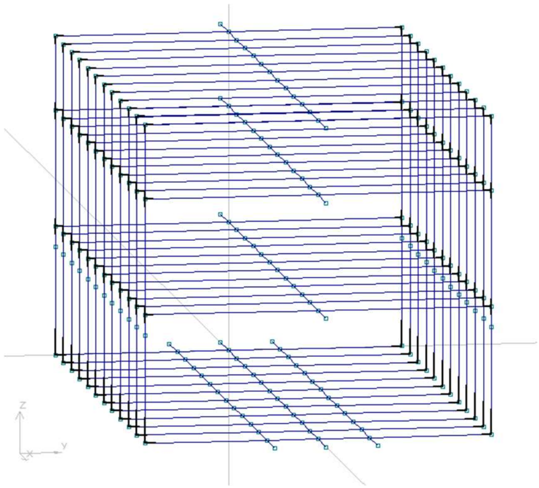


Figure 102 Longitudinal stiffened 3D beam structure

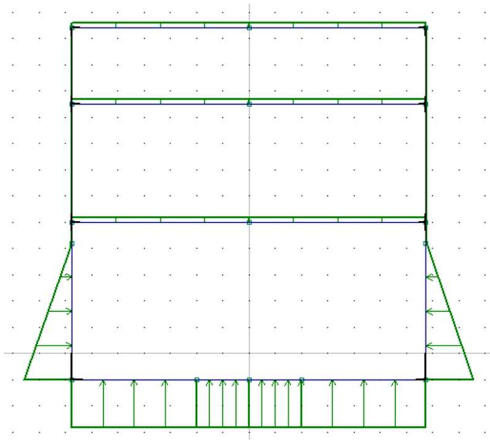


Figure 101 Load applied on a longitudinal stiffened structure. Here is only design loads and static water pressure applied

From the results we received low bending stresses form all parts of the ship, except the hull side. The hull side showed bending stresses from 96N/mm² to 205 N/mm². This is a bit surprising, even if the double bottom is 40cm higher and therefore stiffer in the aft the deviation of bending stresses seems to high. However, in this model the forward of the ship is fixed in z direction, resulting in the moments being in greater part transferred to the longitudinal members.

If we were to remove the bulkhead, the moments would be greater in the forward, thus receive similar bending stresses.

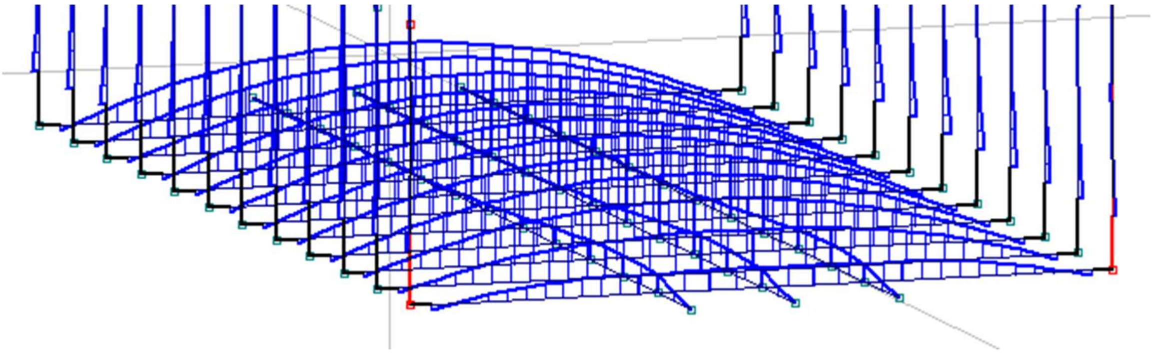


Figure 103 Bending moments with bulkhead in front

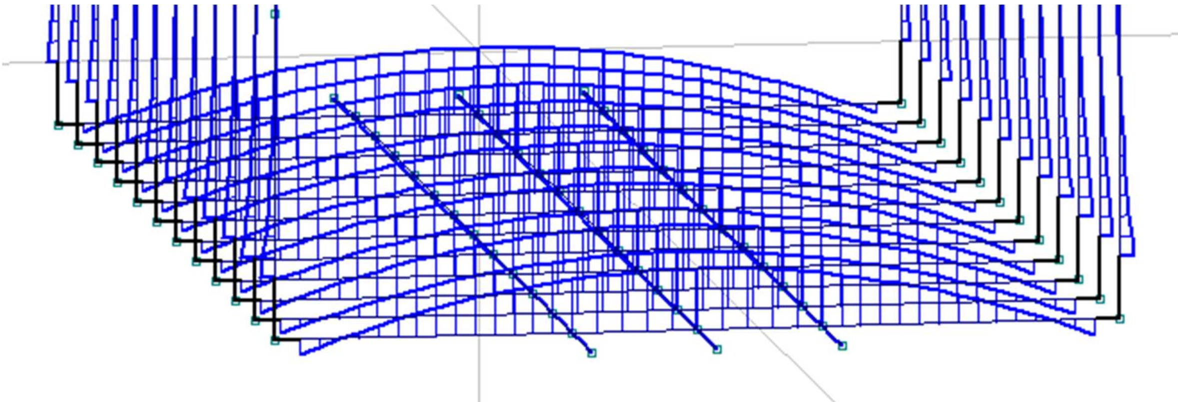


Figure 104 Bending-moments without a bulkhead

After decreasing the beams using the optimum beam theory until most beams in deck reached a web height around 300mm, we stopped reducing the web height since we would need additional height for cut-outs to pluming, ventilation, electrical cables, etc. The deck side girders needed a web height of 500mm; the result showed the axial stresses contributing up to 19% of the combined stresses, so by increasing the area of the profile we could allow a lower web height. With a great enough flange, we reduced the web height to 300mm, with an axial stress near to nothing. However, a beam like this is far from an optimum beam, and is not weight efficient, but the idea of increasing space available is worth exploring.

It was painstakingly clear to us that we had assumed a structure with excessive strength in almost all parts of the ship. The beam webs needed to be at least 300mm and the plates the minimum plate thickness are 5.5mm based on other than strength related issues. We thought that this was not a weight efficient structure; we could increase spacing allowing more load on those elements without increasing the dimensions and ultimately the weight. We also believed that increasing stiffener and girder spacing we could reduce the amount of strength elements, resulting in less welding required which means a lower production cost.

Summary

We figured that our initial assumption was based on ships with length around 100-200 meters, which experiences much greater global loads than our long-liner of 63 meters. We realized that a new approach where needed and bold changes appropriate for a new structure.

Second Structure

The second structure was a transverse stiffened arrangement with 3000mm girder spacing and 600mm stiffener spacing. After reviewing our buckling strength in the first structure it was imminent that buckling would not be an issue for a longitudinal stiffened arrangement, so we choose to change to a transverse stiffened arrangement. We had a little concern with the stiffeners no longer contributing to the hull girder's section modulus, but estimated that the section modulus was so high due to the strength deck's distance from the hull girder's neutral axis and not so much the area added by the stiffeners. The buckling strength where calculated to be 90N/mm², using transverse stiffeners the strengths could be reduced up to a quarter to longitudinal stiffeners. Nevertheless, 90/4 is still 20% more than the initial stress of 18.8N/mm², so we felt confident that the section modulus would not be reduced to such an extent that transverse stiffeners where not an option.

Due to the excessive strength in the first structure, we increased the spacing of both transverse girders and stiffeners to allow a greater load on the elements, hopefully matching the capacity of the 300mm beams and 5.5mm plates.

Local and global

5 Local Rule Requirements - Transverse stiffeners

Stiff. No	ACT / ACT / LOC	Pos Z_a cm³	K c / Z_r cm³	Type Type / excess (%)	h t (mm) / t_{min} (mm)	b_f t_f (mm) / Load Ref.	y z (mm)	σ_F f_1 N/mm² / σ N/mm²	m w_k / ang_PL deg	t_{kw} t_{kf} (mm) / p kN/m²	t_{pl} (mm) / Comp ref.	span spac (mm) / a_{corr} cm²
'tween deck 7600												
1	'tween deck (Trv. stiffener). End points (y,z) = (0, 6000)-(6750, 6000).											
	ACT	'twdk	0.00	20	160	0.0	3375	235.0	10.0	0.0	6.0	6750
	ACT	112		HPbulb	8.0	0.0	6000	1.00	1.00	0.0		600
	LOC		110	2	5.1	Gen		160.0	90	6.4	2	0.0
'tween deck 10500												
1	'tween deck (Trv. stiffener). End points (y,z) = (0, 9500)-(6750, 9500).											
	ACT	'twdk	0.00	20	160	0.0	3375	235.0	10.0	0.0	6.0	6750
	ACT	112		HPbulb	8.0	0.0	9500	1.00	1.00	0.0		600
	LOC		110	2	5.1	Gen		160.0	90	6.4	2	0.0
Outer Shell												
1	Side (Trv. stiffener). End points (y,z) = (6750, 2294)-(6750, 6000).											
	ACT	Side	0.00	20	200	0.0	6750	235.0	10.0	0.0	8.0	3705
	ACT	205		HPbulb	8.5	0.0	4147	1.00	1.00	0.0		600
	LOC		174	18	5.7	Sea		160.0	90	33.5		0.0
2	Side (Trv. stiffener). End points (y,z) = (6750, 5500)-(6750, 6000).											
	ACT	Side	0.00	0	0	0.0	6750	235.0	10.0	0.0	8.0	500
	ACT	0			0.0	0.0	5750	1.00	1.00	0.0		600
	LOC	*	15	-100	5.1 *	Sea		160.0	90	21.8		0.0
3	Side (Trv. stiffener). End points (y,z) = (6750, 6000)-(6750, 9500).											
	ACT	Side	0.00	20	120	0.0	6750	235.0	10.0	0.0	8.0	3500
	ACT	62		HPbulb	8.0	0.0	7750	1.00	1.00	0.0		600
	LOC		60	2	5.1	Sea		160.0	90	13.0		0.0
4	Side (Trv. stiffener). End points (y,z) = (6750, 9500)-(6750, 12400).											
	ACT	Side	0.00	20	80	0.0	6750	235.0	10.0	0.0	8.0	2900
	ACT	26		HPbulb	7.0	0.0	10950	1.00	1.00	0.0		600
	LOC		25	6	5.1	Sea		160.0	90	7.8		0.0
'tween deck 13000												
1	Strength deck (Trv. stiffener). End points (y,z) = (0. 12400)-(6750, 12400).											
	ACT	Strdk	0.00	20	160	0.0	3375	235.0	10.0	0.0	6.0	6750
	ACT	112		HPbulb	8.0	0.0	12400	1.00	1.00	0.0		600
	LOC		110	2	5.1	Gen		160.0	90	6.4	2	0.0

Figure 105 Result from section scantling

'tween deck 13000

1	ACT	6.0	std	0.0	0.0		-	100	6750	600 t	0.0	235.0	1.00
	LOC		5.50	Strdk		Gen		Min	300	12400	2	120.0	6.4
	BUC		7.08 *	1.00				compr	0	12400	32.6	23.4	32.6
2	ACT	6.0	std	0.0	0.0		-	100	6750	600 t	0.0	235.0	1.00
	LOC		5.50	Strdk		Gen		Min	2000	12400	2	120.0	6.4
	BUC		7.08 *	1.00				compr	2000	12400	32.6	23.4	32.6
3	ACT	6.0	std	0.0	0.0		-	100	6750	600 t	0.0	235.0	1.00
	LOC		5.50	Strdk		Gen		Min	4000	12400	2	120.0	6.4
	BUC		7.08 *	1.00				compr	4000	12400	32.6	23.4	32.6
4	ACT	6.0	std	0.0	0.0		-	100	6750	600 t	0.0	235.0	1.00
	LOC		5.50	Strdk		Gen		Min	6000	12400	2	120.0	6.4
	BUC		7.08 *	1.00				compr	6000	12400	32.6	23.4	32.6

Figure 106 Buckling problem in strength deck

5 Hull Girder Strength Requirements

		BOTTOM	DECK	ABOVE DECK	SIDE
Material strength group ....:		NV-NS	NV-NS	NV-NS	NV-NS
Yield point of material ....(N/mm2) :		235	235	235	235
Material factor, f1........:		1.00	1.00	1.00	1.00
Section modulus ratio, Za/Zr........:		6.125	4.935		
	Based on:				
	Za	2.219	1.788		
	Zr	0.362	0.362		

Figure 107 Section modulus from section scantling

6 Layout of transverse stiffeners

Stiffener Bracket	(mm) Arm1	y1 (mm) h1	z1 (mm) bf1	y2 (mm) t1	z2 (mm) tf1	Type	(mm) Arm2	h (mm) h2	bf (mm) bf2	t (mm) t2	tf (mm) tf2
Inner Bottom											
Girder		0	1000	2000	1000	20		220		9.0	
Girder		2000	1000	6000	1000	0		0		0.0	
'tween deck 7600											
Tstif		0	6000	6750	6000	20		160		8.0	
'tween deck 10500											
Tstif		0	9500	6750	9500	20		160		8.0	
Outer Shell											
Girder		0	-300	2000	106	20		240		12.0	
Girder		2000	106	6000	1000	0		0		0.0	
Girder		6000	1000	6750	2294	20		240		14.0	
Tstif		6750	2294	6750	6000	20		200		8.5	
Tstif		6750	5500	6750	6000	0		0		0.0	
Tstif		6750	6000	6750	9500	20		120		8.0	
Tstif		6750	9500	6750	12400	20		80		7.0	
'tween deck 13000											
Tstif		0	12400	6750	12400	20		160		8.0	

Figure 108 Stiffener dimensions

The result looked promising to us. We had reach and exceeded the capacity of the 5.5mm plate thickness, and the stiffener profile seemed appropriate to our knowledge. However, on reviewing the plate thicknesses we saw that the strength deck was prone to buckling but was easily fixed by increasing the plate thickness with 1mm. Even though 1 millimetre do not sound that much, but over an area covering the whole ship, the weight might have a significant effect on the ship's stability.

Transverse

Transverse stiffened arrangement required a different load case than longitudinal stiffeners. The load is transferred to the longitudinal members rather than the transverse girders, which is an important distinction since a point load crate a greater moment than a distributed load. The stiffeners are supported in the outer shell structure, which in 3D-Beam is represented as L- and T-beams with the thickness of the deck and hull side plates, and heights calculated as effective flanges.

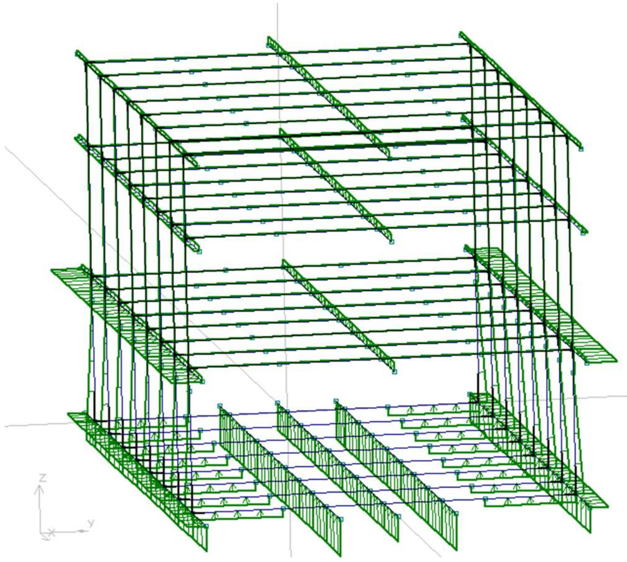


Figure 109 Applied load on transverse structure on 3D-Beam

The loads exceeded the deck and hull side girders' capacity. By using the optimum beam theory, we reach a web height of 450mm to avoid yielding, still within our initial limit of 500mm.

Summary

So far, we have tried to avoid any columns for to keep as much space available in storage for fish. Columns in the deck could reduce the height of the girder profiles by functioning as a support. At this stage we had been made aware that our transverse girders were prone to vibrations.

201

After the structural strength have been explored, there came some changes to the design and most importantly the dimensions of the structure are becoming more than just assumptions. Until this point all the positions and space allocations in the GA have been done with simple estimates, now we can introduce the dimensions that are calculated in the structural analysis. Furthermore, as we design the lines to accommodate the stability and other hydrostatics the GA needs to be revaluated and updated. There will also always come up new information during the design process, and estimates are continually improved throughout this information gathering.

New Beam

Due to a wrongly estimated isolated thickness the estimated beam of the ship has to be revaluated. We had misunderstood some information about the isolation thickness added to the beam, which results in the hull thickness around the main fish storage. The new total thickness of the hull is now estimated to be 550 mm, and the width of the storage is still 12.1 metres. Making the beam of the vessel 13.2 metres due to necessary Bm. This has some important consequences for the rest of the design.

Stability

The beam reduction reduces the stability with it, as already mentioned in the line design. This again means that we have to revisit the hull design in order to increase the GM to what is required. Firstly, by increasing the waterplane area and thereby the BM. This off course has some disadvantages for the hydrodynamics and the seaworthiness of the vessel, at least it lowers their prioritizations. This might not be to an extent that harms the performance too much, on the other hand it can be minimized.

Weight Optimizing

After the structural dimensions were determined, it is time to update the weight estimate in order to make it more accurate. A goal set to achieve an accurate steel weight estimation without spending too much time. That is a compromise since a more accurate weight estimate would give us better-resulting values for the following analyses but requires much work. So, in order to not delay the following analysis, a strategy needed to be developed. Two different strategies were analysed:

First strategy: reduce the total time of the preliminary design by making a more accurate weight estimation in order to limit the number of necessary spirals.

Second strategy: to achieve a faster but less accurate weight optimization. Prioritising speed at this stage, using less time, resulting in higher number of "trips" around the spiral at a faster rate. Versus using more time making fewer designs with higher degree of detail. This strategy is composed of optimizing the earlier plate estimation technique.

Strategy I
In order to achieve a more accurate weight estimation than the plate technique was to utilize different structural software.

The first software analysed was to import the MaxSurf design into Nauticus Hull. In order to import geometries of sections, every 6 meters or less to make an average. So, the values from the critical sections, in order to define plates and stiffeners, from there using the data for the cross-section in order to obtain an estimation of the hull weight. Another method was utilizing 3d software to model the basic ship structure.

Utilizing software came with some problems. Nauticus hull did not give accurate data as the hull form changing quite rapidly, it could give accurate data if enough sections were imported, but the imported sections would sometimes not be able to use in the report. Therefore, each section would need to be modelled by hand, which was a way to time consuming process. Utilizing different 3d software was not possibly as we did not have access to the software necessary. Without necessary software would result in it being time consuming having a lower efficiency as further optimization would be needed. Therefore, second strategy were the only option.

Strategy II
In order to make the plate weight estimation process quicker the hull can be separated into sections to make it both accurate and fast. That would be done by sectioning of the hull consisting of the same plate dimensions and measuring it using siemens NX. In order to make the weight estimation of the double bottom more accurate, since it was highly inaccurate due to it having an incline instead of a flat bottom. The filling degree of the tanks can be utilized to estimate the amount of steel; by approximating 2% of the double bottom consisted of steel. The centres of gravity were estimated by using the centres of the volume obtained in siemens NX.

In order to achieve a better estimate further question about uncertain parts of the estimate were questioned to the supervisor and *Skipteknisk*. Here we got many satisfying answers detailed under.

System weights centres of gravity such as cables and pipes are hard to determine. Our supervisor's advice was to lay 50% of the weight of the cables by the centre of gravity of the main engine. Other 50% was to be placed in the middle of the freezer storage both longitudinally and vertical. Cables was to be placed at the centre gravity of the resulting lightship both longitudinally and vertical.

From *Skipteknisk* we were given the weight of the cranes forward and aft in the ship. Those would both be 6 tonnage per crane. *Skipteknisk* also informed that the height would highly vary if the cranes where either on the deck (Figure 110) or lowered (figure 111), so the earlier estimate were utilized for vertical gravity point for those cranes. Assumption for 1-meter vertical gravity point over deck for auxiliary, hauling and factory where correct assumption.

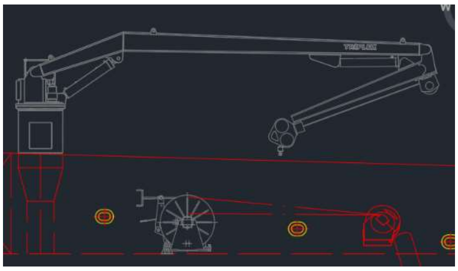


Figure 110: Lowered Crane

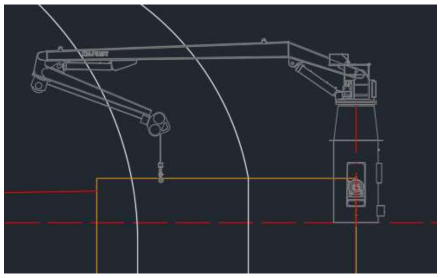


Figure 111: Crane on Deck

The strategy also consisted of utilizing technique of smearing the stiffeners and beams into the plating in approximately plate thickness per m^2 such as decks which has not highly complex shapes appeared.

Our Supervisor told us to add 20% to the total weight of the whole lightship estimate as every weight is just a rough estimate.

Result

The table consist of the different estimates for the lightweight and centres of gravity for the vessel. As one can see underneath that gave us a higher total weight 1971 tonnage. Also, LCG was estimated to be further ahead for the steel weight and other components. The earlier estimate for weight was wrong due to the plate thickness being too thick both for the double bottom, and the side under freeboard. Which combined with other factors gave us a higher vertical gravity point.

Lightweight					
Lightweight	Weight	LCG	VCG	LCG*m	VCG*m
Steel	1068.29	32.12525	7.43	34318.94	7937.72
Pipes and Ventilation	89.60	29.7125	4.928125	2662.24	441.56
Cables	33.60	31.5	7.7	1058.4	258.72
Machinery	137.32	16.08	3.63	2208.129	498.7933
Outfitting	206.5302	32.42319	12.91013	6696.369	2666.332
Factory Equipment	50	44.75	7.6	2237.5	380
Hauling	8.28	21.50	7.60	178.02	62.928
Elevator	5.00	22	6	110	30
Azimuth	16.5	57	4.5	940.5	74.25
MOB-boat	3.50	12.3	11.5	43.05	40.25
Coolers	12.00	50	4.76	600	57.12
Crane aft	6.00	20.5	13.85	123	83.1
Crane fwd	6	52	16.75	312	100.5
Total	1642.62	31.34518	7.689722	51488.15	12631.27
Total +20%	1971.141383	31.34518	7.689722	61785.78	15157.53

Table 8: Lightweight 201

There are other means of increasing the GM, for instance by altering the KG. Lowering the centre of gravity means the BM do not have to be increased. This can be done by lowering the position of heavy weight items in the design. However, when this is not an option, or possible, we can add a constant ballast low down in the design. This added ballast would mean that we are making the vessel heavier than necessary, but this might be less significant than the improvement in stability its results in. For a long-liner a “case keel” is frequently used, this is an added ballast placed under the vessel’s keel in form of a steel filled box. We saw this while visiting Geir, a 600x800 box that followed the keel along the length of the vessel. It only stopped were there where “breaks” in the centreline keel, such as retractable azimuth thruster and moonpool.

Since the design already have a high waterplane area coefficient and knowing that increasing it more or increasing the vessel’s width would typically increase the resistance, we look at other stability increasing measures than making the BM higher.

We add a ballast-keel like the one on *Geir* in order to lower the centre of gravity since the lightship stability condition was critical for the metacentre height. A simple estimation for the keel showed that it needed to between 150 to 250 tonnes which would be explored further.

Back to the Lines

After this new and improved weight estimate have been performed, we take a look back on the lines with these new numbers. We also take into account the effects this will have on the stability.

Beam Effect

It became apparent from the stability calculations that a beam of 13.2 metres brought quite a lot of challenges with it. There would need to be quite a lot of added ballast in the keel to satisfy the DNV stability requirements. Hence, we stated to evaluate if it would be more beneficial to increase the beam a little. Adding enough width to fit another row of pallets would exceed the maximum beam from the requirements, 14 metres. So, this is ruled out, although it could be worth to discuss the opportunity to go beyond with the owner if it continues to be a problem. Nevertheless, before that there are some other alternatives. If increasing the web height inside structure could help the strength calculations, and possibly reducing the necessary flange to reduce the total weight, the two benefits together could make it profitable compromise with the increased resistance. Or if enough weight is saved it could possibly reduce the total resistance, although it is unlikely that we can save that much weight.

For now, we are looking at another little side benefit to increasing the beam slightly. A little increased width in the storage would not make room for more pallets, however it makes it easier to place the pallets and more space for squeezing inn separate fish. Hence, the total beam is increased to 13.3 metres, where this additional 0.1 meter is mostly a precaution. Which will be kept in mind while performing the strength calculations.

Draught

A new draft was estimated to be around 5.6-5.7 meters from the previous lines adding non-buoyant volume. A draft over 5.6 meters would be too much, as that could make the height in moonpool too small, in addition to under 1 metre from the working deck to the waterline. Therefore, making a higher block-coefficient was necessary. It also aids in moving the LCB further forward, since reducing the volume in the aft was impossible without making drastic changes in the GA. The lift position restricted where the lines could start to curve in towards the skeg, and then we need to balance how sharp the lines can be without increasing the resistance or disturbing the flow towards the propeller. Therefore, would it work well for both dilemmas to increase the volume towards the bow. Although, this will also increase the resistance a bit, and since we are not using a bulbous bow the vessel might generate a large bow wave.

The new LCB was approximated to be around 1 to 1.2 meters in front of midship and the draft should be 5.4-5.5 metres maximum. These values were brought back to the line design and changes were made accordingly.

From an educated guess, the aft section was on the limit of being highly inefficient, due to a rapid angle of change from the midsection. As that will cause the water flow to release causing a pressure difference creating a stern wave increasing wave resistance. At the same time, a slow release would increase the resistance by decreasing the hull efficiency for the propeller. To make a compromise, a bit smaller change in the angle was created, and increasing the length of the aft section a little to increase the hull efficiency. The result from this however, is an increase in the volume of the aft section, which conflicts with moving the LCB forward.

Since it would have a higher depth the transom was increased to 5.1 meters. In the forward section more volume added at the front from a wave-piercing bow to a more typical bow. Creating a gentler degree of change into the midsection. Since a gentler degree of change same as in aft section, in theory, it would reduce the resistance for the vessel. Though without a bulbous bow will the degree of change matter for resistance, as the bow wave is not cancelled. Here we wanted to test the difference in resistance in the calm water of our bow design versus a traditional bow. Nevertheless, model tests take a bit of time in the industry it costs a lot, it is therefore important for ship-designer to use their experience and knowledge to make effective designs. We on the other hand, are still students and we wanted to do this test to take full advantage of facilities and learn a bit more about the difference in these to designs. Unfortunately, the Covid-19 virus meant the facilities were closed off and we were not able to perform this test.

As knowledge was obtained on how to define non-buoyant volume in MaxSurf, were both the vertical azimuth in front and the moonpool shaped into the model. The problem faced earlier was that the surfaces did not appear in the boundary condition in MaxSurf stability. For that to perform, the surface for the non-buoyant volume was changed from hull-structure to internal structure (Håvard Vollset Lien, Personal Communication). The non-buoyant volume increased the draft 0.065 meters. Also, since the non-buoyant volume from the vertical azimuth was placing in the forepeak of the ship, compared to the moonpool, just 6 meters behind, did the LCB position decrease 0.1 metre.

Propeller Basic Design

When choosing the optimum diameter, RPM and Pitch is it necessary to choose one that is the limiting factor. In our case that is the diameter, since momentum theory explains that a high propeller diameter leads to high efficiency. The diameter is a limiting factor due to the clearance to the hull. The height restriction from the keel up to the hull where the propeller is situated has a clearance of 4.5 meters. Leading to our maximum propeller diameter of being 3.3 meters (Nerland, Propulsion Part 3 of 4, 2017), (Molland, Turnock, & Hudson, Ship Resistance and Propulsion Practical Estimation of Ship Propulsive Power, 2011).

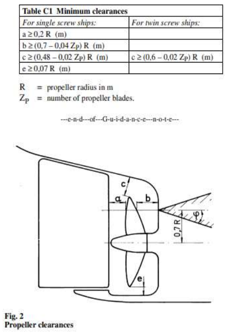


Table C1 Minimum clearances	
For single screw ships:	*For twin screw ships:*
a ≥ 0,2 R (m)	
b ≥ (0,7 – 0,04 Z_P) R (m)	
c ≥ (0,48 – 0,02 Z_P) R (m)	c ≥ (0,6 – 0,02 Z_P) R (m)
e ≥ 0,07 R (m)	

R = propeller radius in m
Z_P = number of propeller blades.

---e-n-d---of---G-u-i-d-a-n-c-e---n-o-t-e---

Fig. 2
Propeller clearances

Figure 112: Propeller Clearance

The propeller is approximately the height of the rudder, since a foil effectivity is given by a larger length/width ratio. Thus, resulting that the effect of a rudder is given by its ratio of height/length. (Håvard Vollset Lien, Personal Communication) informed us that this ratio should not exceed 1.6-1.7. Resulting in a maximum of 1.875-2.1875 length of rudder depending on the propeller diameter. With rudder and clearance gave us a maximum of 4.68 meters from skeg to end of the rudder. To be on the safe side the end of the skeg were placed 5 meters from aft.

Estimating the wake factor and Thrust Deduction Factor

Best determined estimate for wake factor where obtained by (Molland, Turnock, & Hudson, 2011, p. 157 (figure 8.12)). Supervisor deemed this to be a sensible value.

$$W_T = 0.35 - 0.03 - 0.035 = 0.285$$

Thrust deduction where calculated using formula relationship between wake and thrust factor (Nerland, Propulsion Part 3 of 4, 2017).

$$\frac{t}{w} = 1.57 - 2.3\left(\frac{C_b}{C_{wl}}\right) + 1.5 \cdot C_b$$

$$t = \left(1.57 - 2.3\left(\frac{0.633}{0.9}\right) + 1.5 \cdot 0.631\right) \cdot 0.285 = 0.26$$

It is to be noted that the accuracy for utilizing this formula are limited, due to ship having a raked keel. As the block coefficient are reduced than the actual.

Number of Blades

Selection number of blades comes early in the process. Blades number affects several factors such as the optimum diameter, efficiency and vibration. For most vessels have propellers with 4 blades. If one would decrease to 2 blades would that lead to an increase in efficiency, but a far greater increase in diameter of the propeller, also would lead to more vibration. 6 blades would lead to a smaller efficiency but far smaller optimum diameter and vibration. Since a CPP-propeller would 6 blades likely lead to a way larger hub resulting in less efficiency. For this vessel a four bladed propeller was chosen.

Propeller Calculations

So is the goal to find the optimum RPM and Pitch for the different operation speeds. Firstly, necessary thrust has to be calculated. The resistance was taken from Maxsurf resistance using Hooltroop. Also using thrust deduction is required thrust calculated for the vessel under, also reduced speed Va into the propeller equation under.

$$T = \frac{R}{1-t}$$

$$V_a = V_s(1-w)$$

Vs (Kts)	Resistance (KN)	Required Thrust (KN)	Va (m/s)
3	4.8	6.5	1.0963095
9	41.1	55.5	3.2889285
11	74.9	101.2	4.0198015

Table 9: Required Trust (Maxsurf)

Two equations for thrust coefficient and advance coefficient for a propeller in open water has only one unknow that is n rpm. If one restructure advance coefficient equations, one can insert it into the thrust coefficient equation for n, in order to only get the advance coefficient as a variable.

$$K_T = \frac{T}{\rho \cdot n^2 \cdot D^4}$$

$$J = \frac{V_A}{nD}$$

$$n = \frac{J \cdot D}{V_A}$$

$$K_T = \frac{T}{\rho \cdot V_A^2 \cdot D^2} \cdot J^2$$

First part of the equation is a constant, and this is then graphed into an open water diagram for a propeller with suitable blade area ratio.

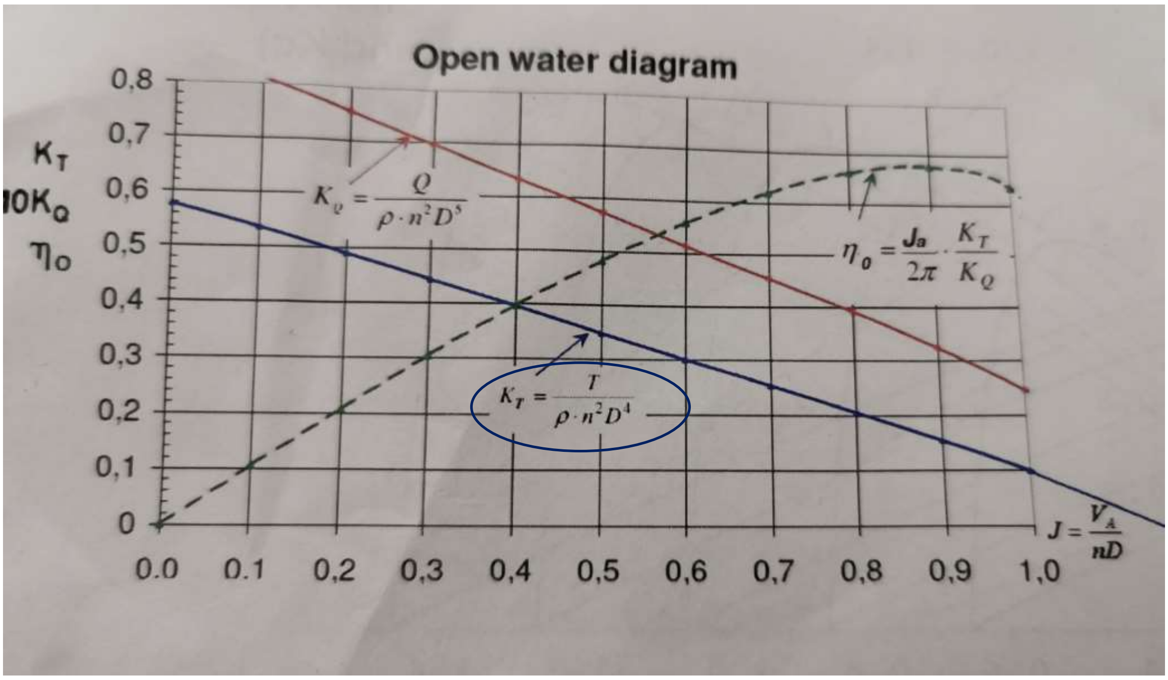


Figure 113 - Open Water Diagram
(Nerland, Marine Hydrodynamics - Propulsion Part 1 of 4)

Blade Area Keller Criterion

Blade area is dependent on cavitation. A higher blade area would give less chance for cavitation, since a smaller blade area ratio gives a higher efficiency for the propeller. It is therefore necessary to check requirements for blade area ratio. To find that requirement one can use the Kellers criterion or Burill diagram. Burill diagram was a bit hard to grasp so to be efficient the Kellers criterion were choosen.

$$\frac{A_E}{A_0} = \frac{(1.3 - 0.3 \cdot Z) \cdot T}{(P_0 - P_v) \cdot D^2} + k$$

(Alto University, 2016)

T = thrust

P0-Pv = difference in pressure at the shaft of the propeller to the pressure from vapor. Pressure from vapour is 1750N/m^2. In our instance by a propeller with 3.3 meter in diameter is 3.65 meters below water line.

D = 3.3

K = factor for type of ship, single screw 0.2

It is important to note that in order to calculate for a CPP-propeller versus an FPP propeller the cavitation should be increased with 5-10% (Alto University, 2016) due to a higher chance of cavitation for a CPP. So here was chosen to add 5% to the blade area ratio.

T = thrust of the vessel, use the minimum requirement of the power on the propeller in order to satisfy the design requirements.

$$T = \frac{P \cdot \eta_D}{(1-t) \cdot V_S}$$

P = Power on the propeller which is 2300 KW, from the requirements.

η_D= quasi-propulsive efficiency, which is the efficiency for the propeller and vessel. That had to be estimated.

The relative rotation efficiency was estimated to be η_R=0.97 since the vessel is not designed specifically for one condition. A higher value would been chosen if it was for example a container vessel with one set design speed. Hull efficiency is obtained by the thrust deduction and wake factor for the vessel. Propeller efficiency $\eta_0 = 0.6$ was determine by looking at what efficiency one should obtain from the open water chart for vessel with blade area ratio (BAR) of 0.8.

$$\eta_D = \eta_0 \cdot \eta_R \cdot \eta_H = 0.6 \cdot 0.97 * \frac{1-0.26}{1-0.285} = 0.6$$

Quasi-propulsive efficiency is estimated to be around 0.6.

$$T = \frac{P \cdot \eta_D}{(1-t) \cdot V_S} = \frac{2300 \cdot 0.6}{(1-0.26) \cdot 11 \cdot \frac{1852}{3600}} = 329.5kN$$

Speed of the vessel here was chosen to be at transit speed of 11 knots, which gave a surprisingly high trust compared to values from Maxsurf resistance Table 9**.** Which indicates that the value obtained from Maxsurf resistance are incorrect. Hence, before a model test has been performed there is a bit uncertainty around this calculation.

It is important to note that calculating for a CPP-propeller, versus an FPP propeller, the cavitation should be increased with 5-10% due to a higher chance of cavitation for a CPP. So, here it was chosen to add 5% to the blade area ratio.

$$\frac{A_E}{A_0} = \left(\frac{(1.3 + 0.3 \cdot 4) \cdot 329.3}{\left(\frac{98100 + 3.65 \cdot 1025 \cdot 9.81 - 1750}{1000}\right) \cdot 3.3^2} + 0.2 \right) \cdot 1.05 = 0.807$$

Nevertheless, putting the blade area ratio into Kellers criterion gave us a very high BAR value, since the maximum BAR ratio is 0.8 (Alto University, 2016). That condition would most likely be a maximum if the ocean conditions where rough. One could also think the designer has thought of going faster which would result in a very different BAR value.

$$\boldsymbol{T} = \frac{P \cdot \eta_D}{(1-t) \cdot V_S} = \frac{2300 \cdot 0.6}{(1-0.26) \cdot 13 \cdot \frac{1852}{3600}} = \boldsymbol{278.8kN}$$

$$\frac{A_E}{A_0} = \left(\frac{(1.3 + 0.3 \cdot 4) \cdot 278}{\left(\frac{98100 + 3.65 \cdot 1025 \cdot 9.81 - 1750}{1000}\right) \cdot 3.3^2} + 0.2 \right) \cdot 1.05 = 0.713$$

In order to be on the safe side a blade area ratio of 0.8 was chosen to accommodate maybe a very violent storm.

The values for the different operation speeds where then calculated in excel, graphed and put into open water diagram obtained for suitable propeller. Orange line represents setting, blue hauling and grey transit.

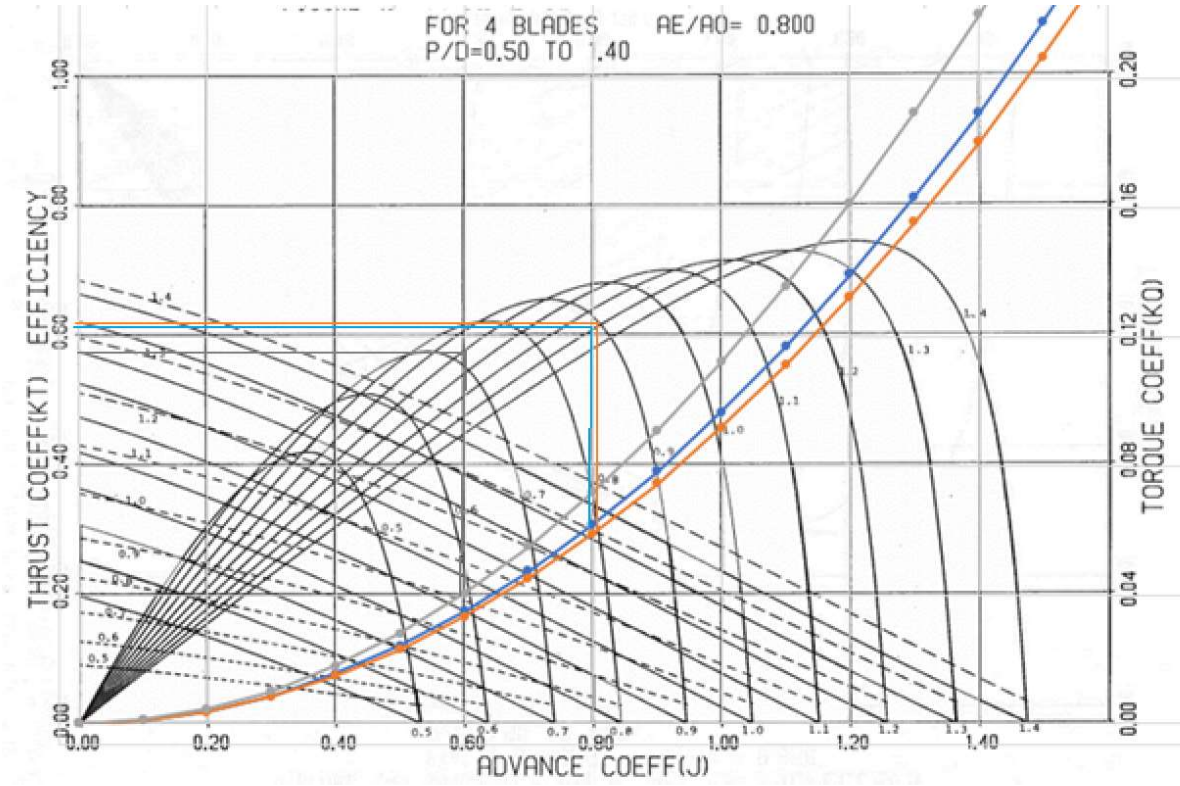


Figure 114: Open water Propeller diagram Wageningen B-series BAR=0.8, with graphed in KT lines *https://deepblue.lib.umich.edu/handle/2027.42/91702*

Intersection point from the graph for Kt and efficiency with different P/D values. Maximum n(eta) (efficiencies are found (Nerland, Propulsion Part 3 of 4, 2017).

Results were deemed quite wrong, most likely due to high error in the resistance measurements from Maxsurf. Therefore, would this be done when more accurate estimates can be achieved later.

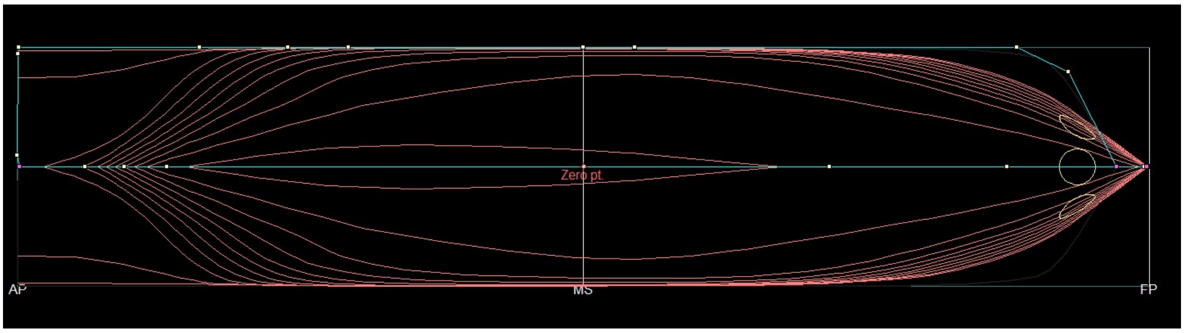


Figure 115: Lines Design Model 203 Beam 13.

Lines Design Model 203 Beam 13.3					
Displacement	2988	t	Waterpl. Area	758.834	m^2
Volume (displaced)	2,914.833	m^3	Wetted Area	1,168.917	m^2
Draft Amidships	5.500	m	Waterpl. area coeff. (Cwp)	0.908	
Immersed depth	5.386	m	Prismatic coeff. (Cp)	0.735	
WL Length	62.914	m	Block coeff. (Cb)	0.648	
Beam max extents on WL	13.283	m	KB	3.280	m
Length: Beam ratio	4.736		BMt	3.536	m
Beam: Draft ratio	2.466		KMt	6.817	m
LCB length	0.931	from zero pt. (+ve fwd) m			
LCF length	-2.531	from zero pt. (+ve fwd) m			

Table 10: Lines Design Model 203 Beam 13.3

New Position for Mag-Packs

The Mag-pack arrangement where moved to a more centralized position, and against the bait storage wall. There were a few factors that went into this decision: the space and transvers trim were the most significant, yet an idea of using the "Anchor Belt" to transport the bait also influenced our decision.

The space around the Mag-packs in the first GA draft can be considered quite cramped on further evaluation. A narrow corridor that goes in an almost s-curve between the bait storage and the Mag-packs is unnecessarily complex to move about in. Therefore, came the idea of moving the Mag-packs away from the "Anchor Belt"-wall and onto the opposite wall. Where we first had planned to have the wardrobe, stairs and exhaust stack. This opens up a straight path from the moonpool-room to the aft section. In addition, it allows the Mag-packs to be moved forward as well, wall-to-wall with the bait storage, which opens up for better space aft of the Mag-packs. As a result, becomes this whole part of the deck easier and more comfortable to move around in, improving the work environment.

It was noted to us as we were guided through Geir that the weight of the hooks and lines are quite significant when it comes to heeling. This is due to the fact that the whole arrangement it placed on one side of the centreline, and it is quite often alternating, since it is released and hauled up again. The new position places the centre of gravity closer to the centreline since it now covers the centreline. This will lessen the effect of transversal trim, and perhaps we can make due with a smaller "heel-ballast" tank.

The idea of potentially using the "Anchor Belt" to transport the bait as well came while we were evaluating the new position. We have discussed earlier that the main point of having the bait storage at this deck is to make it easier to transport it to the unfreezing room and in the end the auto-baiter. Additional ideas of making the transportation even easier, like having a trolley they can use. So, while looking over the new potential layout we realized that the existing belt already covers most of the distance in question. From this came the idea that there could be a hatch into the belt-hall from the corridor, so the bait can simply be lifted from the storage and put through the hatch onto the "Anchor Belt". Then the belt transports the bait down to the stern, and a crewmember lifts it across the beam of the ship to the unfreezing room. This might only cover about half the distance, so we are still speculating if it is a good idea. The only addition we can see is the hatch so there is not a significant increase in complexity and price, however it might seem unhygienic to place the bait on the same belt as the anchors and lines.

Introducing Control Room and Switchboard

As many of the components in the machinery were difficult to find dimensions on, the machinery room in the first GA were quite "naked". So, the owner sent us some example drawings for some of the larger parts: the control room and the switchboard. We had to adapt them a bit to fit them in our design, mainly splitting them up and place the control room above the switchboard room. This allowed us to make use of the already height of the machinery. Although, the GA drawing shows it as only one deck, do we consider it as two because of the height.

Height Restrictions

When presenting this idea to the owner, he commented on the height of the switchboard-room. The switchboards are quite high and there is a need for elevated floor, so it will need 2.8 meters in total height. Since the height in the machinery up to the next deck only is 5 meters there will not be enough height for both the switchboard-room and control room stacked on top of each-other, when taking into account the beam-structure.

Therefore, is there a need to come up with a new design in the machinery. We play with the idea of placing the generator further back, however it would need to be put on an angle to make it fit. Another option is to push the switchboard-room into the storage, taking away our load capacity which we would want to avoid. Therefore, could it be more favourable to move the control room up to the factory deck (deck 3). The problem then is to find space for it at that deck. We could minimize the workshop and put it there, or take away some of the bait storage. Then the rest of the bait would have to be stored in the main storage at the start of the operation cycle. Nevertheless, even if we find space for it at this deck it does not excuse the fact that it still more favourable to have the control room in the machinery. Having both a visual look over the machinery at all times and shorter path to the different equipment is better for workers, and minimizing the length of cables going through the ship decreases the risk of damaging them.

New Layout of Deck 6

The layout of deck 6 have not been revised since one of the first drafts. Even though the intent where to utilize the space and window area, while still having room for the HVAC and bridge electronics, we feel it has not been fully utilized. Therefore, did we come up with 3 new suggestions for GA 201

Traditional/ Simple

The first suggestion revolves around the same principle and aim as the original. Keeping all the rooms, with the lounge facing forward and cabins to the side. The main difference is that the bridge electronics and HVAC are placed side by side at the aft-section of the superstructure. This means we are forgoing one of the potential window walls, and instead making a clean layout and keeping the profile small. The lounge gets a fine view forward at a large angle, almost 180 degrees.

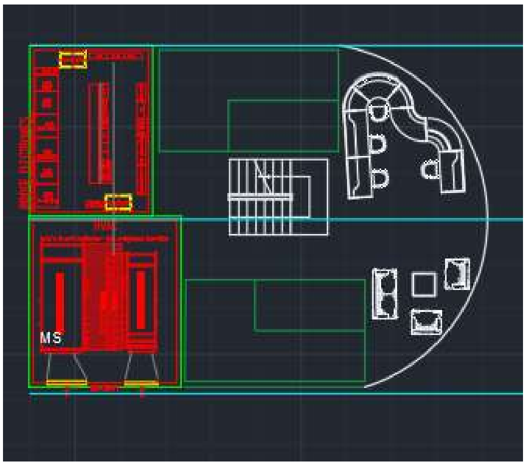


Figure 116: Deck 6 Layout Option I

Forgoing the Cabins

For the second layout we extracted the two cabins, meaning these would have to be placed elsewhere. We suggest deck 5, although this would mean increasing the length of that deck, which is lower and therefore beneficially for the wind area. In this layout the utilization of the window area lacks, and there is only the forward-facing lounge with its almost 180-degree view. The volume and deck area of the lounge is increased a bit compared to the first suggestion, and the seating is slightly expanded.

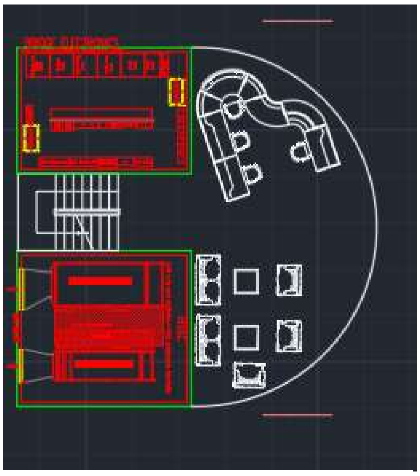


Figure 117: Deck 6 Layout Option II

Maximum Windows

By increasing the size of the whole deck, a bit, all of the potential window area is being utilized. The HVAC only faces outwards with the vents that needs to do so, while the bridge electronics and stairs are gathered in the centre surrounded by a huge u-lounge. Although this lounge does not favour larger seating groups, as we have used before, on the other hand there are numerous of smaller seating, two/three chairs around each table.

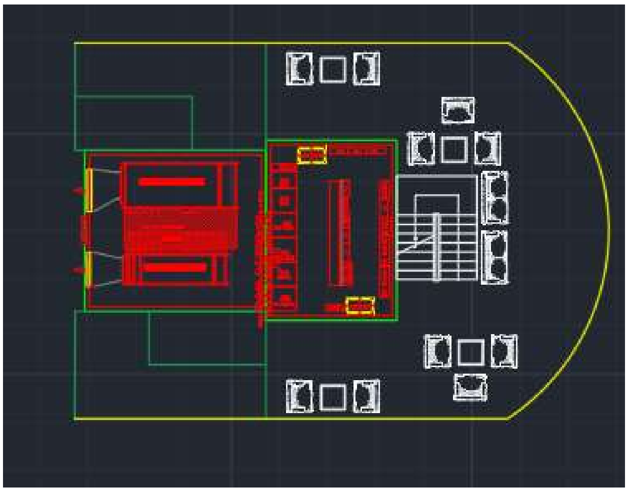


Figure 118: Deck 6 Layout Option III

In order to utilize the forward-facing windows without increasing the profile area, we placed a four-seat couch in front of the stairwell facing forward. All in all, this arrangement is not the most social, nevertheless it is comfortable and relaxing. The idea with this layout is that you sit in a large group around the dining table, and then smaller groups can move up to deck 6 and relax. Allowing the crew to socialize in smaller groups without having to retreat to the cabins.

We do fear that the HVAC system might be quite noisy, and that this could become a problem in this layout. Since it is placed between the two cabins, the noise might be difficult to isolate against or at least it requires more isolation to keep the noise level in the cabins low enough to sleep in.

Pallets

By importing the external lines into AutoCAD, we are able to arrange the GA inside the hull. Then we draw the inside wall according to the space needed for the beam structure, in the main storage there should also be an isolation layer on all the walls. After this is drawn, we start to place drawings of pallets in a systematic order that should maximize the available space. Before in the estimations, the number of pallets is based on the total area and are positioned in a spreadsheet fashion around the centreline. When we now are placing them in an “already designed” space, we decide to start in the front corner and place as many as possible before reaching the centreline. The thought is that we can mirror it afterwards and then see if there are space to fit some along the centreline. We are foremost placing the pallets on the tank top deck and will evaluate if there is space for more at pallet height 2 or 3 afterwards.

It seemed that only that pallets are removed from the first layer, resulting in 565 pallets. Well over the required amount, and can indicate that we have made an oversized storage which might bring on a few consequences. However, at this point in time we keep it as an estimate due to the storage capacity is at the top of our priorities. We evaluated that if there is a possibility to have this capacity, we should explore it.

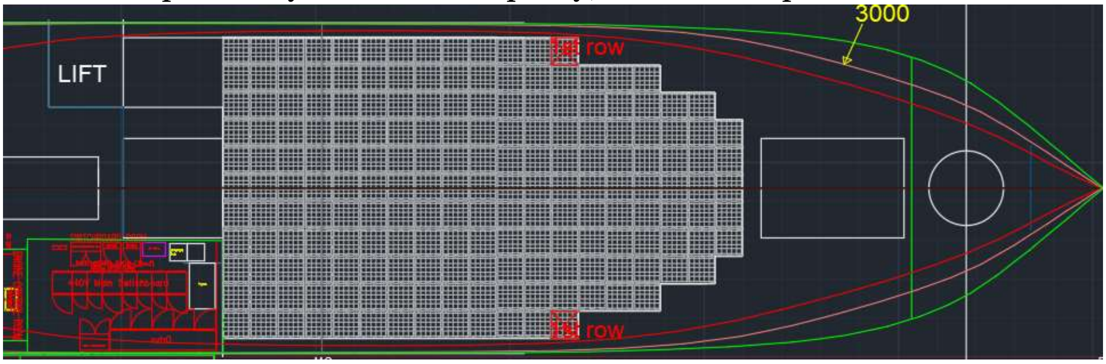


Figure 119: Pallet Arrangement 203

Refrigeration system

The room for the cylindrical coolers has quite a lot of unused volume on its sides, due to the hull-lines. For that reason, we decide to flip the room 90 degrees so that the longest wall is transverse. This allows for more space in the storage room, although it should be noted that this will result in an LCG that is further forward.

At some point, we realized that the volume on the sides could be used for the dump from the Anti-roll tank. Although, even when the cylindrical coolers room was flipped the remaining volume was more than what was needed.

Strength: Third Structure

Due to the girders being prone to vibrations, we were advised to add columns in the transverse girder sections. Our mission for the project is to be able to carry as much fish as possible, so we wanted to find solutions to reduce the space lost in storage. We got confirmed that a reinforced centre girder could replace some columns, so a column on each transverse girder was not necessary. Therefore, we chose centre girders in the storage section, skipping columns on the second and forth transverse girder from moonpool. In the rest of the decks we chose centre columns on each deck in front of moonpool, and two side girders in all decks behind moonpool due to the systems installed there. Due to the extra support we chose to increase the girder spacing to 4000mm, to match the capacity of the beams.

A concern that grew from this change was buckling in the strength deck. By changing the girder spacing we decreased the buckling coefficient. But after a quick check we found the change to be insignificant.

$$Stiffener\ spacing\ 3000 \rightarrow \left(1+\left(\frac{600mm}{3000mm}\right)^2\right)^2 = 1.082 \rightarrow \sigma_{euler} = 27.4\frac{N}{mm^2} > \sigma_{crit}$$

$$Stiffener\ spacing\ 4000 \rightarrow \left(1+\left(\frac{600mm}{4000mm}\right)^2\right)^2 = 1.046 \rightarrow \sigma_{euler} = 26.5\frac{N}{mm^2} > \sigma_{crit}$$

Global and local

Increasing transverse girder spacing does not affect global strength or strength of the stiffeners or plates significantly. Section scantlings tells us the section modulus is roughly the same, and stiffener profiles are identical with the previous structure.

Transverse

By applying the columns, the bending stresses reduced drastically. The transverse girders could be reduced to 300mm with flanges of 150X10mm being the optimum flange to web ratio for an intermedium beam. However, with the deck plate increasing the neutral axis comes closer to the deck plate and we might use Aw/Af=3 as an optimum beam.

The columns were initially inserted as the default 3D beam had to offer. This was to receive the force that worked on the column normal to the column profile, so the value could be used for hand calculation to find the necessary area of inertia the column.

Using column 198 as most critical and applying column therory.

$$P_{euler} = \frac{\pi^2 EI}{\lambda^2} \rightarrow I = \frac{P_{euler}\lambda^2}{E\pi^2} = \frac{94 \quad *5000^2 mm^2}{\pi^2 * 206000 N/mm^2} = 1156cm^4$$

We found a pipe column with diameter of 15 cm and thickness of 12mm being just within the criteria. However, while being within the criteria of buckling, the column failed in yielding. So, by switching failure mode to yield we increased the column to 16cm in diameter and 15mm in thickness. After increasing the strength, the load increased as well. However, the load where increased by 9.5% and the area of inertia was increased by over 300%, knowing the area of inertia and load to be proportionate each other, it was evident that we would have problem with buckling.

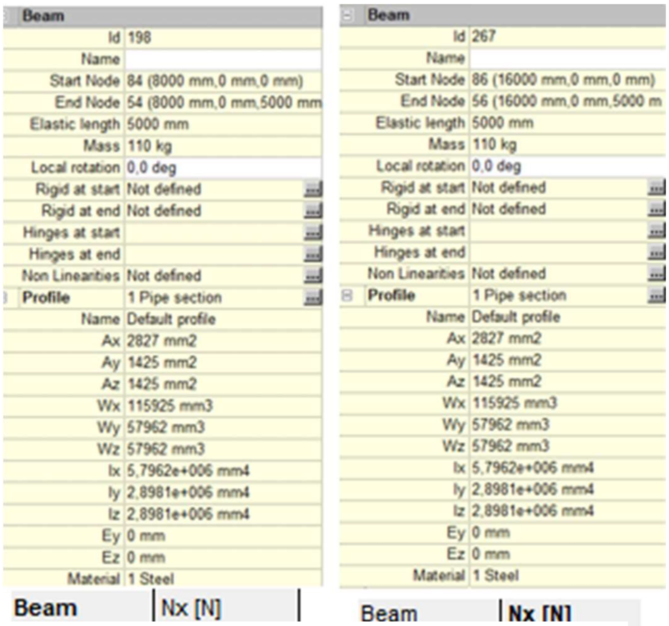


Figure 120 Critical column overview

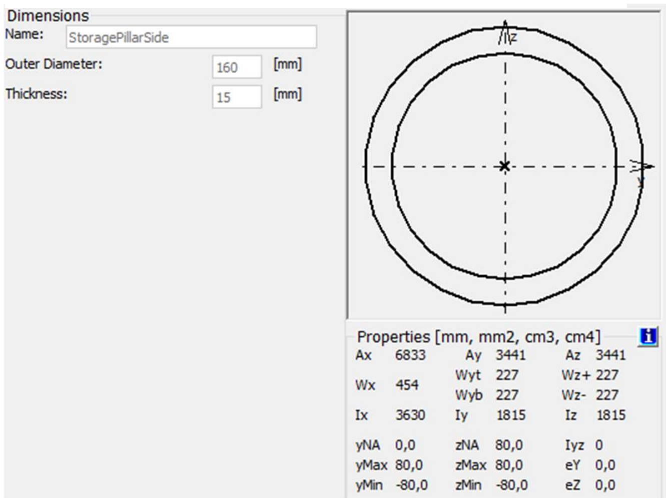


Figure 121 Pillar overview

Another Round

Again, we take another assessment of where we are in the process and how the last changes have affected the other parts of the design. Hence, we take on other spin in the design spiral before we enter Part III of the paper, where land on a final design.

Counting Pallets

The tank top is not reaching the full beam of 13.3 meters and therefore is it not enough space to fit 11 rows transversely, as planned. However, due to the increased length we are able to fit 187 pallets on the first level. Since it is stated in the requirements that we can have 3 pallets in the height we know that we will at least be able to fit 3x187= 561 pallets. From the lines at 3500 mm above baseline (abl) it comes apparent that there is room for more pallets at the top-layer, 4 pallets can be added to the established arrangement. Resulting in 565 pallets, which is actually the same as the last estimate even though there were a lot of changes.

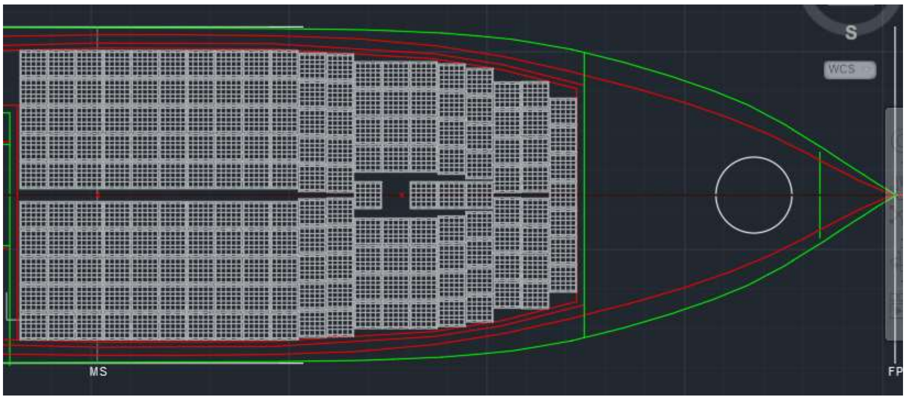


Figure 122: Design-227: Pallets First Layer

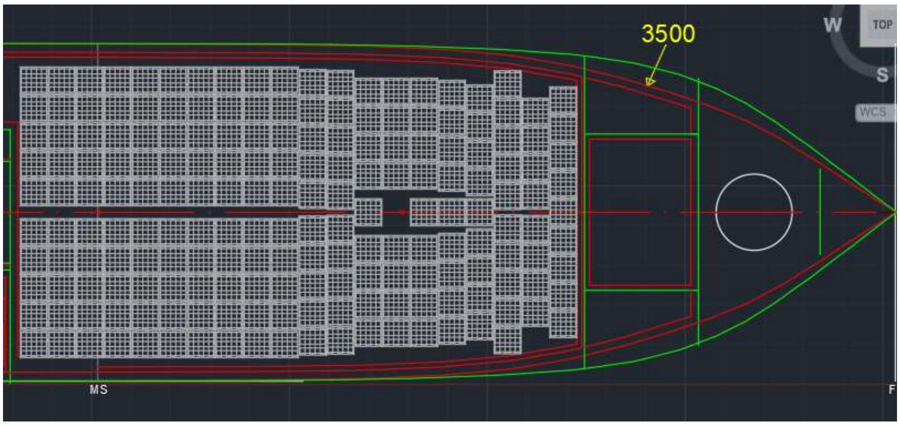
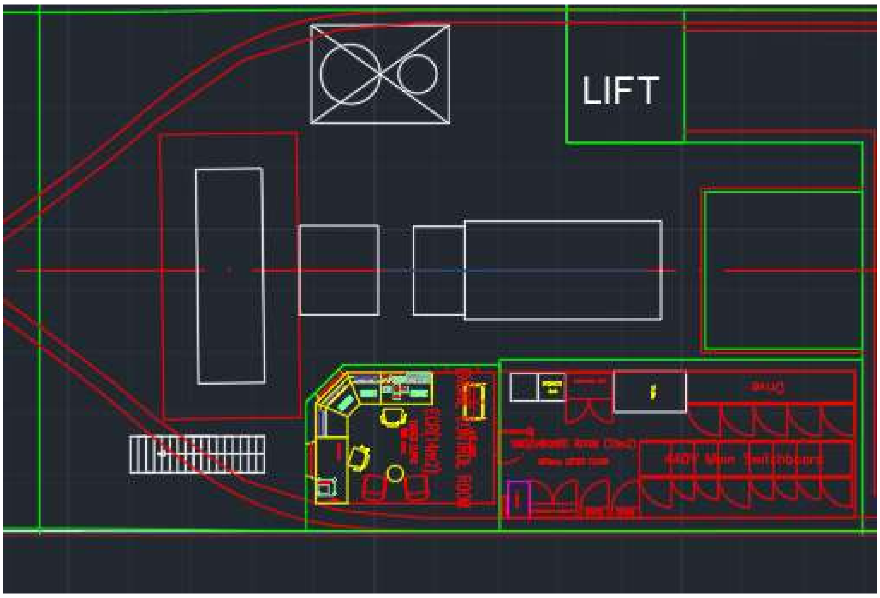


Figure 123: : Design-227: Pallets Top Layer

We also make a separate arrangement from scratch for the top-layer, 3500 mm abl, enabling us to place 11 pallets in the width of the vessel. The top layer is then counted to contain 210 pallets. This is possible; however, it would have to include support for the outer pallets all the way around in addition to more complex stacking. Different factors might affect this arrangement in more ways than the other, as we are using the full beam the size and placement of support beams have greater effect on the utilization. The 187 pallets per layer arrangement has most of the centreline "open" for support beams. In addition, do we fear that the securing of the pallets might be more difficult if the top-layer has a completely different arrangement.

Effecting Other Parts of the Design

Since we started the arrangement from the front, and because the storage starts further forward than first expected, the aft end of the pallets do not line up with the aft wall of the machinery. Therefore, can the moonpool be moved about half a metre forward to eliminated unused volume. The new moonpool centre is 26 metres from AP. This again effects the position of other walls and systems, for instance the control room and switchboards have space enough in the machinery if the generator is placed transversely.



The Factory is moved 500 mm forward and the accommodations-rooms in front of the moonpool are moved 300 forward.

The centre of gravity of fully loaded storage were moved from 37.78 to 39.97 meter forward. Therefore, the lines needed to be redesigned for this purpose since LCB was measured to be needed approximately 1.4-1.5 meters in front of the midsection. That resulted in approximately LCB being placed 4.35% in front of midsection.

LCB position

This created a problem as increasing the LCB so far forward made the forward section highly inefficient due to increase in wave-resistance. So, the lines in the stern needed to be redesigned. The stern water lines were measured to be approximately 40 degrees resulting in separation and bad flow to the propeller. In order to create good flow to propeller the waterline should not have a slope of more than 28 to 30 degrees (Watson D. G., 1998). The problem arrived with the necessary space in the machinery for main engine and generator. New lines were carefully drawn into with that in mind in order to ensure a better flow. The volume in the aft section was reduced and made it possible to reduce the forward trim. In order to have the same displacement the block-coefficient of the mid-section were increase. Result was the resistance highly reduced using the Holtrop statistical calculation from Maxsurf resistance. Since our vessel is both small and unconventional does this not give a very accurate answer and therefore a model-test is necessary. But it can give an idea. Prismatic coefficient was reduced from 0.645 to 0.610.

Rechecking Stability

Following the design spiral, the lines were optimized further in MaxSurf Stability. Since the hull shaped were made larger did the tank also become larger.

During this analysis, the critical conditions for the metacentre changed from not only being the lightweight for the vessel, but also arrival with 20% catch and 10% in tanks. That was most likely due to the low weight from fish storage and tanks, but enough weight to cause by the transom, not immersing in the water. When trying to solve that problem, two options presented. Keel was calculated to be necessary to be 210 tonnage.

What Happens when everything filled in front of LCB

Since the idea behind the storage being filled forward is to reduce the trim, can that have a negative effect since back row of 4 pallets is filled in behind LCB of the vessel. Every weight that is placed forward of the LCB will cause a forward trim and vice versa. Therefore, in theory those 4 rows will cause trim more in the aft section when filled up. Thus, it was necessary to check this. From the excel spread sheet was the new LCG and weight of the cargo calculated. Resulting in 440 pallets with total weight of 469 tons and an LCG of 42.23. To test that current load case of departure from fishing ground was used only changing the cargo, in order to see the difference. Current values from stability gave us approximately even trim with a draft of 5.5 at F.P and 5.45 at A.P. Which resulted in a larger draft at A.P with everything filled in front of LCB. a draft at F.P of 4.31 and A.P of 4.32. The total LCG of weights where moved 0.15 meter forward, but since the forward section of the ship has a larger volume, thus that cause less draft in F.P. Also showing that one can achieve an almost even trim for the vessel at a larger rate than filling the storage from the back. Achieving nearly even trim with a storage filled only to 78%. Achieving the goal set of reducing trim fastest possible.

Strength: Fourth and final

We wanted to find ways to reduce the weight on our structure. So far, we have had the idea that we wanted a simple structure as possible to reduce building cost. By simple we mean less contact faces to weld, fewer cut-outs for stiffener, etc. However, having some issues with the stability of the ship at the time, we were interested in finding ways to save weight.

The first measure where to reduce stiffener profile by adding more longitudinal girders. Decreasing the stiffener span could reduce the profile drastically, and with 4000mm girder with 300mm web and flange of 100x10mm weighing 125kg, the potential was great.

From 3D beam we could easily calculate bending stresses of stiffeners, simulate girders as supporting members as well as receiving the weight of the stiffeners. For the decks we lined up three stiffeners, applied deck load, fixed them in each end and added one fixed node for one stiffener and two fixed nodes for a second stiffener. Then calculated and reduced the profiles until they reach the yield capacity and gathered its weight. The nodes were placed 2000mm from centre to align with moonpool and 4325mm from centre being half the remaining span to the hull side.

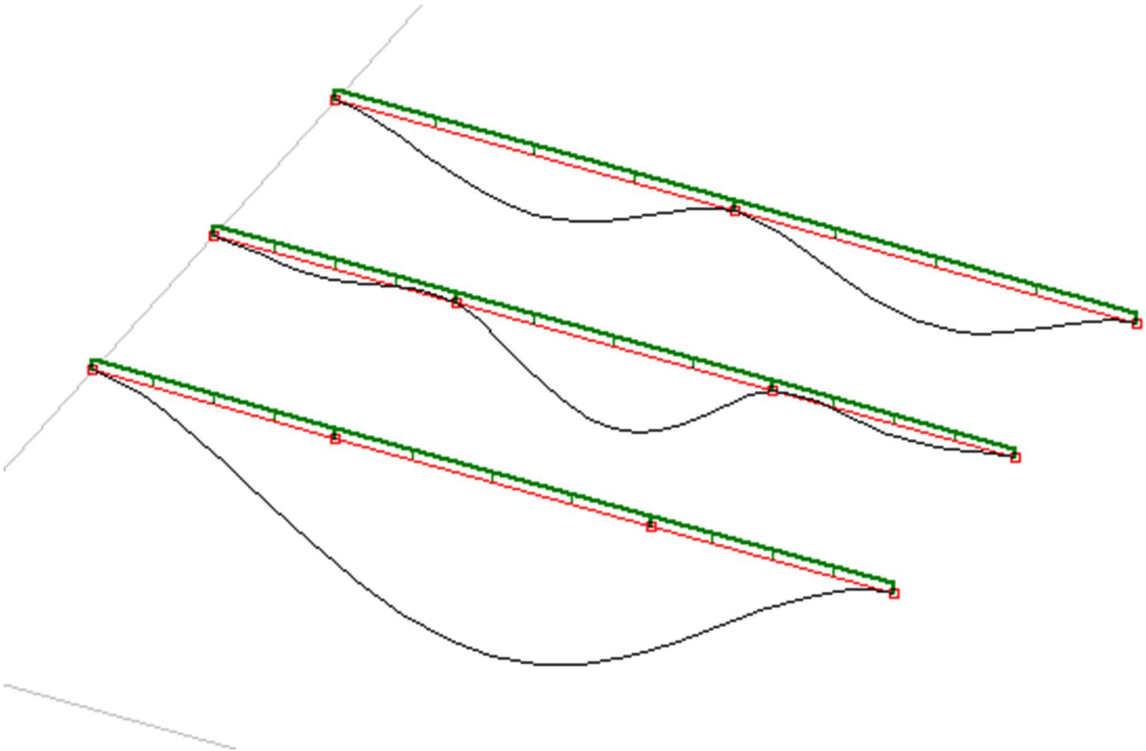


Figure 124 Stiffener supported by longitudinal girders decks

The stiffener with no girder weight up to 273kg *10 stiffeners per transverse girder * 15 girders over the whole ship resulting in a weight of 41.0 tons.

The stiffener with one girder weight up to 244kg*10 stiffeners + 2 longitudinal girders of 125 kg per transvers girder * 15 girder over the whole ship resulting in a weight of 36.5 tons.

The stiffener with two weight up to 203kg*10 stiffener + 4 longitudinal girder * 15 girders resulting in a weight of 31.0 tons.

The hull side stiffener at the top is already almost at bear minimum so no girder can reduce the weight here. In the storage side we have massive stiffeners; reducing these elements is not only advantageous for weight reduction, but since the stiffeners penetrates the isolation and steal conducts heat well, it can be appealing to reduce the stiffener height. Since we want the isolation to be as effective as possible, we will review an arrangement with one longitudinal girder.

Here is three arrangement of the hull side. The first is the previous arrangement, second is one girder in storage and a third with an additional girder in the factory deck.

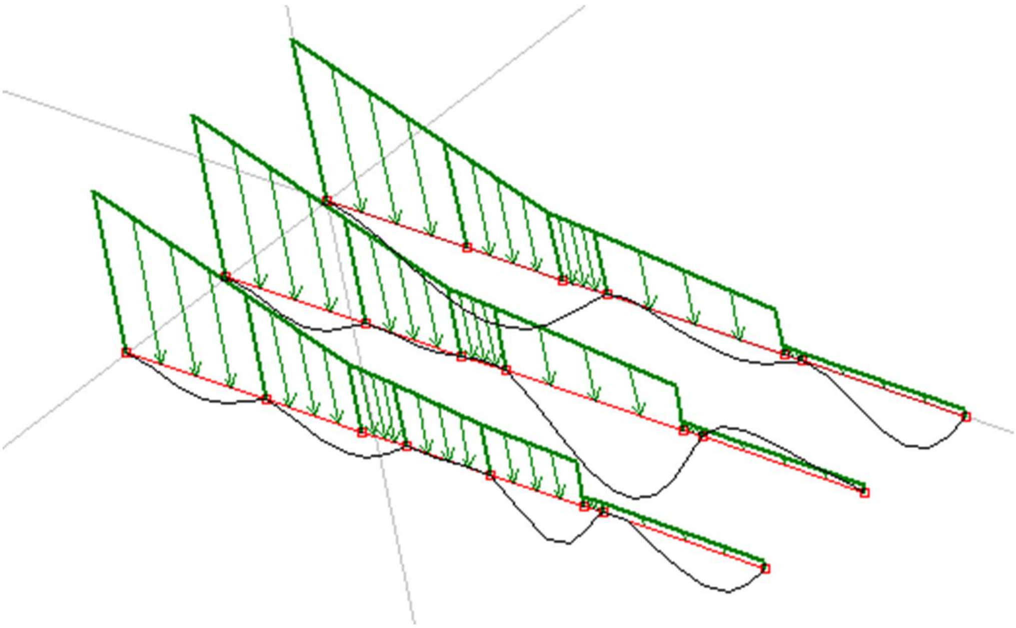


Figure 125 Stiffener supported by longitudinal girders hull side

First arrangement:

$$676kg * 10 * 15 = 101ton$$

Second arrangement:

$$(506kg * 10 + 125 * 2) * 15 = 76.6\ ton$$

Third arrangement:

$$(461kg * 10 + 125 * 4) * 15 = 69.5\ ton$$

Even if the structure creates more complexity and production cost, its more favourable to have a structure that weighs less. A structure that might be too costly to produce ought to be specially considered, while as a rule of thumb, a structure that saves weight is to be pursued.

Our rough estimate showed that by adding girders in the storage and factory hull side, and four longitudinal girders in each deck we could potentially save up to 31.5 tons + 10 tons *3 decks ultimately resulting in a weight reduction of 61.5 tons.

The weight reduction was a success, and without experience on how much production of these arrangement cost we figured this arrangement were our best guess. In addition to the weight loss, we also reduced the girder height. We had earlier set apart 500mm in deck height to the girder and by reducing the webs to about 300mm, we effectively reduced the height of the ship, which had a positive effect on the ship's stability.

The girders were starting to reach the capacity of 300mm webs and in some critical parts of the ship the capacity for an optimum beam where exceeded. Even with the increased flange there what weight to reduce by adding longitudinal girders. We found this to be a great time to proceed to the detailed part of the structure analysis.

Moonpool

The moonpool may be the most critical section of the ship. It is the cause to many disruptions in longitudinal members as well as some stiffeners, and the actual pool caries 48 cubic meters of water in a design load condition. The load is dynamic as the ship receives heave and pitch motions, as well as the water level fluctuates as water rushes in and out of the tube. The pool side and tube experience increasingly pressure at greater depth; this pressure is directed in x and y direction, which caused an unexpected issue.

So far in the structure we had used circular tubes as columns which have received little to no bending moments due to symmetry in the load case and strength arrangement. The tubes retain equal strength in all directions, by the cost of being slightly weaker in comparison to a square tube or an I-beam against bending moments. The column placed to support the storage side of the moonpool experiences several critical loads.

1. The storage skips a column on the first transverse girder. The column is therefore taking the load from 4000mm of the longitudinal girders rather than 2000mm.
2. Moonpool having the possibility to experience the load of 3m water depth.

Being supported by the moonpool side the compression stresses were not as big as the columns in storage, but what fails the column, is the bending moments caused by

3. Uneven distributed loads on each side of the column.
4. The water pressure bending the moonpool walls.

The column is not strong enough to withstand both the compression- and bending stresses.

	Beam	|Sig-Nx| [N/	|Tau-Qy| [N/	|Tau-Qz| [N/	|Tau-Mx| [N/	|Sig-My| [N/	|Sig-Mz| [N/m	**|Min Sig-Ny|**	|Max Sig-Ny|	|Min Sig-Nz| [	|Max Sig-Nz| [N/mm2]
9	193	129	0	2	0	72	0	202	57	129	129

Figure 126 min sig-ny over the requirement

The column transfers the compression stresses to the stiffeners in the moonpool side ultimately failing them.

We choose to remove the centre column and replaced it with two columns on each corner of the moonpools storage side going all the way up to the factory deck. The column was chosen to have an I-Beam profile to withstand the bending moment.

	Beam	|Sig-Nx| [N/m	|Tau-Qy| [N/	|Tau-Qz| [N/	|Tau-Mx| [N/	|Sig-My| [N/	|Sig-Mz| [N/m	|Min Sig-Ny| [	|Max Sig-Ny|	|Min Sig-Nz| [	|Max Sig-Nz| [N/mm2]
1017	1017	54	0	18	0	83	17	137	29	71	37
1018	1018	54	0	18	0	83	17	137	29	71	37

Figure 127 sig-ny within the requirments

The moonpool lies on a different plane than any deck the ship, making a challenge for transferring the load to the beams. To simplify this transfer, we chose to use girder on each side of the moonpool, so the load can co directly from the stiffener to the main structural members. In the moonpool we set transverse stiffeners in the bottom and stiffeners one each side of the pool to carry the hydrostatic pressure.

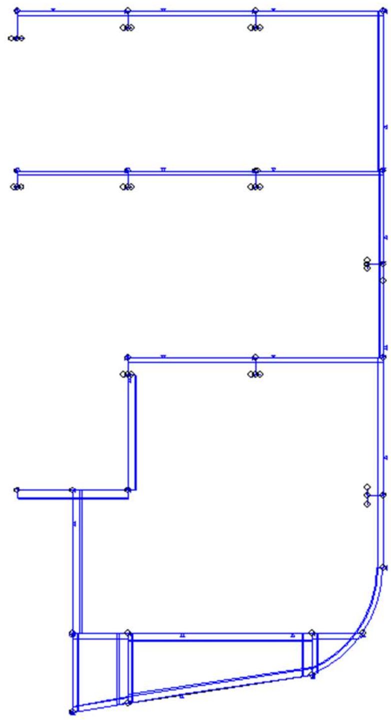


Figure 128 Section scantling model of moonpool section

By using section scantling, we usually calculate local and global strength, but the modelling technique in section scantling does not create a true to reality model of the moonpool. We need to check the local strength in moonpool to get a more accurate answer. However, section scantling can provide us with the global strength and local strength in the non-moonpool members

The result did not deviate from the general structure in the storage section, so we proceeded to review the local strength in 3D-beam. However, the section modulus increased without adding anything to the top… This may be because, by removing the area in the bottom, we heighten the neutral axis, and thereby shifting some of the stress to the bottom.

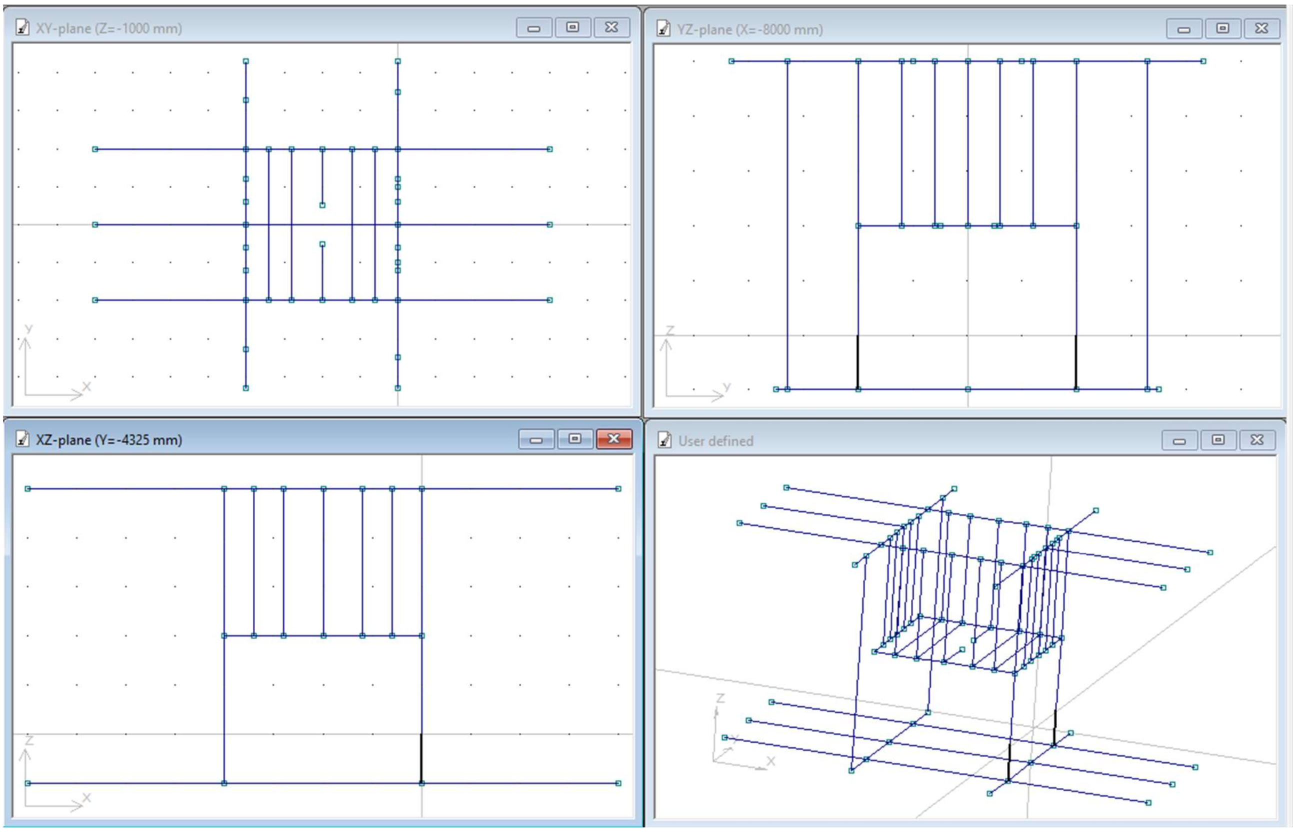


Figure 129 Moonpool structural arrangement in 3D-beam

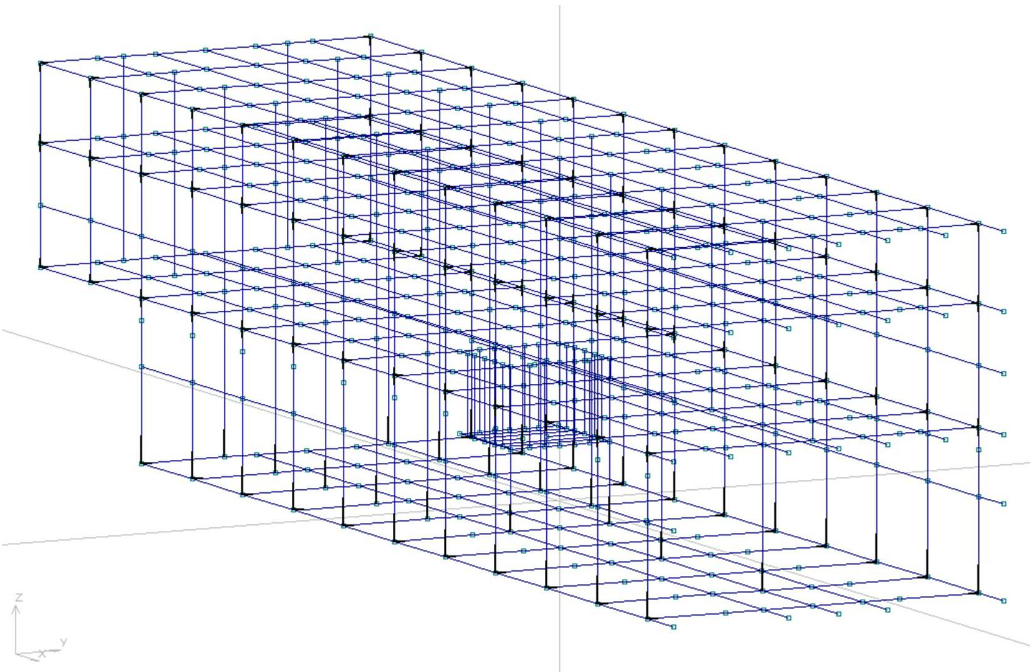


Figure 130 Structural arrangement from aft to fwd bulkhead

The stiffeners are welded to the pool bottom which is represented in the model as L-beams. The load is dynamic, so we imagined a worst-case scenario where the moonpool is filled to the rim creating a hydrostatic load of 3Tonnes/m^3.

Superstructure

The superstructure is made of aluminium, a much lighter and a little weaker material. Aluminium does not share the same material capabilities as steel; the line between plastic and elastic behaviour is not clear, but usually the yield point is determined to be load that creates a permanent deformation around 0.2%. The material factor for aluminium is calculated to be

105 The various formulae and expressions involving the factor f_1 may normally also be applied for aluminium alloys where:

$$f_1 = \frac{\sigma_f}{235}$$

σ_f = yield stress in N/mm^2 at 0.2% offset, σ_f shall not be taken greater than 70% of the ultimate tensile strength.

Figure 131 (DNV,2015, pt.3.Ch2.sec2. C105)

The aluminium in 3D-beam have a tensile strength 110MPa which is within the 70% of ultimate strength of 165MPa. The material factor is then calculated to be 0.47 resulting an allowable stress of 75.2MPa for non-longitudinal strength members. This begs the question; is any of the superstructure a part of the longitudinal strength.

The superstructure does not span continuously over the whole length of the ship and cannot be considered a strength deck and seemingly not part of the longitudinal strength. However, it is considered that the longitudinal stresses are transferred to the superstructure at an angle at 15°. This gives us around 7 meters of the superstructure being part of the longitudinal strength. The rules state that within 0.4L of amidships the allowable stress should be 130MPa*f1 and 160MPa*f1 within the perpendicular. So, we chose 130MPa timed f1 for aluminium for the longitudinal girder in fwd part of the deckhouse; 160MPa timed f1 in rest of the members.

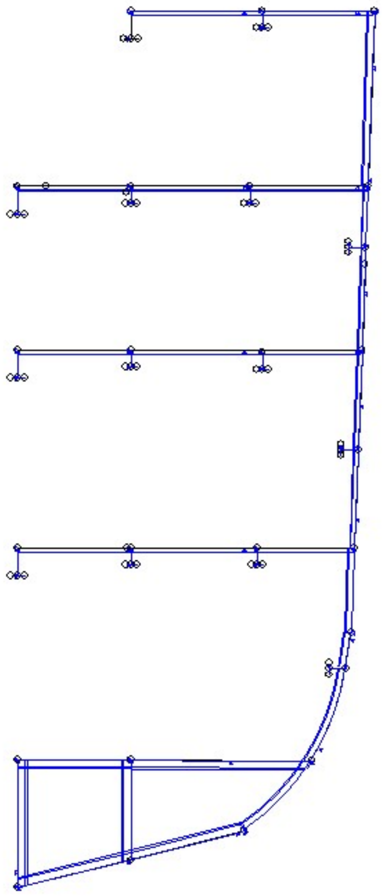


Figure 132 Section scantling model of hatch section frame #85

We could already predict that we would not have any trouble with the global strength. The section modulus was increased with the extra deck, but we did receive stiffener dimensions and plate thickness from sections scantling. However, we were more concerned with the girder arrangement.

In 3D-Beam we simply made the superstructure and fixed it in z directions in the bottom. We put steel columns in dining deck as the weight difference of those elements is not great. The loads on the girder by the hatches increases by 50% due to the hatch dispersing the load over a longer span.

All beams got increased to a web height of 400, the, with deviating flanges to fulfil the strength requirement.

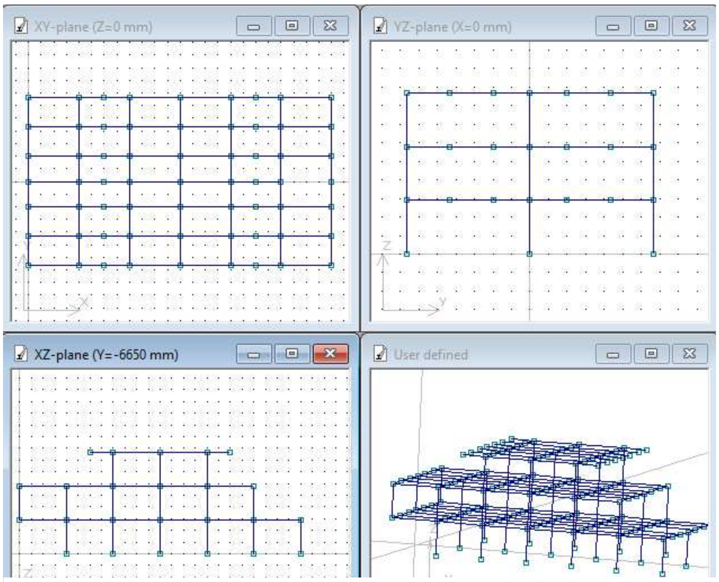
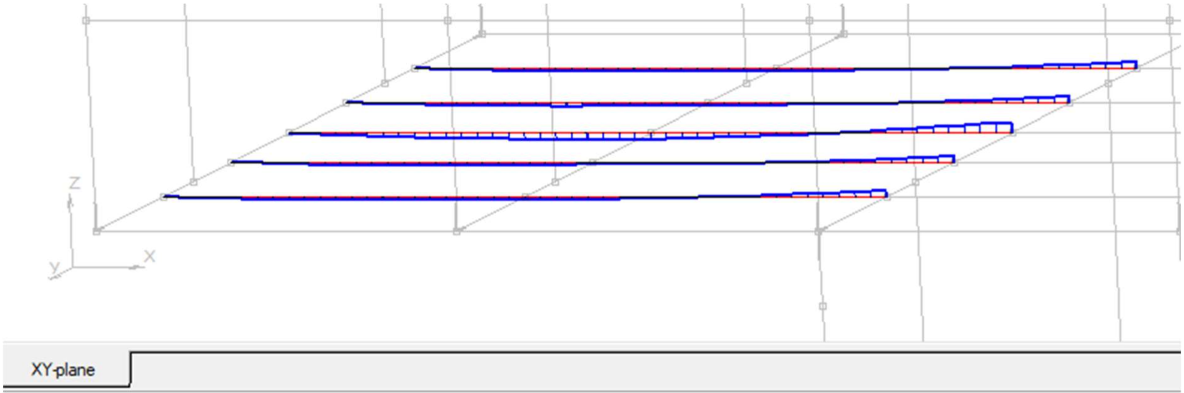


Figure 133 Structural arrangement of the accommodation with hatch in 3D-beam

Stern Overhang

Throughout the ship there is water that supports the structure. The stern has lower depth to than the rest of the ship as its raises, resulting in lover sea pressure that support the structure. We fear the this will result in high local tension in the longitudinal members.

Analyzation shows that we get trouble in the transverse members in the stern.

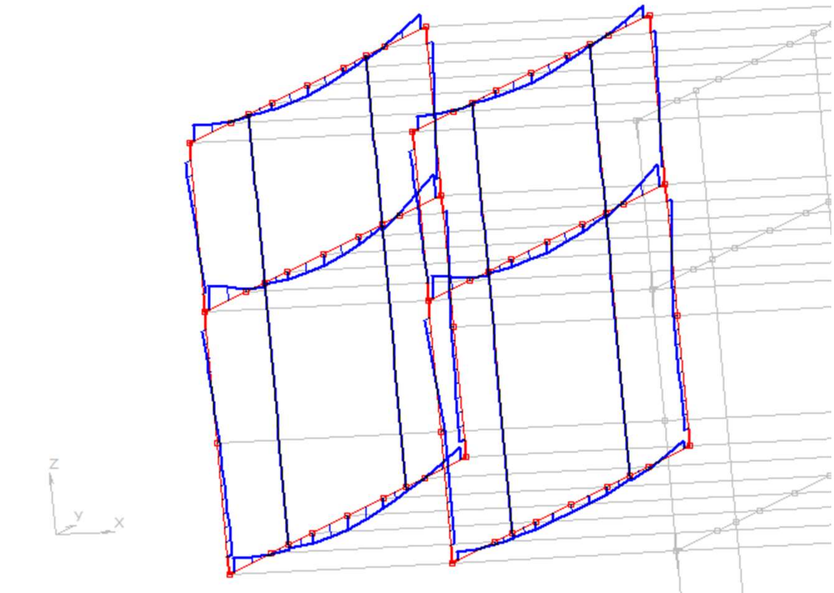


	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/m	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	**|Min Sig-Ny|**	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-
1	893	4	1	15	1	94	18	98	33	22	15
2	894	4	1	15	1	94	18	98	33	22	15
3	898	1	0	14	0	86	2	87	31	3	1
4	897	1	0	14	0	86	2	87	31	3	1

Figure 134 Bending moments and stresses on longitudinal members

Low stresses in the longitudinal members.

The reason for this might be due the column do not transfer the sea pressure to the decks above, effectively increasing the length of the transverse girders. We must therefore increase the profile to the transverse members in the stern.



XY-plane

	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/m	|Tau-Mx| [N/m	**|Sig-My| [N/**	|Sig-Mz| [N/m	|Min Sig-Ny| [N	|Max Sig-Ny|
1	730	1	0	38	0	324	4	325	72
2	741	1	0	38	0	324	4	325	72
3	696	5	0	36	1	304	2	309	63
4	707	5	0	36	1	304	2	309	63
5	734	8	2	25	0	200	14	209	37
6	748	8	2	25	0	200	14	209	37
7	746	1	3	37	0	175	35	176	81
8	747	1	3	37	0	175	35	176	81
9	742	0	4	16	1	166	29	46	167
10	743	0	4	16	1	166	29	46	167

Figure 135 Bending-moments and stresses on transverse members

Part III: The Prestige

The third and last part of this paper is where the results of the design spiral is shown, as it all comes together. We will show our last thoughts on the design of the lines, GA and the resulting stability with the tank arraignment. In addition, to how we had to make some final adjustment to satisfy the requirements and how changes at this stage have to be done differently. Then, we will bring up the conclusion of the strength calculations and the respective class drawings. The final GA and Class Drawings are presented in the Appendix. Moreover, we finally got to do a model testing, which was done with the design we are presenting in this part of the paper. From that test we got some values of the resistance at different speeds, which we will do a quick assessment off and present how we would go about adapting the design to improv it and what consequences this might have. Which is how we would present it to the owner and hear his thoughts on that dilemma, so that we could continue to make the design he wanted.

301

The vessel is moving through the design spiral as it searches for convergence, and now closing in to the final design. There has been a lot of discussion about the challenges we have encountered, changes to solve them and the consequences of these changes. This is something that could go on and on to further improve the design, nevertheless designing a vessel needs to be done on time. Hence, we are now seeking the final solutions that acts as a compromise between the different aims we have been seeking.

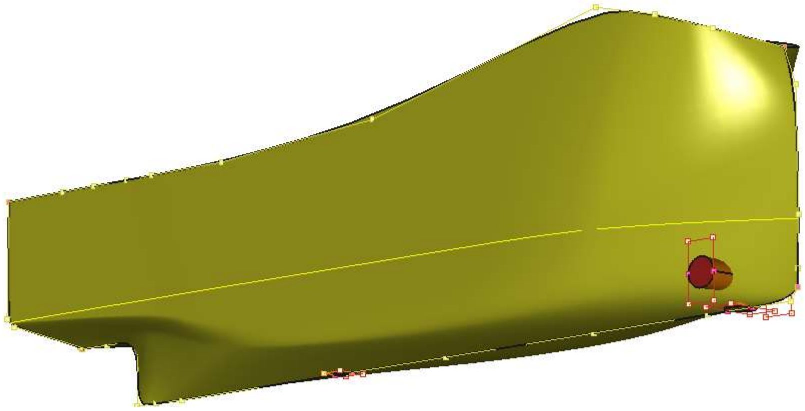


Figure 137: 301 Model

Hydrostatics at DWL

	Measurement	Value	Units
1	Displacement	2920	t
2	Volume (displaced)	2921,228	m^3
3	Draft Amidships	5,500	m
4	Immersed depth	5,500	m
5	WL Length	63,168	m
6	Beam max extents o	13,316	m
7	Wetted Area	1188,570	m^2
8	Max sect. area	65,568	m^2
9	Waterpl. Area	752,794	m^2
10	Prismatic coeff. (Cp)	0,705	
11	Block coeff. (Cb)	0,631	
12	Max Sect. area coeff	0,944	
13	Waterpl. area coeff.	0,895	
14	LCB length	1,416	from z
15	LCF length	-2,760	from z
16	LCB %	2,241	from z
17	LCF %	-4,370	from z
18	KB	3,212	m
19	KG fluid	0,000	m
20	BMt	3,501	m
21	BML	71,621	m
22	GMt corrected	6,713	m
23	GML	74,833	m
24	KMt	6,713	m
25	KML	74,833	m
26	Immersion (TPc)	7,525	tonne/c
27	MTc	34,674	tonne.
28	RM at 1deg = GMt.Di	342,078	tonne.
29	Length:Beam ratio	4,744	
30	Beam:Draft ratio	2,421	
31	Length:Vol^0.333 rati	4,419	
32	Precision	Medium	67 stati

Figure 136: 301 Hydrostatics

From our last design, the main change on the lines where that we increased the radius of the bilge in order to obtain smoother curves and less resistance. The downside to smoothen the lines in the bilge area is that the rolling motions are not damped as much, nevertheless still can be countered by a bilge keel.

Where the transom is raised will increasing the bilge radius to a large extent effect the designed waterline. The area at waterline would be reduced as the radius increases. Therefore, where we limited at the aft by our stability requirements. Although we tried to reach a compromise that we could check the stability again in the stability software.

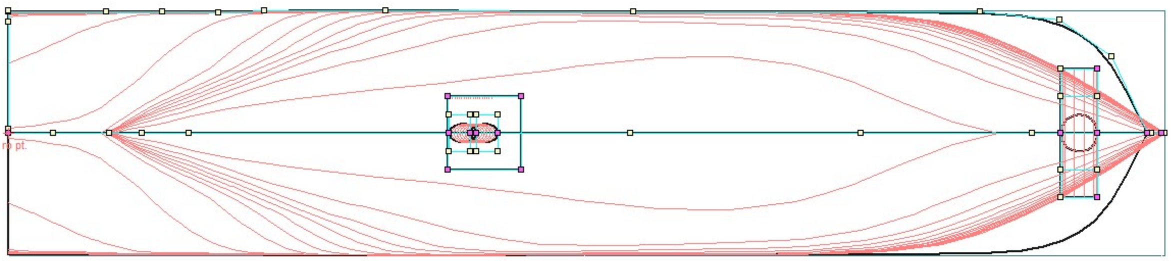


Figure 138: 301 Water Lines

There has been a large focus on the waterlines, keeping them smooth and leading the sea into the propeller. The detailed design around the skeg for example have been down prioritized in favour of the overall design. One would typically design the skeg with a separate surface and then blend it together with the rest of the hull. This aids to make a more detailed design.

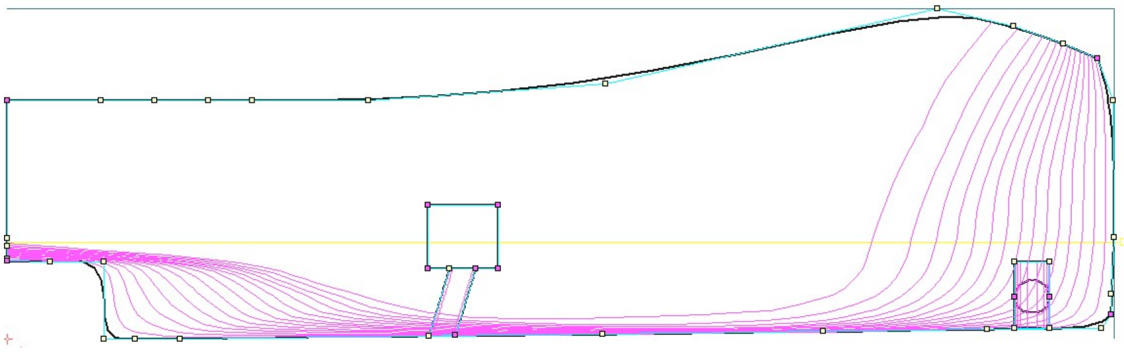


Figure 139: 301 Buttocks

From this perspective the increased bilge radius becomes more apparent. The transom has gotten a shaper angle to hopefully aid in the stability, so that at least the transom goes as deep as it can in the centreline before reducing the space for the propeller. Because as it can be seen the difference in the “edge” when approaching the stern is quite drastic.

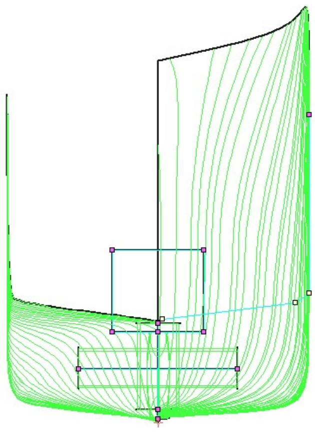


Figure 140: 301 Sections

Deck 1 and 2

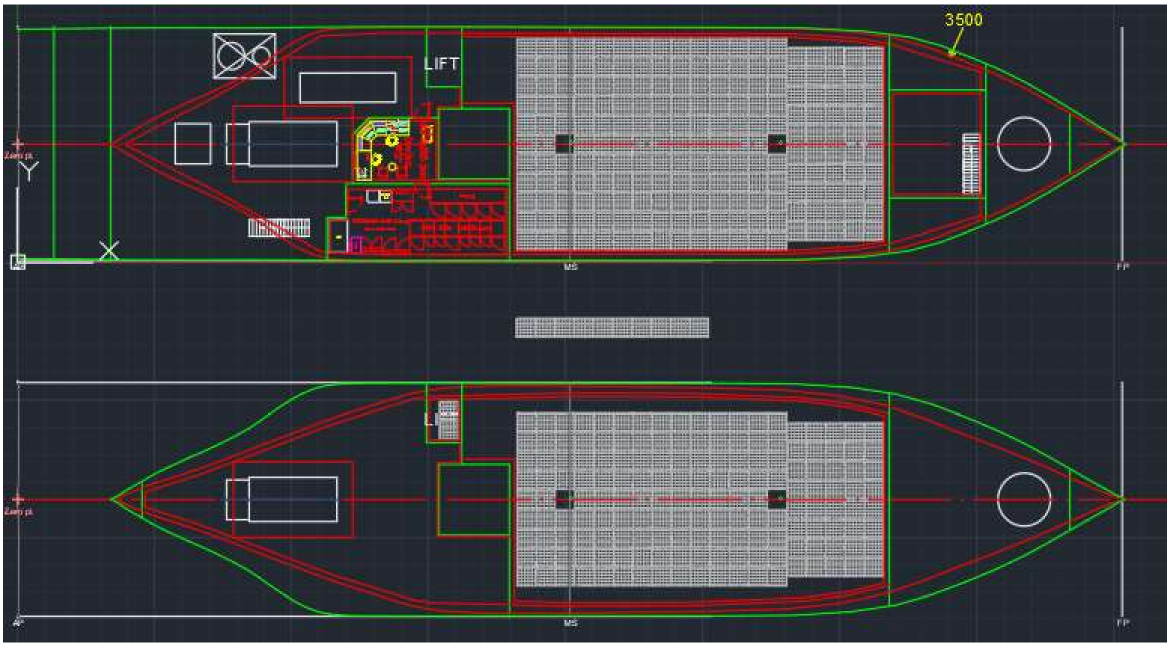


Figure 141: 301 GA Deck 1 and 2

Main Engine Position

In order to accommodate a new layout of the machinery, which should use the space more effectively, the main engine is positioned a bit further aft. Making use of the volume in the start of the skeg and decreasing the needed axel length.

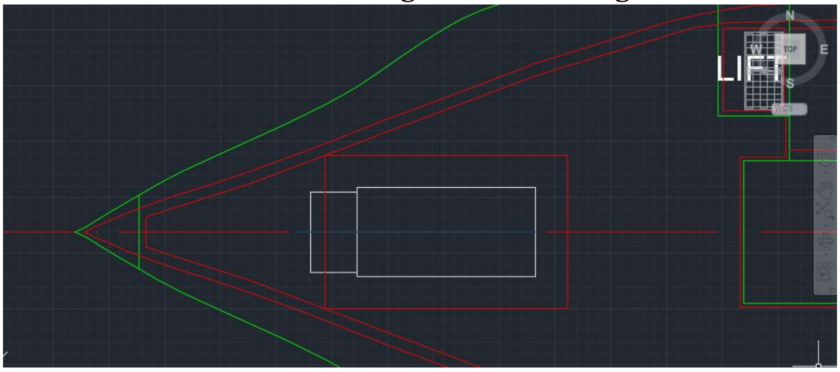


Figure 142: 301 Main Engine Position

Updating Lift Dimensions

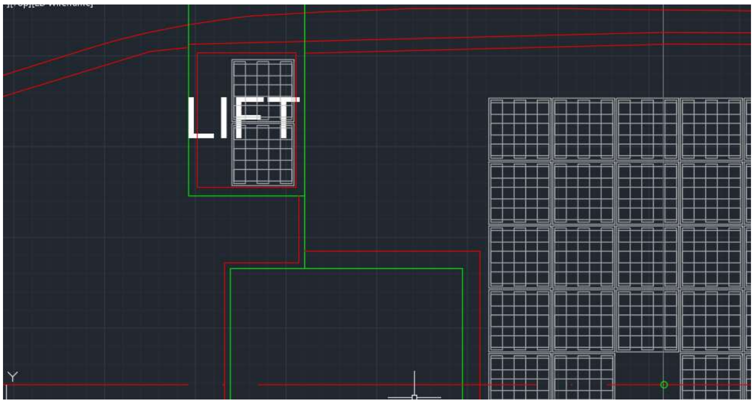


Figure 143: 301 Lift

The extern dimension of the lift shaft has been estimated at a conservative 3m times 3m in the design. Now that we are introducing better and better estimations, we revaluate the dimensions for the lift. We evaluate the dimensions to be too conservative, so we went down a bit on the dimensions, yet still based on the data from *Part I*. Thereby, a lift for two pallets can be 3m times 2m, this is still a bit conservative but less over the top than the last one.

Furthermore, is the truck-manoeuvring area optimized. As the ship has gotten a slimmer beam, the space right outside the elevator has become narrower as well. In addition is the volume between the moonpool and the area in question, ineffective at the moment. Hence, the truck-manoeuvring area is expanded transversely until it is wall-to-wall with the moonpool. In addition, 3 metres between the elevator and the end of the pallets should be sufficient, so we could move the lift a bit forward.

Resulting Layout of Machinery

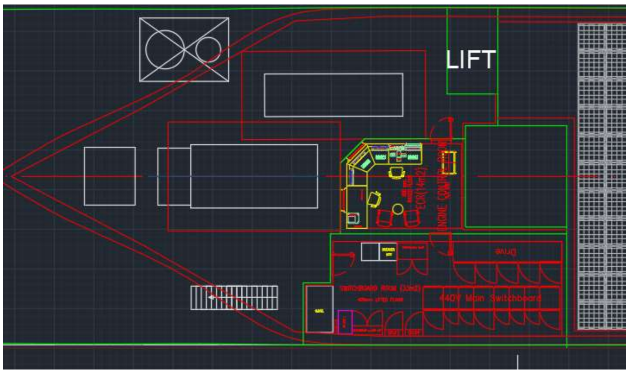


Figure 144: 301 Machinery Layout

The changes above might not seem that significant at first, but when the whole layout of the machinery comes to getter it becomes more apparent. The control room fits aft of the moonpool, with a door straight into the switchboard room at starboard and excellent view of the machines. The Auxiliary generator goes to port parallel with the main engine, yet a little more forward. Then all the components are fitted within the lines 3500 mm abl. Except the exhaust stack, which is fitted close to the "roof".

Refrigeration Room

The front bulkhead of the storage is retracted one metre aft-wards, increasing the size of the cooling room. Where the compressors for all the ship's cooling gears is located. The added metre is to accommodate a ladder from the above deck, which allows for access to the compressors from the factory since the storage becomes closed off when filled. This is sort of a "half" excuse since the ladder have been included in the first size estimate. Although that estimate where purely based on eye-measurement from our trip aboard the Atlantic, in addition is the storage capacity of the vessel comfortably more than the requirement. Due to these aspects have we decided to take a conservative approach to the room size.

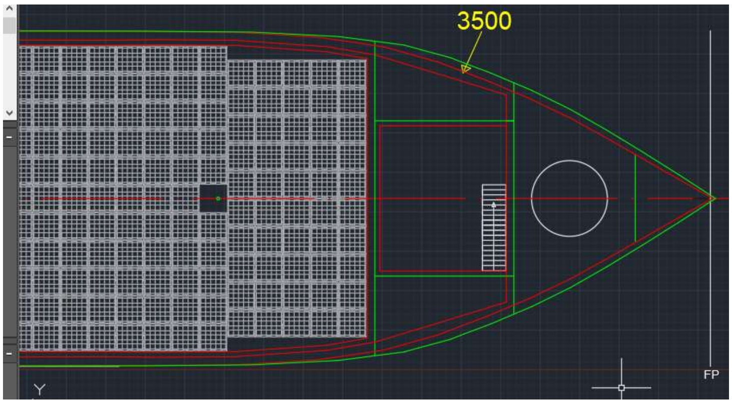


Figure 145: 301 Room for Refrigeration System

Pallet Count

The first and third layer of the pallets are shown in the GA. The second layer is quite similar to the third layer, nevertheless it counts four pallets less.

Output		
Pallets nr.	564	
VCG	4.31256	m
LCG	39.7121	m
Weight Fish	581.151	t
Weight Pallets	14.1	t
Total Weight	595.251	t

Figure 146: 301 Result of Pallets

Deck 3

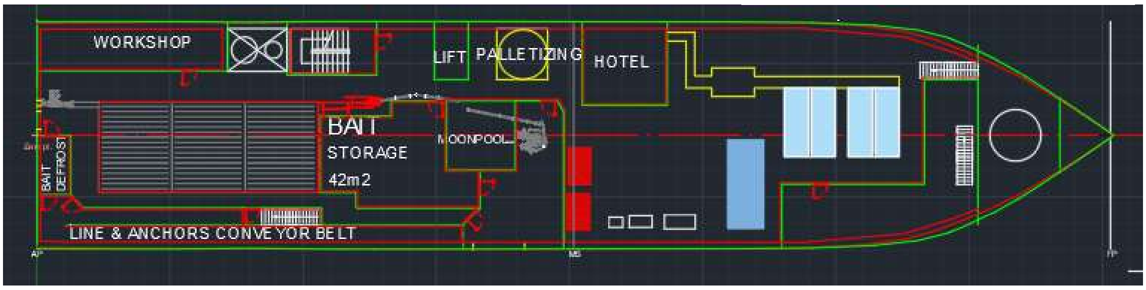
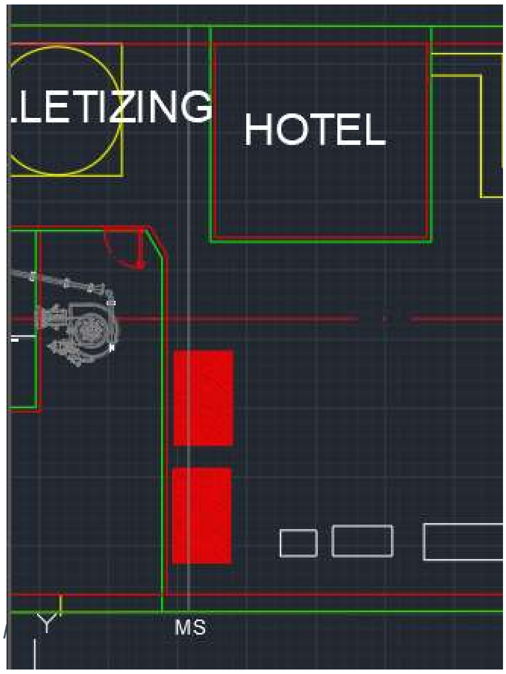


Figure 147: 301 Deck 3

The Factory

The wall between the moonpool and the factory has been slightly altered. It now runs straight into the portside of the moonpool, cutting of the "open" room from the rest of the deck. Letting there be a comfortable and "dry" pathway from the factory, along the palletizing to the lift, bait and Mag-Packs. There is a slight "cut" in the corner to increases the width between the wall and the pallet-hotel. The fish will be cut at the portside of the moonpool and then transported up and over the door, then inside to the bleeding tanks.

At the front of deck 3 there is a large room for the different switchboards and electronics for the factory equipment. Inside this room there is also a stairwell down to the cooling compressors.



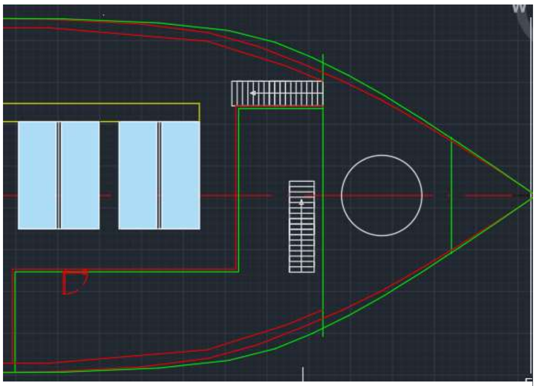


Figure 149: 301 Front of Factory

Bait Storage and Anchors

In order to correspond with the machinery, the stairwell to it needed to be moved forward. This allows us to increase the line and anchor storage space, although it also means the bait storage has to be retracted to make way for the corridor. Then to maintain the same area of bait storage it is extended forward along the moonpool at starboards.

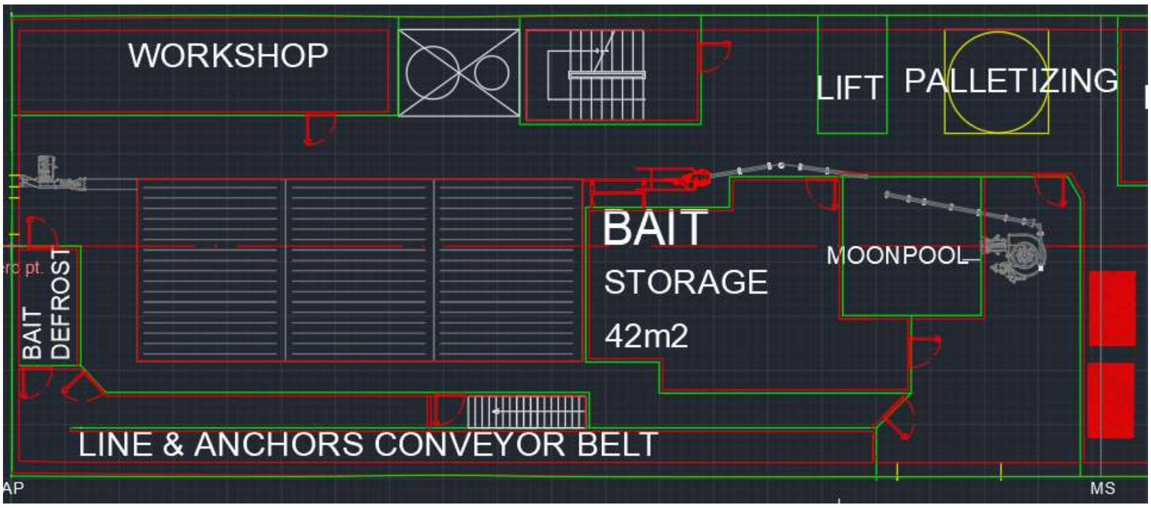


Figure 150: 301 Deck 3 Aft

Deck 4

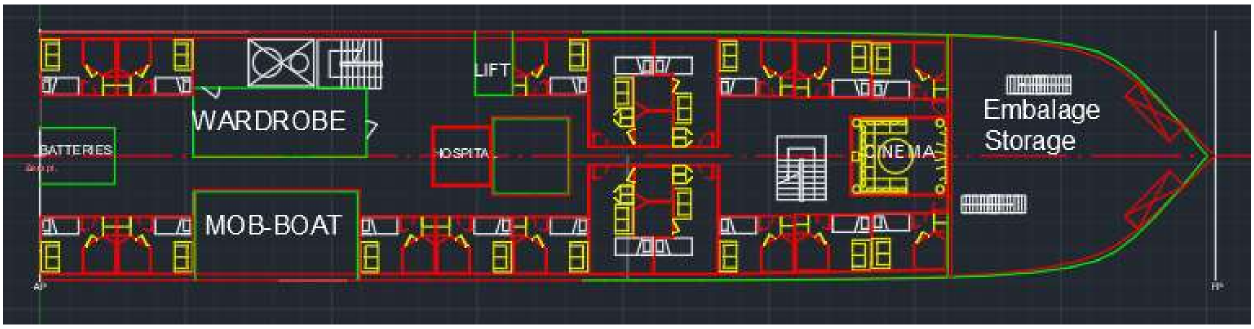


Figure 151: 301 GA Deck 4

There have not been many changes on deck 4, yet there have been some updates on the GA. From the lower decks we have to continue the new placement of the exhaust and aft stairwell. This might mean we would want to extend the wardrobe further forward and “reconnect it with the elevator again. Although it is not strictly necessary, but it helps separate the accommodation from the factory. The position of the stairwell and exhaust also depends on the beam structure, so this should be accounted for in the final design (Appendix).

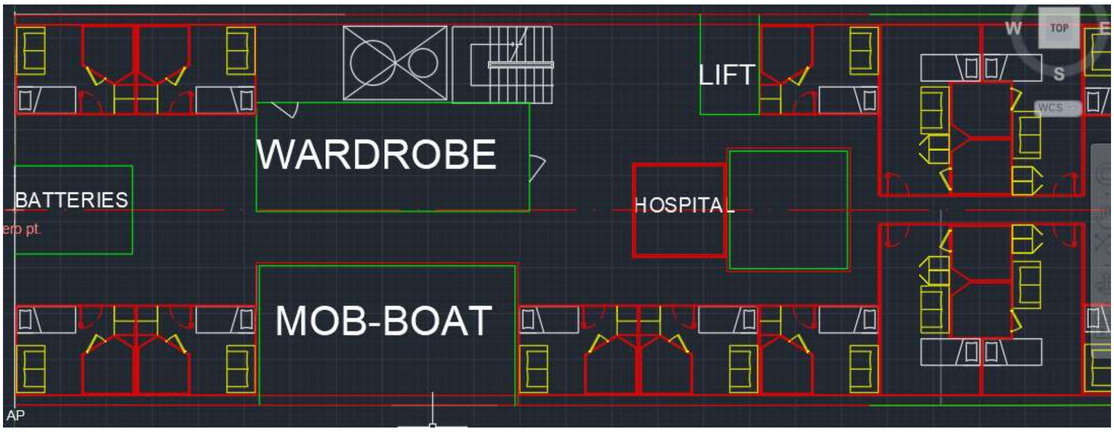


Figure 152: 301 Aft of Accommodation

In the storage for the wrapping, two stairwells have been added to connect with the factory below and the anchor/mooring space above. Due to the placement of the anti-rolling tank, this is the only path to the mooring area. Although we might fit an outside stairwell from deck 6 and down to the mooring, this still leaves the stairwell in the wrapping storage as the only indoor path.

In addition, it is drawn in the two boxes where the anchors would hang.

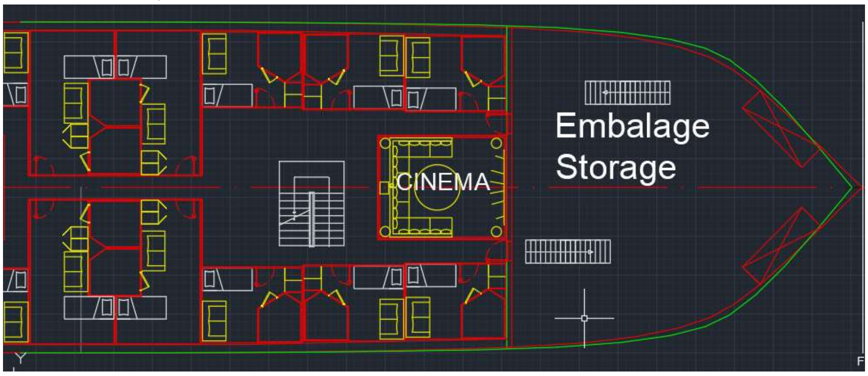


Figure 153: 301 Front of Accommodation

Deck 5

The dry food storage has been adapted slightly to accommodate the stairwell better. Due to the strength calculations for the deck above, we are discussing to expand the dining area. Making better space for the dining in the process, so it should be a win-win. The only thing that might be considered a disadvantage is added wall length, both resulting in higher weight and larger wind area. Although the added wall length would be minimal and should not result in anything drastic.

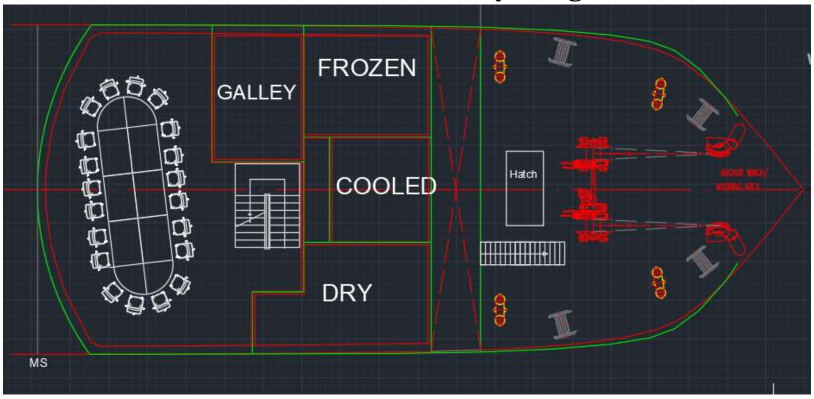


Figure 154: 301 GA Deck 5

Deck 6

In design 201 and its variations we evaluated three different types of layout for this deck and we landed on the third. Which pretty munch stayed the same, with an added "balcony" in the front. Just a little outside space, which gives a good view over the mooring area and front crane.

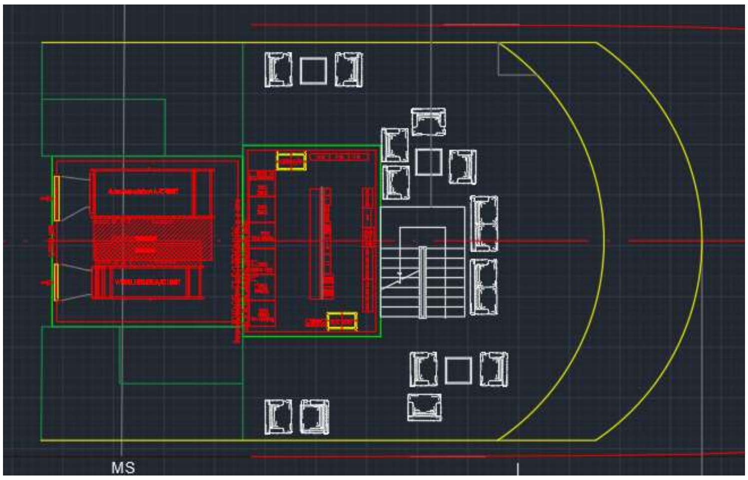


Figure 155: 301 GA Deck 6

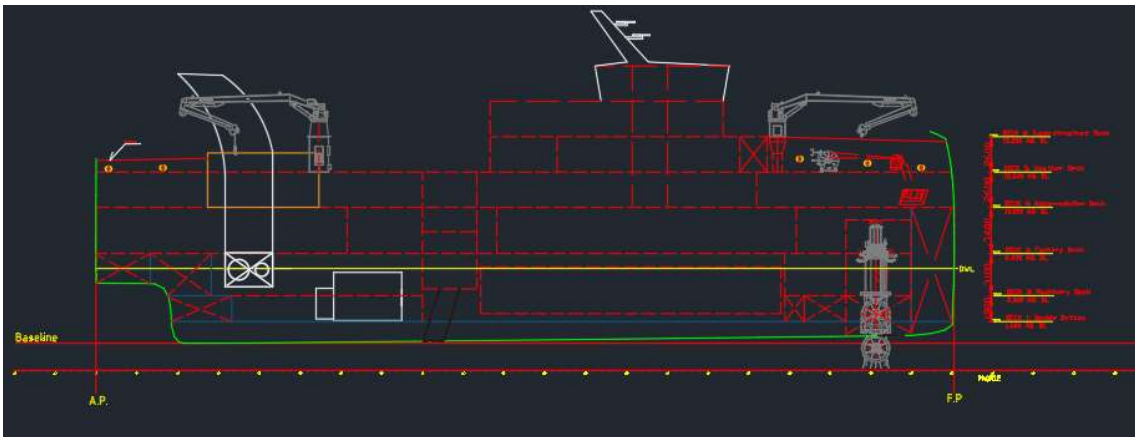


Figure 156: 301 GA Profile

Stability GZ-curve

While going over the 301 design and performing the appropriate calculations and tests it all seemed to go well. That was until we were going to check the GZ-curve for the Arrival 20% load condition. From DNV we gathered that it is required to have a GZ value of at least 0.2 metres at 30 degrees heel, and the area under the graph should be at least 0.055 metre-radians. And no less than 0.09 metre-radians under the graph at 40 degrees heel (DNV GL, 2015, p. 8). Maxsurf stability shows these directly when performing the Large Angel Stability calculations. Although the area under the graph is given in metre-degrees, so we perform some quick calculations to change the unit:

$$0.055 \times \frac{360}{2\pi} = 3.1513$$

Hence, at 30 degrees heel there needs to be at least 3.1513 metre-degrees under the GZ-curve.

$$0.09 \times \frac{360}{2\pi} = 5.1566$$

Hence, at 40 degrees heel there needs to be at least 5.1566 metre-degrees under the GZ-curve.

Now, for the mentioned condition, Arrival 20%, these values were not met. The initial GM was sufficient, but the GZ value at 30 degrees was not and neither where the area under the graph. Then we started “trouble shooting” this condition, which is the one we have had most problems with. So, these results were not unthinkable, yet we preceded to double check that all inputs were correct and that we had optimized the stability of the given condition. The condition has a natural forward trim since the storage is only filled in the front, and therefore is the aft fuel and fresh water tanks prioritized. When all the inputs have been checked, the results are refreshed and still unacceptable. Then we start to play around with the ballast tanks, filling them to increase the draft, and thereby waterline area, in addition to lowering the centre of gravity. Although the existing tanks is not large enough to give us enough stability.

New Ballast

In order to further improve this condition without changing the lines we add more ballast tanks, in spaces that are not necessarily used for other things. First of a large ballast tank aft of the machinery, all the way down to the heeling bunk tanks. This becomes a large tank and actually solves the GZ problems, resulting in both enough GZ and area under at 30 and 40 degrees.

Although, this “solution” does not come without some problems. The resulting draft at AP comes to almost 6 metres, leaving only 0.6 metres up to the working deck. Now, if we go back to the Lovdata for fishing vessel we remember that the working deck must be 1 over the waterline. Fortunately, the draught at FP is only about 4 metres so there is a lot of aft trim, this can be counteracted with filling the front peak, and perhaps increase it a bit. Adding another metre to the front peak brings the AP draught down to 5.6 metres, and we remember that we added one metre to the cooling compressors so perhaps we can arrive at a compromise here. Nevertheless, before we can do that, we see that the area under the GZ-curve

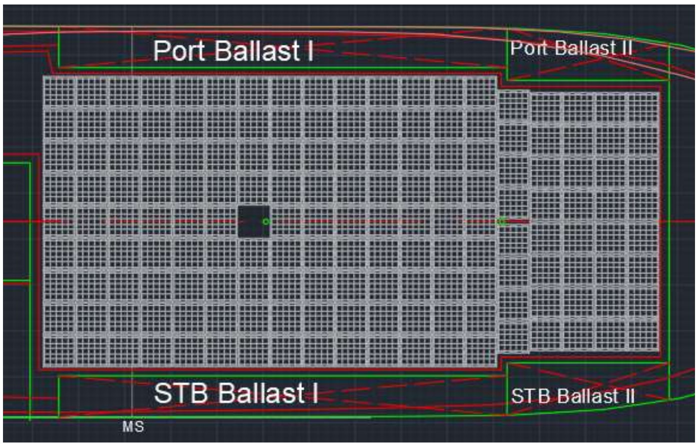


Figure 158: Ballast Tanks in Storage

75	Port ballast	98%	21,793	21,357
76	STB ballast	98%	21,793	21,357
77	Tank074	98%	11,319	11,093
78	Tank075	98%	49,195	48,211
79	STB ballast F	98%	7,187	7,043
80	Port ballast F	98%	7,187	7,043

Figure 157: Weight in added Ballast Tanks

at 30 degrees is below the necessary. This "battle between trim/draught and stability continues until we have added quite a lot of ballast.

We in order to get enough weight without inducing too much trim, we formed tanks at the sides of the bottom layer of the pallets. Due to the bilge there was not enough width to fit another row of pallets at each side, nevertheless it was still enough to form sufficient tanks.

At the end we find that we have to use most off the water ballast tanks in addition to increasing the weight of the keel to at least 212 tonnes. All the ballast tanks are filled with 98%, except the aft heeling tanks. In order to not exceed 5.6 metres draft at AP. The 12 tonnes increase in the keel was a necessary addition to give us enough area under the GZ-curve. In the sense that increasing the water ballast tanks more would reduce the volume that are used for other systems in the vessel.

The result of all this is a GZ of 0.242 metres and 3.16 metre degrees under the graph at 30-degree heel. Then 6.48 metre degrees at 40-degree heel. The displacement at this condition sums up to 2762 tonnes.

	Heel to Starboard deg	0,0	10,0	20,0	30,0	40,0	50,0	60,0
1	GZ m	0,000	0,072	0,129	0,242	0,430	0,672	1,084
2	Area under GZ curve f	0,0000	0,3764	1,3671	3,1569	6,4776	11,9113	20,4893
3	Displacement t	2762	2762	2762	2762	2762	2762	2762
4	Draft at FP m	4,843	4,869	4,978	5,125	5,229	5,109	4,472
5	Draft at AP m	5,566	5,513	5,282	4,893	4,294	3,303	1,498

Figure 160: New GZ Results

This gave us acceptable results, but for the design we are presenting we choose to use 225 tonnes in the keel. Since this have been a reoccurring problem throughout the design, and to present a preliminary design that is not too risky. The results of this will come under *Tanks and Conditions*.

1	Draft Amidships m	5,204
2	Displacement t	2762
3	Heel deg	0,0
4	Draft at FP m	4,843
5	Draft at AP m	5,566
6	Draft at LCF m	5,236
7	Trim (+ve by stern) m	0,723
8	WL Length m	63,160
9	Beam max extents on	13,315
10	Wetted Area m^2	1130,10
11	Waterpl. Area m^2	735,533
12	Prismatic coeff. (Cp)	0,691
13	Block coeff. (Cb)	0,585
14	Max Sect. area coeff. (	0,921
15	Waterpl. area coeff. (C	0,875
16	LCB from zero pt. (+ve	32,380
17	LCF from zero pt. (+ve	28,764
18	KB m	3,051
19	KG fluid m	6,353
20	BMt m	3,779
21	BML m	77,275
22	GMt corrected m	0,476
23	GML m	73,972
24	KMt m	6,830
25	KML m	80,320
26	Immersion (TPc) tonne/	7,539
27	MTc tonne.m	32,416
28	RM at 1deg = GMt.Disp.	22,946
29	Max deck inclination de	0,6576
30	Trim angle (+ve by ster	0,6576

Figure 159: Resulting Hydrostatics

Another Solution?

While we tried solving this problem, we explored other options and studied the graph and the waterlines at different degree heel. And we can see that the initial GM gives a sufficient incline on the graph, however tit flattens out quite a bit at 10-degree heel.

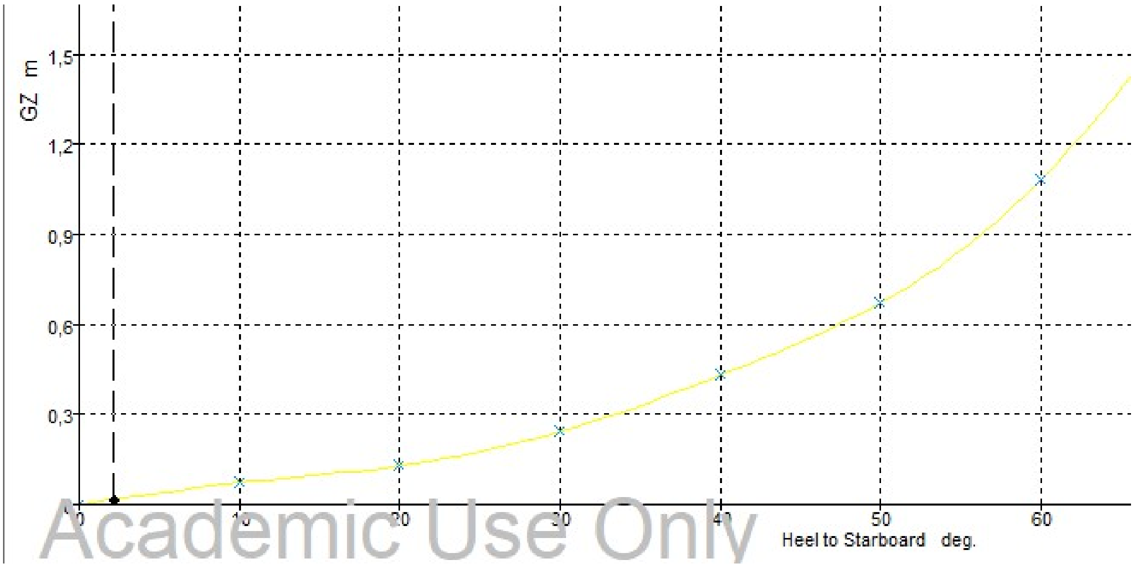


Figure 161: GZ curve of design 301

So, looking at the waterline at 10-degree heel we see that the waterline area at the stern decreased noticeably from no heel. Which can be explained by our transom design, that still has a quite sharp edge and is highly raised (low draft) at the sides.

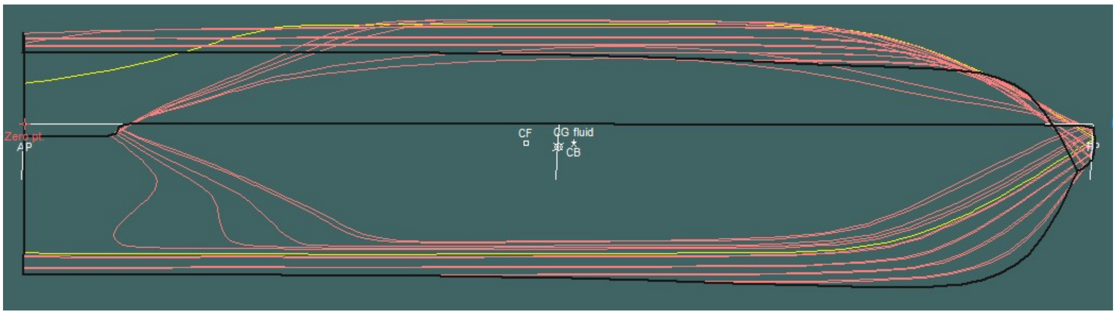


Figure 162: Water Lines at 10-degree heel

Then, as another solution we intend to change the transom a bit, to a more rounded and "normal". Which gave us the required stability result without increasing the weight of the keel, although there still needed to be a bit of ballast water in the tanks.

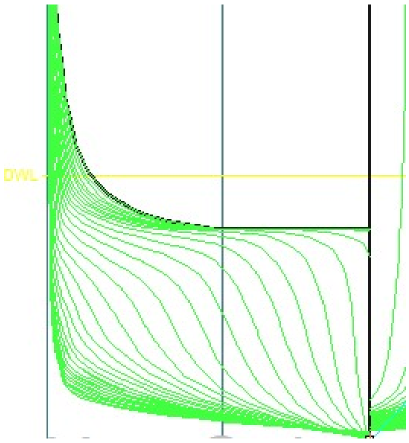


Figure 165: Curved Stern "Test"

1	Draft Amidships m	5,036
2	Displacement t	2564
3	Heel deg	0,0
4	Draft at FP m	4,869
5	Draft at AP m	5,202
6	Draft at LCF m	5,048
7	Trim (+ve by stern) m	0,333
8	WL Length m	63,001
9	Beam max extents on	13,313
10	Wetted Area m^2	1088,75
11	Waterpl. Area m^2	727,583
12	Prismatic coeff. (Cp)	0,677
13	Block coeff. (Cb)	0,578
14	Max Sect. area coeff. (	0,941
15	Waterpl. area coeff. (C	0,867
16	LCB from zero pt. (+ve	33,122
17	LCF from zero pt. (+ve	29,226
18	KB m	3,012
19	KG fluid m	6,469
20	BMt m	3,921
21	BML m	81,435
22	GMt corrected m	0,464

Figure 164: Hydrostatics from Curved Stern

	Heel to Starboard deg	0,0	10,0	20,0	30,0	40,0	50,0	60,0
1	GZ m	0,000	0,075	0,143	0,240	0,382	0,584	0,971
2	Area under GZ curve f	0,0000	0,3830	1,4644	3,3429	6,4161	11,1631	18,6899

Figure 166: Area under GZ for Curved Stern

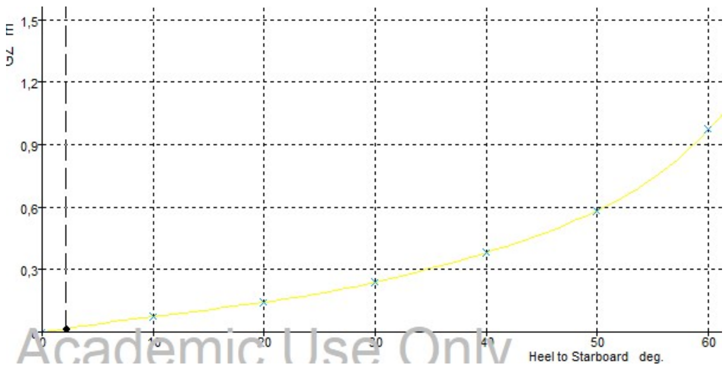


Figure 167: GZ-curve for Curved Stern

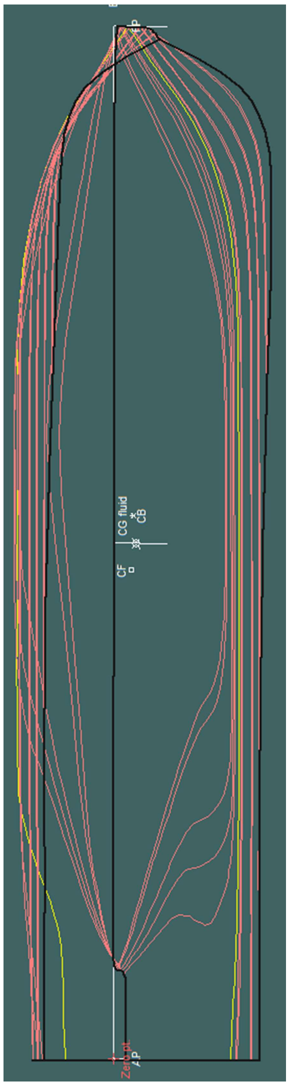


Figure 163: WL at 10-degree, Curved Stern

As can be seen from the results, the stability is good with only 2564 tonnes displacement. Which is almost 200 tonnes less than the other design, and that is mostly "unessacary" ballast water. Including the twelv extra tonnes in the keel, which is added displacement to all conditions and the entire operation profile. Twelv tonnes might not sound like much, neverhteless it is still added weight that are being dragged around all the time only because it is needed at some times. And thinking about the long run and the added fuel cost, might be unacceptable to the owner.

At this moment we do not have the time to continue with this new design, nevertheless it is what we would continue with if we had the time. This changes where made quite quick in order to test out the effect, when implemented in the design it would need to be done with higher attention to details. Then checked again with the other stability conditions, the GA, resistance and compleating another "circulation" of the design spiral.

Tanks and Conditions

The tank arrangement has already been shown briefly in *Part II*, and as stated there it developed in detail as we needed higher degree of accuracy in the stability assessment. Hence, here we will present the arrangement with the data from the 301 design and the conditions thereof.

Figure 169: Tank Arrangement DB

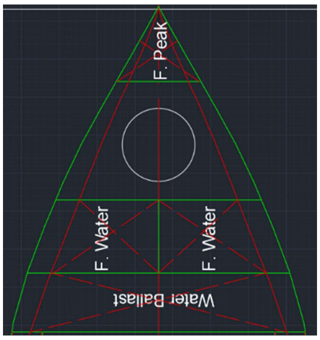


Figure 168:Tank Arrangement Bow over DB

F. Water	DB16	98 %	8,139	7,977
	DB16 Port	98 %	4,762	4,667
	DB16 STB	98 %	4,762	4,667
	DB17	98 %	10,355	10,148
	DB17 Port	98 %	4,442	4,353
	DB17 STB	98 %	4,442	4,353
	Ekstra fwr	98 %	41,448	40,619
	Ekstra fwr	98 %	41,448	40,619
SUM				117,403

Table 12: F. Water Capacity

Urea	DB0	98 %	8,051	7,89
	DB1	98 %	12,179	11,935
	DB1 Port	98 %	1,284	1,258
	DB1 STB	98 %	1,284	1,258
	DB2	98 %	12,625	12,373
	DB2 Port	98 %	2,959	2,9
	DB2 STB	98 %	2,959	2,9
Sum				40,514

Table 11: Urea Capacity

Fluid	Tank	Filling	Unit Volume	Volume Filled
Fuel Oil	DB3	98 %	12,811	12,555
	DB3 Port	98 %	4,575	4,483
	DB3 STB	98 %	4,575	4,483
	DB4	98 %	12,848	12,591
	DB4 Port	98 %	6,172	6,048
	DB4 STB	98 %	6,172	6,048
	DB5 Port	98 %	7,683	7,529
	DB5 STB	98 %	7,683	7,529
	DB6 Port	98 %	8,897	8,719
	DB6 STB	98 %	8,897	8,719
	DB5	98 %	10,699	10,485
	DB7 Port	98 %	9,72	9,525
	DB7 STB	98 %	9,72	9,525
	DB8	98 %	12,648	12,395
	DB8 Port	98 %	10,243	10,038
	DB8 STB	98 %	10,243	10,038
	DB9	98 %	12,538	12,288
	DB9 Port	98 %	10,556	10,345
	DB9 STB	98 %	10,556	10,345
	DB10	98 %	12,419	12,171
	DB10 Port	98 %	10,71	10,496
	DB10 STB	98 %	10,71	10,496
	DB11	98 %	12,258	12,013
	DB11 Port	98 %	10,654	10,441
	DB11 STB	98 %	10,654	10,441
	DB12	98 %	7,068	6,927
	DB12 Port	98 %	6,141	6,018
	DB12 STB	98 %	6,141	6,018
	DB13	98 %	11,927	11,689
	DB13 Port	98 %	10,165	9,962
	DB13 STB	98 %	10,165	9,962
	DB14	98 %	11,657	11,424
	DB14 Port	98 %	9,489	9,299
	DB14 STB	98 %	9,489	9,299
	DB15	98 %	14,096	13,815
	DB15 Port	98 %	10,122	9,92
	DB15 STB	98 %	10,122	9,92
Sum				353,999

Table 13: Fuel Oil Capacity

Fluid	Tank	Filling	Unit Volume	Volume Fille
Ballast	DB18	98 %	9,648	9,455
	DB18 Port	98 %	2,421	2,372
	DB18 STB	98 %	2,421	2,372
	Peak	98 %	52,166	51,12268
	Anti Roll DU	98 %	38,39	37,6222
	Anti Roll Du	98 %	38,39	37,6222
	Aft Bulk	98 %	29,193	28,60914
	Aft BUlk2	98 %	29,193	28,60914
	Anti Roll	98 %	52,356	51,30888
	ballast Aft	98 %	124,017	121,53666
	Port ballast	98 %	21,262	20,83676
	STB ballast	98 %	21,262	20,83676
	Tank074	98 %	11,043	10,82214
	Tank075	98 %	47,995	47,0351
	STB ballast F	98 %	7,011	6,87078
	Port ballast F	98 %	7,011	6,87078
Sum				483,90222

Table 14: Ballast Capacity

To compare the actual tank capacity to the required, there needed to be 350 m^3 fuel oil and our vessel has a total capacity of 354 m^3. The fresh water capacity exceeds the 100 m^3 requirement with over 17 m^3, and the urea capacity is just over the requirement 40 m^3. When it comes to the ballast capacity was there no direct requirements, nevertheless as mentioned in *New Ballast* we needed it to satisfy the stability requirements. Hence, it became quite a large volume, 484 m^3.

Departure

1	Draft Amidships m	5,289
2	Displacement t	2826
3	Heel deg	0,0
4	Draft at FP m	4,922
5	Draft at AP m	5,655
6	Draft at LCF m	5,321
7	Trim (+ve by stern) m	0,733
8	WL Length m	63,163
9	Beam max extents on	13,316
10	Wetted Area m^2	1140,84
11	Waterpl. Area m^2	736,039
12	Prismatic coeff. (Cp)	0,695
13	Block coeff. (Cb)	0,589
14	Max Sect. area coeff. (	0,922
15	Waterpl. area coeff. (C	0,875
16	LCB from zero pt. (+ve	32,287
17	LCF from zero pt. (+ve	28,764
18	KB m	3,101
19	KG fluid m	6,044
20	BMt m	3,699
21	BML m	75,636
22	GMt corrected m	0,757
23	GML m	72,693
24	KMt m	6,800
25	KML m	78,732
26	Immersion (TPc) tonne/	7,544
27	MTc tonne.m	32,596
28	RM at 1deg = GMt.Disp.	37,314
29	Max deck inclination de	0,6662
30	Trim angle (+ve by ster	0,6662

Figure 171: Departure Hydrostatics

	Heel to Starboard deg	0,0	10,0	20,0	30,0	40,0	50,0	60,0	70,0	80,0	90,0	100,0
1	GZ m	0,000	0,125	0,236	0,399	0,636	0,927	1,380	2,004	2,455	2,674	2,747
2	Area under GZ curve f	0,0000	0,6421	2,4382	5,5478	10,6806	18,4190	29,7666	46,6829	69,1904	95,0059	122,177
3	Displacement t	2826	2826	2826	2826	2826	2826	2826	2826	2826	2826	2826
4	Draft at FP m	4,922	4,943	5,042	5,185	5,292	5,180	4,555	2,491	-4,161	n/a	-22,984
5	Draft at AP m	5,655	5,614	5,402	5,026	4,440	3,468	1,692	-1,670	-11,221	n/a	-25,808
6	WL Length m	63,163	63,165	63,163	63,163	63,164	63,164	63,147	63,053	60,605	61,012	61,624
7	Beam max extents on	13,316	13,520	14,157	15,073	15,390	15,733	17,103	18,944	18,010	17,465	17,283
8	Wetted Area m^2	1140,84	1126,23	1110,35	1113,77	1125,21	1143,25	1185,84	1204,01	1188,12	1180,95	1177,77
9	Waterpl. Area m^2	736,039	724,358	721,195	747,319	779,385	823,180	902,583	902,398	850,433	816,157	804,736
10	Prismatic coeff. (Cp)	0,695	0,696	0,706	0,720	0,729	0,728	0,718	0,698	0,712	0,703	0,695
11	Block coeff. (Cb)	0,589	0,592	0,504	0,436	0,410	0,404	0,395	0,402	0,519	0,639	0,467
12	LCB from zero pt. (+ve	32,287	32,291	32,304	32,326	32,354	32,379	32,404	32,406	32,401	32,388	32,363
13	LCF from zero pt. (+ve	28,764	29,492	30,499	31,016	31,177	31,208	31,411	32,968	33,199	33,243	33,154
14	Max deck inclination de	0,6662	10,0179	20,0023	30,0002	40,0037	50,0073	60,0085	70,0053	80,0019	90,0000	99,9997
15	Trim angle (+ve by ster	0,6662	0,6105	0,3272	-0,1452	-0,7740	-1,5556	-2,6017	-3,7777	-6,3920	-90,000	-2,5656

Figure 170: Departure GZ

For the departure fuel, fresh water and urea are filled to the maximum and we filled up the anti-roll tank. We see that the GM is 0.735 metre even though the anti-roll tank is filled up 80 percent, so stability is good. Although, such a high GM may result in a bit uncomfortable ride. And we see the same tendencies in the GZ.

Departure from Fishing Ground

1	Draft Amidships m	5,571
2	Displacement t	3010
3	Heel deg	0,0
4	Draft at FP m	5,651
5	Draft at AP m	5,491
6	Draft at LCF m	5,564
7	Trim (+ve by stern) m	-0,160
8	WL Length m	63,170
9	Beam max extents on	13,317
10	Wetted Area m^2	1184,13
11	Waterpl. Area m^2	737,448
12	Prismatic coeff. (Cp)	0,698
13	Block coeff. (Cb)	0,633
14	Max Sect. area coeff. (	0,945
15	Waterpl. area coeff. (C	0,877
16	LCB from zero pt. (+ve	33,079
17	LCF from zero pt. (+ve	28,811
18	KB m	3,240
19	KG fluid m	6,139
20	BMt m	3,479
21	BML m	71,371
22	GMt corrected m	0,580
23	GML m	68,472
24	KMt m	6,719
25	KML m	74,611
26	Immersion (TPc) tonne/	7,559
27	MTc tonne.m	32,704
28	RM at 1deg = GMt.Disp.	30,452
29	Max deck inclination de	0,1455
30	Trim angle (+ve by ster	-0,1455

Figure 172: Dep. From F. Ground Hydro.

	Heel to Starboard deg	0,0	10,0	20,0	30,0	40,0	50,0	60,0	70,0	80,0	90,0	100,0
1	GZ m	0,000	0,091	0,172	0,314	0,553	0,870	1,357	1,969	2,383	2,582	2,642
2	Area under GZ curve f	0,0000	0,4672	1,7689	4,1224	8,3914	15,4089	26,3730	43,0414	65,0105	89,9852	116,172
3	Displacement t	3010	3010	3010	3010	3010	3010	3010	3010	3010	3010	3010
4	Draft at FP m	5,651	5,679	5,788	5,946	6,091	6,059	5,548	3,809	-1,446	n/a	-20,042
5	Draft at AP m	5,491	5,438	5,213	4,830	4,243	3,263	1,482	-1,899	-11,712	n/a	-26,403
6	WL Length m	63,170	63,176	63,178	63,182	63,188	63,193	63,187	63,118	61,533	61,294	61,921
7	Beam max extents on	13,317	13,521	14,162	15,160	15,905	16,403	17,877	18,893	18,064	17,559	17,426
8	Wetted Area m^2	1184,13	1159,14	1144,13	1147,68	1161,63	1182,24	1225,64	1235,43	1222,40	1213,83	1210,28
9	Waterpl. Area m^2	737,448	722,321	721,808	752,224	793,787	843,325	921,086	905,102	863,277	828,277	817,749
10	Prismatic coeff. (Cp)	0,698	0,699	0,702	0,707	0,711	0,709	0,700	0,684	0,688	0,681	0,672
11	Block coeff. (Cb)	0,633	0,596	0,510	0,442	0,405	0,396	0,385	0,409	0,512	0,615	0,458
12	LCB from zero pt. (+ve	33,079	33,084	33,097	33,117	33,141	33,165	33,184	33,186	33,188	33,179	33,161
13	LCF from zero pt. (+ve	28,811	29,681	30,669	31,204	31,472	31,589	31,896	33,143	33,600	33,622	33,556
14	Max deck inclination de	0,1455	10,0023	20,0058	30,0117	40,0172	50,0195	60,0172	70,0100	80,0040	90,0000	99,9984
15	Trim angle (+ve by ster	-0,1455	-0,2190	-0,5228	-1,0147	-1,6805	-2,5406	-3,6918	-5,1751	-9,2524	-90,000	-5,7648

Figure 173: Dep. From F. Ground GZ

When departing from the fishing ground with 100 percent catch there is a bit of a forward trim. One would want to avoid this by adding some ballast, however we wanted to show this condition as light as possible. since the displacement is naturally high due to the storage and 30 percent fuel, fresh water and urea. Furthermore, is the trim still less than the raked keel and we evaluated from the waterlines that it still should work. It should also be noted that we kept 60% in the anti-roll dump tanks as a precaution.

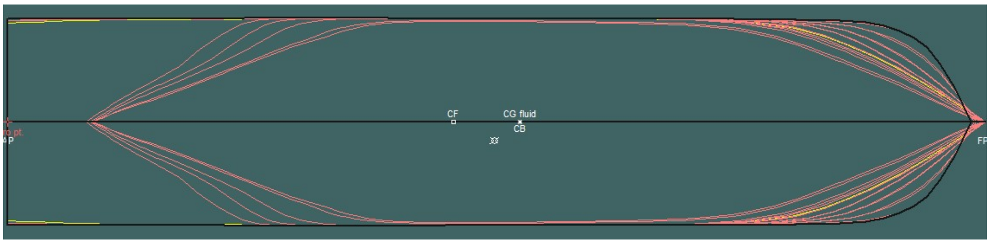


Figure 174: Dep. From F. Ground WL

Arrival 100%

1	Draft Amidships m	5,356
2	Displacement t	2857
3	Heel deg	0,0
4	Draft at FP m	5,289
5	Draft at AP m	5,423
6	Draft at LCF m	5,362
7	Trim (+ve by stern) m	0,133
8	WL Length m	63,160
9	Beam max extents on	13,316
10	Wetted Area m^2	1155,91
11	Waterpl. Area m^2	736,209
12	Prismatic coeff. (Cp)	0,694
13	Block coeff. (Cb)	0,613
14	Max Sect. area coeff. (	0,937
15	Waterpl. area coeff. (C	0,875
16	LCB from zero pt. (+ve	32,958
17	LCF from zero pt. (+ve	28,804
18	KB m	3,121
19	KG fluid m	6,278
20	BMt m	3,652
21	BML m	74,884
22	GMt corrected m	0,495
23	GML m	71,727
24	KMt m	6,773
25	KML m	78,005
26	Immersion (TPc) tonne/	7,546
27	MTc tonne.m	32,519
28	RM at 1deg = GMt.Disp.	24,697
29	Max deck inclination de	0,1213
30	Trim angle (+ve by ster	0,1213

Figure 176: Arrival 100% Hydrostatics

	Heel to Starboard deg	0,0	10,0	20,0	30,0	40,0	50,0	60,0	70,0	80,0	90,0	100,0
1	GZ m	0,000	0,071	0,130	0,251	0,460	0,732	1,178	1,801	2,240	2,458	2,536
2	Area under GZ curve f	0,0000	0,3697	1,3602	3,1944	6,6952	12,5708	21,9257	36,8225	57,2430	80,8865	105,919
3	Displacement t	2857	2857	2857	2857	2857	2857	2857	2857	2857	2857	2857
4	Draft at FP m	5,289	5,324	5,442	5,601	5,731	5,662	5,097	3,219	-2,626	n/a	-21,305
5	Draft at AP m	5,423	5,357	5,111	4,713	4,108	3,106	1,283	-2,263	-12,507	n/a	-27,224
6	WL Length m	63,160	63,170	63,171	63,175	63,181	63,185	63,176	63,090	61,098	61,185	61,792
7	Beam max extents on	13,316	13,519	14,156	15,084	15,587	16,009	17,435	18,920	18,037	17,512	17,356
8	Wetted Area m^2	1155,91	1128,28	1114,62	1119,40	1131,42	1150,47	1194,16	1209,46	1194,57	1185,98	1183,96
9	Waterpl. Area m^2	736,209	716,503	716,228	745,833	781,142	826,762	908,641	903,839	855,162	819,739	810,131
10	Prismatic coeff. (Cp)	0,694	0,696	0,702	0,708	0,713	0,710	0,700	0,682	0,689	0,680	0,672
11	Block coeff. (Cb)	0,613	0,590	0,502	0,435	0,404	0,396	0,386	0,400	0,509	0,614	0,453
12	LCB from zero pt. (+ve	32,958	32,963	32,979	33,002	33,028	33,054	33,075	33,078	33,078	33,070	33,049
13	LCF from zero pt. (+ve	28,804	29,810	30,755	31,225	31,441	31,513	31,738	33,078	33,463	33,459	33,420
14	Max deck inclination de	0,1213	10,0000	20,0019	30,0074	40,0133	50,0163	60,0151	70,0092	80,0037	90,0000	99,9987
15	Trim angle (+ve by ster	0,1213	0,0300	-0,3001	-0,8071	-1,4746	-2,3227	-3,4633	-4,9721	-8,9111	-90,000	-5,3661

Figure 175: Arrival 100% GZ

In the arrival conditions there is only 10 percent of fuel, fresh water and urea remaining in the tanks. Since the storage is forward in the vessel, the little that remains in the tanks is placed as far aft as possible. This actually results in a stern trim, which we chose to keep since it gave better stability. Due to larger beam at the stern than the bow, which have come up quite a bit throughout the report.

Arrival 20%

1	Draft Amidships m	5,222
2	Displacement t	2775
3	Heel deg	0,0
4	Draft at FP m	4,866
5	Draft at AP m	5,578
6	Draft at LCF m	5,253
7	Trim (+ve by stern) m	0,712
8	WL Length m	63,160
9	Beam max extents on	13,315
10	Wetted Area m^2	1132,35
11	Waterpl. Area m^2	735,638
12	Prismatic coeff. (Cp)	0,692
13	Block coeff. (Cb)	0,586
14	Max Sect. area coeff. (	0,921
15	Waterpl. area coeff. (C	0,875
16	LCB from zero pt. (+ve	32,377
17	LCF from zero pt. (+ve	28,765
18	KB m	3,061
19	KG fluid m	6,324
20	BMt m	3,763
21	BML m	76,938
22	GMt corrected m	0,499
23	GML m	73,675
24	KMt m	6,823
25	KML m	79,994
26	Immersion (TPc) tonne/	7,540
27	MTc tonne.m	32,438
28	RM at 1deg = GMt.Disp.	24,183
29	Max deck inclination de	0,6473
30	Trim angle (+ve by ster	0,6473

Figure 177: Arrival 20% Hydrostatics

	Heel to Starboard deg	0,0	10,0	20,0	30,0	40,0	50,0	60,0	70,0	80,0	90,0	100,0
1	GZ m	0,000	0,077	0,139	0,257	0,450	0,698	1,116	1,729	2,180	2,404	2,488
2	Area under GZ curve f	0,0000	0,4002	1,4643	3,3751	6,8664	12,5277	21,3923	35,5915	55,3409	78,4198	102,941
3	Displacement t	2775	2775	2775	2775	2775	2775	2775	2775	2775	2774	2774
4	Draft at FP m	4,866	4,892	4,999	5,146	5,251	5,133	4,500	2,432	-4,273	n/a	-23,098
5	Draft at AP m	5,578	5,527	5,299	4,911	4,314	3,326	1,525	-1,943	-11,810	n/a	-26,436
6	WL Length m	63,160	63,163	63,162	63,162	63,165	63,164	63,146	63,053	60,564	61,000	61,615
7	Beam max extents on	13,315	13,519	14,154	15,038	15,299	15,638	17,006	18,947	18,006	17,457	17,267
8	Wetted Area m^2	1132,35	1115,07	1100,05	1104,26	1114,97	1132,57	1175,47	1195,74	1179,35	1172,24	1169,04
9	Waterpl. Area m^2	735,638	720,445	717,760	744,611	774,688	817,390	898,829	902,567	848,748	814,506	802,938
10	Prismatic coeff. (Cp)	0,692	0,694	0,704	0,718	0,726	0,724	0,714	0,694	0,707	0,698	0,690
11	Block coeff. (Cb)	0,586	0,591	0,501	0,434	0,409	0,402	0,393	0,398	0,515	0,634	0,462
12	LCB from zero pt. (+ve	32,377	32,383	32,399	32,424	32,453	32,480	32,503	32,507	32,502	32,488	32,464
13	LCF from zero pt. (+ve	28,765	29,605	30,587	31,065	31,212	31,242	31,417	32,971	33,217	33,254	33,159
14	Max deck inclination de	0,6473	10,0160	20,0016	30,0005	40,0044	50,0082	60,0092	70,0059	80,0022	90,0000	99,9996
15	Trim angle (+ve by ster	0,6473	0,5776	0,2731	-0,2137	-0,8515	-1,6424	-2,7025	-3,9713	-6,8199	90,0000	-3,0323

Figure 178: Arrival 20% GZ

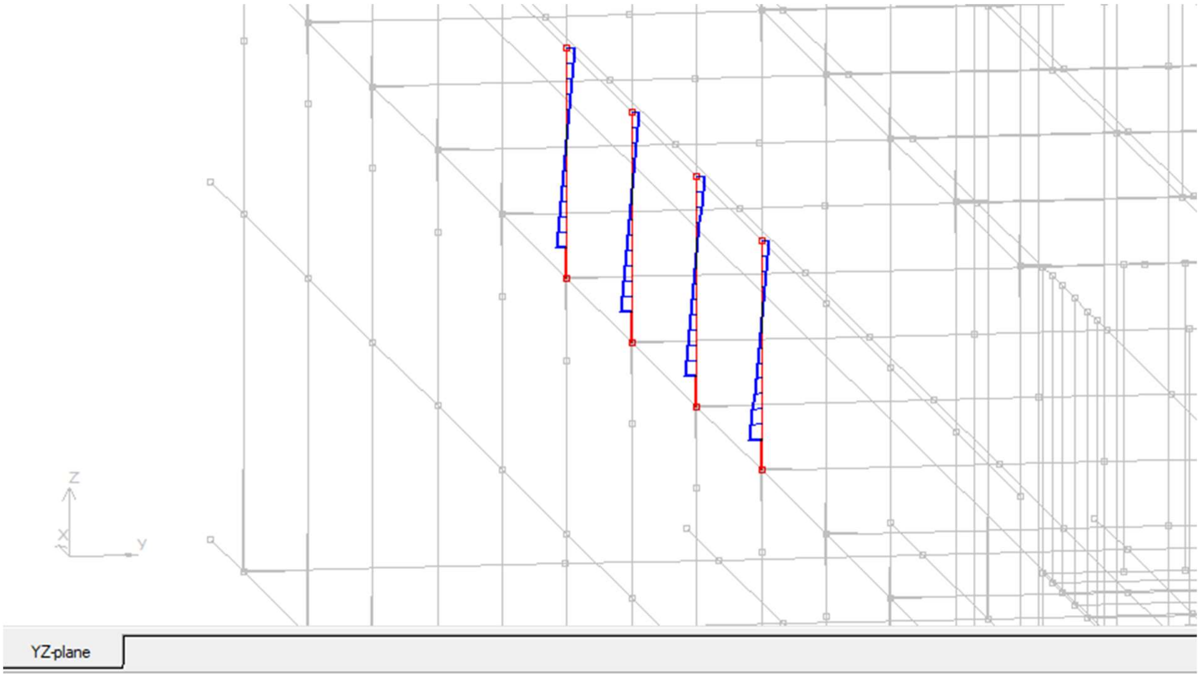
Arriving with only 20 percent of the storage filled means we have to use quite a lot of ballast, as discussed in *New Ballast*. In total, with 98 percent in the anti-roll dump, there is about 370 tonnes of ballast water.

Strength and Class Drawing

These are the 3D-Beam result for our Girder system. This is an overview of the strength in the most critical elements of the structure, as well as the highest stresses in each part of the ship. The results from section scantling is presented in the appendix**.**

Hull Side Stresses

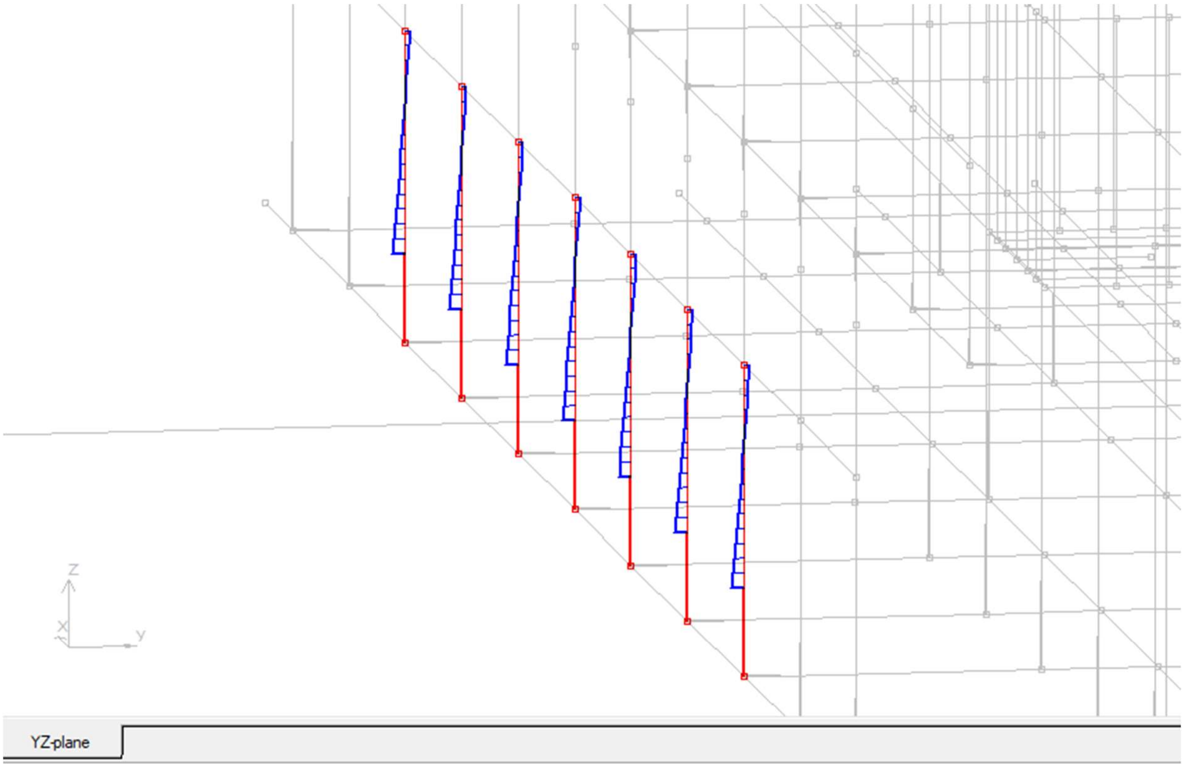
The hull side has been a constant problem throughout this process. The sea pressure applying great loads on the girders, and the moment being transferred through the height of the ship. The Girders ought to be as low as possible due to the storage space, makes the hull side girders the most critical.



	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/m	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	**|Min Sig-Ny|**	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-Nz| [N/mm2]
1	474	51	1	31	0	93	6	144	10	58	45
2	515	50	1	32	0	94	8	144	13	58	41
3	570	47	1	32	0	94	10	141	15	57	36
4	122	55	0	29	0	85	4	139	4	58	51

Figure 179

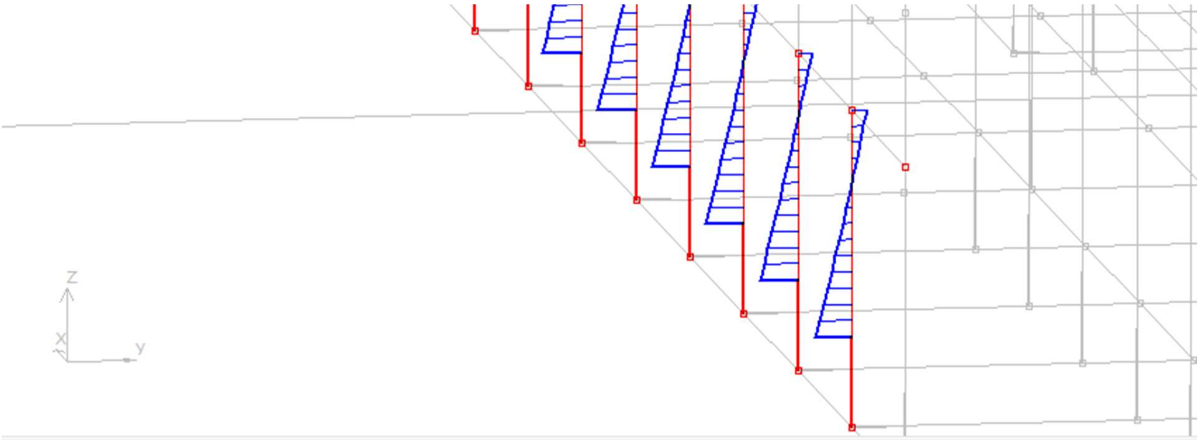
These are the highest stresses in the hull side are well within the requirements of 160MPa. The von mises stresses reaches 155MPa is reasonable.



	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/m	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	|Min Sig-Ny| [N	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-Nz| [N/mm2]
1	5	44	0	63	0	110	1	155	44	46	43
2	20	45	0	61	0	109	3	155	42	48	43
3	32	42	0	62	0	111	5	154	46	48	37
4	112	43	0	64	0	108	0	151	44	43	42
5	482	39	0	64	0	106	2	145	46	41	37
6	507	39	0	64	0	105	1	144	46	40	37
7	578	37	0	64	0	103	4	140	45	41	33

Figure 180

In the bottom we see the hull side is just within the 160MPa requirements, high shear forces give us von mises of 190MPa. This is critical, as the model receives unrealistic forced in deck that induces a moment that counteracts the moment induced by the sea pressure. If we remove the rule load, and apply the actual load the deck experiences, we will see that the stresses exceed the allowable bending stresses.

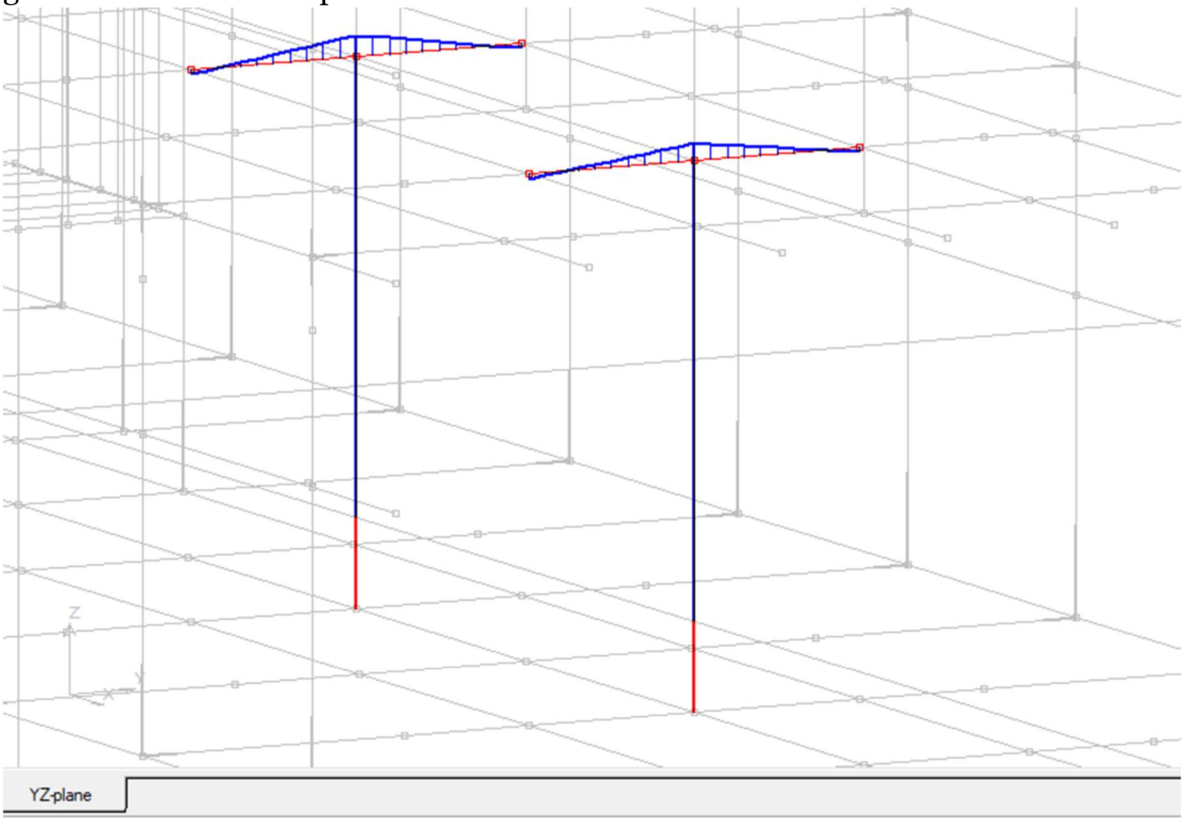


	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/m	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	**|Min Sig-Ny|**	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-Nz| [N/mm2]
1	20	53	0	72	0	138	4	191	58	56	49
2	32	49	0	73	0	139	7	188	62	56	42
3	5	51	0	72	0	136	2	187	57	54	49
4	112	51	0	72	0	131	2	182	54	53	49
5	507	49	0	72	0	129	4	179	54	53	46
6	482	49	0	72	0	129	2	178	54	51	47
7	578	49	0	72	0	128	3	177	54	52	45
8	44	43	1	70	0	125	13	168	57	56	30
9	626	45	0	71	0	121	6	166	52	51	39

Figure 181 An example of a load-case without any load in the decks, resulting in excessive bending stresses.

Connection Stresses

Columns in the storage transfer the loads from the double bottom to the deck above. In storage we skip some columns, meaning the transverse members experiences a much greater load than their peers.



	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/m	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	**|Min Sig-Ny|**	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-Nz|
1	254	141	0	0	0	15	0	157	126	141	141
2	193	152	0	0	0	3	0	155	148	152	152
3	379	20	1	37	0	122	7	142	23	27	13
4	63	20	1	37	0	122	7	142	23	27	13
5	67	21	0	31	0	99	1	120	14	22	20
6	383	21	0	31	0	99	1	120	14	22	20

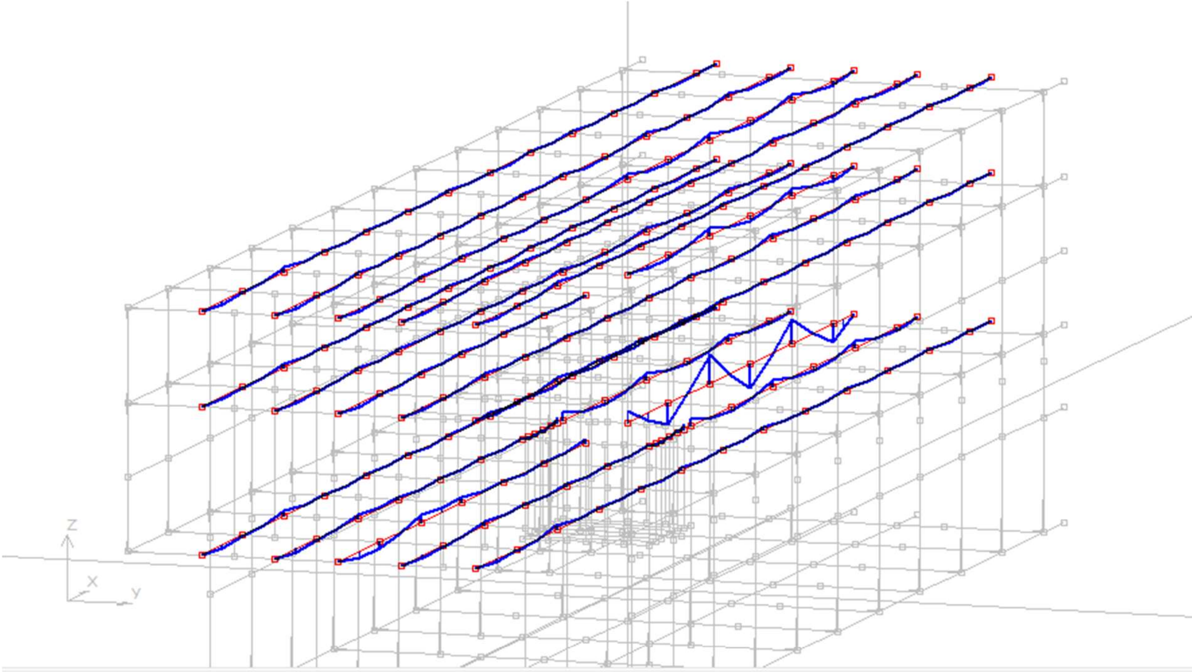
Figure 182

Withing the stress requirements, and von mises of 156MPa

Stress overviews:

Stress Overview of the Longitudinal

Not to exceed 130MPa



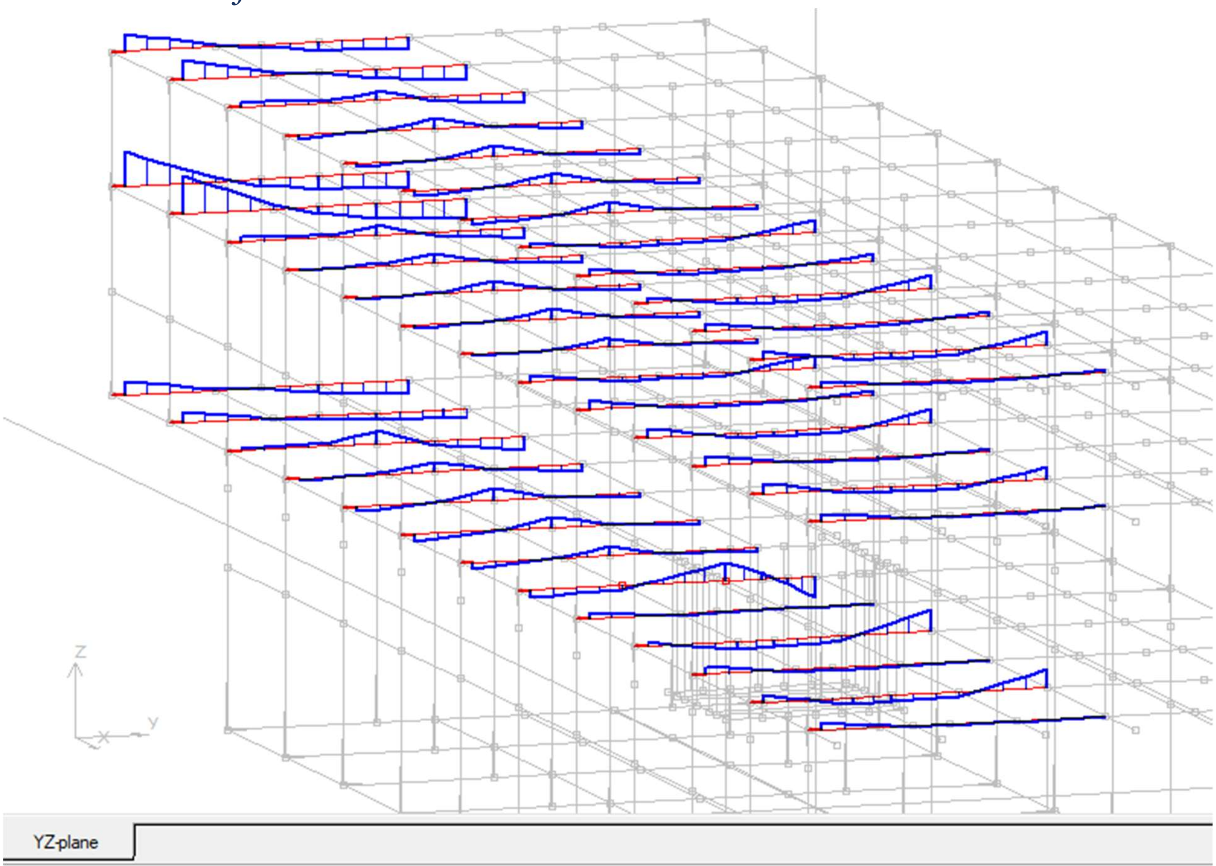
YZ-plane

	Beam	Sig-Nx [N/mm	Tau-Qy [N/m	Tau-Qz [N/m	Tau-Mx [N/m	Sig-My [N/m	Sig-Mz [N/m	**Min Sig-Ny**	Max Sig-Ny [	Min Sig-Nz [N	Max Sig-Nz [N/mm2]
1	64	2	0	39	0	124	0	127	106	2	2
2	62	5	0	36	0	122	0	127	87	5	5
3	74	2	0	19	0	124	0	122	79	2	2
4	72	1	0	17	0	122	0	121	54	1	1
5	376	2	0	18	0	113	5	111	71	3	8
6	362	2	0	18	0	113	5	111	71	3	7
7	76	2	0	18	0	111	0	110	77	2	2
8	78	1	0	17	0	111	0	109	67	1	1
9	68	2	0	32	0	106	0	109	75	2	2
10	66	2	0	36	0	107	0	108	106	2	2

Figure 183

Within by some margin. Von mises of 141MPa.

Stress Overview for Transverse Members

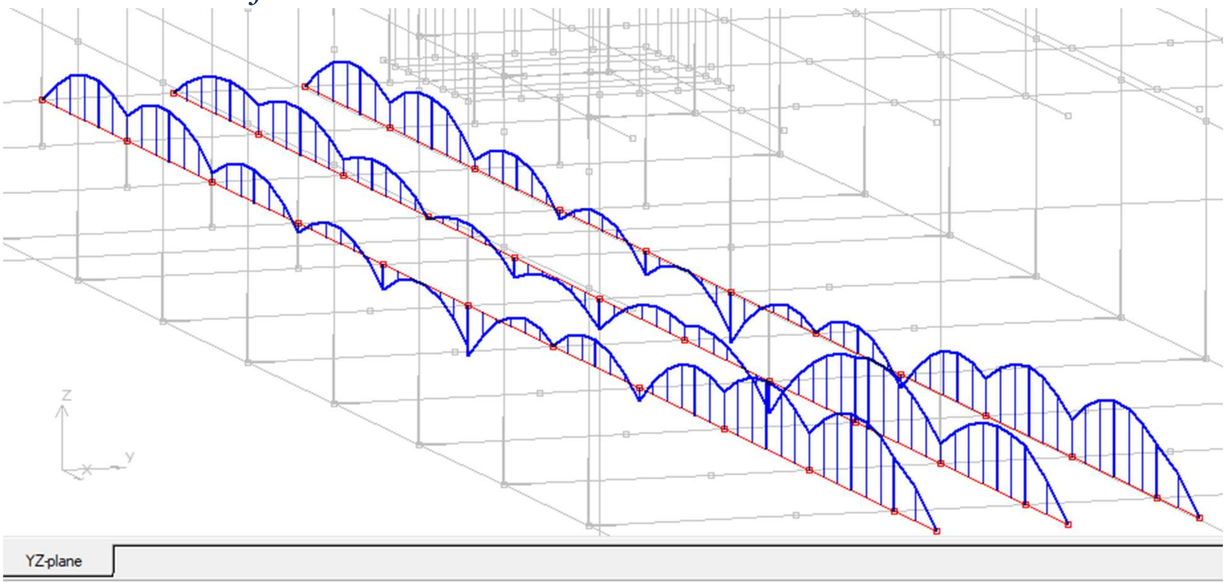


	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/m	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	**|Min Sig-Ny|**	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-N
1	331	5	0	28	0	151	1	156	48	6	4
2	283	2	0	28	0	153	1	156	48	3	1
3	706	4	0	34	1	149	1	153	58	5	3
4	287	2	0	27	0	140	1	142	55	3	1
5	379	20	1	37	0	122	7	142	23	27	13
6	335	5	0	26	0	135	1	140	53	6	4
7	327	5	1	24	1	134	7	139	32	13	2
8	279	1	0	23	0	136	1	136	34	2	0
9	740	0	1	30	0	131	7	131	54	8	7
10	713	4	1	19	0	116	11	120	29	15	7

Figure 184

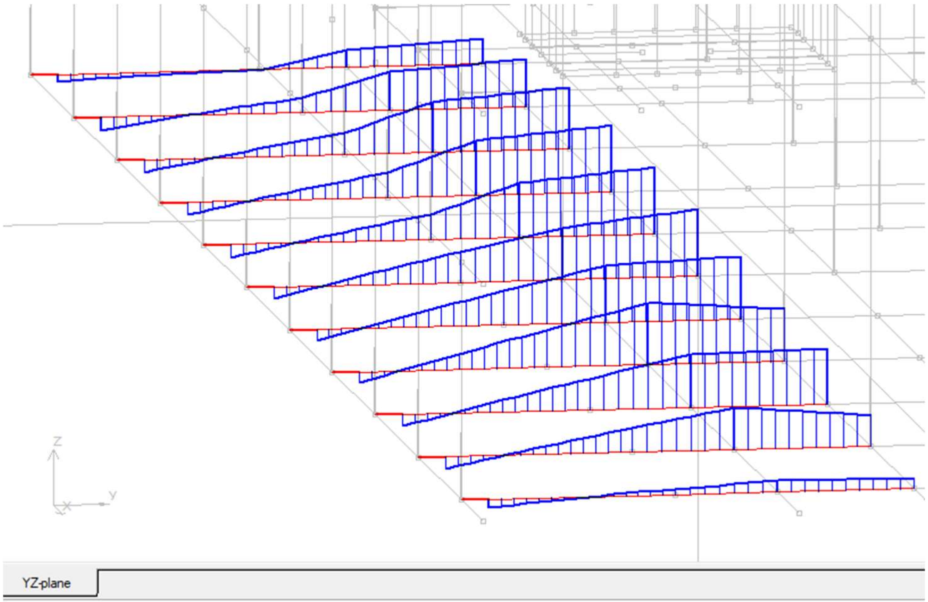
Everything looks good by some margin. Von Mises stresses up to 163MPa.

Stress Overview of Bottom Girders



	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	**|Tau-Qz| [N/**	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	|Min Sig-Ny| [N	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-Nz| [N/mm2]
1	104	2	0	32	0	26	0	28	24	2	2
2	138	2	0	31	0	18	0	20	16	2	2
3	144	2	0	31	0	18	0	20	16	2	2
4	140	2	0	30	0	18	0	20	16	2	2
5	134	2	0	30	0	18	0	20	16	2	2
6	137	2	0	28	0	19	2	21	17	4	0
7	143	2	0	28	0	19	2	21	17	4	0
8	110	2	0	27	0	23	0	25	21	2	2
9	850	0	1	26	0	17	3	17	17	3	3
10	849	0	1	26	0	17	3	17	17	3	3

Figure 185 Longitudinal

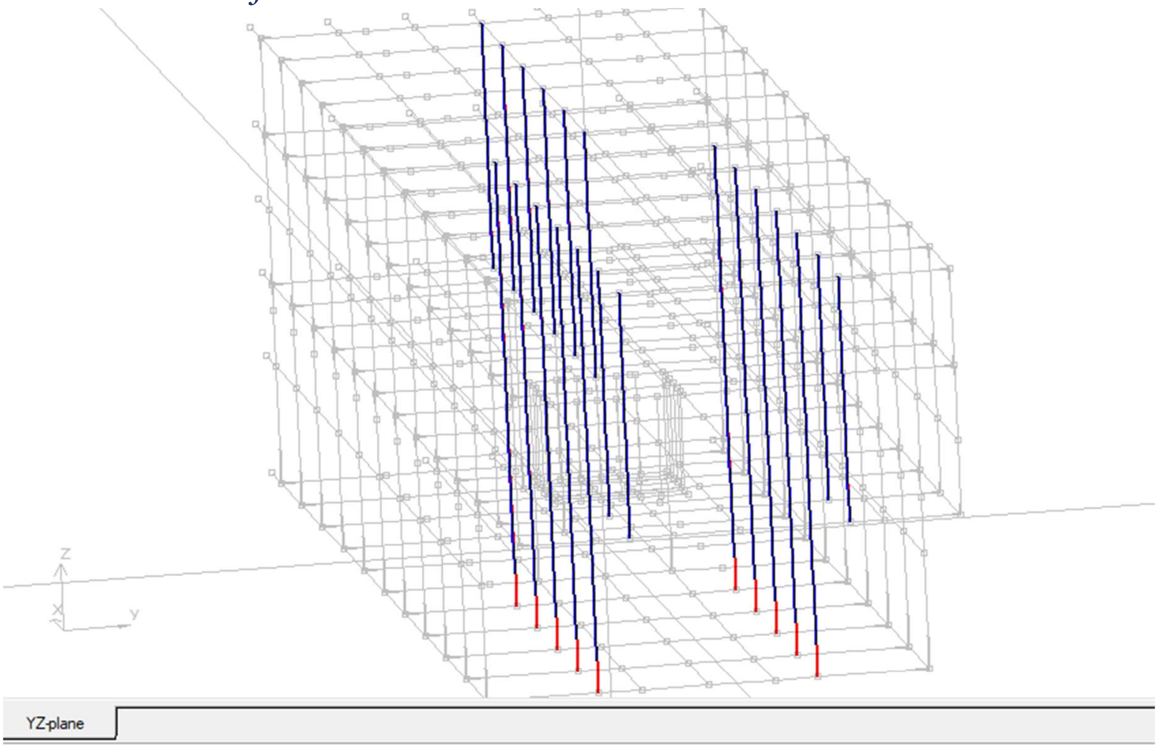


	Beam	|Sig-Nx| [N/mm	|Tau-Qy| [N/m	|Tau-Qz| [N/	|Tau-Mx| [N/m	|Sig-My| [N/m	|Sig-Mz| [N/m	|Min Sig-Ny| [N	|Max Sig-Ny| [	|Min Sig-Nz| [N	|Max Sig-Nz| [N
1	530	5	0	59	0	42	2	47	36	7	4
2	459	5	0	58	0	42	1	47	36	6	4
3	557	5	1	57	0	39	3	44	33	8	2
4	15	5	0	51	0	31	0	36	25	6	5
5	27	5	0	50	0	28	0	33	22	6	5
6	1	5	0	48	0	28	0	33	23	5	5
7	605	5	1	48	0	29	4	35	24	10	1
8	185	5	0	45	0	51	0	56	46	6	5
9	187	5	0	43	0	47	1	53	42	6	5
10	111	5	0	42	0	23	0	29	18	6	5

Figure 186 transverse

Although there is no problem with bending stresses, we want to bring the shear stresses for a further look at cut-outs.

Stress Overview of Columns



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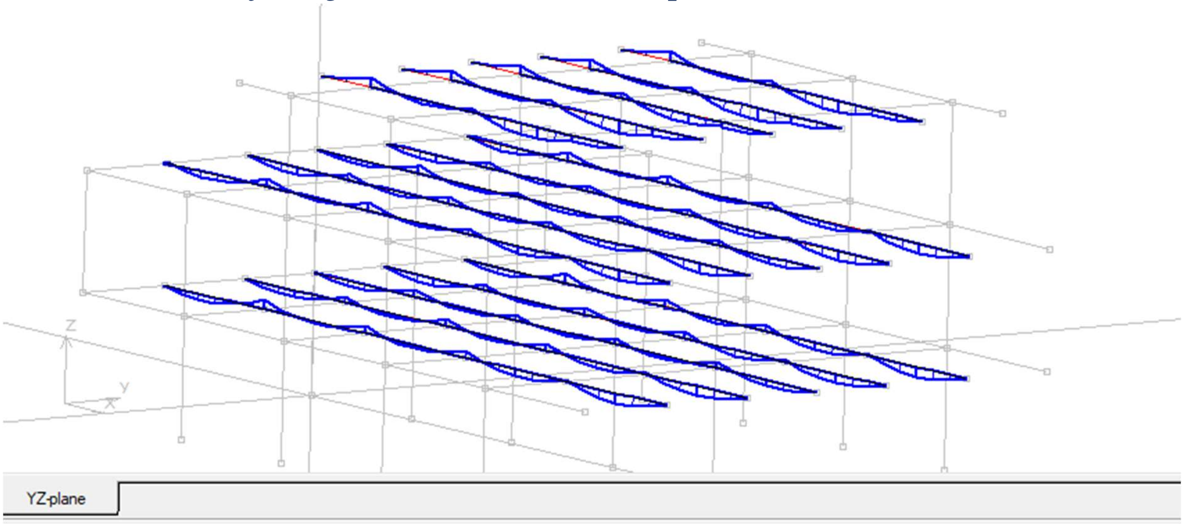
<tr><th></th><th>Beam</th><th>|Sig-Nx| [N/mm</th><th>|Tau-Qy| [N/m</th><th>|Tau-Qz| [N/m</th><th>|Tau-Mx| [N/m</th><th>|Sig-My| [N/m</th><th>|Sig-Mz| [N/m</th><th>|Min Sig-Ny|</th><th>|Max Sig-Ny| [</th><th>|Min Sig-Nz| [N</th><th>|Max Sig-Nz| [N/mm2]</th></tr>
<tr><td>1</td><td>202</td><td>136</td><td>0</td><td>0</td><td>0</td><td>8</td><td>0</td><td>144</td><td>128</td><td>136</td><td>136</td></tr>
<tr><td>2</td><td>196</td><td>137</td><td>0</td><td>0</td><td>0</td><td>6</td><td>0</td><td>143</td><td>131</td><td>137</td><td>137</td></tr>
<tr><td>3</td><td>198</td><td>133</td><td>0</td><td>0</td><td>0</td><td>2</td><td>0</td><td>135</td><td>131</td><td>133</td><td>133</td></tr>
<tr><td>4</td><td>204</td><td>133</td><td>0</td><td>0</td><td>0</td><td>1</td><td>0</td><td>134</td><td>132</td><td>133</td><td>133</td></tr>
<tr><td>5</td><td>200</td><td>86</td><td>0</td><td>1</td><td>0</td><td>42</td><td>0</td><td>128</td><td>44</td><td>86</td><td>86</td></tr>
<tr><td>6</td><td>194</td><td>87</td><td>0</td><td>1</td><td>0</td><td>38</td><td>0</td><td>126</td><td>49</td><td>87</td><td>87</td></tr>
<tr><td>7</td><td>685</td><td>94</td><td>0</td><td>0</td><td>0</td><td>1</td><td>11</td><td>96</td><td>93</td><td>106</td><td>83</td></tr>
<tr><td>8</td><td>682</td><td>94</td><td>0</td><td>0</td><td>0</td><td>1</td><td>11</td><td>96</td><td>93</td><td>106</td><td>83</td></tr>
<tr><td>9</td><td>589</td><td>86</td><td>0</td><td>0</td><td>0</td><td>0</td><td>13</td><td>86</td><td>85</td><td>99</td><td>73</td></tr>
<tr><td>10</td><td>586</td><td>86</td><td>0</td><td>0</td><td>0</td><td>0</td><td>13</td><td>86</td><td>85</td><td>99</td><td>73</td></tr>
</table>

Figure 187

Within by some margin.

However, these columns should also be checked for buckling.

Stress Overview of Longitudinal Members in Superstructure



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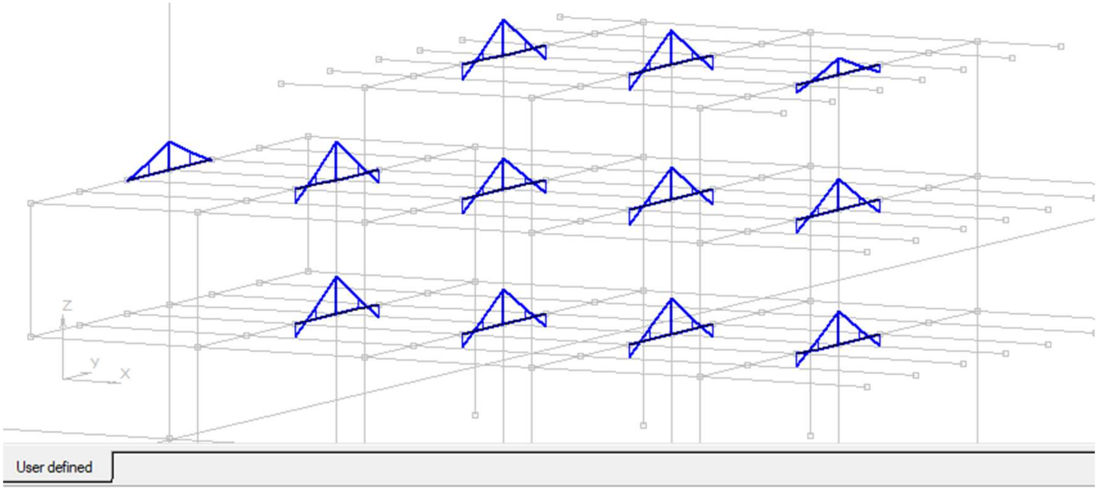
<tr><th></th><th>Beam</th><th>|Sig-Nx| [N/mm</th><th>|Tau-Qy| [N/m</th><th>|Tau-Qz| [N/m</th><th>|Tau-Mx| [N/m</th><th>|Sig-My| [N/m</th><th>|Sig-Mz| [N/m</th><th>|Min Sig-Ny| [N</th><th>|Max Sig-Ny|</th><th>|Min Sig-Nz| [N</th><th>|Max Sig-Nz| [N/mm2]</th></tr>
<tr><td>1</td><td>183</td><td>0</td><td>0</td><td>7</td><td>0</td><td>18</td><td>0</td><td>6</td><td>18</td><td>0</td><td>0</td></tr>
<tr><td>2</td><td>173</td><td>0</td><td>0</td><td>7</td><td>0</td><td>18</td><td>1</td><td>6</td><td>18</td><td>1</td><td>1</td></tr>
<tr><td>3</td><td>182</td><td>0</td><td>0</td><td>7</td><td>0</td><td>18</td><td>2</td><td>5</td><td>17</td><td>2</td><td>2</td></tr>
<tr><td>4</td><td>174</td><td>0</td><td>0</td><td>7</td><td>0</td><td>17</td><td>0</td><td>6</td><td>17</td><td>0</td><td>0</td></tr>
<tr><td>5</td><td>122</td><td>1</td><td>0</td><td>6</td><td>0</td><td>15</td><td>0</td><td>4</td><td>15</td><td>0</td><td>1</td></tr>
</table>

Figure 188

Stresses well within the requirements.

Stress Overview for Transverse Members in Superstructure



<table>
<tr><th></th><th>Beam</th><th>|Sig-Nx| [N/mm</th><th>|Tau-Qy| [N/m</th><th>|Tau-Qz| [N/m</th><th>|Tau-Mx| [N/m</th><th>|Sig-My| [N/m</th><th>|Sig-Mz| [N/m</th><th>|Min Sig-Ny|</th><th>|Max Sig-Ny| [</th><th>|Min Sig-Nz| [N</th><th>|Max Sig-Nz| [N/mm2]</th></tr>
<tr><td>1</td><td>196</td><td>5</td><td>0</td><td>16</td><td>0</td><td>62</td><td>1</td><td>67</td><td>27</td><td>7</td><td>4</td></tr>
<tr><td>2</td><td>192</td><td>5</td><td>0</td><td>16</td><td>0</td><td>61</td><td>1</td><td>67</td><td>27</td><td>6</td><td>5</td></tr>
<tr><td>3</td><td>31</td><td>1</td><td>0</td><td>16</td><td>0</td><td>65</td><td>0</td><td>66</td><td>33</td><td>1</td><td>1</td></tr>
<tr><td>4</td><td>53</td><td>1</td><td>0</td><td>16</td><td>0</td><td>64</td><td>0</td><td>65</td><td>32</td><td>2</td><td>1</td></tr>
<tr><td>5</td><td>180</td><td>5</td><td>0</td><td>15</td><td>0</td><td>60</td><td>1</td><td>65</td><td>26</td><td>6</td><td>4</td></tr>
</table>

Figure 189

Critical stresses within 74 MPa

Sudden Problems with 301

The longitudinal strength was based on the bending moments retrieved by the 101 design and since the values was below the rules with some margin, we were confident that we could retain the rule bending moment through the design process. However, the changes in the storage room, hull form, and the ballast has significantly altered the buoyancy and gravitational forces, possibly changing the longitudinal bending moments. We checked the bending moments for the new design, and it turned out it had increased by almost the double the initial values. The new values did not affect the longitudinal strength, but caused buckling problems in the top deck, which Nauticus hull suggested to counter with a 1mm plate thickness increase. At this point we had already increased the plate thickness by a millimetre due to buckling and we were hesitating to increase it even further due to the fragile stability. We therefore reviewed other measures.

From Euler's formula of plate buckling we know there is 3 main factors one can alter: The plate slimness, material factor and the buckling coefficient determined by stiffener span and spacing. Seemingly there is no way to change the buckling strength without altering the topology outside the plate thickness. However, there are ways to hack the buckling coefficient by adding bars to the plate, effectively reducing the stiffener span ultimately increasing the buckling coefficient. This is an appealing solution, as we do not have to increase the plate thickness over the whole ship, but rather save some weight and use a thinner plate.

Details for Class Drawings

As we stat to draw the class drawings, there has to be calculated some details due to other design factors. In the beams we need to establish if and where there can be cut-outs for pipes and cables. There are some other conditions that we have to check for buckling. In addition to calculate the strength of the connections between stiffeners and Beams.

Shear Strength for Cut-outs

Cut-outs are necessary for pipes and electrical cables can go hidden within the structure. This reduces the shear strength of the element with a cut-out web. Shear strength is most accurately calculated by the formula $\tau_{act} = \frac{QS}{It}$. However, its generally accepted to use $\tau_{act} = \frac{F}{A}$. This formula is not as accurate and calculate lower actual shear stresses than the accurate one. If the stresses build up to critical stresses, one might have to use the accurate formula.

We wanted to review the bottom girders; They are a single web with ship bottom and tank top as flanges, and since they are tanks, they need to be accessible for people for inspections. The webs need therefore openings big enough for inspectors to transverse the tanks, its generally accepted that opening of 600x400mm is grate enough. We chose to review a girder with the combination of a small web and high shear forces, and ended up reviewing a girder with a web of 1100mm and shear forces of 500kN. The transverse bottom girders have a near to uniform shear stress, so we assume this force in all sections of the girder. $\tau_{act} = \frac{50000}{(1100mm-600m\quad)*10mm} = 100\frac{N}{mm^2}$ this is not accepted by the DNV rules. We have to lower the stress to at least 90N/mm², but preferably more due to the formula being inaccurate. By increasing the web thickness to 12mm we receive a shear stress of 83N/mm². This about 10% within the rules, and we assume this is ok, because the load case is rear and unrealistic.

Similar calculations must be performed on other girders with cut-outs. A conservative approach could be comparing the stress of a girder without cut-outs and a girder with; the forces should be the same, but the stress and area altered.

$$\tau_{Whole} * A_{Whole} = F = \tau_{Cut} * A_{Cut}$$

$$\tau_{Whole} * h_w * t_w = \tau_{Cut} * h_{max} * t_w$$

$$\frac{\tau_{Whole} * h_w}{\tau_{Cut}} = h_{max}$$

In the top deck we have a case with experiencing a shear stress of 28N/mm² and a web height of 300mm. Saying we allow a shear stress of 85 N/mm gives us a required height of 100mm leaving 200mm to a possible cut-out. The cut-outs needed for stiffeners and pipes are well within the calculated requirement. However, the beam is receiving bending stresses up to 160 N/mm², combined with shear stresses up to 90 N/mm² means a von mises stress reaching the yield point of the material. On the other hand, this is a rear and unrealistic load case, so we assume its ok. A closer and more accurate look could be taken to reduce the web thickness.

Plate buckling as initiated by transverse loads

It is not only longitudinal loads that may cause buckling in plates. Bending moments in each girder is transferred to the deck and hull side plating risking the plates to buckle. Plates and beams are fairly similar through the ship; This makes for a uniform buckling strength in most decks, and a similar section modulus allowing us to mainly focus on the moment of each element.

Every plate with transverse loads has a buckling coefficient of 4, stiffener spacing of 660mm and the material factor is constant. Form this we can calculate a critical stress for each plate.

8mm=> 109MPa

7mm=> 83MPa

6mm=> 62MPa

The hull side with a plate thickness of 8mm has a section modulus 2200cm² in bottom and 1600cm² in top; the factory deck has a section modulus 1330cm² with a 6mm plate and 1500 cm² with thickness of 7mm; the accommodation and weather deck has a section modulus 1200cm² with a 6mm plate and 1370 cm² with thickness of 7mm.

From this we can find the critical moments that will cause buckling and the elements we need to fix.

It is not only the plating that are prone to buckling, but the girders as well. The girder's web is practically just a plate keeping the flanges distance. We assume the greatest problem lies in the storage, as a high web and great compression forces in forces in form of cargo and sea pressure results critical buckling case.

Estimation of the pressure; 0.06 N/mm2 from sea and 0.045 N/mm^2 form a cargo with a dynamic add-on. This gives us an actual buckling stress of 15 N/mm^2. The plates have buckling coefficient of almost 1, creating a buckling strength of 14N/mm^2. We can solve this problem by either increasing the web thickens, or adding the buckling rods to increase the buckling coefficient.

Stiffener and Girder Connection

Stiffeners are required to take up great loads, but those loads are transferred to beams. We must make sure that the material is able to hold the load in these points. An approach to this is finding the area connection the stiffener to the girder and the load it experiences, then compare it to the critical shear stress.

$$\tau_{act} = \frac{F}{A}$$

$$F = s * l * P$$

$$A = t_w * l_c$$

$$\tau_{act} = \frac{s * l * P}{t_w * l_c}$$

We wanted to review the stiffeners in the tween decks, since the stiffener height are low, resulting in a low possible value for the connection length.

$$\tau_{act} = \frac{660mm * 2325mm * 5 * 10^{-3}N}{10mm * 80mm} = 9.6\frac{N}{mm^2}$$

This value is well within the requirements for shear stresses. We also wanted to review the connections in the ship bottom, due to the high sea pressure.

$$\tau_{act} = \frac{660mm * 2325mm * 6 * 10^{-2}N}{12mm * 180mm} = 42\frac{N}{mm^2}$$

This value is also well within the requirements.

If we would get problems, we could apply a bracket on the other side of the stiffener, effectively double the connection length.

Evaluation

The strength assessment is far from complete but the time available has caused us to wind-up the assessment. This leaves us to evaluate what we have done, what we should have done and what further work we could have done.

We started sort of haphazardly trusting our own experience with longitudinal stiffened structure, assuming that the least time-consuming way to perform the assessment was to undergo the spiral as quickly and many times as possible. However, it's obvious to us now that this was a time-consuming process. A more appropriate approach would be to start by calculating the load accepted on a girder with a web of 300mm. Out initial value of 500mm was an uneducated estimation based on greater ships, however the height of the web is determined by the load applied which can be altered with the girder spacing. We were told that the minimum web height should be no less than 300mm, and with optimum beam theory we could use the calculated section modulus to find the acceptable load which can be used to find the girder spacing. From this it is easier to assume the most weight efficient stiffener spacing. In the end we had to change the spacing to 660 mm, since 4000 divided by 600 is far from a whole integer, this was a realisation that happened last minute so the frame spacing is still 600. In hindsight would we choose a relationship between stiffeners and girders that gave a whole integer.

Rather than going straight for the safe bet of longitudinal stiffeners, we should have taken the risk of transverse stiffeners earlier and use longitudinal as a last resort against buckling. Literature concerning topology are not consistent when stating the length of which to change to a longitudinal stiffened structure; length from 70 to 90 meters have been stated. We believe however that these length marks the grey area of which the counteract becomes too excessive, and the length varies for each design and/or engineer.

Now this is the beauty of hindsight, we can judge our own previous ignorance with our newfound experience. However, mistakes were made and now we must recess what the way forward is. We ate not pleased with the girder profiles we ended up. They are not weight efficient according to the optimum beam theory and restricted in height due to stability. The next move would be to reduce the girder spacing, to more effectively make use of the capacity of a beam, with 300mm web and flange according to an optimum beam. Then an assessment would be made of what stiffener spacing would be most weight efficient, in regard of stiffener yielding and longitudinal plate buckling.

Due to the tools we used, the rules available from the DNV-GL web site and the rules we are trained in, we ended up using a combination of DNV rules form 2012, 2015 and DNV-GL rules from 2020. Preferably, we would have taken more time in properly classifying the ship, and chosen a singular rule set, rather than a combination of three.

We thought the great load we applied would result in a conservative result and our dimensions could been quite smaller, however the load gave us a false sense of security. We discovered too late that we would get failure with a 'Lighter' load case. In a more thorough strength assessment, we would have checked more load cases. As well as reviewed the strength of the moonpool tube. Our current assumption is it requires rings of stiffeners, like on land water towers.

Model Test
A model test was done in order to test the resistance of the vessel. During the entire design process, it is crucial to understand how the ships reacts with the surrounding water, so it can be optimized. In the early stage of the design spiral and preliminary design are the hull created on several design assumption from experience of similar ships. More difference in design will lead to higher degree of uncertainty of those assumptions. Even though them exist many good computer programs, are those also based on assumption, due to complexity of fluid mechanics. So, to this day the most accurate method for testing and understanding how the fluid interacts with the hull are still done from a model test. Most model test are performed in towing tanks, but other tanks specially designed for conditions such as manoeuvring, cavitation and ice tanks area also utilized. For this project, a towing tank where utilized.

Towing Tank
A towing tank is a large basin of water which often have several meters of width, depth and hundreds of metres long. A carriage runs along the length of the pool either towing the model or following a self-propellered model measuring data such as resistance for example. The purpose of the large width and depth are to simulate the ship operating at open seas, so neither the bottom or side effects the test. Accurate measures are performed when the ship moves at a constant speed, the longer time at a constant speed a better average and less effect are induced from the sensor noise. The test can be done both for testing resistance in still water, regular or irregular waves. Also, other data can be measured such as ship motion in waves.

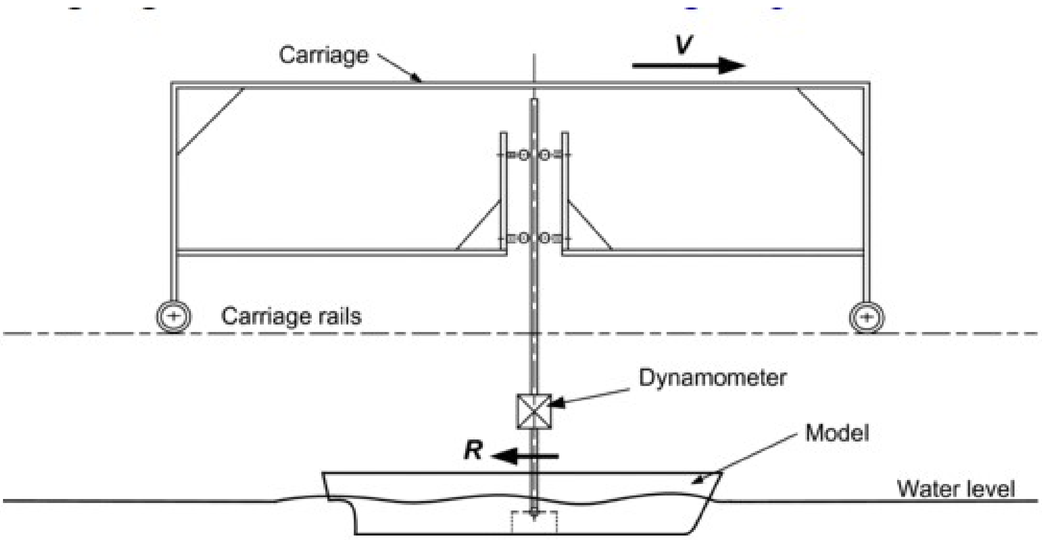


Figure 190: Towing Tank

Total resistance of a vessel

A vessel experiences several different components of resistance adding up to the total resistance of a vessel. First main component is viscous resistance which are created from the friction of the fluid tangential to the hull, and the viscous effects of water on the hull. Second component are the wave resistance, which consist of a serious of different waves created by the shape of the hull, such as the main ones bow and stern waves. Thirdly the ships resistance created from moving through air at certain speeds. The total resistance of a vessel can be expressed as follows.

$$R_T = R_F + R_W + R_{AAS}$$

- R_F Viscous (friction) resistance

- R_W Wave resistance

- R_{AAS} Air resistance

Dimensionless Coefficients

In most instances is it not practical to test the resistance of the ship itself. A model test is performed instead. In order to scale the resistance from the model to the full scale, the values have to be practically applicable over different sizes of vessels. Therefore, naval architectures utilize dimensionless numbers.

Viscous Resistance

(ITTC, 2017)The forces result in tangential shear forces that makes it necessary to create a counter acting force to move at a constant speed. Friction resistance acts on the entire wetted surface of the hull.

The boundary layers cause typically a turbulent flow. Reynolds number explains whether or not the water flows either laminar or turbulent. The flow is laminar with Reynolds number under 5X10^5, or turbulent above 1X10^6 while in between the flow is a transitional phase between laminar and turbulent flow. The frictional resistance is increased when the ships moves at a higher speed causing a larger boundary layer.

Geometrical Similarity

For the model to measure accurate values is it necessary to scale down every geometry so they are proportionate to the full-scale model. For this purpose, a scaling factor (λ) which represent ratio for the length of the model to the full-scale length.

$$\lambda = \frac{L_m}{L_s}$$

- Where L_m is the length of the model and L_s is the length of the ship.

For area the scale factor has to be squared (λ^2), or for the volume it has to be cube of the factor (λ^3).

Dynamical Similarity

Dynamic values are velocity, acceleration, gravitation, density and viscosity of the fluid. To make these values similar is impossible as it is impossible to scale down the properties of water. William Froude found out that if a ship that if a ship is geometrical similar and it can be scaled down by the speed divided by the square root of the ship. Found out that if the viscous resistance where estimated and subtracted from the resistance. The resulting residual resistance consisted of the wave resistance, and that were dynamically similar to the coefficient of wave resistance for the full scale.

$$\frac{V_S}{\sqrt{L_s}} = \frac{V_m}{\sqrt{L_m}}$$

$$C_{Wm} = C_{Ws}$$

- Where C_{Wm} is the coefficient of wave resistance from the model test, C_{Ws} are the wave resistance of the full scale of the vessel.

So approximately dynamically similarity is achieved when scaling down velocities by square root of the scale factor $\lambda^{0.5}$. This is called Froude scaling.

Model Test for Optimizing Hull Shape

We wanted to do a model test for our vessel, to see how the hull shape that were designed affected the resistance and pitch motion of the vessel. This process should have happened earlier, in the preliminary design, but were delayed by CORONA.

The model test was done in towing tank of NTNU-Ålesund. Resistance in still water was only measured due to the sensors in the tank did not function properly to give correct wave parameters, for testing resistance in regular waves. Idea behind pitch test where to test the pitch motion against a normally shaped bow as a reference. Pitch motion where only measured by eye, due to Corona reducing the time for testing more than one model.

The model was reduced to a small-scale factor due to the tank being small in both length, width and depth. In order to achieve a small period of constant speed to measure the resistance. The model had an LWL of 94 cm which gave us a scale factor calculated below.

$$\lambda = \frac{94(cm)}{6300(cm)} = 1/67$$

Scale factor where further used to calculate the wetted surface area of the model, as this where important factor for scaling the resistance, as it is a factor of the equation for the total coefficient of resistance for the model.

$$Wet\ Surface\ of\ Model = 1188.57(m^2) \cdot \left(\frac{1}{67}\right)^2 = 0.265\ (m^2)$$

The idea where to measure 3 tests at intervals of 1 knot between 14kt to 3kt of the full-scale speed. The resistance achieved from the lower speed of 3 and 4 knots gave us wrong answers. So, the speed where set to in between 14kt to 5kt. The Froude scaling where used to calculate the speed of the model in the towing tank to achieve dynamical similarity. Reynolds number where calculated for both the ship and model, and Froude number for the model for utilizing to scale the resistance to full scale.

$$Re = \frac{u * LWL}{v} \qquad Fr = \frac{u}{\sqrt{g * LWL}}$$

Where u is the velocity, LWL length of the waterline and g = 9.81m/s^2 gravitational force. v is the kinematic viscosity. 17 degrees where assumed for the fresh water in the tank since it should be nearly equal to the room temperature. 7 degrees where chosen for the tank, due to operation in or near artic seas. Fresh water at 17 degrees Celsius ($v = 1.08 * 10^{-6}\ \{m^2/s\}$) (26th ITTC Specialist Committee on Uncertainty Analysis, 2011, s. Tabel 1), Sea water at 7 degrees Celsius ($v = 1.48 * 10^{-6}\ \{m^2/s\}$) (26th ITTC Specialist Committee on Uncertainty Analysis, 2011, s. Tabel 3)

Velocity Ship (kn)	Velocity Ship (m/s)	Velocity Model (m/s)	Froude nr. Model	Reynolds nr. Model	Reynolds nr. Ship
5	2.56	0.31	0.102811	2.55E+05	1.36E+08
6	3.07	0.37	0.123373	3.06E+05	1.63E+08
7	3.58	0.44	0.143936	3.56E+05	1.91E+08
8	4.09	0.50	0.164498	4.07E+05	2.18E+08
9	4.60	0.56	0.18506	4.58E+05	2.45E+08
10	5.11	0.62	0.205622	5.09E+05	2.72E+08
11	5.62	0.69	0.226184	5.60E+05	3E+08
12	6.13	0.75	0.246747	6.11E+05	3.27E+08
13	6.64	0.81	0.267309	6.62E+05	3.54E+08
14	7.16	0.87	0.287871	7.13E+05	3.81E+08

Table 15: Model Test

The average resistance measured from the 3 datapoints at a speed where further converted to calculate the total coefficient of performance for the model. To further scale it to full scale utilizing 1978 ITTC method explained below (ITTC, 2017). Also, the dimensionless Froude number for the model and the ship where calculated to be used for.

$$C_T = \frac{R_T}{\frac{1}{2} \cdot \rho \cdot V^2 \cdot S}$$

- Where C_T is the total coefficient of resistance, R_T resistance from the towing test, V is velocity of the model, S wet surface area and ρ is the density of fresh water (1000kg/m^3).

Scaling the Resistance

In order to scale the resistance correctly, several procedures have been created. The scaling procedure method utilized are ITTC 1978 (ITTC, 2017). This method is created based from the trend of cargo vessels full scale resistance to their model resistance. Their model says that the total coefficient of resistance on the vessel is given by the equation below.

$$C_{TS} = C_{F0s}(1 + k) + \triangle C_F + C_w + C_A + C_{AA}$$

Coefficient of Wave Resistance

Coefficient of wave resistance where found from the Froude method explained above. Equation are given below from ITTC 1978 procedure.

$$C_w = C_{Tm} - C_{F0m} \cdot (1 + k)$$

Coefficient of resistance ($\boldsymbol{C_{F0s}}$)

Hull friction coefficient is calculated from an empirical formula from utilizing the Reynolds number.

$$C_{F0m} = \frac{0.075}{(Log(Re) - 2)^2}$$

Formfactor (K)

To calculate the total coefficient of friction it is necessary to determine the form factor from the model test. ITTC 1978 discovered that this factor can be very accurately calculated from a trend between the full scale and the model test explained by the equation below.

$$\frac{C_T}{C_{F0m}} = k + \alpha\left(\frac{Fr^n}{C_{F0m}}\right)$$

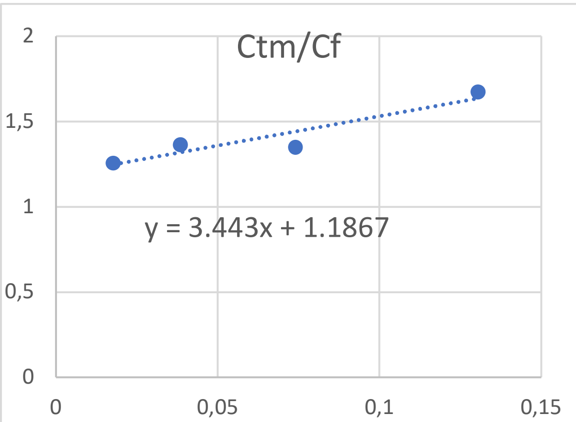


Figure 191: Ctm/Cf

Utilizing lower speed region, in order to limit the effect of water resistance, one can determine the form factor (K), from regression analysis. Inserting points ($y = \frac{C_T}{C_{F0m}}$) and (x = $\frac{Fr^n}{C_{F0m}}$). Doing a regression analysis between the points with equation (y=k+αx). Gives the form factor for the full scale of the ship and the model. (Fr^n) Froude number n-root since n is between 4-6 depending on the Froude number for the model. 4 lowest Froude number values where between (0.1<fr<0.2) ITTC says to utilize (n=4). Resulting form factor regression analysis are represented in figure 191. Results where (k=1.187), and (α = 3.443).

Roughness allowance ($\triangle C_F$)

$$\Delta C_F = 0.044\left[\left(\frac{k_S}{L_{WL}}\right)^{\frac{1}{3}} - 10 \cdot Re^{-\frac{1}{3}}\right] + 0.000125$$

Are an empirical formula for calculating the coefficient of resistance from roughness of the hull. k_s is the value of the roughness of the hull. This depends on the coating and life span of the hull. If no value is measured or given one should utilize standard value of 150 microns for the roughness. But for modern ship with modern coating needs to determine their own factor.

Correlation Factor (C_A)
Factor utilized for correction of resistance based on similar vessels and the model tests. They say each institution should have their own correlation factor. That is due to each basin will give different values, depending on how much they affect the model during the test. Also, since the correlation factor given from ITTC method are determined based on cargo ships, and can vary depending on the vessel. In this project the standard roughness factor above where utilized and no information where given on the correlation factor of the NTNU-Ålesund towing tank. So, the equation illustrated below they gave where used for calculating the correlation factor.

$$C_A = (5.68 - 0.6 \log \mathrm{Re}) \times 10^{-3}$$

Coefficient of air resistance (C_{AAs})
Coefficient of air resistance where only used on the full-scales ships total resistance, due to it being so small (1/67) therefore negligible.

$$C_{AAs} = 0.001 \cdot \frac{A_T}{S}$$

After calculating the coefficient of the total resistance of the vessel, where the total resistance calculated from the equation below.

$$R_{TS} = C_{TS} \cdot \left(\frac{1}{2} \cdot \rho_S \cdot V_S^2 \cdot S_S\right)$$

- Density of water utilized was sea water (1025kg/m^3).

Resistance Factors not Included in this Analysis
Factors that were not included in this analysis where such as added friction resistance from the bilge keel and keel. Also, the transom stern where not introduced into the calculation as this change after the model was created, due to the stability problems explained below. For a more accurate resistance those factors should be included.

Result from Model Test

Results from scaling the resistance are shown below. To represent the full-scale resistance a trendline where graphed in excel. The trendline with the most optimal fit through the points where an exponential function.

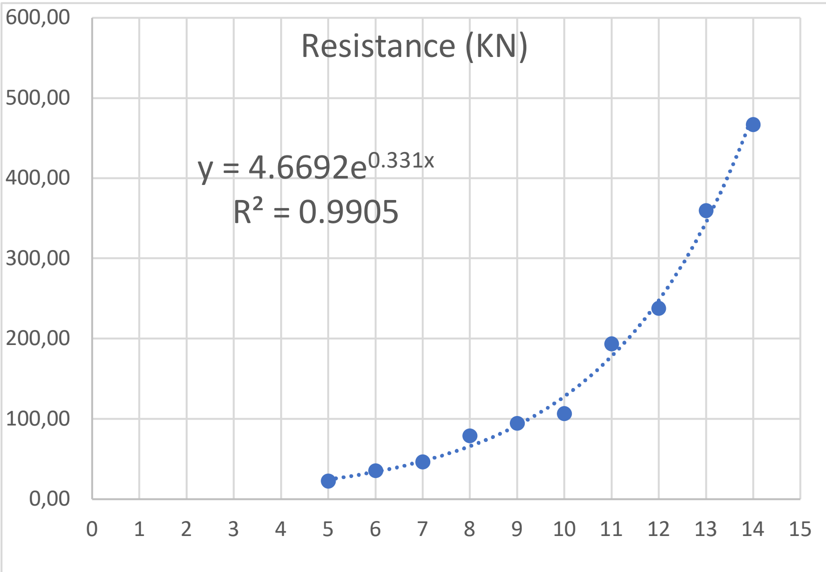


Figure 192: Resistance from test

V(kn)	Re (KN)
5	22.33
6	35.31
7	46.30
8	78.67
9	94.35
10	106.33
11	193.37
12	237.84
13	359.67
14	466.82

Table 16: Resistance for Full-Scale

Visual Assessment

Another part of doing a model test is that it allows us to visually inspect how the hull behaves and the waves form along it. This will help making the assessment of how the hull can be improved. We took videos while we did the test so that we could further inspect the wave pattern after the test, we even took some in slow motion. There was a clear exponential growth of wave creation and the front of the vessel seemed to generate the most significant wave.

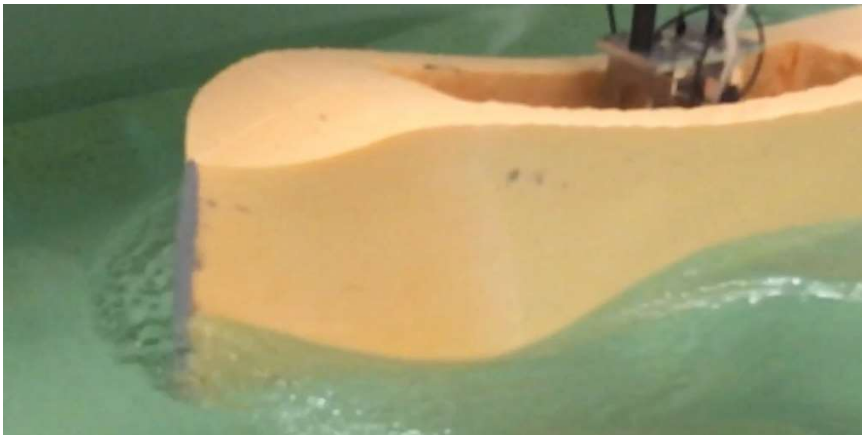


Figure 193: Bow Wave 14 kn

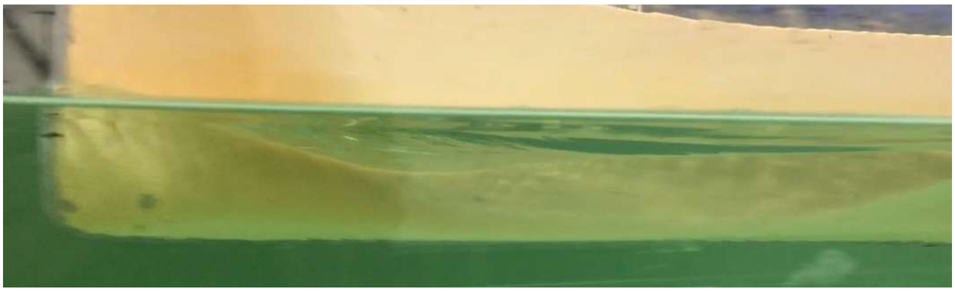


Figure 194: 14 kn Wave at Front Shoulder.

Figure 195: 14 kn Wave Profile

The aft section created a smaller wave compared to the forward section. Although, at high speed (14/15 knots) there was "bobbles" forming at the transom. Indicating low pressure, meaning there is a lot of separation, which is undesired

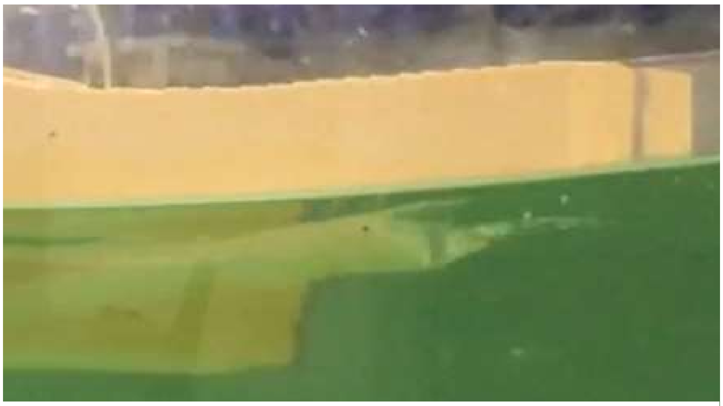


Figure 196: 15 kn Bubbles at Transom

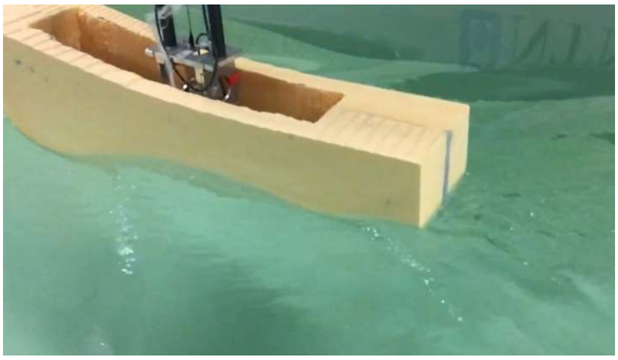


Figure 197: Waves at 14 kn

Furthermore, could we see that the waves at around 14 knots created quite sharp lines, and it looked as the hull was not "comfortable" running at these kinds of speeds. At 11 knots on the other hand it is a different story.

11 knots

Although, there are still noticeable waves at 11 knots they are significantly smaller. The bow wave still rides up quite high at the stem (figure 198), moreover the rest of the pattern seems to be cancelled somewhat out. So, aft shoulder the waves were reduced (figure 199), nevertheless there is still a bit disturbance (figure 199) at the transom which was to be expected.

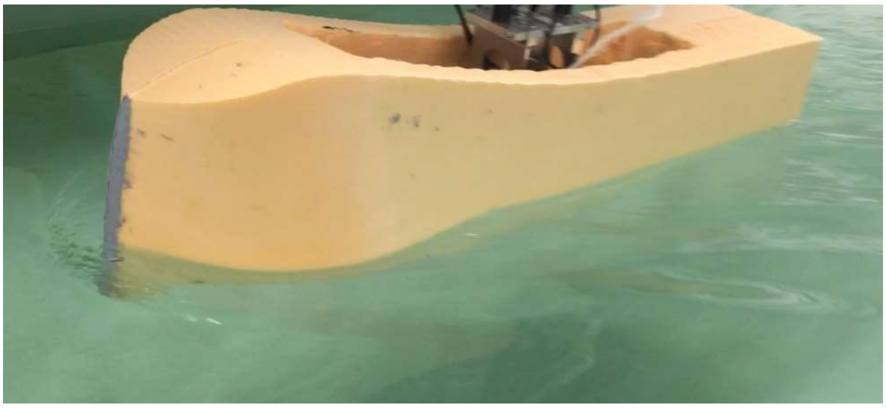


Figure 198: Bow wave 11 kn

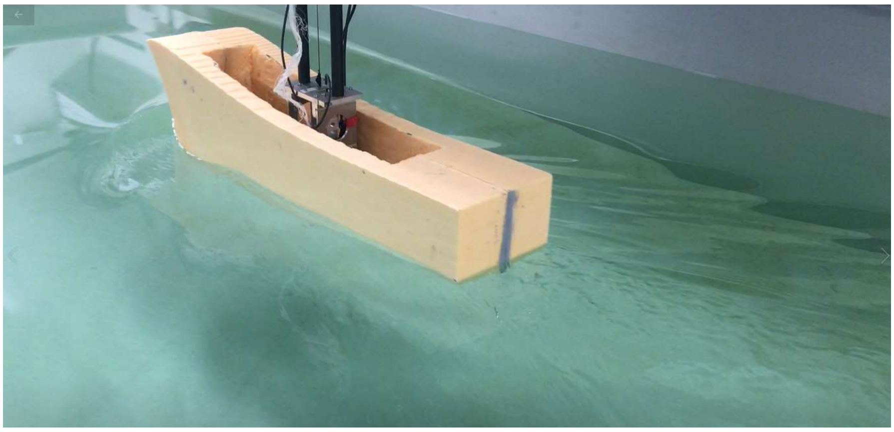


Figure 199:11 kn

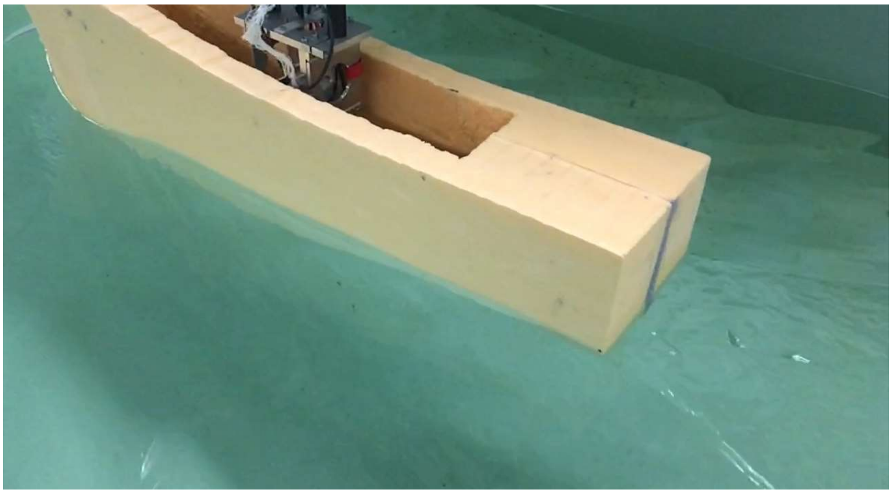


Figure 200: Transom 11 kn

In Waves

The model was also run in waves, we did not measure any data from this as it was more just for us to see how it handled. We did three runs where the vessel held a speed of 11 knots (full scale) and three runs where it was stationery. The waves produced had a height between 80 and 90 millimetres, meaning they simulate full-scale waves between 5.36 and 6.03 metre.

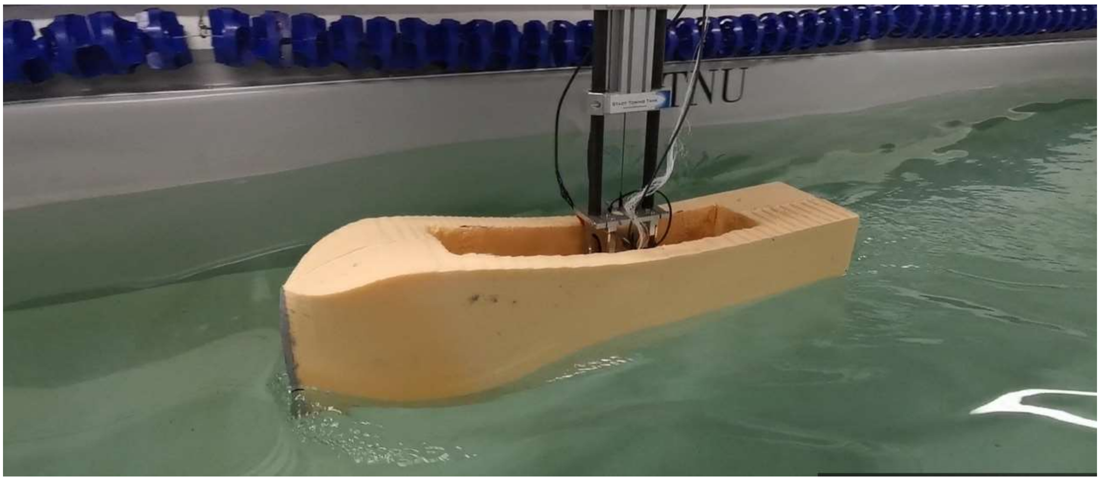


Figure 201: 11 kn in waves

Pitch motion from visual analysis, when running the model into waves, was improved assumed to be reduced at 11 knots and zero speed. It had a higher degree of acceleration and range of motion in zero speed compared to 11 knots.

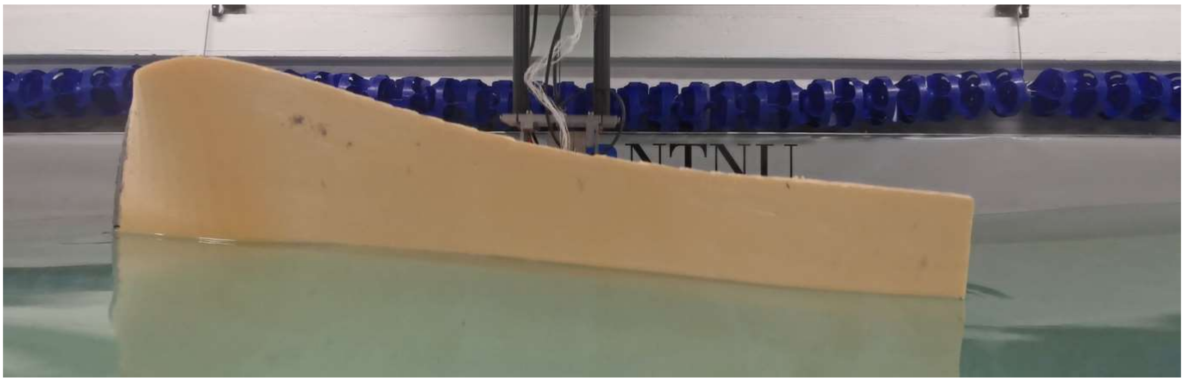


Figure 202: Stationary in waves, max pitch

The draft in the bow changed quite a bit, in opposition to the stern which never lifted out of the water. Because the way the transome is “lifted” means that when the vessel pitches it would have quickly lose the bouyancy and therefore the weight keeps it level.

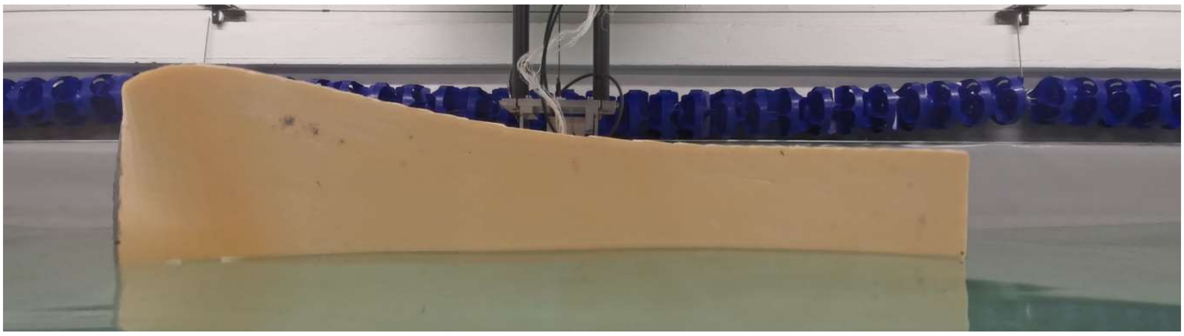


Figure 203: Stationary in waves, lowest draught at FP

Figure 204: Stationary in waves, lowest draught Aft

Understanding the Results

The accuracy of each resistance calculation is wrong due to several factors, but main correlation for the resistance curve is accurate enough for concluding on design. Firstly, the resistance should be higher due to correlation factor error and form factor error. Correlation factor from ITTC were developed for merchant vessel with a higher L/B ratio, decreasing the resistance. Also, since the water properties are not scale combined with a small model creates a lesser degree of realistic flow. By realistic flow means for example less separation in flow, viscous resistance in the model compared to the full-scale vessel.

Form factor of 0.18 were assumed to be higher earlier. That is due to the vessel having a high level of fullness even with a low Cb due to the low L/B ratio. A value of 0.18 seemed way to low, since mostly very slender ships achieve such a low value. A research paper (Min, 2009) was that investigated the accuracy of utilizing the regression method from ITTC 1978. They showcased that the calculation method is highly accurate for large models. Since the frictional resistance increases until the flow around the hull is fully turbulent, leading to a higher form factor until the terminal factor are achieved. The results were at low speeds are achieved at a model size of approximately 6-8 meters in length. With such a small-scale model in the tank, with a low Reynolds number almost reaching Re of 10^5. Which makes the form factor (k) significantly smaller due to the flow around the hull at lower speeds be assumed to be laminar. Additionally, why ($\alpha = 3.44$) as the frictional resistance are increased as vessel operates at a higher speed with a higher Reynolds number. In all conclusion the frictional resistance should be assumed to be quite higher in the resistance.

Therefore, model test resistance values can only be assumed to showcase how the hull affects the wave resistance. Since the wave resistance are not an exact exponential curve, but are affected by different wave lengths, a deeper analysis on how the different waves from bow, aft and stern around the were made figure 205. That was done by analysing the growth of increase in resistance per knot increase in between correlating points.

- Line1 : Point (5,6,7)kt: Resistance growth factor: 12
- Line2: Point (7,8)kt: Resistance growth factor: 32.3
- Line3: Point (8,9,10)kt: Resistance growth factor: 13.8
- Line4: Point (10,11)kt: Resistance growth factor: 87
- Line5: Point (12,13,14)kt: Resistance growth factor: 114.5

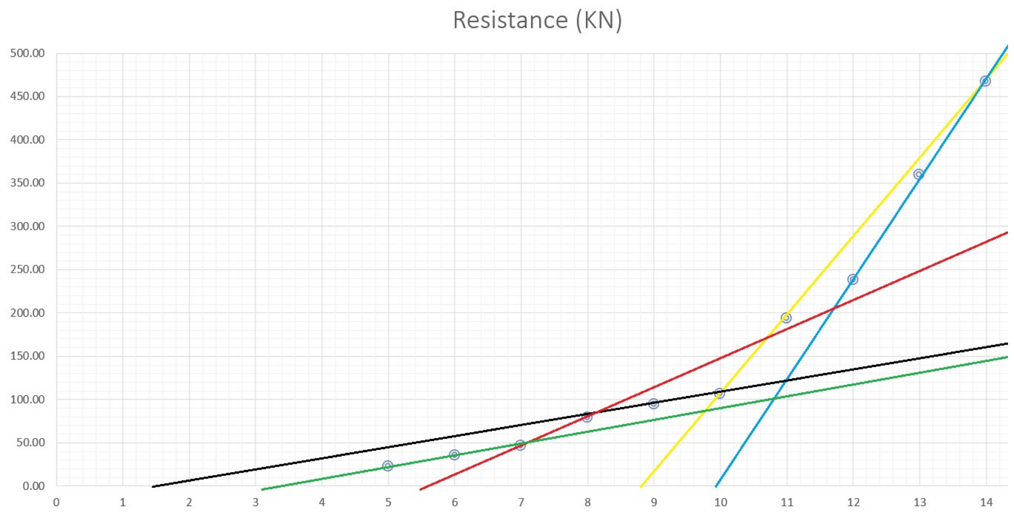


Figure 205: Analysis of Total Wave Resistance

Increase of resistance in line1 represents how the hull started generating more wave resistance as well as more frictional resistance from 5 to 7 knots. In between 8-10 knots the increase of resistance where reduced, due to less wave resistance. Less wave resistance was due to how humps and hollow from bow wave cancelled out the aft or stern wave. Before sharply increasing between 10-11knots. Which showcase how the bow wavelength are different at different speeds. From 10 to 11 knots there where an almost 200% increase in resistance. This shows that the efficient speed were in between 8-10 knots, while highly inefficient at 11knots.

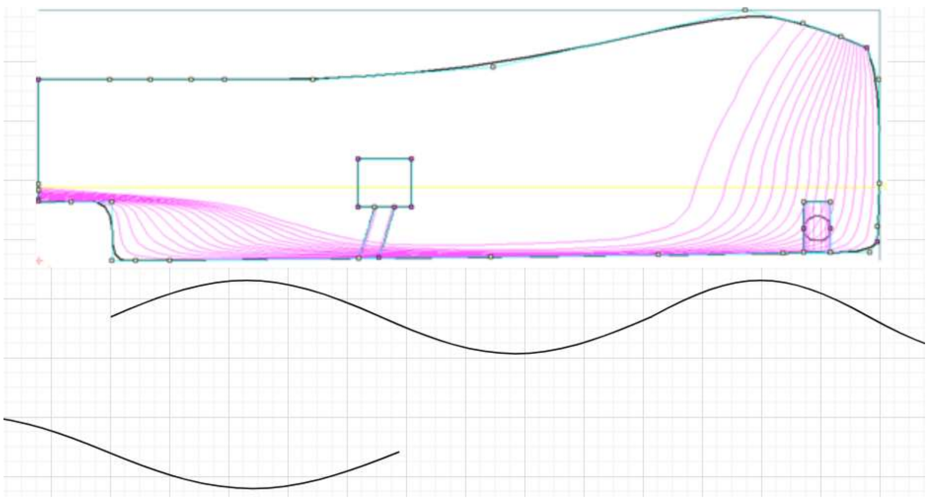


Figure 206 - Illustrating Efficiency Hump from Bow wave and Hollow from stern wave at 9 knots

Danish Seine Evaluation

Earlier it where decided that if Danish Seine should be implemented, did GA 102 represent the best way to do this. Task where to analyse the effect of implementing Danish Seine into the design. So firstly, wanted to analyse the effect on the stability. Therefore, the weights for the Danish seine equipment such as winches, driveable gallow and vacuum pump where input into the 301 load conditions, which represented the most realistic weight calculation. Condition arrival with 20% where analysed since that where the critical condition.

A problem encountered utilizing stability, is that the values for VCG and LCG would change when opening the load cases on another computer. Even though the same hull, same load conditions and compartments where used. That can be seen in figure 208 and figure 207, as figure 207 represent when the load condition where opened on another computer. Since this would analyse the difference between with or without Danish seine equipment, did this not matter, as the ratio where analysed.

	Heel to Starboard deg	0,0	10,0	20,0	30,0
1	GZ m	0,000	0,072	0,129	0,242
2	Area under GZ curve f	0,0000	0,3764	1,3671	3,1569
3	Displacement t	2762	2762	2762	2762
4	Draft at FP m	4,843	4,869	4,978	5,125
5	Draft at AP m	5,566	5,513	5,282	4,893
6	WL Length m	63,160	63,162	63,162	63,162
7	Beam max extents on	13,315	13,519	14,153	15,029
8	Wetted Area m^2	1130,10	1112,33	1097,49	1101,84
9	Waterpl. Area m^2	735,533	719,698	717,081	743,997
10	Prismatic coeff. (Cp)	0,691	0,694	0,704	0,717
11	Block coeff. (Cb)	0,585	0,589	0,500	0,433
12	LCB from zero pt. (+ve	32,380	32,386	32,402	32,428
13	LCF from zero pt. (+ve	28,764	29,624	30,601	31,070
14	Max deck inclination de	0,6576	10,0164	20,0016	30,0005
15	Trim angle (+ve by ster	0,6576	0,5853	0,2771	-0,2118

Figure 208: Established GZ

	Heel to Starboard deg	0,0	10,0	20,0	30,0
1	GZ m	0,000	0,083	0,150	0,273
2	Area under GZ curve f	0,0000	0,4303	1,5830	3,6307
3	Displacement t	2762	2762	2762	2762
4	Draft at FP m	4,821	4,847	4,953	5,100
5	Draft at AP m	5,584	5,533	5,306	4,918
6	WL Length m	63,160	63,162	63,161	63,161
7	Beam max extents on	13,315	13,519	14,153	15,030
8	Wetted Area m^2	1129,88	1112,64	1097,62	1101,85
9	Waterpl. Area m^2	735,538	720,367	717,532	744,185
10	Prismatic coeff. (Cp)	0,691	0,694	0,704	0,718
11	Block coeff. (Cb)	0,583	0,588	0,500	0,433
12	LCB from zero pt. (+ve	32,332	32,337	32,353	32,378
13	LCF from zero pt. (+ve	28,761	29,598	30,580	31,054
14	Max deck inclination de	0,6937	10,0187	20,0022	30,0003
15	Trim angle (+ve by ster	0,6937	0,6245	0,3204	-0,1653

Figure 207: GZ-values when opening again

1	Draft Amidships m	5,202
2	Displacement t	2762
3	Heel deg	0,0
4	Draft at FP m	4,821
5	Draft at AP m	5,584
6	Draft at LCF m	5,236
7	Trim (+ve by stern) m	0,763
8	WL Length m	63,160
9	Beam max extents on	13,315
10	Wetted Area m^2	1129,88
11	Waterpl. Area m^2	735,538
12	Prismatic coeff. (Cp)	0,691
13	Block coeff. (Cb)	0,583
14	Max Sect. area coeff. (	0,921
15	Waterpl. area coeff. (C	0,875
16	LCB from zero pt. (+ve	32,332
17	LCF from zero pt. (+ve	28,761
18	KB m	3,051
19	KG fluid m	6,297
20	BMt m	3,780
21	BML m	77,274
22	GMt corrected m	0,534
23	GML m	74,028
24	KMt m	6,831
25	KML m	80,320
26	Immersion (TPc) tonne/	7,539
27	MTc tonne.m	32,441
28	RM at 1deg = GMt.Disp.	25,730
29	Max deck inclination de	0,6937
30	Trim angle (+ve by ster	0,6937

Figure 210: Hydrostatics to figure 207

1	Draft Amidships m	5,240
2	Displacement t	2798
3	Heel deg	0,0
4	Draft at FP m	4,739
5	Draft at AP m	5,742
6	Draft at LCF m	5,284
7	Trim (+ve by stern) m	1,003
8	WL Length m	63,162
9	Beam max extents on	13,316
10	Wetted Area m^2	1134,67
11	Waterpl. Area m^2	735,820
12	Prismatic coeff. (Cp)	0,693
13	Block coeff. (Cb)	0,577
14	Max Sect. area coeff. (	0,911
15	Waterpl. area coeff. (C	0,875
16	LCB from zero pt. (+ve	31,994
17	LCF from zero pt. (+ve	28,742
18	KB m	3,084
19	KG fluid m	6,361
20	BMt m	3,736
21	BML m	76,320
22	GMt corrected m	0,459
23	GML m	73,043
24	KMt m	6,819
25	KML m	79,394
26	Immersion (TPc) tonne/	7,542
27	MTc tonne.m	32,432
28	RM at 1deg = GMt.Disp.	22,412
29	Max deck inclination de	0,9122
30	Trim angle (+ve by ster	0,9122

Figure 209: Hydrostatics with Danish Seine Equipment

	Heel to Starboard deg	0,0	10,0	20,0	30,0
1	GZ m	0,000	0,076	0,140	0,255
2	Area under GZ curve f	0,0000	0,3961	1,4651	3,3699
3	Displacement t	2798	2798	2798	2798
4	Draft at FP m	4,739	4,755	4,848	4,986
5	Draft at AP m	5,742	5,708	5,506	5,136
6	WL Length m	63,162	63,163	63,160	63,158
7	Beam max extents on	13,316	13,520	14,157	15,060
8	Wetted Area m^2	1134,67	1122,12	1105,69	1108,65
9	Waterpl. Area m^2	735,820	726,702	722,581	747,187
10	Prismatic coeff. (Cp)	0,693	0,695	0,705	0,722
11	Block coeff. (Cb)	0,577	0,580	0,503	0,435
12	LCB from zero pt. (+ve	31,994	31,999	32,014	32,038
13	LCF from zero pt. (+ve	28,742	29,382	30,402	30,930
14	Max deck inclination de	0,9122	10,0360	20,0076	30,0002
15	Trim angle (+ve by ster	0,9122	0,8665	0,5983	0,1369

Figure 211: Danish Seine GZ

After analysing the new load case, saw an 0.15-metre decrease in GM when including Danish seine equipment. Area under GZ curve drastically reduced with value of 0.3 at a heel of 30 degrees. To achieve the same area under GZ curve as without Danish seine equipment the keel needed approximately 20 tons more weight. Also, the draft at AP increased with 0.15 meters approximately.

This analysis only included adding the weight of Danish seine Equipment. Due to the weight of increasing accommodation in deck 5 from increasing the length of deck. Would further increase the KG, decreasing the GM. As doing weight estimation is labour intensive work. Therefore, if one would include this would one need to add more tons to the keel. Since increase in deck 5 would approximately be centred around LCB, would that not increase the draft at AP.

Also, if one would need to add a significant percentage more area above waterline. Therefore, a significant more effect on directional stability from the wind.

Evaluation

When looking back at how we approached the design and comparing it to the result, we see that even though we intended to lean towards a bottom-up approach the result has a lot of similarities to what a top-down approach resolves in. Hence, when performing a design time could be spent more efficiently by dialling back the research and brainstorming session, however for this paper it allowed us to explore more possibilities and experience deeper what goes into a design.

Part 1 required much research about system we had not encountered before in our education. At the same time, we managed to utilize our theoretical knowledge to understand systems like machinery, refrigeration system and etc. Understanding these systems in enough detail where very time consuming, which resulted in a shorter time of design process. A vessel consists of a vast number of different systems that requires different areas of knowledge, hence a team with specific people for specific areas can shorten the time spent on research. This can also be accomplished by hiring in consults for designing a new type of vessel. We had contact with a few people that consulted us regarding different aspects, Wilmar Æsøy for machinery is one of them. Without them it would have been next to impossible for us to design this vessel, at this goes to show how important a good network is.

Evaluating how ideas from the brainstorming session has been incorporated into the final design brings up the two main solutions, lift position and factory flow. It is fun to look back and see how these ideas came up seemingly random and have functioned as the backbone for the final design. The lift position has moved slightly forward throughout the process as the level of detailing has increased as more place where necessary for machinery, although the principal with having one lift amidships have stayed put. We are still not certain that one lift would work well enough, and the unloading in ability to unload the fish pallets while loading the bait. Although, we see that even in the latest GA the width is not a problem and that having two lifts side by side or a paternoster lift can still be a solution. In order to assess this further we would calculate the loading/unloading time differences. How a clear path of the factory would aid the factory flow was also linked to the lift position, in addition to how all the components have been laid out. Taking a look at the drafts from the brainstorming, we saw how the layout have stayed more or less the same.

Some parts of the placement of the elevator, can be questionable. As it is placed more towards one side of the vessel. When the vessel roll, the motion inside the elevator will be increased, same regard operating forklift. At the same time, we had no method to do test that with our ability. Although the shipowner did not mention anything wrong about it, or our supervisor. We think that this would be interesting to explore in further work.

When placing the bait storage at deck 3 we discussed how this would negatively affect the stability, and we had a lot of challenges with stability. However, the stability problems came at the conditions where the bait was nearly empty. Hence, we never decided to move it down, and it seems like that placement works quite well. While evaluating the final result this idea is something we think worked quite well, there are some negatives sides to it, mostly because it takes up a lot of space in. Nevertheless, how it all came together in the end seems for us to function great for overall work efficiency in the hauling area.

Another storage where accessibility is an essential part, is the food storage. Where it first was planned to have access from the front, but due to the anti-roll tank we could not have opening through. Then we had to assess if the anti-roll tank or the food storage should be moved, if not there was another way to access. Since this is “on” the weather deck we assessed that the food could be lifted up to the deck and simply carried through the dining hall and galley into the storage. This might not be the shortest path, jet we evaluated that this is an easy enough loading process.

As stated in *Part I*, we evaluated that the first priority would be to maximize the storage space for the pallets of frozen fish. Which at 580 tonnes of fish exceeded the required capacity of 450 tonnes by 130 tonnes. Unfortunately, this had some drawbacks, even though we tried to make compromises the machinery is perhaps a bit cramped. Another drawback became apparent in the model test as the bow wave built itself up, so it should be talked over with the owner if the additional capacity is desired. Having more capacity than the they are able to fish, before running out of supplies, will not give added revenue.

When considering the accessibility to the main storage, there has been focus on getting the pallets in and out with the lift. An oversight though is the personnel’s access and safety as the detailing level increased, there has been placed a few staircases around the vessel although not into the main storage. The idea was to keep it together with the lift and inserted as an own component when the less conservative estimation of the lift, although in all the excitement it was unfortunately forgotten. We still think the best place for it would be somewhere near the lift, perhaps next to the moonpool, yet it would have to be checked that it would not block the corridor. Another option could be to place the stairwell into the hull-side, although here it may come in the way for the pelletizing machine. We remember that the measurement of the palletizing machine was based purely on eye-measurements, so with specified dimensions it would easier to assess. This is something that would need further considerations, because it is an essential part for operation and safety.

Many aspects of the design process became apparent through the different weight estimations. Both showcasing how technical areas are interconnected, lack of research material, and the importance of experience. The weight estimation changed a lot during the process, due to main factor such as lack of experience and lack of necessary components weight. Both since the structural components were overestimated in the first process, since most of our experience was with bigger vessels, for example the double bottom. Another factor is how preliminary formulas of other ships can lead to an error in result; for example, the machinery was overestimated significantly by schnekluth formula. As well in some sense of understanding the total estimate where wrong, therefore compensating by overestimating some systems in order to get a more realistic result. When in theory, most of the weight estimation errors were probably due to lack of small components in the estimates, as lack of understanding for estimating more than the significant components included in the design. Therefore, adding a total of 20% to the entire lightweight of the ship, then only the hull, was determined to be most accurate. Most of the correct scale was obtained by asking our supervisor to estimate. All this resulted in a tremendous difference between VCG and LCG estimates throughout the design spiral. Resulting in a necessity of redoing work of the hull design both for structural, stability, and resistance more than optimizing, resulting in a more inferior result.

At the same time resistance should be a more important factor, during the design process. Too much focus, were put into trying to reduce the weight and optimization of stability. In this case, probably would been better to focus less on the added weight from the keel. If an added weight in the keel would allow us to design a hull with less resistance the consumption would be less and the vessel would be more economical to run. Lowering the consumption of MDO is also beneficial for the environment. Comparing to other vessel could give a better starting point for this than how we went about with the design. By trying first without a ballast-keel, without any good estimations for resistance and aiming for maximum storage we got on a path that underestimated the effects on resistance.

From the model test we saw that the pitch acceleration was reduced at 11 knots in regular waves. In irregular waves, having the same rate of change of the bow volume can cause rapid pitch acceleration from small waves (Håvard Vollset Lien, Personal Communication). In addition, was pitch more severe at no speed, assumed to be similar for 3 knots. Since large increase in wave resistance at 11 knots would leave the shipowner dissatisfied. The bow needs to be redesigned slimmer with a change in volume, that decreases the ship motion from small waves, and resistance, or a bulbous bow to extend the efficiency to 11 knots.

Before our project we had worked with ships over 100m and L/B's well over 5, this was damaging for our design as we made assumptions based on our previous experience. Knowing these assumptions would have such effect on our result, we would have been more careful with our decision making.

There has been a bit of an absence regarding the fish-remains in this paper. This is because as we started to gather intel, about how they could be used and what had to be done with them, it became apparent that it would not be valuable. Most of the remains are there next to no-demand for, and although there is a demand for the liver and roe, they would have to be hermitized onboard. This means the vessel would need another machine (to hermitize) and space to store them (Grete Hansen Aas, personal communication, 23. Jan. 2020). The head can simply be frozen and sold, but here is there also a lack of demand. Certain countries desire them, however then they would have to be transported further.

What we definitively should have worked more on though is comparing the structural design with the GA. When we made the class drawings, we saw that the stairwells exhaust-stack and lift did not line up with the beam structure as well as it should/could. Since this revelation came last minute the alterations in the GA and the class drawings have not been revised. The lift was moved in the GA since we did not want to "break" up the structure in the midship, moreover where the it made a compromise in the aft section. The one of the beam-spacing (*l*) was reduced, in addition to small alterations with the MOB-boat, stairs and exhaust placement in the GA. These alterations should be calculated to assure they are strong enough, ideally there should be compromise between the structure and the GA which we would have wanted to explore if we had the time. Hence, in hindsight we should have compared the two parts in a higher degree earlier and throughout the design spiral.

In the beam structure we had to use quite a bit of support pillars, which have been done as a precaution. The inside walls of the vessel have not been accounted for, and in practice they would help supporting the rest of the structure. In addition, can strategically placed walls reduce the open span and reduce the risk for vibration, which was our main reason for the support pillars. The placement of the pillars and walls is something we would want to further analyse before drawing any conclusion. Hence, the pillars in the aft section of the class drawings have been excluded due to insufficient analysis.

Remembering that the owner's requirement stated that the vessel should be good looking and have a low wind area, we have to evaluate to what extent the final result meets this. The wind area has been a point that has been brought up throughout the process and it seems we have reached a compromise between that and the inside volume. Evaluating the looks is quite ambiguous, our vessel as a quite different appearance than *Geir* and *Atlantic* while still looking good in its own way. The bow is a little bit high, although after testing the model it seems it can be lowered, yet the superstructure blends it into the rest of the profile.

We believed earlier that reduced building cost would be an important factor, but as our supervisor told us, this would be specially considered, and we should go for the lightest structural weight as possible. However, concerning the building cost we could add that the hull shape requires little double curved plates. Single curved plates are cheaper in construction, combined with the weight reduction (Olav Småriset, Personal Communication, May 19 2020), therefore we have ended up with a cheap structure.

Conclusion

In conclusion, we made a preliminary designed of a vessel that in principal inherits to the specified requirements, yet it is far from “perfection”. Meaning that it is a design that is presentable, nevertheless there are a lot of dark spots and discussable compromises. The results from the model test is a clear example where it can be argued that the storage capacity was prioritized too much compared to the resistance and sea-worthiness, but in the end we evaluated that this is a dilemma we would present to the potential owner and discuss it before going forward. In opposition to the stability dilemma, of adapting the lines in order to reduce the need for ballast, which is strictly a problem that could be solved with more time. However, the task includes the time restriction and that is why we ended up with a bit of a hasty solution, just adding more ballast to the keel. This means the vessel would “drag” around additional weight, which contradicts with both the economical and the environmental aspect. Although, this is something that could be improved after the preliminary design when weight estimations are more accurate. In addition, should it be investigated if the added weight increases the resistance, or if it allows for smoother lines and actually reduces the overall consumption.

Regarding the inclusion of Danish Seine, we came to the conclusion that this is a choice that should be made early in the process. In a way that there is available space to fit it no lower than deck 4, right above the line arrangement/ factory. In addition, would we regard this size of vessel a bit small to incorporate both, especially when taking into account the wind area.

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