



TRA Visions 2016

Idea Number:

L1-25

PILLAR / TRANSPORT MODE			
ROAD ☐	RAIL ☐	MARINE ☒	CROSS MODAL ☐
Project Related to:	☐ RA1: Environment – Decarb., Sustainability & Energy Efficiency ☒ RA2: Vehicle & Vessels Technology, Design & Production ☐ RA3: Urban and Long-Distance People Mobility – Systems and Services ☐ RA4: Freight Transport & Logistics ☐ RA5: Safe, Secure and Resilient Transport Systems ☐ RA6: Transport Infrastructures ☐ RA7: Human Factors, Socio-Economics and Foresights ☐ RA8: Automation and Connectivity ☐ RA9: Enabling Environment for Innovation Implementation		

Project Title:

Closed Ballast Dynamic Depth Ship



TRA VISIONS 2016
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*Closed Ballast Dynamic Depth (CBDD) Ship
Report*

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1. Introduction

The purpose of ballasting is to increase stability or seakeeping by adding weight, in a similar way to a hot air balloon using sandbags, a ship uses seawater for such operations.

The main function of ballasting is to keep the ship balanced when there is insufficient or unevenly stowed cargo. Extra ballast is also taken on when sailing through rough seas in order to increase the ship's stability, or to make the ship sit lower in the water when it needs to pass under a bridge for example.

Once cargo has been unloaded from a ship, the ship takes on ballast in order to replace the weight. When another cargo is loaded, the water is pumped back into the local environment. (Simm, ND)

Shipping moves over 80% of the world's commodities and transfers approximately 3 to 5 billion tonnes of ballast water internationally each year. (GloBallast Partnerships, 2014)

From Table 1, it can be seen that bulk carriers transfer the largest proportion of ballast water compared to other ship type.

Vessel Type	DWT	Ballast Conditions			
		Normal (tonnes)	% of DWT	Heavy (tonnes)	% of DWT
Bulk Carrier	250,000	75,000	30	113,000	45
Bulk Carrier	150,000	45,000	30	67,000	45
Bulk Carrier	70,000	25,000	36	40,000	57
Bulk Carrier	35,000	10,000	30	17,000	49
Tanker	100,000	40,000	40	45,000	45
Tanker	40,000	12,000	30	15,000	38
Container	40,000	12,000	30	15,000	38
Container	15,000	5,000	30	n/a	
General Cargo	17,000	6,000	35	n/a	
General Cargo	8,000	3,000	38	n/a	
Passenger/RORO	3,000	1,000	33	n/a	

Table 1. Ballast water capacities for different types of ships (GloBallast Partnerships, 2014)

1.1 Need

Ballast water is considered as one of the main methods responsible for introducing invasive species into marine ecosystems. The effects of invasive species in marine environments are seen as one of the greatest threats to aquatic ecosystems and are almost always irreversible. Any form of marine life small enough to be sucked through the pumping system, including microbes, small invertebrates and the eggs and larvae of larger species, can be taken in with the ballast water and transferred to another part of the world when the water is discharged. The local environment where the ballast water is discharged has not evolved to cope with the invasive species, this means that the local environment can die out due to no predators of the invasive species.

An example of invasive species is the introduction of zebra mussels in the east coast of United States of America (USA) in 1986 and has now spread through rivers towards central parts of USA (see Figure 1). Zebra mussels are notorious for their biofouling capabilities by colonizing water supply pipes of hydroelectric and nuclear power plants, public water supply plants, and industrial facilities. They colonize pipes constricting flow, therefore reducing the intake in heat exchangers, condensers, firefighting equipment, air conditioning and cooling systems. In 2000 it was estimated that costs associated with damage caused by 50,000 invasive species introduced into the USA amounted to \$137 Billion.



Figure 1. Spread of zebra mussels in USA (Nonindigenous Aquatic Species, 2014)

Introduction of invasive species can also cause major ecological damage. For example, Chinese mitten crabs from Northern Asia which were introduced to Western Europe, The Baltic Sea and the West Coast of North America. They undergo mass migration for reproductive purposes and prey on native fish and invertebrate species causing local extinction. In addition, invasive species that are associated with ballast water can also harm local communities, such as various strains of cholera bacteria which cause cholera epidemics. There can even be an economic impact as local fish stocks can be depleted due to the invasive species. (IMO, ND)

Due to the danger and potential damage of invasive species, some countries have adopted more stringent ballasting requirements. For example, the Australian government deems all salt water from ports and coastal waters outside Australia's territorial sea to be a 'high-risk' and capable of introducing exotic marine pests into Australia. The discharge of high-risk ballast water in Australian ports or waters is prohibited. All international vessels intending to discharge ballast water anywhere inside the Australian waters are required to manage their ballast water in accordance with Australia's mandatory ballast water management requirements. (Australian Government Department of Agriculture, 2011)

Additionally, the International Maritime Organisation (IMO) introduced the Ballast Water Management (BWM) Convention in 2004 which will come into force after 30 states and 35% of the world fleet tonnage signs the agreement that limits the flexibility of ballasting operations globally. As of 23rd October 2014, 43 states and 32.54% of world fleet tonnage has agreed and signed the convention. (IMO, ND)

1.2 Invasive Species - Statistics

Over the last century, there has been an ever growing increase in transportation of invasive species which correlates with the growth of the world seaborne trade which can be seen in Figure 2 and 3.

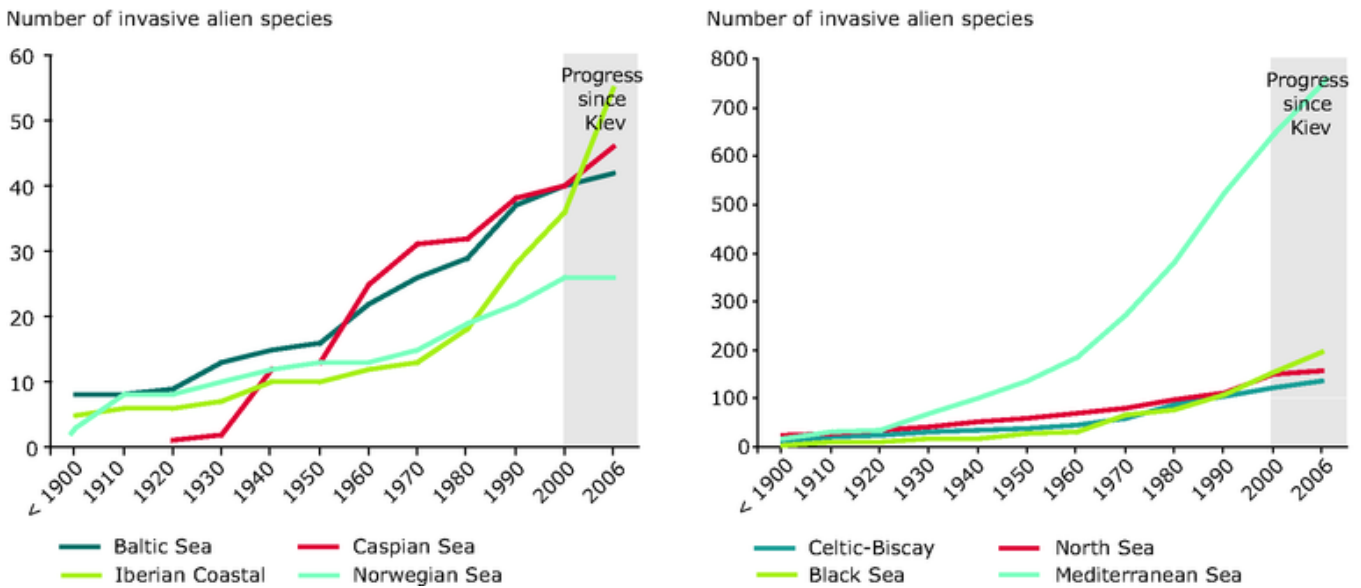


Figure 2. Growth of invasive species transportation (European Environment Agency)

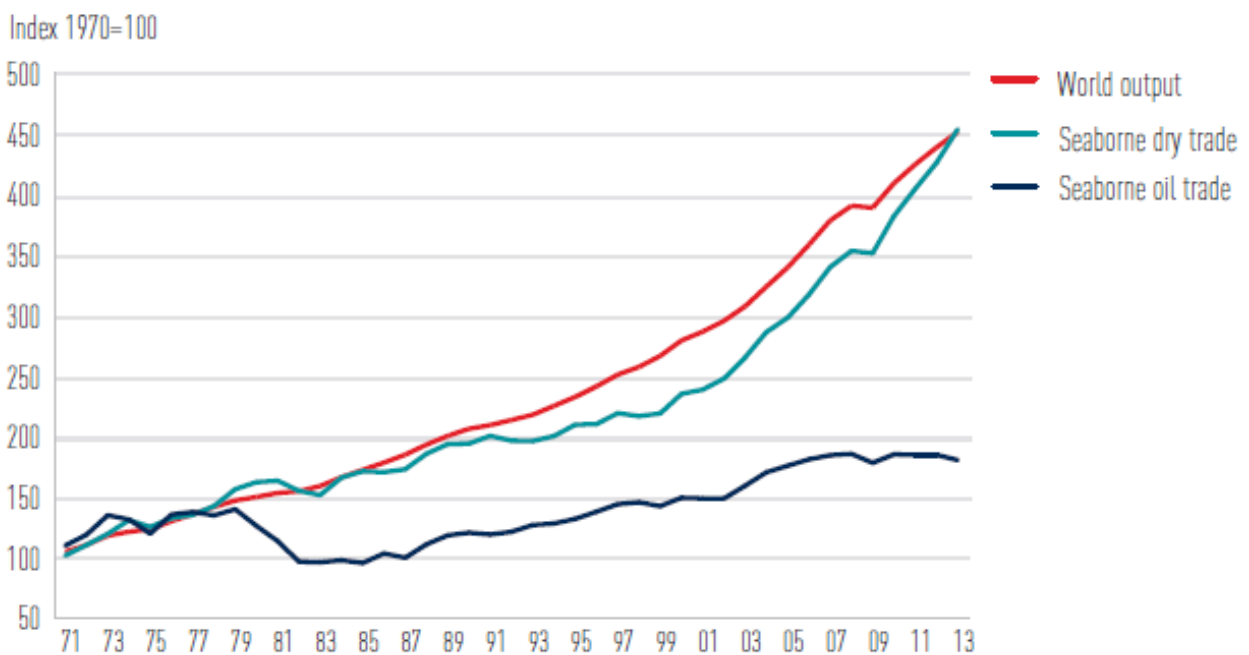


Figure 3. World Seaborne Trade and Economic Growth 1970-2013 (Platou)

It is noticeable that the Mediterranean Sea is affected the most by invasive marine species because it lies on the main shipping route from Asia to Europe and America, as shown in Figure 4.

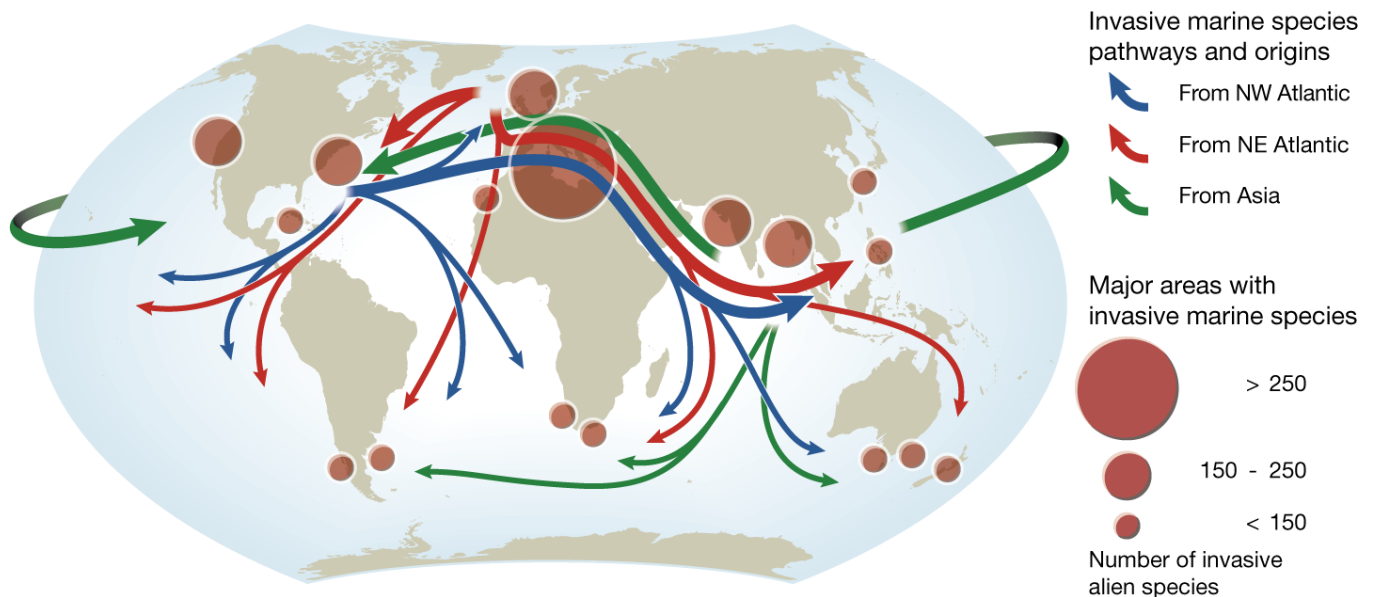


Figure 4. Invasive marine species pathways and origins (International-Marine, ND)

Currently, there are multiple methods approved by the IMO to combat the transportation of invasive species in ballast water, see Figure 5. Even with these methods being used, there is still an ever growing increase in incidents regarding invasive species transportation in ballast tanks, see Figure 2. This suggests that these treatment methods are not 100% effective. Furthermore, the treatment methods kill organisms rather than preventing the transportation of non-indigenous species.

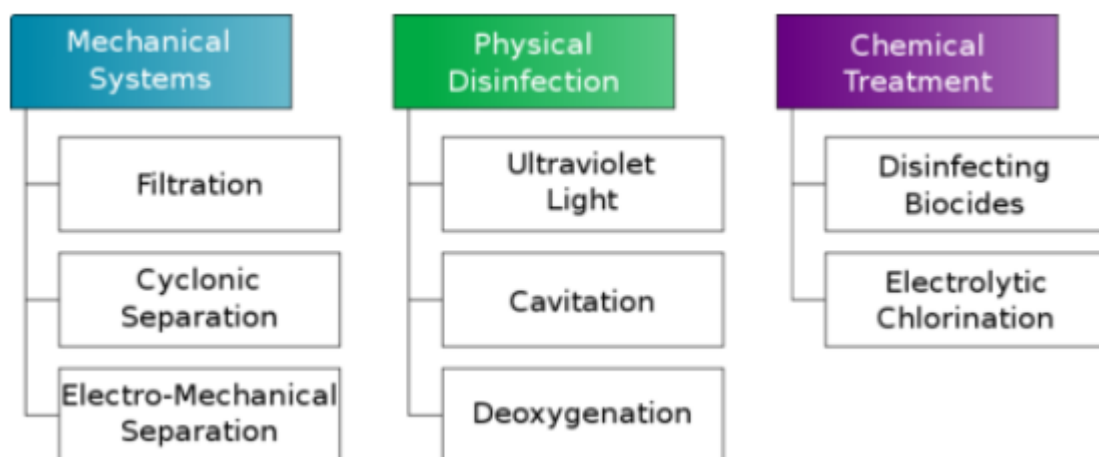


Figure 5. Ballast water treatment methods

1.3 Current Research

Following the adoption of the IMO BWM Convention 2004, interest in these alternatives to conventional BWM has increased since they offer ways to avoid, or substantially reduce, the possibilities of transporting invasive species in ballast tanks.

The range of alternatives to traditional ballast methods can be categorised into the following two groups: (GESAMP, ND)

- No Ballast or Zero discharge methods such as the following:
 - ‘Monomaran Hull’ – Delft University of Technology (DUT), Netherlands, Figure 6.
 - ‘Volume Cargo Ship’ – Det Norske Veritas (DNV), Norway, Figure 7.
 - ‘Solid Ballast Ship’ – Daewoo Shipbuilding & Marine Engineering (DSME), Korea
- Continuous flow methods, Figure 8.

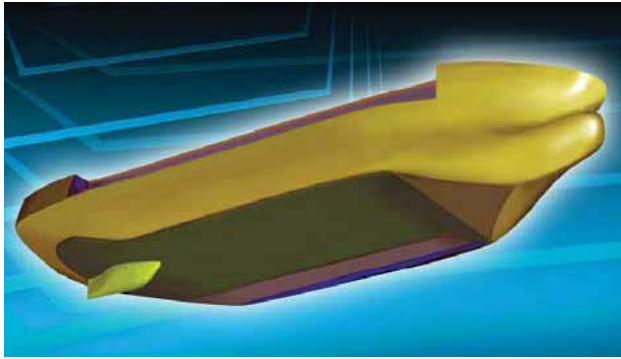


Figure 6. Monomaran hull

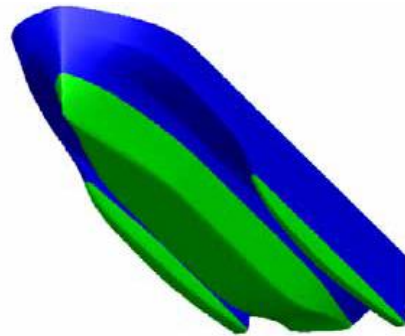


Figure 7. Volume cargo ship

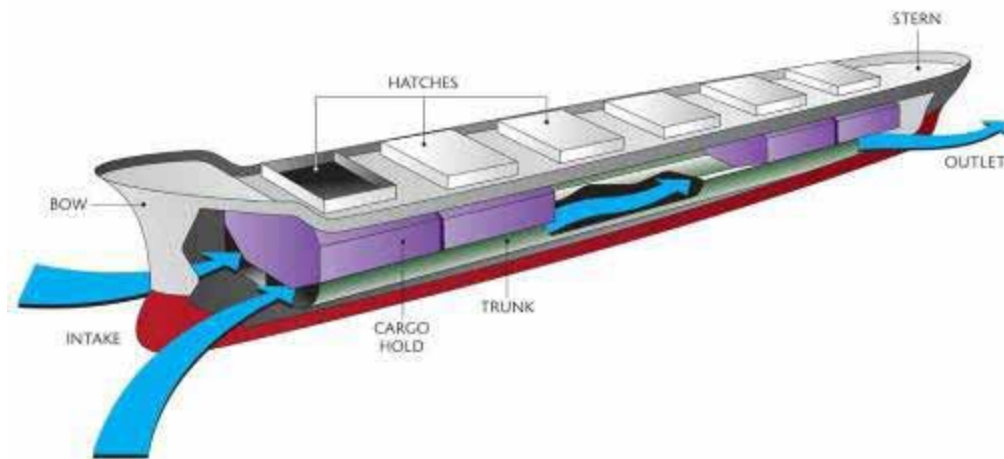


Figure 8. Continuous flow method

2. Concept Design

The proposed design aim is to remove the need for conventional ballasting operations during loading and unloading of cargoes, negating the risk of transporting invasive species. This can be achieved by altering the ship depth to improve the stability and cope with changing displacement, compensating for the addition or removal of cargo. When the ship is fully loaded, the depth of the ship will be fully extended to its maximum position and when unloaded, the depth of the ship is at its minimum, to increase the ship's stability by lowering the centre of gravity of the lightship resulting in overall increase in GM.

In order to improve the sea keeping aspect of the ship, a closed ballast system is introduced with the design purpose of shifting fluid to achieve the required trim and stability. It is important to note that this closed ballast system will have no interaction with the sea and will solely be used for trim and stability.

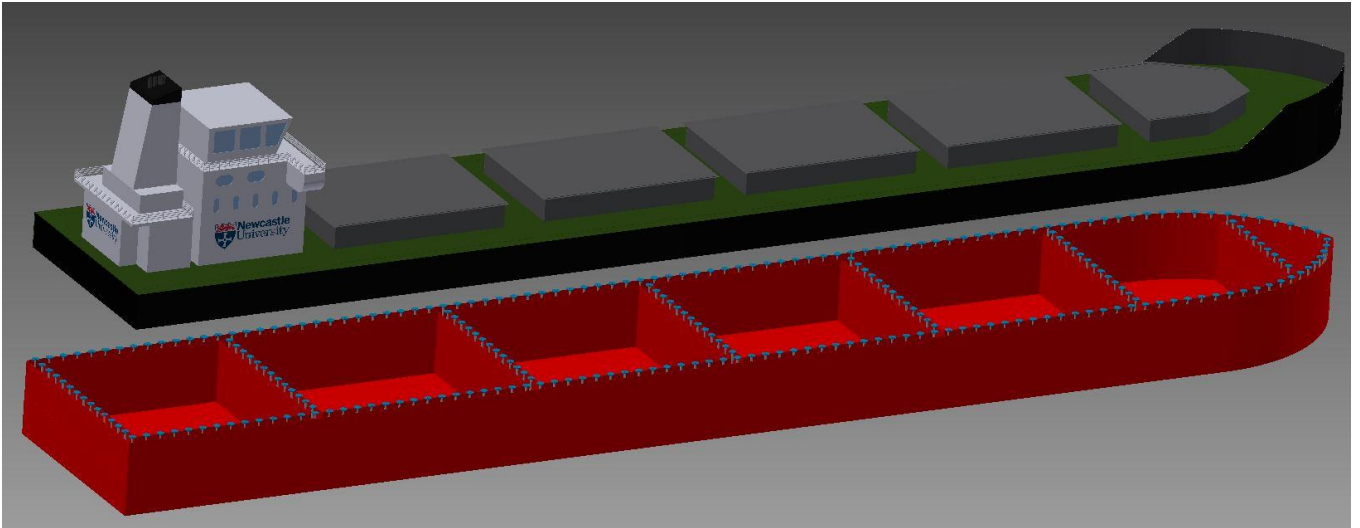


Figure 9. Concept design with lifting mechanism

2.1 Ship Selection Statistics

The ship type selection was based on data from Table 1 (page 4) which indicates that bulk carriers transport the largest volume of ballast water due to the nature of their operation, which largely requires them to travel unloaded with ballast on their return journey. While the selection of size of the ship was based on the data from Rotterdam port (Annex A), indicating that 20,000-30,000 gross tonnage (GT) ships are the most frequent users of the port. The combination of both type and size of the ship fall into the range of a handysize bulk carrier category which will be used for this design project.

2.2 Basis Ship and Alterations



Figure 10: Sea Horse 35 bulk carrier - Dragonera

The basis ship selected was the Dragonera, see Figure 10, as it falls in the handysize bulk category. The Dragonera has an optimised hull with a high block coefficient, which also has no bulbous bow, as it is unnecessary if the vessel's bow dimension is subject to change regularly. More details for the Dragonera's dimension and shape can be found in Annex B. A blocky hull shape with minimum curvature is essential for the initial design as it will minimise the complexity of adjusting the depth.

In order to apply the concept design of adjustable depth, the ship was split along the waterplane area into 2 structures, the hull and the top deck including superstructure. The adjustability of the depth can be achieved by installing a lifting mechanism in between the double skin hull and bulk heads. Figure 9 (page 9) demonstrates this.

3. Design Aspects

It is important to recognise that this is a marine engineering project and therefore advanced structural strength, stability and seakeeping of the concept design will not be carried out at this stage. However, basic parameters have been calculated to show justification of the design.

3.1 First Principles of Ship Design

3.1.1 Structural Strength

To determine the structural strength of the ship, a mid-ship cross-section is assumed for both the expanded and collapsed cases using the basis ship dimensions, see Figure 11. The cross-section contains a double-skin with a small gap of 50 mm to allow the top side to move up and down with respect to the bottom hull without any contact, minimising friction and risk of damage. A plate for the lifting mechanism to sit on is also provided between the double-skin of the bottom hull. The steel used in these calculations is assumed to have a yield stress of 315 N/mm^2 . The material type was chosen based on a reference capesize ship where the material had the lowest quality steel used. Using the basic load equations, demonstrated in Annex C, the section moduli for the hatch, keel and deck were calculated, as they are typically subjected to the most bending stress. The section modulus for the respective parts was calculated with an assumption of 40% stiffener factor on certain plates that are typically stiffened on bulk carriers. In addition, an assumption of a thickness factor for certain parts of the cross section to optimise the strength and weight was made. These values were then compared with the IACS minimum section modulus requirements, extracted from IACS Common Structural Rules for Bulk Carriers, and theoretically proven to be safe, see Table 2.

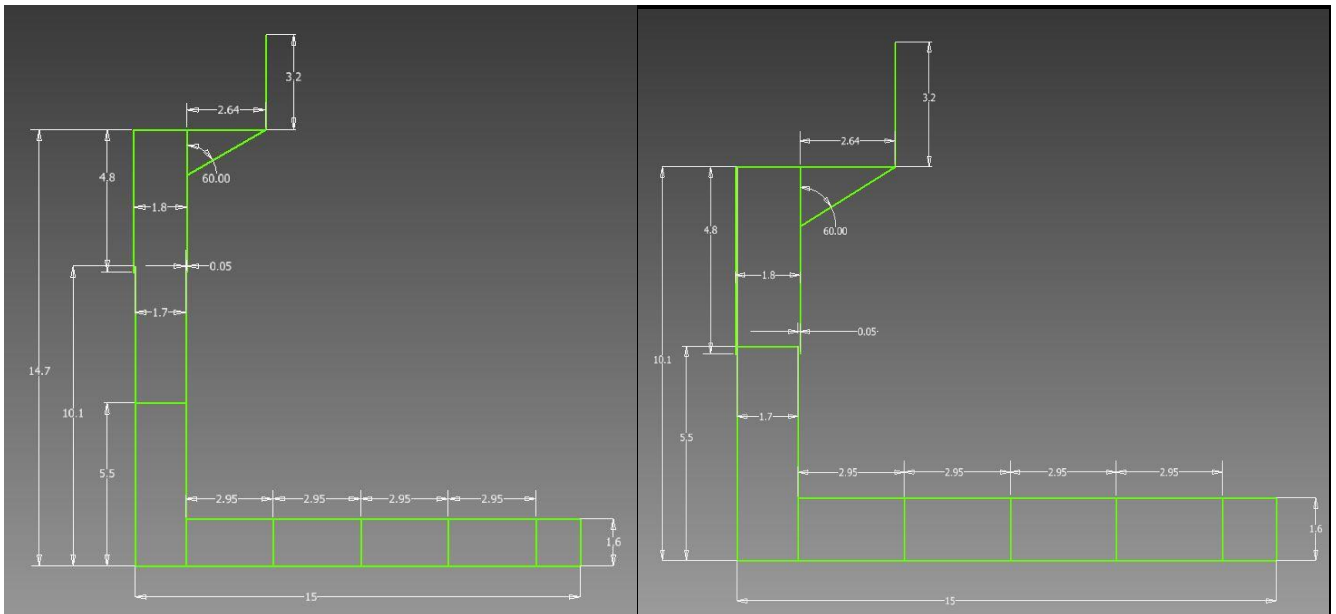


Figure 11. Mid-ship cross-section for expanded case (left) and collapsed case (right)

IACS Z_{MIN} (m^3)		Z (m^3)	
		Expanded Case	Collapse Case
9.875038776	Hatch	14.8513	9.9533
	Keel	31.1023	19.4901
	Deck	20.1820	15.6368

Table 2. Comparison of section moduli of the ship's cross-section with IACS minimum criteria

After meeting the minimum sectional modulus requirement, the assumed stiffener coefficient of 40% was validated. These validation calculations were made for the deck as it is always stiffened on bulk carriers and is subjected to the maximum bending stress. When selecting the type and size of the stiffener a reference was taken from a capesize bulk carrier which had flat bar stiffening on the deck with a web height and thickness of 390mm and 27mm respectively. Using these stiffener dimensions the spacing of stiffeners was calculated to be 634mm as compared to the capesize ship where the spacing is 880mm, see Table 3. This resulted in a deck with more densely packed stiffeners than the reference ship, which further validates that the 40% stiffener factor is a safe assumption. The assumed stiffener dimensions were then proven to meet the IACS stiffener dimension criteria.

	Dragonera Seahorse	Capesize ship
Web height h_w (m)	0.39	
Web thickness t_w (m)	0.027	
Area of stiffener (m^2)	0.01053	
Deck length (m)	4.44	12.32
Number of stiffeners	7	14
Stiffener spacing (m)	0.6343	0.88

Table 3. Stiffener spacing comparison

All the equations and calculations used are provided in Annex C – First Principles of Ship Design (Structural Strength).

3.1.2 Stability

3.1.2.1 The need for stability calculations

As mentioned in Section 3, it is still important to determine whether the vessel design will float and float upright. Therefore it is required to carry out intact stability calculations of the vessel, in both its designed, extended loaded operating condition as well as its collapsed unloaded case for return voyages. To go about this, stability calculations were used to find the intact stability of the ship in both the collapsed and extended case. This was important to justify the design of the project.

3.1.2.2 Calculations

As shown in Annex D, the calculations for the stability of the ship were carried out and can be summarised by Table 4

Parameter	Extended Case	Collapsed Case
KB	5.05m	1.16m
BM	7.43m	32.95m
KG	9.01m	4.55m
GM	3.47m	29.54m

Table 4 - Stability of ship in Extended and Collapsed Case

3.1.2.3 Analysis

These results show a positive value for GM, and this indicates that the design ship floats upright which is the main reason for conducting these initial stability calculations. The GM of the vessel in both conditions also meets the requirements set by the IMO that the GM of the vessel must be greater than 0.15m.

The GM for the extended case can be related to handysize bulk carriers to compare the results. It was found that the GM of such a vessel can vary dramatically, from 1.3m up to 8.5m whilst carrying cargo, and when un-laden can reach values of 22.5m This shows that the calculations used to find the GM of the extended case of the ship are reliable and therefore both results can be trusted. The reason for the GM of the collapsed case being greater than usual is due to the fact that there is a significant mass of steel weight being lowered when the ship collapses as well as cargo being removed. See Annex D.

3.2 Lifting Mechanism

3.2.1 Technology Available

One of the main considerations for the project was the method in which the top deck and superstructure would be lifted and dropped. After conducting some research five main methods were found:

- Ferrofluid
- Thermal expansion valve
- Gears
- Pneumatics
- Hydraulics

A ferrofluid is a fluid that becomes strongly magnetised in a magnetic field. Research has been done into this field as using magnetism seemed like a potential way to lift the structure. However, it is still in the stages of development and it is therefore not known if this method would actually work within the project. Furthermore, the process would require constant electrical power, making it a costly method with no fail safe mechanism.

A thermal expansion valve (commonly used in refrigeration) is an appropriate research topic because it is a potential solution to moving the top deck. It works by heating a liquid, subsequently changing the density. As the density is reduced the liquid expands, pushing against the structure causing it to lift. A potential positive is that the system may be able to run from the waste heat of the engine, saving energy and reducing cost. However, this method has too many negatives to be considered. Firstly, it requires constant heating in order to work; therefore, if one of the heating systems failed the structure would drop. Furthermore, it has not been proven in large scale operations and therefore poses a risk making it unsuitable.

A gear system used as a lifting mechanism would work by using an arrangement of five lifting gears and replicating this throughout the ship. The benefits of this system include it being a proven, relatively simple design. However, if one gear failed there is no fail safe and the lifting mechanism would fail to work. After calculating the probability of one gear failing out of the whole system, it proved to be too unreliable to consider. There were some other negatives as well, including regular and expensive maintenance as gears wear away easily and would require constant lubrication, in addition a risk of corrosion and water damage causing rust.

In the end it came to a choice between pneumatics and hydraulics. Pneumatics use air at a high pressure to produce mechanical motion. The advantages of using a pneumatic system include good reliability and the fact it is a proven technology. On the other hand, this project would require the system to use very high pressures in order to deal with the lifted weight, therefore, making it potentially unsafe.

A hydraulic system is very similar to a pneumatic system, but it uses a hydraulic fluid rather than a gas. In the end, hydraulics have been chosen to be used for the lifting mechanism because in addition to being a proven, reliable system, it can also handle larger forces in comparison to a pneumatics system. One drawback of the system is the potential reduction in safety due to a higher fire risk, however, this is reduced by ensuring the material of the cylinders and pistons using the hydraulic fluid is non-flammable.

Overall, hydraulics seemed to fit best into the project considering safety, simplicity and maintenance.

3.2.2 Engineering Requirements

3.2.2.1 Positioning

A hydraulic system requires use of pumps and cylinders. After calculating weight to be lifted and pressure required, the number of cylinders could be found. This is explained in detail in Annex E, Appendix I. The system requires a total of 280 cylinders. Due to the positioning of bulkheads, the ship needed to be split into sections with an appropriate number of cylinders in each section. After studying the schematic drawing and considering the dimensions of the cylinders, 14 sections with an average of 20 cylinders in each section has been chosen. This gives an even distribution of weight and a suitable distance between each cylinder. In addition, 3 systems are positioned at the back of the ship in order to cope with the added weight of the superstructure, shown in Figure 12.

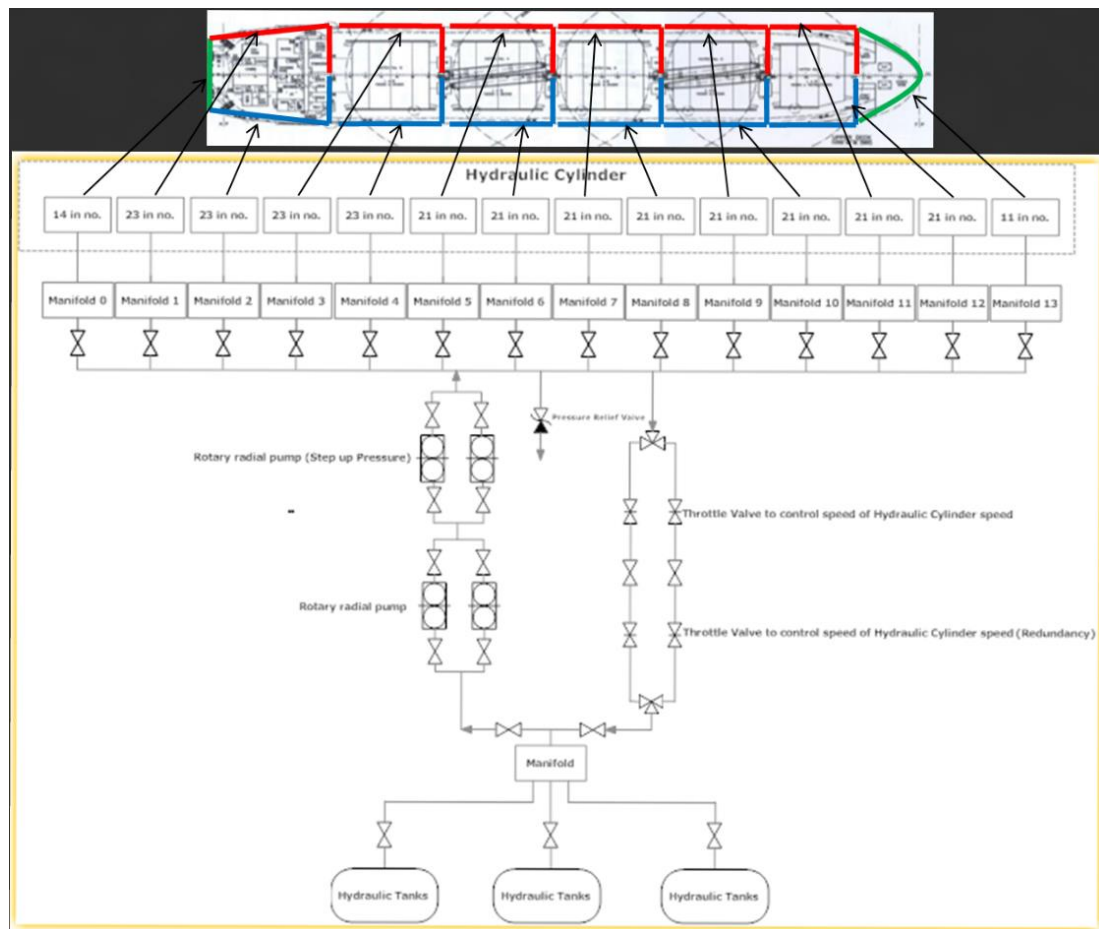


Figure 12: Hydraulic System Schematic Diagram

Due to the fact this design is a brand new concept, it is difficult to find a hydraulic system able to cope with the requirements of the project and therefore, it is necessary to make a new design. The first part of the design involves calculating the pressure within each cylinder. In basic terms, weight and number of cylinders are assumed, the weight is estimated using half the lightship weight and a significant safety factor of 3, this allows for redundancy in all pistons. Then using the area of each cylinder a total pressure of 112bar is found (for more details refer to Annex E, Appendix I).

3.2.2.2 Material Selection

Materials for the piston and cylinder can be found using CES EduPack software. In Annex E, Appendix II it is explained how the material Stainless steel, austenitic, A151 205 is found. Using the selected material cylinder dimensions and piston thickness can be calculated giving the values listed in Table 5. Using the piston thickness calculated the Euler buckling load can be calculated and compared to the yield stress given in the material properties. This shows a safety factor against buckling of 2.11 proving the piston will not fail under buckling.

Item	Thickness (mm)
Cylinder	3.303
Cylinder end plug	12.355
Piston	8.830

Table 5. Material Thickness

Added features to the cylinder are important to the project. The first added feature is the webbed cap at the ends of the cylinder. This is done to limit the point load, meaning the load is distributed more, therefore reducing the risk of any damage to the structure above or below. The next added feature is a pin used to join the webbed cap to the cylinder piston. This means, if the piston fails, there is an easy method of removing the webbed cap. A drawing of this is shown in Annex E, Appendix IV.

3.2.2.3 Redundancy and Reliability

Figure 12 (page 14) shows a diagram of how the hydraulic system is arranged. It shows two high pressure and two low pressure pumps all capable of 100% flow rate. It is arranged in this way so the flow can travel diagonally as well as vertically creating a bypass, consequently a better reliability is achieved as when one pathway fails an alternative is available.

3.2.2.4 Pump Selection

It is now essential to find the flow rate of the hydraulic fluid in order to begin pump selection. This is found using the volume of hydraulic fluid being moved due to the movement of the cylinder. This moving volume is then divided by the time it takes to expand or collapse the structure giving a flow rate of 10.5m³/hr. With the flow rate and pressure in each cylinder the 60 Frame Plunger Pump was selected. Details shown in Annex E, Appendix III.

3.2.2.5 Operating Procedures

This is a completely new concept and no other ship has this system on it, so a new operating procedure must be suggested:

The cylinders should be inspected daily and recorded within the watch log, they should be inspected for wear, corrosion and leakage, which can be done relatively easily by walking down the double hull.

When unloading the ship, the flow rate must be controlled from either the bridge or engine room and all of the pressures within the cylinders monitored. The side of the ship should be brushed down to remove any barnacles or marine growth before collapsing commences.

When loading the ship, all of the cylinders must be monitored and if one is not functioning correctly, the top web can be removed meaning that the cylinder will not be attached to the top section of the ship, this should not be done for two cylinders or more in a row. The cylinder is easily able to be connected again when the ship is in collapsed or expanded state.

Major collapsing and extending the ship should take place within port, this will give greater safety for the ship, it is designed to take 6 hours and cannot be sped up, it can however take longer than that if necessary.

3.3 Water Tight Integrity

3.3.1 Engineering Requirements

3.3.1.1 Positioning

It is important that the vessel is not dependent on one sole system to ensure its water tight integrity which means dividing the ship up into sections. The most logical solution for this is to split the ship up into port and starboard side and then split these into sections. Using the bulkheads as sections, this allows for 6 sections along the port and starboard side with an additional section running the whole length around the front compartment. This compartment only requires one system due to its much smaller size as compared to the other sections. It is important to note that the bulkheads are protected from two independent systems to avoid a breach in watertight integrity. This division is shown in Figure 13.

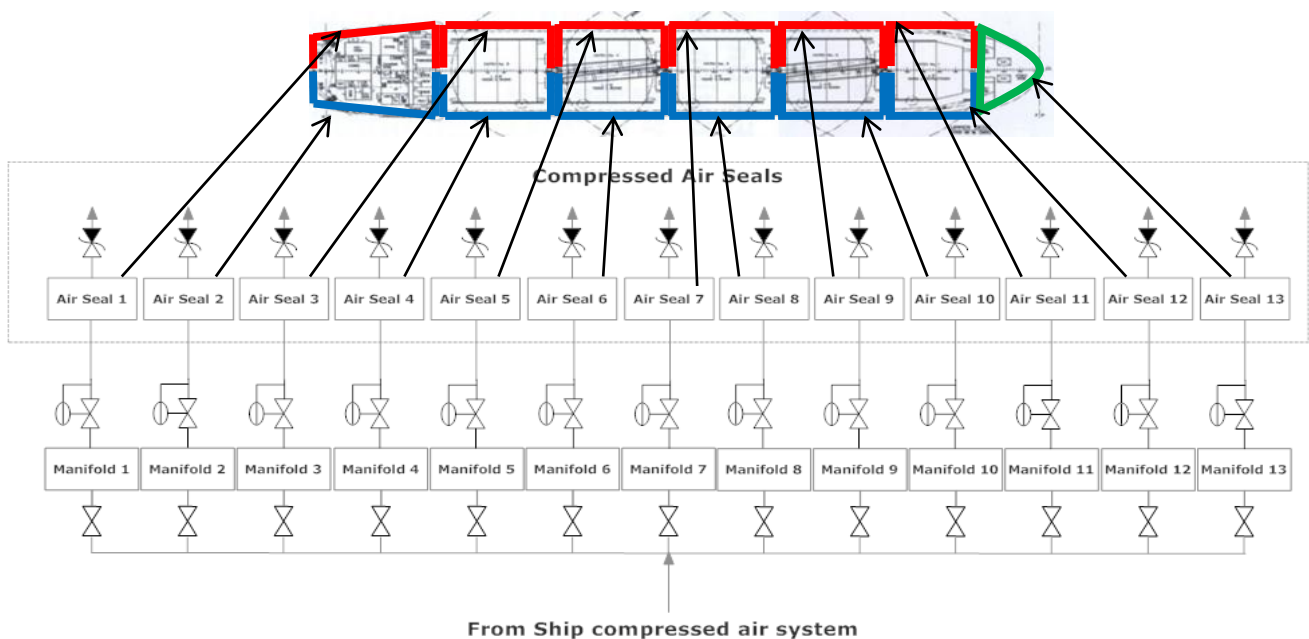


Figure 13. Water Tight Compress Air System

With the ship divided up in this manner, a total of 13 systems will be required. Within these systems lengths of 2m seal will be installed which is a manageable length to enable inspections to be carried out and replacement lengths fitted. The cross section of the collapsed and inflated seals is shown in Figure 14.

With respect to the ship's hull, these seals will be located either side of the stationary part of the hull as shown in Figure 15 This will ensure that there are two systems operating at one place at any one time, with a system on the outside and inside of the hull respectively.

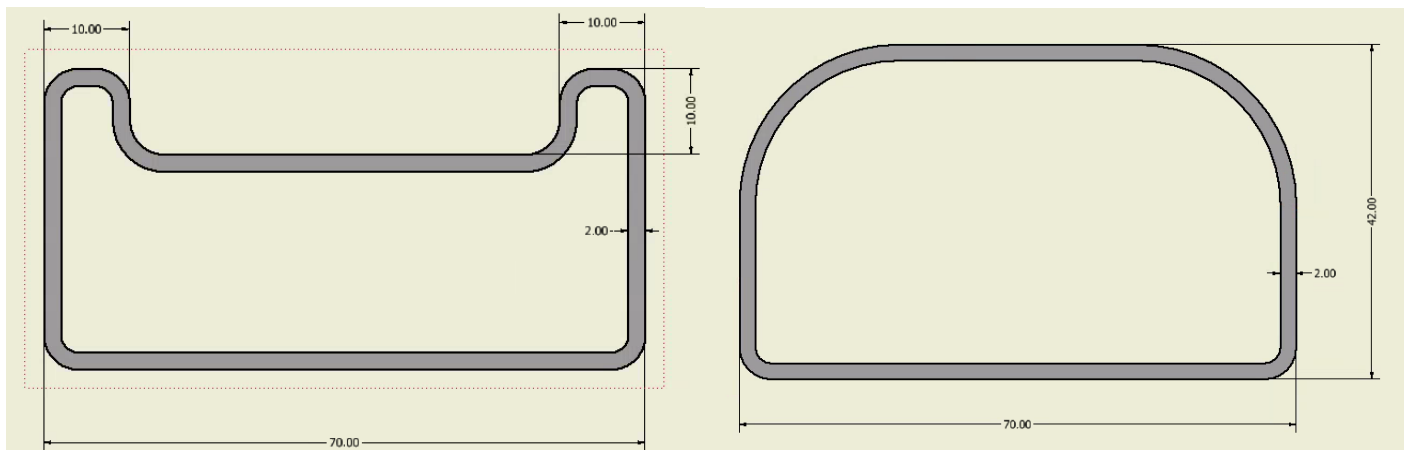


Figure 14. Water Tight Seals Design

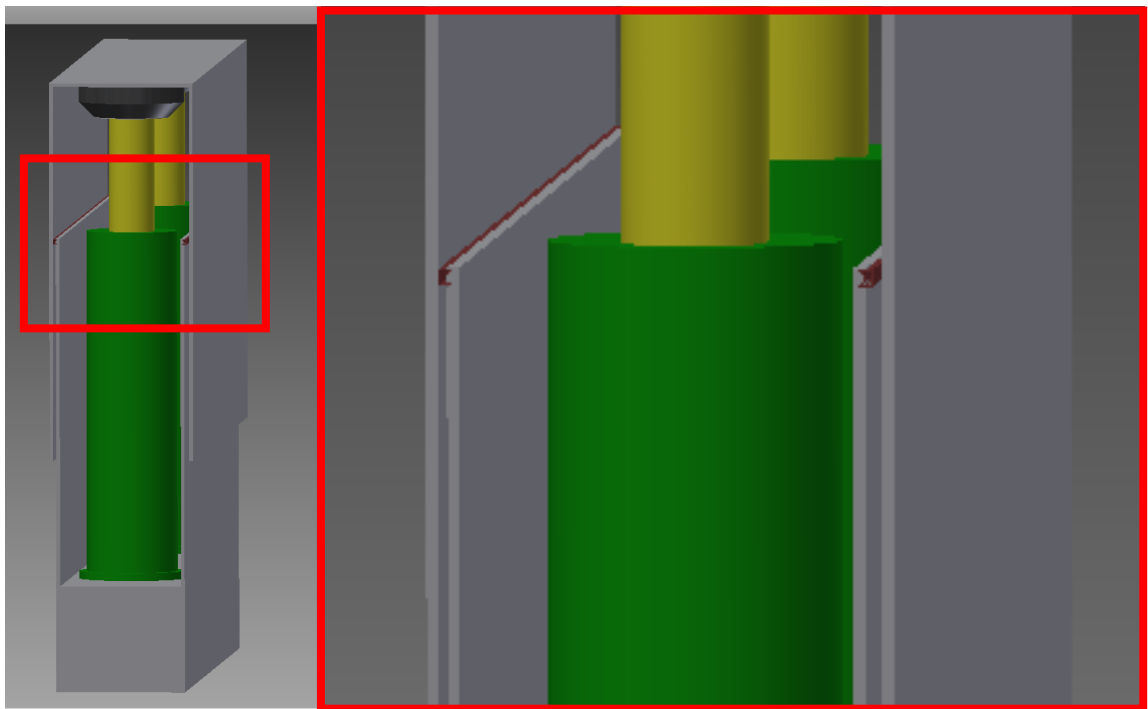


Figure 15. 3D Visualization of Water Tight Seal Position

3.3.1.2 Material Selection

Using CES EduPack (a materials information software), the desirable material properties for the seals could be input to the software which can then able to search for and select a most suitable material for use within the design. A full description of how this is carried out is given in Annex F. In summary, the material chosen is Perfluoro elastomer which was chosen because of its durability in water (both salt and fresh), its durability in oil and fuel and its ability to be injection moulded. Perfluoro elastomer is currently used in extreme chemical environments such as seals in offshore oil supply which reassures the selection of the material.

It is important in this section to include a description of what the hull conditions should be around the seal. It is important that the area around the seal is kept clean and free from any debris that may build up here as this is likely to damage or tear the seal. One method to prevent damage occurring to the seals whilst the steel of the ship moves up and down is to wash around the seal before carrying out this operation, however this is time consuming and costly. Alternatively, this can be done by applying a high specification anti-fouling paint to the area of hull that will be in contact with the seals. A paint that would be suitable for this application could be anti-abrasive paint such that is used currently in the offshore industry designed to protect surfaces subject to unusual abrasive and wear conditions such as pipe lines and drill pipe areas. An example of this paint is Devgrip® 238 produced by Devoe High Performance Coatings.

3.3.1.3 Redundancy and Reliability

As described in the positioning of the systems throughout the ship, 13 sections will be required. This will help the reliability of the concept and help in the case of any redundancies that may occur. This is because if a rip or a defect were to occur then a constant positive pressure will be able to be applied to that specific point in the system by using the two compressors either side of the tear. This will prevent water ingress to the seal and maintain the watertight integrity of the ship.

3.3.1.4 Air Compressor Selection

3.3.1.4.1 Normal Operating conditions

The ship air system will be used to fill the compressed air tank located in the engine room. The distribution of the compressed air is illustrated in Figure 13 (page 16).

3.3.1.4.2 Emergency Operating conditions

There are 3 main considerations to take into account whilst selecting a compressor; these are the type of compressor, capacity of the compressor and working pressure of the system.

Firstly, calculating the capacity of the compressor is needed. All calculations are shown in Annex F and result in a capacity of 590 Cubic Foot per Minute (CFM) which is the common units used for calculating compressor capacity.

Considering the type of compressor, as described in Annex F, there are two types of compressors which are Reciprocating air compressors and Rotary Screw air compressors. The Rotary screw air compressor was chosen as it works at a higher capacity than the Reciprocating compressor which is required as shown by the high capacity of 590 CFM.

Finally, the pressure that the system must operate at must be found. From researching inflatable seal technology, similar systems are filled to 8 bar or 116 psi so this seems a reasonable value to assume such a system would operate at.

These parameters have resulted in a final choice of compressor. Each system will require one of these compressors, so a total of 13 will be needed.

Parameter	Value
Manufacturer	Almig
Type of Compressor	Rotary Screw Compressor
Capacity	152 – 602 CFM
Pressure	75 – 190 psi

Table 6. Emergency Air Compressor Selected

3.3.1.5 Operating Procedures

A basic set of instructions should be written up as to the operation of the new vessel concept, this is as follows:

Collapsing the Vessel

1. Decompress the air in the seals to enable the hull to be moved
2. Inspect the seal around the vessel for any debris that could cause damage to the seal
3. Wash around the seal if required to remove debris or sediment
4. Follow operating procedures for collapsing the top of the ship whilst simultaneously unloading the cargo
5. Inflate the air seals for un-laden voyage
6. Inspect the seals around the ship, checking the watertight integrity

Extending the vessel

1. Decompress the air in the seals to enable the hull to be moved
2. Inspect the seal around the vessel for any debris that could cause damage to the seal
3. Wash around the seal if required to remove debris or sediment
4. Follow operating procedures for extending the top of the ship whilst simultaneously loading the cargo
5. Inflate the air seals for laden voyage
6. Inspect the seals around the ship, checking the watertight integrity

4. Power Requirements and Distribution

There are two power requirements for this concept in the two distinct operating environments, the collapsed case and the expanded case. This means that if possible, a system should be suitable for both operating environments equally.

4.1.1 Empirical Power Requirements

The basis ship, Sea Horse 35, has an installed power of 7.5MW and therefore with an additional factor included, our design ship will require no more than the 8MW of power to be installed. This additional power will be used for auxiliary systems related to the hydraulics such as temperature control. Further power will be required for the movement and pumping of the closed ballast system on board.

According to the International Towing Tank Conference (ITTC57), the total resistance of a vessel is made up of a number of different factors primarily friction and residuary considerations. Using the basis ship, Sea Horse 35, the frictional resistance accounted for roughly 30% of the total resistance. The residuary resistance made up 60% of the total resistance. In the collapsed case the residuary resistance dropped by 10% making it approximately a 5% drop in the total resistance from the expanded case. This means that the power required in the collapsed state is found as the difference between the powers required in the expanded case minus 45% of the ratio in drafts. This assumption has been due to the drop in friction and residuary resistance is the principle cause for the lower power requirements. The outcome of these calculations has led to a total fuel saving of 25.03% in the ships life time assuming an empty backhaul. This is shown numerically in Annex G, Appendix I.

4.1.2 Electrical Distribution

Due to the two operating conditions having very different power requirements, the use of generators supplying an electrical distribution connected to a motor is a viable option thanks to its ability to cope with a range of operating states. This allows for a variable power and rotational speed, while maintaining the engine generators at their optimum speeds enabling a variable power supply to the propulsion system. To allow the efficient variable speed of the electrical system, a converter of some form has to be used. For this an induction motor could be selected as it is a modern component in power electronics and is capable of working within the power and speed ranges while producing very low levels of harmonics into the system which would cause a disturbance.

4.1.3 Propeller Requirements

The propeller for a vessel such as this poses some interesting challenges, it has to be big enough to power the fully laden ship while being able to move slowly enough as to not over power the vessel in the collapsed case. To enable the requirements to be met, a propeller of 5.2m diameter was selected and a blade area ratio of 0.92. This lead to a rotational speed of 222rpm in the expanded case and just 23rpm in the collapsed case, both numbers are suitable and achievable with the selected electrical distribution system.

Another challenge with such a large propeller, and such a small draft in the collapsed case, is to make sure the propeller can spin freely with minimal interference from the hull. This has led to the propeller having to extrude from the aft end of the ship. To make this safe for workers, divers or a man overboard the propeller has to be sheltered, in which case a transom flap or a ducted propeller would be the obvious options. In this instance, a ducted propeller has been chosen as it should help to create a smoother, laminar flow into the propeller. Since the propeller extrudes the aft end of the ship, there is no room for a conventional rudder. This means the propeller must be capable of manoeuvring the ship. To make up for the lack of rudders, it is a sensible idea to use an azipod propeller.

4.1.4 Podded Propulsion

The podded propeller will be able to rotate in order to give maximum manoeuvrability. The figures below are a representation of the podded system in profile view. The aft end of the ship will be more ship shape however the Figures 16 and 17 illustrate the concept rather than a final ship shape design.

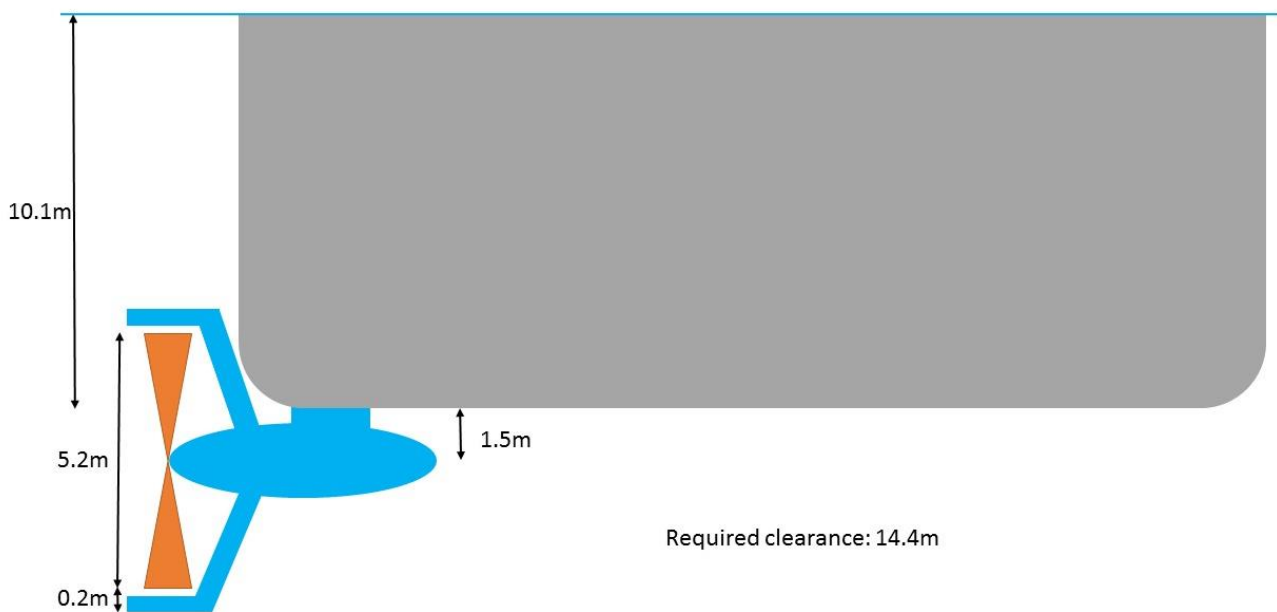


Figure 16. Expanded case podded propeller diagram

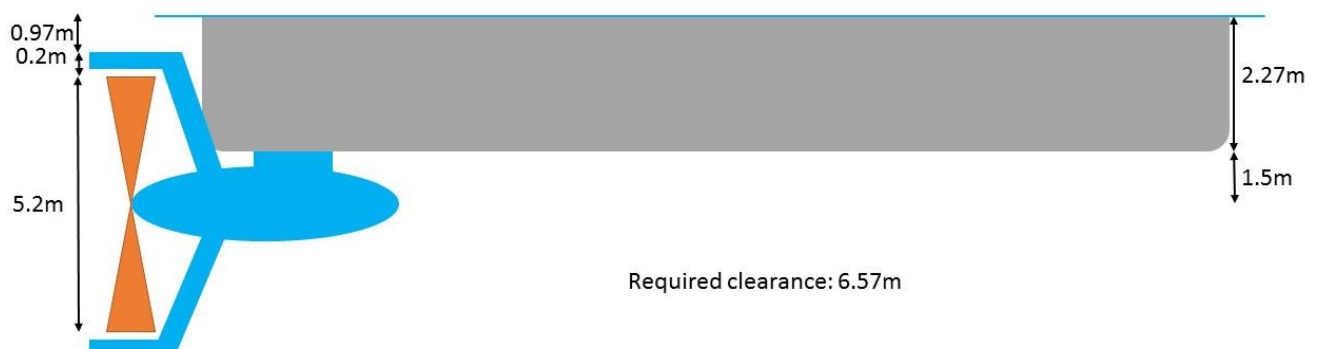


Figure 17. Collapsed case podded propeller diagram

The draft values shown display the clearance of the ship and not the functional draft of the ship affecting stability, sea keeping or resistance.

5. Evaluation

5.1 Economic Evaluation

The estimated additional cost of materials and equipment for this project can be split into five major sections; additional steel mass, hydraulic pumps, hydraulic cylinders, air compressors and watertight seals. The majority of this cost is initial capital cost which will be reduced if brought in bulk or on a regular basis. Some of these materials, such as seals and hydraulic fluids are reoccurring however they are low cost jobs which can be minimised and only occur on inspection or as part of a standard maintenance schedule

The capital costs will be outweighed by the fuel saving due to the power reduction in the collapsed state meaning an estimated saving of \$11,800,000 over the ships 25 year lifetime, this is summarised in Table 7. The capital costs of the items in Table 7 were estimated using basic quotation methods, the cost saved due to fuel are explained in Annex H.

Item	Estimated Initial Cost
Hydraulic Pumps	\$100,000
Hydraulic Cylinder	\$151,000
Hydraulic Seal	\$15,600
Air Compressor	\$786,500
Additional Steel	\$127,000
Total Material Cost	\$1,180,100
Estimated Labour Cost (30% of Materials)	\$354,030
	\$1,534,130
Estimated Fuel Savings over 25 years	(\$13,400,000)
Total saving from capital cost	(\$11,800,000)

Table 7. Summary of economic saving

5.2 Advantages of Design

As shown above in table 7 there is a significant economic benefit to the design of this ship, however there are more than just economic advantages to this design. This project has demonstrated a method that would eliminate the need for ballast water and result in the prevention of transportation of marine species. This will have a substantial positive impact upon marine habitats, and one that must be made to prevent irreversible environmental damage from occurring. With the inevitable implementation of the BWM Convention, ships will require suitable ballast water management treatment systems and this design is a marine engineering solution that eliminates the problem all together.

5.3 Future Improvements

This report details a proof of concept rather than a fully designed and sea worthy vessel therefore there are finer details yet to calculate, however the concept has been proven in this report.

Areas of improvement include:

- More detailed structural and hydrodynamic analysis.
- Various ship shapes including the potential for bulbous bows.
- In depth piping design for hydraulic and pneumatic systems.
- Optimisation of estimates.

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