Liquid Ammonia Fuel Storage Alternatives and Minimizing Integration Impact on Ship Design

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Liquid Ammonia Fuel Storage Alternatives and Minimizing Integration Impact on Ship Design

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MASTER'S THESIS

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List of Abbreviations

AFC	Alkaline Fuel Cell
ASME	American Society of Mechanical Engineers
BOG	Boil Off Gas
BOR	Boil Off Rate
CAGR	Compound Annual Growth Rate
CNG	Compressed Natural Gas
CTO	Chief Technology Officer
EMSA	European Maritime Safety Agency
EU	European Union
FCAW	Flux Core Arc Welding
FW	Flat Walled
GMAW	Gas Metal Arc Welding
GTAW	Gas Tungsten Arc Welding
HCCI	Homogeneous Spark Compression Engine
HDPE	High Density Polyethylene
ICE	Internal Combustion Engine
IGC	International Code for Gas Cargo Carriers
IGF	International Code for Low Flash point Fuels
IMO	International Maritime Organization
LNG	Liquid Natural Gas
LPG	Liquid Petroleum Gas
LPV	Lattice pressure Vessel
MARPOL	International Convention for the Prevention of Pollution from Ships
MARVS	Maximum Allowable Relief Valve Setting
MAWP	Maximum Allowable Working Pressure
MGO	Marine Gas Oil
P	Pressurized
PEM	Polymer electrolyte membrane
RC	Round Cornered
RW	Round Walled
SAW	Submerged Arc Welding
SCC	Stress Corrosion Cracking
SMAW	Shielded Metal Arc Welding
SOLAS	International Convention for the Safety of Life at Sea
SOFC	Solid Oxide Fuel Cell
SP	Semi-Pressurized
TIG	Tungsten Inert Gas Welding
VLCC	Very Large Cargo Carrier

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Abstract

Modern developments in both ammonia internal combustion engines and fuel cells have created the possibility of using ammonia as a carbon-free fuel for propulsion onboard ships. Coupled with these developments has been the recent funding for projects to develop ammonia crackers which offer on-site hydrogen production, enabling the use of a fuel with a high mass energy density without sacrificing large volumes onboard. In this study the requirements for storing ammonia as a fuel are investigated with application to a small passenger ferry. Ammonia, known for its toxicity, needs special containment arrangements and ventilation systems to safely store the substance on manned platforms. These systems affect the space available for ammonia storage in a fixed boundary system and as a consequence may require major changes on a ship to maintain operation specifications. Most scientific research to date has focused on feasibility cases of using ammonia fuel on large ships primarily where ammonia is assumed available from cargo tanks. Storage arrangements, tank types and methods of storage on board have yet to be investigated in detail. In this work, a case study focused on retrofitting a fully electric city ferry to use ammonia as the energy vector instead of lithium-ion batteries. The work offers insight into safety requirements that should be considered, and a detailed comparison of key performance indicators (KPIs) based on tank type and corresponding storage method. KPIs of greatest importance were the mass of fuel that could be stored, the cost of implementing the storage method and the additional space available for extra cargo. Using a multi-criteria analysis seventeen alternatives were analysed to find that for the case under study an arrangement of three horizontal cylinders would provide the most well-suited alternative primarily due to lower cost. For larger ships the use of lattice pressure vessels is highly recommended as an ammonia fuel tank as these tanks reduce sloshing, allow pressurized storage and can be built using relatively cost-effective methods.

1 Introduction

1.1 The Need for Ammonia As A Fuel

The merchant shipping industry began propelled by clean natural wind energy. This wind brought sailors of imperial kingdoms such as the Spanish, Portuguese, Dutch and English across vast oceans to conquer and trade with nations around the world. Since the late fifteenth century, the shipping industry has seen at least three major revolutions: adapting from wooden hulls to steel, wind to steam propulsion and the development of very large cargo vessels running on conventional fossil fuels [1][2], yet now the world is seeing a new revolution returning to using clean renewable fuels.

Technological development in the shipping industry has been driven by the need to travel faster, travel further, or increase the amount of cargo that can fit onboard. In recent times, however, the drive for new technological development is on conserving the environment in line with the Paris Agreement of 2015 to "hold the increase in the global average temperature to well below 2°C above pre-industrial levels"[3]. Ship owners are now under pressure from new International Maritime Organization (IMO) and MARPOL regulations to reduce their greenhouse gas emissions and carbon footprint. Energy efficiency and carbon emissions are mandatory for all ships to report as of 1 January 2023. This data is attained from the Energy Efficiency Existing Ship Index (EEXI) and carbon intensity indicator (CII)[4].

In addition, de-carbonization has received great attention from European Union (EU) policymakers through the recently released "Fit for 55" packages of policies with particular emphasis on the FuelEU proposal [5][6]. The way the EU hopes to cut down carbon emissions is by using the EU ETS (Emissions Trading System) from 2024 in Europe. The goal of the EU ETS system is to reduce carbon emissions by 55% from 1990 to 2030. By implementing this tax system the EU expects ship owners to invest in low-carbon technologies and promote alternative fuels.[7]

As a response to new regulations, the shipping industry has taken to the idea of using cleaner fuels to reduce carbon emissions. In the grand scheme of things, the hope is to reduce carbon emissions to net zero by the year 2050, however, it is not possible to do so with most conventional hydrocarbon fuels. There are a handful of existing fuel options to reduce carbon emissions to acceptable levels yet only two options exist to reach the final goal of net zero emissions.

Ammonia (NH_3) and hydrogen (H_2) qualify as carbon-free fuels, however, due to NH_3's higher volumetric energy density and the extremely low temperatures required to store hydrogen, ammonia is the current preferred alternative of the two. Ammonia is also favoured as there is already an existing global supply infrastructure for it and commercial viability. Notably, hydrogen has a far lower boiling point temperature than ammonia making it difficult to store in liquid condition. Hydrogen also has a wide flammability range from low concentrations in air up to very high concentrations making it a more explosive substance to handle than ammonia[8] [9].

Dual fuel engines present the necessary bridge between using conventional fuels and integrating ammonia into ship propulsion. Internal Combustion Engines (ICE) power 95% of the world's shipping fleet which means that any rapid change to the shipping industry would only be possible by retrofitting existing systems [10]. Anhydrous ammonia is known to be a difficult fuel to combust due to narrow and high flammability limit in air, low flame speed and high minimum ignition energy compared to other fuels which means a pilot fuel often needs to be used to promote combustion. Typically a pilot fuel is mixed into the primary fuel stream in small quantities to raise the flame temperature which facilitates combustion. Table 1 demonstrates the less favourable combustion characteristics of ammonia in comparison to other fuels. The main positives of using ammonia as fuel are its high octane number and volumetric energy density when taking into consideration density.

Table 1: Ammonia and potential pilot fuel properties[11]

	Energy Content (LHV) [MJ/kg]	**Octane Number**	**Flame-velocity [m/s]**	**Flammability - limits [vol/%]**	**Minimum Ignition Energy [mJ]**
Ammonia	18.6	>130	0.067	15-28	8
Hydrogen	120	>130	3.25	4.7-75	∼0.016
Diesel (n-dodecane)	44.11	<20	∼0.80	0.43-0.6	∼0.23
Gasoline (iso-octane)	44.34	100	0.41 ∼0.58	0.6-8	1.35 ∼0.14

The alternative to using combustion engines and pilot fuels is to use fuel cells to produce electricity by consuming hydrogen. There is a great deal of research going into developing a fuel cell technology that can use ammonia directly in the fuel cell without first having to crack the ammonia into hydrogen and nitrogen. Although much research is being conducted into direct ammonia fuel cell technology, the implementation on existing ships would require further space onboard to accommodate large fuel cell modules and heavy battery storage[12][13][14]. Combustion engines are favoured over fuel cells for larger ships as they have a high power density, load response, robustness, and lower cost[15]. Direct Ammonia Fuel cell technology is still under development at this stage and has not seen the technological readiness level of internal combustion engines for commercial applicability[12].

The use of PEMFC and AFC fuel cells which use stored hydrogen as a primary fuel source, on the other hand, have been developed and tested by the automotive industry[16]. In 2013 Hyundai launched the *ix35 Fuel Cell*, the first mass-produced hydrogen fuel cell vehicle in the world[17]. This kind of technology is promising for smaller vessels where vessel range is not an important key performance indicator (KPI). In the maritime industry there has also been demonstrated proof of feasibility for PEMFC in projects such as the NEMO H2 and FCS Alsterwasser projects[18].

To improve the range of these vessels hydrogen needs to be stored in a different form to enable greater quantities of energy per unit volume. One such form is to use ammonia as the hydrogen carrier. Ammonia can be cracked under high temperatures to produce hydrogen which can then be used for fuel cells or dual-fuel engines on board. The main problem with using ammonia as a fuel on vessels is that the current regulations set by IMO International Code for Low Flashpoint Fuels (IGF) and IMO International Code for Gas Cargo Carriers (IGC) do not explicitly recognise ammonia as a fuel source onboard ships due to its high toxicity compared to other liquefied gas fuels[19]. In response, classification societies have put together tentative rules that allow for ship designs to be approved in principle (AiP)[20].

1.2 Ammonia Fuel Storage On Vessels

Ammonia may not be recognised yet as a fuel by the IMO but that does not mean ammonia has not been carried on ships before. Ammonia storage as cargo is well documented in the IGC guidelines which forms a good baseline for standards and regulations. Storage solutions for ammonia on land have been present for many years thanks to the fertilizer industry and as a component of many cleaning products. Ammonia has also been transported by ship in large vessels for decades by modified LPG carriers, however, the storage of ammonia onboard as a fuel for propulsion has only become a topic of great research and development in the last decade.

The new risks foreseen for ammonia stored as a fuel are the increased mobility of ammonia, the more spaces ammonia will be present onboard the vessel, and the additional ventilation required. All these risks must be carefully evaluated and mitigated as necessary. Recognised existing studies quantifying risks of installing ammonia fuel supply systems onboard vessels are masters theses from TU Delf students N. De Vries in 2019 [21] and L.N.Henderik in 2020 [22]. The former focused on an ammonia-fuelled 54000

DWT ammonia carrier and the latter focused on an ammonia-fuelled solid oxide fuel cell (SOFC) propelled small vessel. These studies described how different propulsion systems could be used for powering a ship with ammonia, however, they neglected to mention anything about the fuel tank alternatives. The study by De Vries proposed a layout for an ammonia-hydrogen dual fuel ICE fuel system concluding that this propulsion option would be most suitable for the large ship under study where the fuel was assumed available from a large cargo tank. Henderik gave a detailed feasibility study of using a SOFC for a small vessel concluding that the technology level of SOFCs in 2021 was not suitable for the vessel under study.

The best-known study relating to alternative storage arrangements for ammonia fuel tanks is that of the European Maritime Safety Agency (EMSA) published in 2022 [23]. This report highlighted many risks of storing ammonia as fuel but did not elaborate well on why certain storage tanks would be better than others. Instead, the focus was on the location of the tanks and the risk analysis associated with the positioning of the tanks. As mentioned in the EMSA report ammonia can be stored using one of three methods: refrigerated, semi-pressurized (likewise, semi-refrigerated) or pressurized. The refrigerated method is usually used to store ammonia in large quantities at low temperatures below -34 ° C. The pressurized and semi-pressurized alternatives are usually used to store ammonia in smaller quantities at higher temperatures and pressures for transportation on land by road or rail. The EMSA performed a case study investigating integrating ammonia fuel tanks into three large ship types using general type C cylindrical tanks or a rectangular-shaped type A tank.

1.3 Motivation and Objectives

The knowledge gap identified is the need for a study which focuses on the design of fuel storage systems which minimise integration impact on a ship taking into account the use of new dual fuel engines or fuel cells. Previous efforts in tank placement and selection have been focused on high-level system layouts for large ships yet the applicability of ammonia propulsion technology to smaller vessels has largely been neglected. Current feasibility studies do not specify how different methods of storage (refrigerated, semi-pressurized or pressurized) can alter general arrangements. There is still a degree of uncertainty about which types of ammonia fuel tanks should be used onboard, the rules surrounding ammonia storage and the fuel supply systems. Suggestions by the EMSA indicate that similar fuel handling systems to existing LPG and LNG fuelled carriers may be modified to integrate ammonia. These suggestions from the study need to be investigated further from a more technical point of view.

In this master's thesis, the aim is to find various storage alternatives for using ammonia as a fuel onboard vessels that meet the IMO IGF criteria for alternative design and coupled to that investigate how the corresponding fuel system can be integrated into a ship with minimum impact on human safety and operational characteristics of the vessel. A new topic addressed in this study is the use of lattice pressure vessels as an ammonia fuel tank. This new technology is evaluated and ranked against other known ammonia fuel tank alternatives. The case study not only looks at general arrangements but also pieces together the necessary elements for the ship fuel system to function effectively. The integration impact can be measured in terms of volume, mass, cost and safety.

There are three primary objectives of this master's thesis and one secondary objective. The first of the three objectives is completely research-based and forms a requirement to gain the knowledge to accurately formulate solutions of how to store ammonia.

Primary objectives

- Establish limitations and complexities of storing ammonia safely for a manned platform including existing regulations.
- Minimize platform integration impact of energy vector.
- Maximize energy density on-board per volume

Secondary objective

- Investigate using alternative materials for ammonia tanks.

1.4 Structure of the work

This master's thesis is split into five chapters with several appendices to document calculations and drawings.

- Chapter 1: Introduction
  This chapter gives an overview for the importance of this study and provides motivation and objectives of the work. It highlights why ammonia is in demand as a fuel and gives a foreshadowing of what is to follow later in the thesis.
- Chapter 2: State of the Art
  This chapter culminates the work of various authors from different fields in a literature review to gain a firm understanding of what ammonia is and how it is used currently. The chapter is divided into a series of subsections which focus on different elements contributing to the main aim of the thesis. The chapter starts off

with ammonia´s physical properties and toxicity risks to humans and the environment (section 2.1.2). Following this, there is an in-depth study into how ammonia is usually stored on land (section 2.2) and on ships (section 2.4), the regulations surrounding its storage and how ammonia storage compares to the storage of other known gas fuels. Finally, the chapter looks at the various ammonia fuel consumers likely to be found on an ammonia-fueled ship which are important for quantifying fuel consumption and the integration impact of using ammonia as the energy vector.

- Chapter 3: Case Study
  This chapter states the methodology for tank selection and system design. The methodology is applied to a case study of a self-propelled ferry with the ability to carry ammonia as an energy vector. The scope is narrowed to focus on a dual fuel HCCI combustion engine in section 3.1 in an attempt to evenly compare the fuel tank options. The scope of study encompasses all potential tank designs recognised by IMO codes and pressure vessel standards including lattice pressure vessels.

- Chapter 4: Results and Discussions
  Relevant KPIs identified from the case study are computed and key findings are recorded in results tables. A multi-criteria analysis helps to decide the best tank alternative based on a series of analyses. Discussions on the most suitable tank types and applicability to other ship types are subsections in this chapter. Discussions on regulatory development and regulatory gaps conclude the chapter.

- Chapter 5: Conclusions
  This final chapter summarizes all that was discovered from the literature review and the case study, highlighting the important points. Ending off the thesis with recommendations for future work.

Calculations given in the appendices are primarily to estimate the size of equipment for the general arrangement. The focus is solely on the storage and distribution of ammonia to the propulsion system. The propulsion systems currently commercially available on the market are at most limited if not still in development therefore, where necessary, assumptions for operational characteristics and specifications are estimated based on scientific research.

2 State of the Art

This section aims to cover what has been previously researched in the field of ammonia storage, what is known about ammonia as a substance, the current best practices for the storage of ammonia in a number of different industries, and how ammonia is used as a fuel source. This section focuses primarily on scientific research and developments covered in the past five years centred around ammonia-propelled ships. The information generated from this study is foreseen to aid the development of new solutions to the primary objectives of this work.

2.1 Ammonia Properties and Complexities

The physical properties of ammonia are important to know when designing a robust,safe and maintenance-free system capable of containing the substance. Ammonia is similar to many other low flash-point marine fuels such as LPG, LNG and hydrogen with the added danger of high toxicity. This section serves to gain an understanding of the challenges and advantages of using ammonia as a fuel.

2.1.1 Forming process

Ammonia has been produced commercially for over 75 years using the Haber-Bosch Process[24][25]. This process synthesises nitrogen and hydrogen in an exothermic reaction to produce NH_3. The hydrogen component can be produced in many ways, however, one method to ensure that well-to-wake carbon emissions remain at 0% is to produce hydrogen through electrolysis which proves as an advantage to ammonia in relation to other alternative fuels. The nitrogen component is freely available via the separation of air molecules. After the ammonia gas is formed it needs to be compressed or refrigerated to produce liquefied ammonia which can be stored for use in the industry[23].

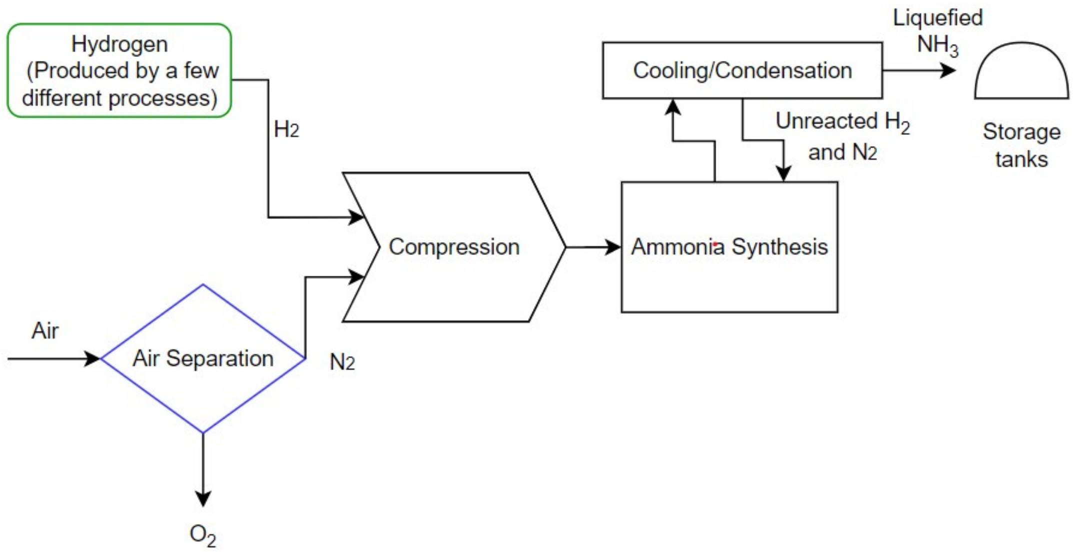


Figure 1: Haber-Bosch Process Diagram

2.1.2 Ammonia Properties

The table below summarizes the properties of ammonia in comparison with Marine Gas Oil (MGO) as stated by the European Marine Safety Agency. [23]

Table 2: Ammonia Physical Properties in Comparison to MGO[23]

Item	**Ammonia**	**MGO**
Energy density by Volume (MJ/L)	12.9	35.95
Lower Heating Value (LHV) (MJ/kg)	18.8	42.8
Heat of vapourisation (kJ/kg)	1371	250-450
Autoignition temperature (ºC)	651	250
Liquid density (kg/m3)	696 (at -33 ºC)	840 (at 15 ºC)
Adiabatic flame temperature at 1 bar (ºC)	1800	2000
Molecular weight (g/mol)	17.031	54
Melting point (ºC)	-77.7	-26
Boiling point (ºC)	-33	154
Flash point (ºC)	132	60
Critical temperature (ºC)	132.25	654.85
Critical pressure (bar)	113	30
Flammable range in dry air (%)	15.15 to 27.35	0.7 - 5
Minimum ignition energy (mJ)	8	0.23
Cetane number	0	40
Octane number	~130	15-25

Ammonia is easily compressible and usually transported in liquid form in steel tanks [26] but can be hazardous (see definition of hazardous in appendix J) due to its explosiveness at high heat and toxicity (addressed in detail in section 2.1.4). The substance is sought after as fuel not because it outperforms MGO but instead for its higher volumetric energy density in comparison to liquid hydrogen (9 MJ/L)[27]. Ammonia has a flammability range of 15.15% to 27.35% in dry air and can be extinguished using dry chemicals,CO_2 or water spray (precaution must be taken to prevent runoff)[28]. The lower flammability limit translates to about 150 000 ppm which is significantly higher than the upper tolerable human exposure limit mentioned later in Table 3. Ammonia has a relatively low energy density of 22.5 MJ/kg when compared to 55 MJ/kg of natural gas and 45 MJ/kg of Diesel[24], however, the key that makes ammonia so appealing as a fuel is its high octane levels which make it interesting for use in combustion engines, discussed further in Section 2.1.3.

It is important to understand the phase diagrams of ammonia and how its density changes due to temperature and pressure in the gas and liquid phase. Later these phase diagrams will be used as a reference to design storage solutions for ammonia.

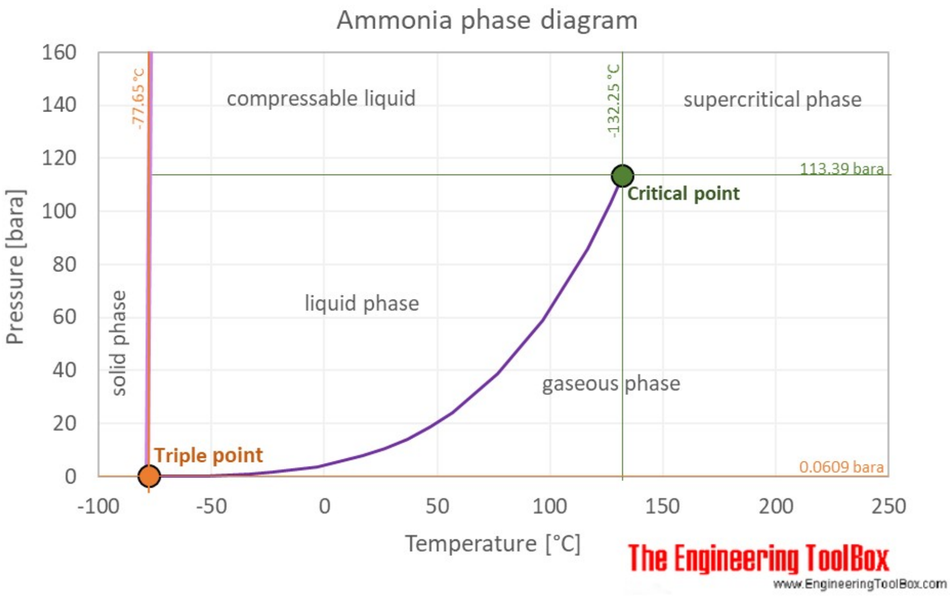


Figure 2: Ammonia phase change diagram[29]

It is quite clear that if maximum expected ambient temperature is 45 °C [30] then the required storage pressure is somewhere between 16-20 bara (approx 1.8 MPa) for fully pressurized systems. Under fully pressurized conditions ammonia is expected to have a density of +-570 kg/m^3 (indicated in Figure 3) which is much lower than a density of 693 kg/m^3 for fully refrigerated conditions of -34 °C and atmospheric pressure.

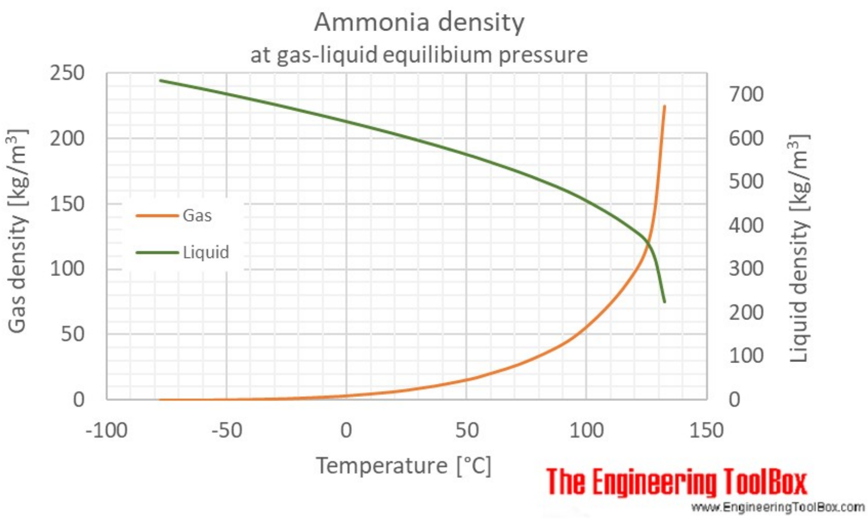


Figure 3: Ammonia density change according to temperature[29]

Stress Corrosion Cracking

One of the biggest concerns with storing ammonia is its incompatibility with various industrial materials. It is a reducing agent with the ability to react with acids, halogens and oxidising agents. Stress corrosion cracking may become present when carbon manganese steel comes into contact with ammonia contaminated with excess oxygen, oxygen acts as the oxidizing agent to cause corrosion. In addition, ammonia is very corrosive to metals such as nickel, nickel based steel, zinc, mercury, copper and cadmium[31], therefore galvanized steel or copper tubing may not be used in ammonia systems. IGC rules (17.12.6) state that for ammonia carriers nickel steels with a nickel content lower than 5% should be used to limit stress corrosion cracking (SCC)[32]. Steels with up to 5% nickel may be used when the carriage temperature of ammonia is below -20 °C (IGC 17.12.7)[30].

Stress corrosion cracking can be modelled as a function of water content and oxygen content. Ideally if oxygen concentration increases then so should the water vapour concentration to avoid stress corrosion cracking. There are two problems with adding water to the stored ammonia tank, one is that it generally remains in liquid phase so the gas part of the tank will still corrode and secondly since ammonia will be used for combustion, a high water content in the fuel is not favorable. Figure 4 demonstrates SCC behaviour with varying oxygen and water concentrations. Generally, when oxygen content is above 0.5% the water content should be above the green limit line to prevent corrosion from developing.

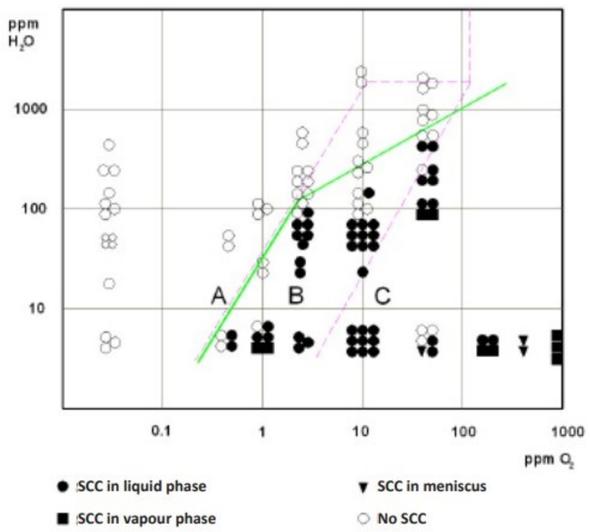


Figure 4: Stress Corrosion Behaviour in Carbon Manganese materials [33]

2.1.3 Ammonia Combustion

Ammonia combusts to form water and nitrogen under ideal conditions. The difficulty is to reduce the production of nitrogen oxides under realistic conditions which form as a result of burning ammonia in excess oxygen. Nitrogen dioxide is a greenhouse gas three hundred times more environmentally damaging than CO_2 if not handled correctly[34]. The decomposition of ammonia into nitrogen oxides can be complex and may be determined by specific fuel composition and thermodynamic conditions. The basic balanced chemical equations according to Erdemir et al [35] are shown below for 100% oxygen (Equation 1) and excess oxygen (Equation 2):

$$NH_3 + 0.75(O_2 + 3.76N_2) \longrightarrow 3.32N_2 + 1.5H_20 \tag{1}$$

$$NH_3 + [X(0.75)](O_2 + 3.76N_2) \longrightarrow [1.5]H_2O + [2.82X + 0.5]N_2 + [(X-1)\cdot 0.75]O_2 \tag{2}$$

Where X is the percentage of theoretical air. As said these are the basic equations for NH3 combustion reaction. Equation 3 represents the full combustion of ammonia to form nitric oxides [35].

$$\begin{aligned}[Y]NH_3 + [X\cdot Y(0.75)](O_2 + 3.76N_2) \longrightarrow [1-C\cdot Y]NH_3 + [Z\cdot C\cdot Y]NO_x \\ +[1.5Y - [1.5\cdot[1-C\cdot Y]]]H_2O + [([1-Z]/2)\cdot C\cdot Y]N_2 \\ +[[(X\cdot Y\cdot 0.75\cdot 2) - (Z\cdot C\cdot Y\cdot X) - [1.5\cdot Y - (1.5\cdot(1-C\cdot Y))]]/2]O_2\end{aligned} \tag{3}$$

The more complex reaction forms NO_x and is dependent on X (percentage of air), Y

(Number of input moles of NH3), C (Percent conversion of NH3) and Z (percent of NOx formation).

The planned approach by MAN to reduce NO_x production is to use SCR (Selective Catalytic Reduction) systems to capture nitrogen oxides and un-burnt ammonia turning them into harmless nitrogen gas and water[36]. It is understood that these SCR systems require a significant space allowance so application on very small combustion systems such as motor vehicles has been said to generally not be feasible. The basic equations for catalytic reduction of NO_x gases are as follows [37]:

$$4NO(g) + 4NH_3(g) + O2 \longrightarrow 4N_2(g) + 6H_2O(l) \tag{4}$$

$$2NO_2(g) + 4NH_3(g) + O2 \longrightarrow 3N_2(g) + 6H_2O(l) \tag{5}$$

$$NO_2(g) + NO(g) + 2NH_3(g) \longrightarrow 4N_2(g) + 3H_2O(l) \tag{6}$$

From Equation 4 to Equation 5 it is evident that a small portion of the ammonia will be required to be diverted to the SCR to reduce NO_x production which will add to the storage space required for the ammonia fuel when a dual fuel engine is used onboard. The exact amount of additional fuel is a function of the engine's combustion characteristics. The tests for these types of engines burning ammonia are currently under study therefore with future NO_x emission data this additional amount of fuel can be quantified[37]. Another problem with using ammonia in combustion engines is the high auto-ignition temperature. This significantly high auto ignition temperature (as seen in Table 2) means that ammonia will not ordinarily combust in a compression engine or by regular spark ignition. The solution is to use a pilot fuel to start the combustion process. The behaviour of the combustion engine varies greatly dependent on the type of pilot fuel used and mixture ratio of pilot fuel used. Most favorable in terms of reducing carbon emissions from tank to wake is to use hydrogen. Hydrogen has a low auto-ignition temperature and can be produced by cracking the ammonia already stored onboard. Although using hydrogen eliminates the creation of CO2 it promotes the creation of larger quantities of NO_x in the exhaust streams due to higher flame temperatures[38]. Therefore SCR systems are required to avoid NOx emissions.

A stoichiometric equation used by Lhuiler et al.[38] to model the combustion of ammonia and hydrogen mixtures is as follows:

$$\begin{aligned} &(1 - x_{H_2})NH_3 + x_{H_2}H_2 + \frac{3 - x_{H_2}}{4}(O_2 + 3.76N_2) \longrightarrow \\ &(\frac{3 - x_{H_2}}{2})H_2O + (\frac{1 - x_{H_2}}{2} + 3.76 \cdot \frac{3 - x_{H_2}}{4})N_2 \end{aligned} \tag{7}$$

Where x_{H_2} is the molar percentage of hydrogen in the mixture. Furthermore, simplified modeling of ammonia and hydrogen combustion can be achieved by separating the two reactions into the ammonia part and the hydrogen part. Such a case was done in the study by De Vries [21]. The equations are as follows:

$$4NH_3 + 11N_2 + 3O_2 \longrightarrow 13N_2 + 6H_2O \tag{8}$$

$$2H_2 + 4N_2 + O_2 \longrightarrow 4N_2 + 2H_2O \tag{9}$$

2.1.4 Risk to Humans and Toxicity

In order to safely store ammonia, one must know its chemical and physical properties. Ammonia at ambient temperature is naturally a colourless, dangerous gas to humans. It is known to cause skin irritation along with having a strong suffocating odour [39]. Ammonia is commonly encountered by humans in the form of ammonium hydroxide after it has been exposed to water when used in house cleaning products. Due to the hygroscopic[1] properties of ammonia it easily dissolves in water to become ammonium hydroxide which is a weak base yet in high concentrations is still dangerous to humans. Notably in a presentation on hydrogen and ammonia safety(Drager,2023) it was mentioned by presenter Hine that in the event of an ammonia leakage in more humid environments, there would be a tendency to form a low-lying toxic cloud of ammonia. This cloud could sustain levels of ammonia of 20 000 ppm which is far above the safe human exposure limits. The human exposure limits to anhydrous ammonia are shown in Table 3 below where the long-term exposure is over 10 hours and the short-term exposure is over 15 minutes[40][41]. Exposure to concentrations of anywhere between 2000-3000 ppm can be fatal to humans within 10 minutes[23][21].

Table 3: Exposure limits to ammonia[40][41]

Region	Legislation	Long term exposure limits		Short term exposure limits	
		mg/m3	ppm	mg/m3	ppm
European Union	OEL	14	20	36	50
USA	NIOSH	18	25	27	35

Ammonia is recognised as a toxic corrosive gas and if in direct contact with humans can result in severe injuries or death. Workers in direct contact with the substance should always wear the correct personal protective equipment. In zones containing high levels of ammonia or in the event of a leakage the area should be well ventilated to prevent explosion risk. Humans can detect ammonia in concentrations as low as 5 ppm meaning that in case of a leakage, any workers in the nearby vicinity of the leak should be able to detect ammonia and immediately take preventative safety measures to avoid long-term exposure[28]. Another favourable characteristic of ammonia is when it is released into the

[1] Easily absorbs water

atmosphere it becomes a gas with a density less than air and thus dissipates rapidly in an open space [9] given the air is of low water vapour content. In closed spaces, however, special care should be taken to avoid an accumulation of ammonia in areas near the ceilings and adequate ventilation should be used.

A study by Christensen et al. in 2005 showed the possibility of storing ammonia in metal amine complexes to reduce the toxicity risk of pure ammonia in the event of an accident. This method ensures that ammonia will only be released at temperatures above 350°C [42].

Modelling of the gas dispersion in a ship engine room was completed by Yadava and Jeong in 2022 to perform a safety evaluation in the event of an ammonia leak onboard an ammonia-fueled vessel[15]. The model simulated several different scenarios in which a hose rupture occurred, notably the dispersion cloud change characteristics given if the leak was a vertical or horizontal stream flow. Key observations were that within 2 minutes anybody inside the engine room would need to evacuate before exposure limits became critical. The simulation of the pooling of ammonia at the ceiling showed if current ventilation systems would be appropriate to prevent fire and explosion risks. It was noted that further work needed to be done to account for different scenarios and leakage types to fully access risks in the engine room or other important compartments in the ship containing ammonia[15]. A dispersion study was also presented by DNV in 2021 in which two ammonia bunkering situations were analysed for a passenger vessel. The first of which is through a ship-to-shore connection and the second of which is ship-to-ship. The PHAST modelling software was used to predict the affected areas. In this study, it was concluded that the ship-to-shore connection had a greater potential for widespread affected zones[43].

Ammonia Detection

Ammonia leakage can be detected by a number of apparatus. The most prominent in the industry is known as the catalytic bead detector[44]. This detection device can be calibrated to detect levels of ammonia as low as 10 ppm. The device is said to need service checks every 6 to 10 months to prevent sensor poisoning. These units cost approximately 1400 to 1600 EUR per piece (March 2023)[44].

Another method is to detect leaks using acoustic monitoring devices. Such devices have a listening time of 9 seconds before raising an alarm in which the high-frequency noise of a pressurized gas leak can be detected. These systems are not known to be able to pinpoint a failure but rather to alert the user that there is a potential leak. They can also only be used on high-pressure systems.

2.1.5 Risk to the Environment

Ammonia combustion in internal combustion engines is a current topic of interest around the world which has a low technological readiness level for commercial use[37][45]. As a result, there is not a great deal of emission data to confidently model the effect of ammonia engines on the environment. Estimated levels of nitrogen oxide emissions come from theoretical equations and benchmark engine tests[36][46] but actual data may only be obtained after many more tests. After a few years, the true environmental impact will be able to be assessed[47].

On the other hand, the threat of ammonia spillage from tanks or piping has been studied in quite great detail on land. The effects of ammonia in water have primarily been focused on coastal areas and freshwater systems. Samie Parkar of Lloyd's Register presented his findings of modelling ammonia spills into water in February 2023. His research focused on three different cases, namely: a container ship with fully refrigerated storage, a bulk carrier with pressurised storage and a tanker with semi-refrigerated storage. He modelled two different scenarios: one in the case of a tank failure in the event of a collision and the other in the event of bunkering leakage using Process Hazard Software (PHAST), with Raj and Reid model for ammonia and water interaction. In these models, he included multiple different weather scenarios. He concluded that the worst-case scenario was a hole in the tank due to collision, however, the probability of this was extremely low therefore more likely would be a ruptured bunkering line with low wind conditions to disperse the ammonia. It was noted that further simulations should be done with changes in PH levels and salinity of the water[48].

US Coastguard conducted tests in the 1970s to test the dispersion of ammonia in water. They tested in a lab, in a swimming pool and in a lake. They discovered boiling ammonia forms on the surface and 70% disperses into the water[47].

In terms of fauna and flora, ammonia has the most detrimental effect on fish as it cannot be excreted via their gills in high quantities,causing mortality either directly from the ammonia or later due to nitrite poisoning (brown blood disease)[49]. In most aquatic environments the release of ammonia leads to an increase in algal growth and biochemical oxygen demand which is generally not healthy for the environment. Some bacteria convert ammonia to nitrites (harmful to fish) whilst others convert nitrites to nitrates (not harmful to fish). These nitrates can be absorbed by plants, however, excess ammonia in an ecosystem can lead to an imbalance which starves certain organisms of oxygen. Notable is if the concentration of chloride in water is higher (higher salinity) there is less of a tendency for the fish to absorb nitrates through their gills, thus an ammonia spill could possibly be worse for fish in fresh water than in seawater [49]. A recent example of how sensitive river ecosystems can be to algae blooms was the Oder river mass fish deaths of September 2022, although the initial cause was not clear, it is evident that introducing an imbalance in the fresh water ecosystem can lead to disastrous effects [50].

Birds ,reptiles and marine mammals also suffer from physiological effects[48].A study by Franklin and Edwards in 2019 on the effects of ammonia on sea fishes revealed that the concentration of non-ionized ammonia increases with water temperature[51]. In general, when ammonia dissolves in water, non-ionized ammonia and ionized ammonia exist in equilibrium, the latter of the two is not toxic to fish[52]. A method to combat the spill of ammonia would be to spray a mild acid to dissolve the ammonia into its ionized form, thus reducing the pH to 6 resulting in less than 0.1% of the ammonia remaining in toxic form[51].

2.2 Land Based Ammonia Storage Solutions

Before the use of ammonia as a fuel, it has been widely produced and used in the fertilizer industry, as a refrigerant, a component of cosmetic products and in many household cleaning products. This prior knowledge is a great advantage to build modern solutions for using ammonia as a fuel in shipping. Traditionally ammonia has been transported by ship, rail and road using a few different storage solutions.

2.2.1 Fixed Storage Tanks

80% of all ammonia produced on earth is used for fertilizer products[53]. The fertilizer industry has arguably pioneered the storage of ammonia for the rest of the world. Ammonia is used primarily in the production of ammonium nitrate, a more explosive substance than pure anhydrous ammonia, used to enhance plant growth. The consequences of improper storage of this substance was made undeniably clear by the Beirut explosion in August 2020. Ammonia has been used as a common refrigerant gas due to its ability to absorb heat. Consequently, the handling of ammonia in piping systems is well documented and practices for safe use are well established.

The common practice for storing ammonia on land is to store it in large quantities using refrigerated tanks (up to 50000 tonnes). For this purpose tanks have traditionally been manufactured from low temperature carbon-manganese steel with a low tolerance to corrosion. Common design standards used are API 620 and more recently API 625 while API 2000 is also used for pressure relief standards.

For refrigerated storage on land there are generally 2 types of tanks: Single-wall steel tanks with external insulation and double walled tanks with perlite insulation in between the walls. The former often having concrete rings surrounding it to contain the entire contents of the tank. Double walled tanks can be further defined as double walled double integrity tanks (DWDI) with insulation either around the outer tank or in the annular space between tanks. Insulation on the outer walls generally allows for a longer tank lifetime but often costs more than a tank with internal insulation. Generally single walled tanks have been discontinued due to the higher risk levels as opposed to DWDI tanks. Commissioning requires purging the tank with nitrogen until there is less than 4% oxygen by volume thereafter purging with ammonia vapor until oxygen levels are below 0.5%. Ammonia is then injected to the tank at a low cooling rate of less than 2°C per hour using a spray system[54].

2.2.2 Transportable Tanks

Ammonia has long been transported by rail and road. Fertilizers Europe[33] issued a set of guidance rules for transporting ammonia by rail in 2007. The document states that ammonia is normally transported in cylindrical pressure vessels in amounts of 50-110 m^3 of ammonia. Ammonia rail tanks are usually designed for pressures up to 2.6 MPa but normally operate at pressures in the region of 0.5-1.2 MPa. It is said that even in the event of a derailment or collision the probability of a tank rupture is extremely low. Special care must be taken not to use any copper materials as ammonia is highly corrosive. The International Carriage of Dangerous Goods by Rail (RID) sets common standards for international transportation of ammonia. Mentioned is that the tank shell should be made of suitable metallic materials which shall be resistant to brittle fracture and to stress corrosion cracking between -20° and +50°C under regulation RID 6.8.2.1.8. This was illustrated previously in Figure 4. Additionally, nitrogen purging is only required on first use of the ammonia tank or if another gas is to be stored in the tank [33].

The United States Department of Transport issued an advisory on the safe use of anhydrous ammonia nurse tanks in 2008. These tanks are horizontal cylinders mounted on trailers primarily used by farmers for dosing crops directly with anhydrous ammonia. The tanks usually store a maximum of 3000 gallons (11.35 m^3) of ammonia under a design pressure of 250 psi (1.7 MPa) designed for temperatures in the range of 125 ° F (51 ° C)[55].

Figure 5: Ammonia Nurse Tanks application as a fertilizer distributor[56]

Nurse tanks have been used for many years therefore there is a history of design development as a result of faults. What should be evaluated are the events where failures did occur and why. A study by Russel et al in 2014 highlighted four major accidents involving pressurized transportable ammonia tanks where stress corrosion cracking contributed to tank rupture [57]. The first incident in 2003 (Calamus,Iowa Incident)[57] involved a pressure vessel designed to store ammonia at 250 psi (1.7 MPa) and 3/8 inches (9.5 mm) thick using SA 455 steel. The tank was 27 years old when it ruptured along a longitudinal weld seam during a filling operation causing the death of 2 people. It was determined that inadequate welding and lack of periodic radio-graphic inspection were the reason the tank achieved catastrophic failure.

The second incident happened in 2005 (Morris,Minnesota Incident)[57] where a 1000 gallon tank (3.8 m^3 tank) ruptured 3 hours after filling to recommended 85% filling limit. A portion of the tank head blew off turning the tank into a deadly rocket which split a tractor in half. The third incident in 2007 (Silver lake,Minnesota Explosion)[57] involving a 1000 gallon (3.8 m^3 tank) which was 28 years old ruptured due to a crack originating from an area of the tank head which had previous impact damage.

The fourth accident mentioned in the article by Russel et al [57] happened in 2003 where a 26 year old 10600 gallon (40 m^3) cargo tank ruptured at its head while it was being filled. The tank was constructed of ASTM A516 grade 70 quenched and tempered steel. It had a nominal thickness of 0.399 inches (10.13 mm) and a minimum head thickness of 0.25 inches (6.35 mm) designed for a working pressure of 265 psig (1.8 MPa). An extensive post incident study revealed that corrosion was present in cracks in the tank head which contributed to the catastrophic failure. The overall diagnosis was that the manufacturer should have only allowed ammonia containing a minimum of 0.2% water to be stored in a tank made of ASTM A516 grade 70 steel with a quenched and tempered treatment.

What was found in common between all these failures was that periodic X-Ray inspection was not performed resulting in missed detection of stress corrosion cracking in areas of weld seams or previous defects.

Automobile Fuel Tanks

Ammonia as a fuel for automobiles is a topic revisited for commercial use after the last major use in 1943 when Belgium experimented fuelling 100 buses using a blend of ammonia and coal gas as fuel. At that time there was a shortage of diesel due to World War 2 which lead to the further development of commercial ammonia combustion engines. Shortly after the war, when diesel became available again, the industry reverted back to the cheaper, energy rich fuel and further development of ammonia engines was shelved for a later day[58].

In the automobile industry there is recent research ongoing for storing ammonia for use in hydrogen fuel cell vehicles. A study by Reynaldo et al. in October 2020 investigated the use of polymers and composites for the construction material of liquefied ammonia tanks for mobility vehicles[59]. In 2013 South Korean researchers of KIER successfully tested an ammonia fuelled dual fuel propelled passenger car called the AmVeh. This vehicle uses a 70% ammonia to 30% gasoline spark ignition engine[60].

The study by Reynaldo et al.[59] highlights the typical use of a Type IV pressure vessel for an automobile fuel tank as it is the lightest option compared to other pressure vessels. In the study, the material combinations (liner and composite) are studied by finite element analysis. Materials identified as the liner part compatible for use with ammonia were Polyethylene terephthalate (PET) and polypropylene (PP). These materials possess the required ultimate tensile strength and demonstrate low cost. Carbon-fibre-reinforced polymer (CFRP) and glass-fibre-reinforced polymer (GFRP) are adopted as composite skins. The results of the study recommended,that for a 37.2L prototype tank, PP-CFRP with stacking arrangement $[90/+-30/90]_{3s}$ for lowest stress in the liner during a burst test while an arrangement of $[90/+-0/90]_{3s}$ was best for impact loads. International rules for pressure vessel design are documented in detail in ASME VIII division 1 (2019) standards.

2.3 Relevant Regulations about Ammonia Storage

This section focuses solely on regulations for storing ammonia and using it as a fuel on vessels. There exist many rules and regulations about ammonia handling and storage but most rules are related to fixed installations, process plants or non-mobile applications. For example, the International Fuel Gas Code (IFGC) contains rules about storing gas primarily for household use and it specifically states that it is not applicable to the use of hydrogen, LPG and CNG on vehicles[61].

2.3.1 Class Societies and Maritime Rules

Current IMO IGC code forms the base of all class society rules in regards to Liquefied Gas Carriers. This code is used for gas carriers while IMO IGF code for Low Flash-point Fuels is used to design vessels capable of using ammonia as a fuel. As mentioned in the section 1, these codes do not include the use of ammonia as a fuel as of yet due to its novelty. IGC section 16.1 specifically states that methane (LNG) is the only cargo whose vapour or boil-off gas may be utilized in machinery spaces of category A [30]. In September 2021 the IMO sub committee for Carriage of Cargoes and Containers (CCC) re-established a correspondence group to find amendments to the IGF Code and develop guidelines for low-flashpoint fuels such as ammonia[19].The CCC re-convened in September 2022 for the 8th session to discuss further amendments to the IGF and IGC codes with the intention of incorporating ammonia as an alternative fuel source, in the end a work plan was agreed on with the intention of finalizing the guidelines for use of ammonia as a fuel by CCC 10 in September 2024. A new IGC and IGF codes are expected to come into force by January 2028 with many changes due to be made[19].

Meanwhile there are great developments being made in ammonia-propelled vessels by shipbuilding companies such as NYK (Japan) [62] , Amogy (USA) [63], Dalian Ship Yard (Korea), VARD (Norway) and others as class societies have continued to amend their guidelines to achieve Approval in Principle (AiP) for new pilot project ammonia vessels.

The following class societies have made an attempt at tentative rules and suggestions for using ammonia onboard ships as a fuel:

- ClassNK- Guidelines for ships using alternative fuels (September 2021)
- Bureau Veritas -Ammonia Fuelled Ships Tentative rules NR671 (July 2022)
- American Bureau of Shipping (ABS) -ABS Guide for Ammonia Fuelled Vessesls (September 2021)
- Det Norske Veritas (DNV)- Rules for Ammonia in Part 6 Chapter 2 Sec 14 (July 2014)
- Korean Register (KR)- Guidelines for Ships Using Ammonia as Fuels (July 2021)

Key design changes to regular gas fuelled ships are identified and summarized in these guidelines. Important aspects to consider are ventilation requirements and arrangement of accommodation spaces (since the intended use of the ammonia is onboard a manned platform). All societies base their recommendations off existing IGC and IGF codes with exceptions made to cater in for ammonia toxicity.

A study by the European Maritime Safety Agency (EMSA) found that there are a number of relevant standards set out by the International Organization of Standards (ISO) that relate to handling ammonia. Most of these are for general use on land and the handling of ammonia as a fuel for marine application is not yet documented by the ISO. The EMSA report states that ISO 8217: 2017 for marine fuels is widely used for petroleum product handling on ships and parts of it may be applicable to ammonia. ISO 23306:2020 standard for the specification of liquefied natural gas as a fuel for marine applications and the ISO/AWI 6583 'Specification of methanol as a fuel for marine application' are examples of other low-flash point fuels which recently received sets of standards, therefore it is foreseen that ammonia will follow suit. It is foreseen that future IGF and IGC codes will incorporate such rules and take into consideration feedback from class societies which have a direct link to shipbuilders documenting the design processes of integration of ammonia onboard vessels.

2.4 Liquefied Gases: Storage Solutions on Vessels

Section 2.2, *Land Based Ammonia Storage Solutions*, can be used as reference when designing a tank capable of containing ammonia but, on a vessel, requirements for storage can be more stringent as outlined by class rules and code from section 2.3.1. Attention must be paid towards containing toxicity risks in smaller manned areas within a sensitive surrounding environment. For over 50 years liquefied gas has been transported by ships[64] so much can also be learnt from storage methods for gas as cargo.

2.4.1 Storage Methods

There are 3 distinct ways to store liquid ammonia onboard vessels as cargo according to IMO IGC regulation [30]. These are:

1. Refrigerated
2. Semi-Pressurized or Semi-Refrigerated
3. Pressurized

The decision of which of the three methods to use depends greatly on the quantity of ammonia to be stored and the space available to install systems to maintain tank storage conditions. Table 4 summarizes the different storage types and some of their characteristics.

Table 4: Table of different storage types[65][31]

Method	Pressure storage	Fully Refrigerated	Semi-pressurized
Tank types	Type C	Type A,Type B	Type C
Capacity	±2000t	±50000t	±2500t
Temperature	Ambient	-33.3 °C	-33.3 °C to 10 °C
Pressure	1.6-1.8 MPa	0.045-0.7 MPa	0.65-0.85 MPa
Method	Compression	2 stage refrigeration compressors	Single stage refrigeration

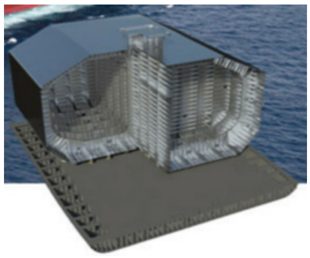
2.4.2 Tank Types

For each of the storage methods in section 2.4.1, there is a common preferred tank type used based on the requirements to safely store liquefied NH_3. There are three types of independent tanks according to IMO regulation (see Table 5 and Figure 6) and one integrated tank (see section 2.4.5) currently used by the liquid gas shipping industry. Type A, B and C tanks are all self-supported and prefabricated independently of the hull construction. Typically types A and B are large and prismatic suited for refrigerated storage with little tolerance to over pressure while type C has been traditionally suited for pressurized storage in smaller quantities than type A, type B or membrane tanks [66].

Type A tanks differ from type B in that they require a complete secondary barrier while type B tanks only require a partial secondary barrier (in the lower part of the cargo hold to recover leaks)[67]. This distinction means that out of the two, type A tanks would be more suitable for storing ammonia with a high toxicity risk. Figure 6a shows the internal structure of the tank and the gap between the interior boundary and the secondary barrier which was supported by the ship's double bottom. Figure 6b illustrates the partial secondary barrier only existing in the lower part of the tank.

The type C tank (Figure 6c), on the other hand, does not require a secondary barrier[67]. Type C tanks are said to be over-designed to manage very large pressures that can cause damage to the inner layers of type A and type B tanks. Generally, inner barriers of type A tanks are not crack propagation resistant which is why the second barrier is required [23].


(a) Type A Tank


(b) Type B Tank


(c) Type C Tank

Figure 6: Typical Ammonia Cargo Tank Types

There are advantages and disadvantages for each tank type which works on a case-by-case basis for storing ammonia. The simplest way to store liquefied ammonia is arguably to use type C pressurized storage as it does not require any auxiliary maintenance systems to maintain pressure in the tank, unlike type A tanks where an additional gas compression system is needed to cater for 2-stage refrigeration. The use of pressurized storage comes with its own set of risks as noted by Dräger [44] that pressurized ammonia liquid at ambient temperature and high pressure will expand in the event of a leak causing icing around the tank, a frostbite risk for any worker needing to attend to a leakage to contain toxic emissions. Coupled with this the ship structure may be damaged by the low temperatures.

Wärtsilä claims that semi-refrigerated storage is the most common and suitable option for small gas carriers using type C tanks[66], Technological developments from *Lattice Technology* support the use of type C tanks and indicate that type C tanks with semi-pressurized storage may also be suitable for larger vessels (see section 2.4.3).

Classification society Bureau Veritas (BV) mentions in their Tentative Rules on Ammonia fuelled ships [20] that there are three possible fuel tank possibilities for ammonia, namely:

- Type A tanks at or near atmospheric pressure and refrigerated to a temperature of -33.3 °C (Fully Refrigerated)
- Type C tanks under pressure at ambient temperature (fully pressurized)
- Type C tanks under pressure lower than vapour pressure at ambient temperature (semi-pressurized tank)

The third option implies that the tank is kept at a temperature somewhat lower than ambient temperature. Therefore semi-pressurized storage would also require some level of thermal insulation to maintain lower temperatures. Notably, the use of Type B tanks with a partial secondary barrier is excluded due to the higher toxicity risk of carrying ammonia. The full secondary barrier ensures that in the event of a leak the toxic gas can be held for a minimum of 15 days [20].

Table 5 summarizes each different tank and their respective benefits:

Table 5: Different tank types and their characteristics[23]

	Type A	**Type B**	**Type C**	**Membrane**
Shape	Prismatic	Prismatic or Spherical (Moss)	Cylindrical or Bi-lobed or Tri-lobed or LPV Prism	Thin membrane supported by adjacent hull
Design Req	Classical ship structure design rules	Fatigue analysis and model tests required (Leak before failure)	Design Based on modified pressure vessel codes	Structural design assessment of containment system
Volume Optimization	Medium	Medium	Low (better with bi or tri-lobed)	High
Max Gauge Pressure	0.07 MPa	0.07 MPa	>0.2 MPa	0.07 MPa
Secondary Barrier	Yes	Partial	No	Yes
Inerting Requirements	Inert inter barrier	Hold filled with dry air	None (full pressure) Inert gas (semi-pressurized)	Inert inter barrier
Volume/Weight Ratio	Medium	Medium	Low	High
Inspection	Easy access	Easy access	Easy access	Special testing and inspection

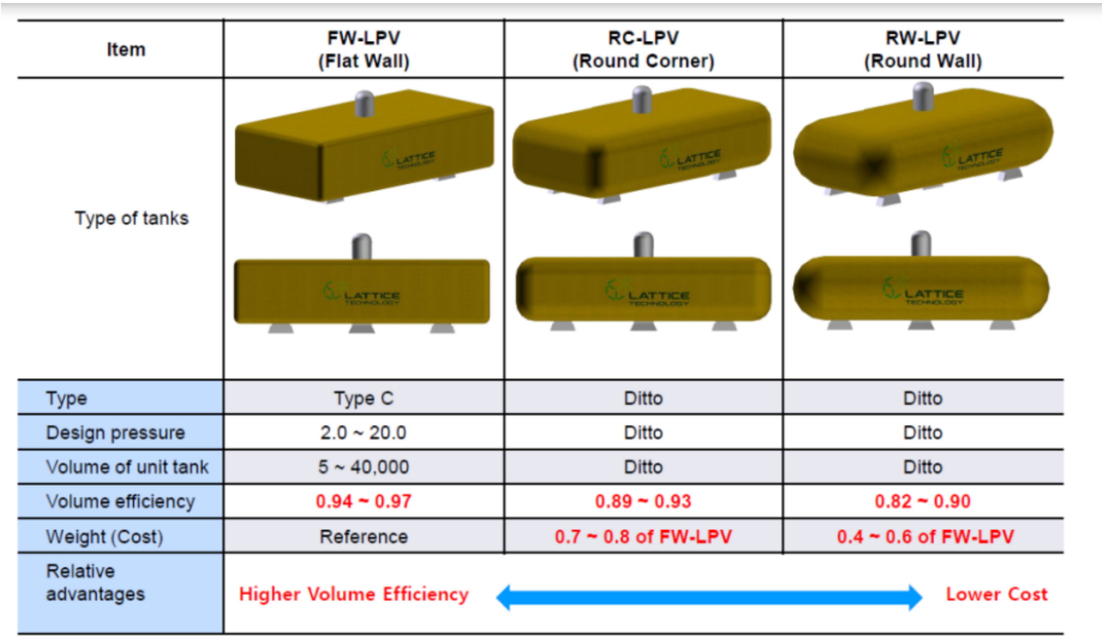
2.4.3 Lattice Pressure Vessels

Arguably the show stopper in terms of pressurized and semi-pressurized fuel storage innovation is the lattice pressure vessel (LPV). The invention patented in 2013 (Korea Patent No.10-1231609 and Korea Patent No.10-1254788) and in 2019 (US patent No. US 10,429,008 B2) by researchers at the Korean Institute of Science and Technology (KAIST) was originally designed for the storage of natural gas. This technology, often referred to as a ¨type C equivalent¨ tank by DNVGL, offers the ability to use pressurized or semi-pressurized storage without the restriction of using a cylindrical, lobed or spherical outer shell. The tanks are constructed with an internal lattice structure of stiffeners and frames (see appendix B). The lattice structure distributes the internal stresses over the whole internal structure which allows the outer shell to be much thinner than traditional large type C cylindrical pressure vessels.

When tested for storing refrigerated LNG the need for re-compression and refrigeration auxiliary systems was avoidable. Custom-shaped tanks such as rectangular flat-walled pressure tanks, which better utilise the storage space, are possible with this kind of technology. Essentially the volume of the tank can be increased from limitations of hundreds of cubic meters of storage space to thousands of cubic meters of storage space per tank

[68]. See section 2.5.1 on patents for details on the tank design.

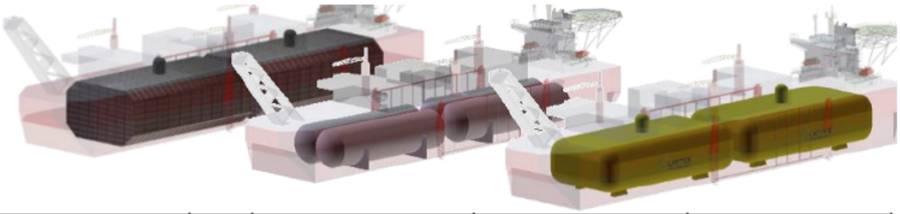
Experimental tests have been conducted on LPV tanks of volumes ranging from 22-80 m^3 with pressures as high as 2 MPa without observing any indication of weakening. Furthermore, non-linear simulations on the 80 m^3 tank indicated that pressures up to 4 MPa could be sustained before failure. Therefore use of these tanks for fully pressurized ammonia storage may be possible as the required design pressure for liquid ammonia at ambient temperature is in the range of 1.8 MPa. Notably, the manufacturing of prismatic pressure tanks of moderate thickness is in many ways simpler than manufacturing thick curved plates for traditional type C pressure vessels according to Bergen(Bergen,2017). Currently, the company Lattice Technology has created 3 main alternative tanks shown in Figure 7 with the capability to create tanks of variable size and shape[68].



Item	FW-LPV (Flat Wall)	RC-LPV (Round Corner)	RW-LPV (Round Wall)
Type of tanks			
Type	Type C	Ditto	Ditto
Design pressure	2.0 ~ 20.0	Ditto	Ditto
Volume of unit tank	5 ~ 40,000	Ditto	Ditto
Volume efficiency	0.94 ~ 0.97	0.89 ~ 0.93	0.82 ~ 0.90
Weight (Cost)	Reference	0.7 ~ 0.8 of FW-LPV	0.4 ~ 0.6 of FW-LPV
Relative advantages	Higher Volume Efficiency	↔	Lower Cost

Figure 7: Lattice Pressure Vessel Types [68]

Lattice Tanks have been compared to using similar membrane alternatives and bi-lobed type C pressure vessels. The results showed that in comparison to Bi-lobed tanks the round wall lattice tank gives more than 18 % additional volume while still maintaining the same design pressure (Figure 8).



Category	Item	Unit	FSRU With Membrane tanks	FSRU With Bi-lobe Tanks	FSRU With RW-LPV
Ship	Length O.A.	m	100.0	Ditto	Ditto
	Breadth	m	20.0	Ditto	Ditto
Cargo tanks	Type		Membrane	Type C	Ditto
	Design pressure	barg	0.7	3.0 (↑430%)	Ditto
	Volume of two tanks	m3	30,000	21,800 (↓27%)	27,300 (↓9%)
	Weight of two tanks	ton	-	-	1,160
Other Comparison measures	Inner hull Secondary barrier Gas detection system Heated cofferdam Pump tower BOG handling		Required	Not required	Not required
	Sloshing risk		Probable	No risk	No risk

Figure 8: Lattice Pressure Vessel Comparisons [68]

Lattice Technology has not only focused efforts on ammonia storage but has also gained approval in principal from many classification societies such as Loyds Register,Korean register, ABS, DNV, ClassNK and BV for designs for liquid hydrogen (LH2), liquefied petroleum gas (LPG), carbon Dioxide (CO2) and liquid natural gas (LNG) Tanks. The company put the first commercial lattice pressure vessel for natural gas into service in 2019 on the Ulsan Cleaning ship. Proffesor Daejun Chang explains that this year (2023) the company will demonstrate their latest liquid hydrogen tank with a volume of 12500 m^3 at 0.2 MPa internal pressure. The tank uses specialised scalable vacuum insulation to maintain tank storage conditions[69].

A masters thesis by a student of the Korean Advanced Institute of Science and Technology supervised by Professor Chang completed a work investigating the optimal shape and boil off gas generation of a fuel tank for an LNG fueled tugboat in 2019 [70]. This work used a lattice pressure vessel of a custom shape to fuel a 3.7 MW LNG-Fueled tugboat. Four horizontal cylindrical tanks were compared against using one trapezoidal shaped prism tank. The cylindrical tanks were said to use 17 m^3 of the space while the prismatic tank alternative used 37.5 m^3 of the space therefore the lattice pressure vessel alternative was 55% more space-efficient. The tank was designed using IGF (2015) ,IGC (2014) and ASME BPV (2013) codes and modelled using Abaqus finite analysis software. The design pressure was set to the MARVS LNG limit of 1 MPa.

The study investigated boil off gas rates (BOR) for various insulation materials including aerogel blanket, perlite powder, glass bubble and polyurethane foam under both atmospheric conditions and under vacuum conditions. The results showed that vacuum insulation conditions led to a significant reduction in BOR. Important notes from this study are how the BOR was calculated and the most efficient insulation mechanism being the glass bubble with vacuum insulation. The equation used to determine the BOG assumed a steady state model with constant atmospheric temperature. The thermal resistance was simplified to only consider the insulation layer thickness and neglect convection and radiation phenomena on the outside and inside of the tank. The thermal resistance of the tank walls was also neglected. Temperature gradient throughout the tank was assumed negligible. The one dimensional heat transfer equations used were:

$$R = \frac{t}{k * A}[K/W] \tag{10}$$

Where R is the thermal resistance of the insulation, t is the thickness of the insulation (m), k is the thermal conductivity of the insulation ((W/(m.K))) and A is the heat transfer area. The heat ingress, q, into the tank is given by Equation 11:

$$q = \frac{T_{amb} - T_{LNG}}{R}/1000[kW] \tag{11}$$

Where T_{amb} (K) is the ambient environment temperature , T_{LNG} (K) is the averaged constant temperature of the LNG. The amount of boil of gas is defined by Equation 12 in the study, the same methodology used by Al-Breiki and Bicer (2020)[71]. The amount of boil off gas (BOG) is given by Equation 12.

$$BOG = \frac{q}{h_{fg}} * 3600 * 24[kg/day] \tag{12}$$

Where h_{fg} (kJ/kg) is the latent heat of evaporation of the LNG. The boil off rate (BOR) as volume percentage of liquid evaporated per day is given by Equation 13

$$BOR = \frac{BOG}{LLF * \rho * V} * 100[\%/day] \tag{13}$$

Where LLF is the liquid filling level of the tank, ρ is the density of the liquid LNG (kg/m^3), V is the volume of the tank (m^3).The results of the BOG study are reproduced as per the source in Table 6 and Table 7 to demonstrate the differences in thermal conductivity of various insulating layers. The storage mechanism identified is semi-refrigerated storage to enable lower pressure storage below 1 MPa.

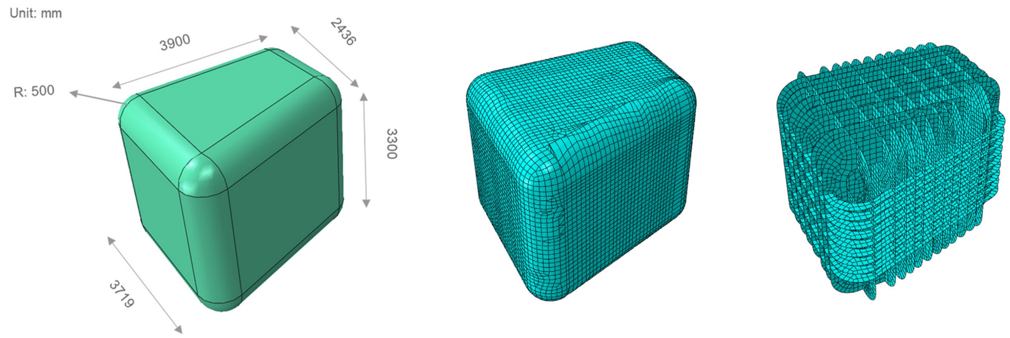


Figure 9: LPV tank specifications as presented by Kim in 2020 [70]

Table 6: Insulation types and BOR under atmospheric conditions as presented by Kim in 2020 [70]

Items	Unit	Aerogel blanket	Perlite powder	Glass bubble	Polyurethane foam
Thermal conductivity	W/m·K	0.011	0.035	0.025	0.023
Thickness	m	0.3	0.3	0.3	0.3
Thermal resistance	K/W	0.342	0.107	0.15	0.163
Heat ingress	kW	0.606	1.927	1.377	1.266
BOG	kg/day	78.729	250.5	178.929	164.614
BOR	%/day	0.518	1.649	1.178	1.084

Table 7: Insulation types and BOR under vacuum conditions (0.013 kPa) as presented by Kim in 2020 [70]

Items	Unit	Aerogel blanket	Perlite powder	Glass bubble	Polyurethane foam
Thermal conductivity	W/m·K	0.003	0.004	0.002	0.009
Thickness	m	0.3	0.3	0.3	0.3
Thermal resistance	K/W	1.253	0.94	1.88	0.418
Heat ingress	kW	0.165	0.22	0.11	0.496
BOG	kg/day	21.471	28.629	14.314	64.414
BOR	%/day	0.141	0.189	0.094	0.424

It is clear from tables 6 and 7 that the vacuum insulation alternative reduces the thermal conductivity of the insulation layer significantly and hence increases thermal resistance. The most significant improvement is with the glass bubble insulation, improving from having the second highest BOR to the lowest with vacuum insulation.

2.4.4 Comparisons With Other Liquefied Gas Tanks

The key focus of this thesis is on liquefied ammonia storage, however, much can be learnt about how to store ammonia gas on a vessel from liquefied gases with similar properties. One such gas is LPG which does not require to be stored under strict cryogenic conditions (refrigerated below -150 °C) unlike other gases such as Oxygen, Nitrogen, Hydrogen, LNG requiring extremely low temperature storage. Table 8 compares the properties of ammonia with LPG ,LNG and standard marine gas oil. The relative volume of fuel column in the table does not take into account insulation or secondary barrier requirements.

Table 8: Liquid Ammonia Properties Compared to LNG and LPG[72][23].Refrigerated (R), Semi- Pressurized (SP) and Fully Pressurized (FP).

Fuel	**Temp (deg C)**	**Storage Press (MPa)**	**Specific Energy (MJ/kg)**	**Energy Density (MJ/L)**	**Relative Vol Fuel**
MGO	ambient	atmosheric	42.7	38.4	1
LNG (R)	-162	atmospheric	48	21.6	1.8
LNG (SP)	(-162) or higher	0.5-1	48	17.3	2.2
LNG (FP)	ambient	20-25	48	9.8	3.9
LPG (R)	-48	atmosheric	46.3	26.9	1.4
LPG (SP)	(-48) or higher	0.5-0.7	46.3	23.6	1.6
LPG (FP)	ambient	1.8	46.3	20.6	1.9
Ammonia (R)	-33	atmosheric	18.6	12.9	3.0
Ammonia (SP)	(-33) or higher	0.5-1	18.6	11.6	3.3
Ammonia (FP)	ambient	1.8	18.6	10.6	3.6

Classification societies have based decisions about ammonia tank selection on previous discoveries with LNG and LPG. A recent study on the feasibility of ammonia as a marine fuel by Machaj et al.[72] evaluated LNG Type C cryogenic tank compatibility with ammonia. Important findings were that LNG tanks should be suitable to store ammonia due to the use of austenitic stainless steel (SS316L and SS304) which contains Cr content higher than 10% providing corrosion resistance. In addition, the design pressures and temperatures are also much more extreme for LNG therefore from a strength point of view the LNG tanks are suitable.

Typical independent tanks used for LNG storage are the IHI-SPB type A tank (manufactured by IHI corporation) made from aluminium alloy 5083 with a maximum thickness of 30 mm and insulation of 270mm polyurethane foam or the spherical Moss type B tank made of the same material with maximum thickness of 50 mm and 250 mm of polyurethane foam insulation[73]. A company called *Torgy LNG* has developed a type A tank certified by *DNVGL* which has internal reinforcement and baffles supported by a secondary layer attached to the ship´s hull. The design uses a combined insulation of an air gap and polyurethane foam.

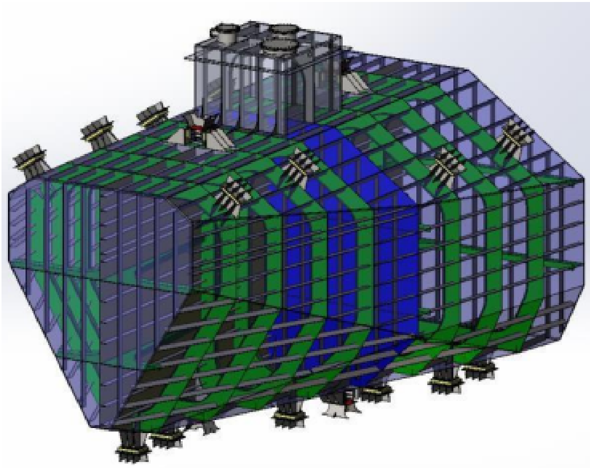


Figure 10: Torgy Type A Baffled Tank Inner shell[74]

The secondary barrier was originally designed as a composite layer in 2013 but after experiencing some issues it was changed to carbon steel. The unique insulated connection design between the primary stainless steel inner shell and the outer carbon steel shell prevents heat transfer to the cold stored fluid and also limits galvanic corrosion.

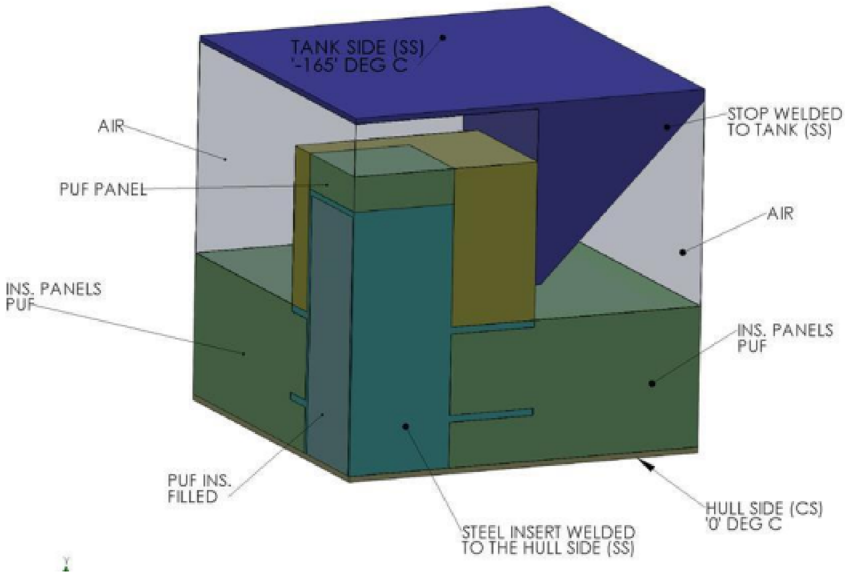


Figure 11: Torgy Type A Tank insulated connection [74]

The next question of tank compatibility is if the tank size needs to be made larger to cater for ammonia since logically ammonia fuel is less energy dense than LNG as shown in Table 8. Ammonia is said to require about 1.6 to 2 times more volume than LNG according to the EMSA 2022 report[23] yet Machaj et al [72] took a more detailed look into tank holding times which revealed 2 important aspects. Firstly that re-liquefaction may only be necessary for longer journeys given that for a set tank of 80 m^3 pressurized ammonia could be held for 532.7 days while refrigerated ammonia could be held for 349.9

days and methane could only be held for 14.3 days under a pressure relief valve setting of 1.1 MPa and a controlled atmospheric temperature. Secondly that the volume required for ammonia is in actual fact lower than what the EMSA report[23] suggested and is instead closer to 1.47 times the volume of LNG and not as large as 2 times the volume.

This result was confirmed by a study by Al-Breiki et al [71] where boil off gas (BOG) rates of LNG versus ammonia were studied and LNG BOG rate was calculated as five times higher than ammonia. This all means that less insulation will be needed for ammonia tanks and for long distance travels. This is especially true when compared to LNG tanks using polyurethane foam insulation where the thickness can be 30 to 40 cm on the outside of the tank [68]. The significantly lower boil off rate due to higher latent heat of evaporation[32] means that less fuel will be lost as BOG.

Ammonia has a relatively low viscosity compared to conventional MGO. This low viscosity can make ammonia more prone to sloshing than MGO and can also make it more challenging to handle during transport and storage. Its viscosity is 256.4 μPa.s in liquid phase at -33.55 °C [29]. Fortunately, solutions for low viscosity fuels have already been developed. LNG has a viscosity of 117.2 μPa.s at -161.64 °C, therefore systems designed to handle LNG should be able to be applied to ammonia storage. Studies in BOG generation characteristics in type C cylindrical LNG tanks have recently been studied by CFD analysis[75]. The sloshing behaviour at multiple excitation frequencies was studied by Ju et al [75]to understand the temperature and pressure changes within the tank and how this affects BOG production. It was noted that when sloshing frequency is constant the larger the amplitude and the stronger the movement of the fluid in the tank. Larger amplitudes give rise to severe interface fluctuation which boosts heat and mass transfer between vapor and liquid phase. The greater the rate of increase in mass averaged internal energy the more BOG produced [76]. It should be noted, however, that ammonia has a higher density than methane under storage conditions therefore the impact of sloshing loads could be larger than with LNG.

2.4.5 Membrane Tanks

Membrane tanks have been around for many years serving the liquefied natural gas industry to store LNG at cryogenic temperatures. The system uses a flexible stainless steel inner layer supported by insulation layers and the internal structure of the ship. The ship is required to have a double bottom for this kind of tank specifically (a requirement generally fulfilled by most large ships). Membrane tanks offer flexible geometry, low weight and low BOR with certain insulation specifications[64]. The most prominent membrane tank provider in the market is Gaztransport and Technigaz (GTT). GTT has equipped over 200 gas carrying vessels with their membrane tank patented technology. The types of membrane commonly used are the GTT Mark III or the GTT NO 96 membrane. Mark III usually made of SS304L with a thickness of 1.2 mm and R-PUF (glass fiber reinforced polyurethane foam) insulation of 270 mm. On the other hand the GTT NO 96 made of Invar (36% Nickel) with a thickness of 0.7mm and an insulation layer of plywood and perlite (530 mm thickness). In February 2021 the GTT Mark III membrane tank gained approval in principle from Bureau Veritas for use as an ammonia tank without any major design changes [77]. The tank was approved based on the compatibility of the primary barrier with ammonia and the higher design pressure of 0.1 MPa previously designed for LNG.

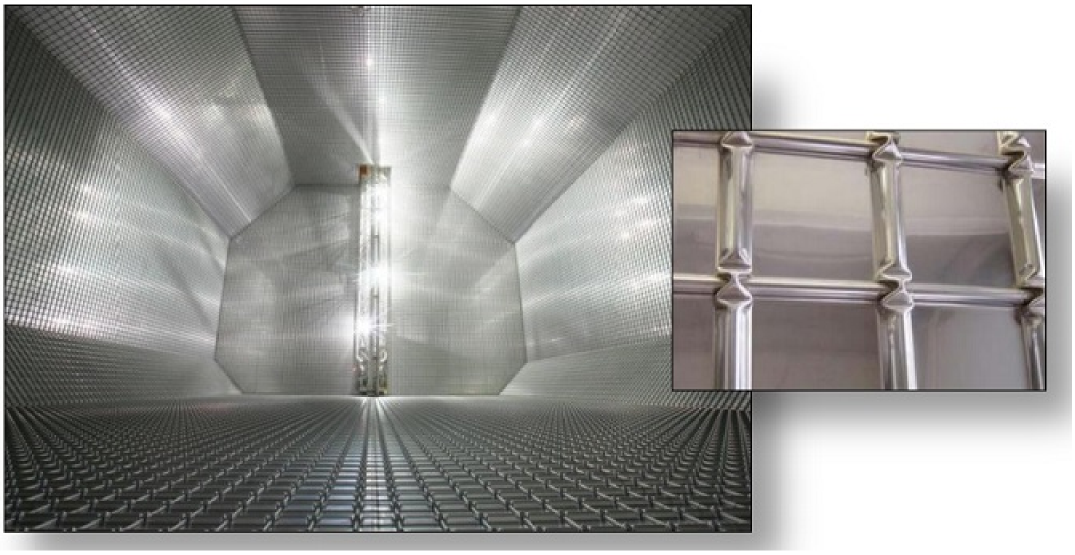


Figure 12: Mark III membrane tank from GTT [64]

The mark III membrane has a second membrane layer made of triplex material (see Figure 13) to comply with IGF code regulations and to prevent leaks. The flexible nature of the insulation allows the membrane to deform without cracking upon collision or grounding of the ship. This unique feature means that storage space is maximised while at the same time maintaining safe containment of the gas unlike other independent tank types which would be more likely to crack on grounding or collision[64].

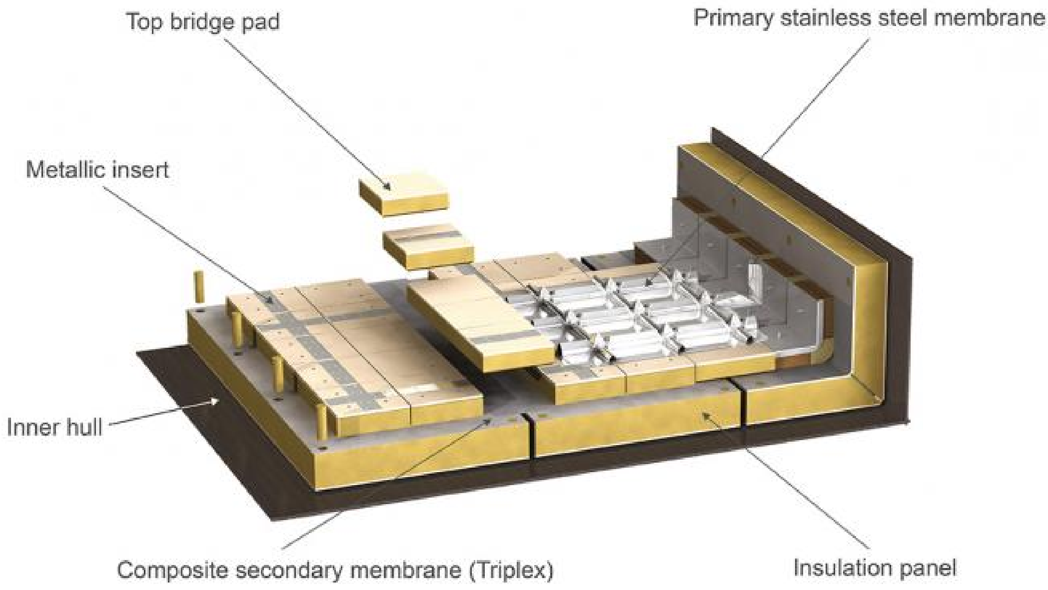


Figure 13: GTT Mark III Insulation Detail [64]

Membrane tanks are known to have problems with sloshing induced loads, incidents best known are: the Polar Alaksa incident (1969) ,the Arctic Tokyo incident (1971) and the Larbi Ben M´Hidi incident (1978)[78]. All of these incidents resulted in tank deformation due to sloshing induced resonance. As a result tank designers addressed the sloshing problem by adopting a hexagonal shape to mitigate wave formation as seen in Figure 12. Another incident was not recorded with a membrane tank until the Catalunya Spirit incident (2006). Previously it was assumed that only the worst environmental conditions would lead to large sloshing loads yet it was found through further studies that even medium wave heights can induce sloshing loads at critical wave periods. Experimental tests showed that internal waves within the 2 to 5 meter range had the worst effects on the tank sides. The previous incidents noted were all of Invar type membranes, the Mark III membranes have experienced less incidents with three Mark III membranes slightly deformed in 2008[78] yet to this day there has not been one case of liquid leakage from the membrane[64].

Independent type SPB (type A) (Figure 10) and Moss (type B) were designed as a response to reduce fluid movement within the tank. Membrane tanks cannot use the same baffle system as SPBs as baffles would transfer heat directly from the hull structure into the tank since the inner structure is not isolated from the ship hull.

More effective for membrane tanks is the use of an anti-boil off gas anti sloshing blanket (ABAS). The ABAS is a flexible membrane structure which lies on the surface of the liquid LNG and restrains fluid motion [73]. Further studies on sloshing reduction have been done by evaluating floating ball baffles against horizontal and vertical baffle systems by Sygal et al (2018) [79]. The study revealed the potential to greatly reduce tank weight while simultaneously reducing pressure fluctuations due to sloshing on the tank walls. A concern worth noting is that the material of the ball baffle shall be extremely wear resistant to avoid deposition of fuel contaminating material into the tank. Also the wear on the membrane should be evaluated from internal friction due to dynamic behaviour of the baffle system.

2.5 Relevant Patents

This section details two patented inventions that are important to consider when selecting a tank capable of storing ammonia. These two inventions demonstrate some of the previous challenges that were overcome in storing anhydrous ammonia or similar gases such as LNG.

2.5.1 Prismatic Pressure Tank Having Lattice Structure (2019)

In October 2019 the Korea Advanced Institute of Science and Technology was awarded the patent (US 10429008 B2) for a lattice pressure vessel. Inventors Dr. D.J Chang and Dr. P.G Bergan made 12 claims in the patent. Summarized here are the twelve claims: First of which mentions the orthogonal cell beam support structure inside a prismatic pressure tank that accommodates high-pressure fluid. Secondly mentioning quadrangular holes in the cell walls. Third mentioning the stiffeners have girders with flanges. Fourth mentioning that the cell beams may have circular cross sections. Fifth mentions beams may have diamond-shaped cross sections. Sixth mentioning that cell structures intersect each other and are attached to the cell wall to produce the pressure tank. Seventh mentions that at least one of the inner or outer walls contains stiffening members having lattice forms. Eighth mentions that flanges of the girders are welded to the outer wall. Ninth mentions there are gas sensors between the outer and inner walls of the tank. Tenth mentions one or more wall surfaces of the inner and outer wall. Eleventh mentions concrete or heat-insulating material between inner and outer walls can provide improved heat insulation. The final claim summarises all other claims mentioning that cell structures are connected to an inner wall at a predetermined distance from an outer wall. The space in between containing a plurality of girders having a plate shape, such girders align with portions of inner cell walls. Appendix B.1 shows images of the internal lattice structure of the invention.

2.5.2 Anhydrous Ammonia Storage tank (1960)

In may 1960 Mr. A Christensen was awarded a patent for an anhydrous ammonia storage tank (US patent no.29383960) this invention related to the storage of liquid anhydrous ammonia at atmospheric pressures. It was noted that ammonia should not be mixed with air as this could make the mixture explosive. This invention also attempted to reduce the cost of storage of ammonia by using refrigerated storage instead of pressurized and ensured that the toxicity risks of ammonia were controlled.

In the late 1950s the concern about using storage systems at atmospheric conditions were that sensors would not reliably be able to detect slight changes in pressures which could lead to immediate failure of the tank. Consequently this invention proposed a method to maintain atmospheric pressure inside the tank without the risk of high oxygen levels. Appendix B.2 shows a detailed drawing of the invention.

The way he proposed to equalize pressure and maintain oxygen content in the stored ammonia was using a condenser combined with a saturator all connected by pipes to the storage tank. When pressure in the tank is above normal pressure the gas flows through to a condenser (2) which causes condensable material to liquefy and drain to the saturating section (5). The remaining gas leaving the condenser is sent to the saturator where liquid from the storage tank is sprayed into the gas stream. The liquid and condensate then flow back into the storage tank. Excess gas is then cooled, scrubbed and vented to the atmosphere. Notably this system is focused primarily on preventing outside air from raising the oxygen percentage in the tank to an explosive level. The tank still relies on a separate compressor (17) and evaporator (18) to effectively maintain liquid refrigerated temperatures.

2.6 Ship Fuel System Integration

Integral to the fuel storage system used onboard vessels is the corresponding propulsion mechanism. For smaller vessels less than 5000 tons the weight of the propulsion system will be significant as fuel storage space and ship design specifications may be directly affected. With larger ships the weight may not be so much of a problem as will be the volume taken up by storage tanks and additional ammonia handling machinery. The propulsion mechanism will dictate the fuel efficiency and consumption rate of fuel , hence the quantity of fuel that needs to be stored onboard to achieve a desired range. The following subsections cover the most recent advances in the maritime industry in terms of ammonia combustion engines, direct ammonia fuel cells and a brief look at ammonia crackers.

2.6.1 Internal combustion Engines

Types of Combustion Engines

The majority of current internal combustion engines run primarily based on the Otto or Diesel cycle. The first of the two uses a spark ignition (SI) system to ignite fuels with low minimum ignition energy and low critical pressure. Spark ignition engines usually use compression ratios in the region of 4:1 to 8:1. Diesel engines on the other hand burn fuels using compression ignition (CI) with compression ratios in the range of 16:1 to 20:1. There is also a third variant of the combustion engine which is known as the Homogeneous Charge Compression Ignition Engine (HCCI) or similar Spark Controlled Compression Ignition (SPCCI) which is a modified Otto cycle engine.

This engine type takes advantage of relatively high compression ratios (usually around 16:1) and pre-mixing fuel before injection, the SPCCI variation still using a spark to initiate combustion. Since they operate at higher compression ratios the efficiency of such engines are higher than standard SI engines while at the same time producing lower nitrogen oxide emissions. HCCI engines have the advantage of being able to run on most alternative fuels including ammonia and hydrogen. Exhaust gas re-circulation (EGR) has been used by engine manufacturers to reduce NOx emissions [80] and studied by Pochet et al. for use with an ammonia/hydrogen fueled engine[81]. These type of engines have been developed and commercially produced by automobile manufactures such as Mazda with the SkyActive-X 2L model released in 2019 which burns regular gasoline [80]. The use of ammonia with these engines is still currently being researched.

Ammonia is being tested in both Otto and Diesel engines [36] [46] [38] [81] [35]. The most recent developments documented by Cardoso et al in a 2021 review paper[37]. There is a problem with using hydrogen as the pilot fuel in traditional high compression engines known as hydrogen ringing. Ringing occurs when the combustion intensity is too high and gas expansion faster than the speed of sound induces a pressure shock. Noted in the study by Pochet et al. [81] that in order to use both hydrogen and ammonia fuel effectively, a balance between high compression ratio to facilitate ammonia combustion and a limited compression ratio to prevent hydrogen ringing is needed. The favorable condition is to use a low equivalence ratio (oxygen content in the reactants) to avoid ringing, therefore leaner fuel mixtures are preferable for the HCCI engine. Pochet et al. tested a one cylinder HCCI with inlet pressures ranging between 1 to 1.5 bar and pre heated gas mixtures with inlet temperatures ranging from 428 K to 475 K, similar was done in a study using an Otto engine preheating the hydrogen-ammonia-air mixture to 323 K [38]. These high temperatures can reduce the mixture density and thus the volumetric efficiency of the engine which the study by Pochet et al. addressed.The study recommended higher intake pressures to be used with ammonia to minimize the required intake temperature and maximize indicated mean effective pressure. Noted was the exponential increase in intake temperature required for higher volume percentages of ammonia in the mixture, the highest tested volume percentage of ammonia was 70%. At this mixing percentage an equivalence ratio of 0.28 was used.

Dual Fuel Engines Using Gas and Alternative Fuels

MAN ,*Wärtsilä*, WinGD and other engine manufacturers are currently developing a range of large scale dual fuel engines suitable for use with ammonia, methanol and methane. According to a presentation by MAN in March 2023 the technologies are ready but a greater push from regulation will be needed to drive the uptake of dual-fuel engines[82]. The selection of which alternative fuel to use works on a case by case basis dependent on vessel type, size, trading patterns and fuel availability. Ammonia fueled internal combustion engines are the latest technology to be tested out of the alternative low flash-point fuel category. MAN mentioned in a presentation in March 2022 that their ammonia engines would be retrofits of existing engines they have such as: the ME-C engine, the ME-GI engine, the ME-GIE engine, the ME-LGIM engine and finally the ME-LGIP engine which is currently under testing stages.

Methanol engines are at a slight advantage over ammonia and hydrogen due to their higher technological readiness level and as a result 33% of MAN's new shipbuilding projects involve methanol fuelled systems [82]. As of 2023 the W32 methanol engine from *Wärtsilä* has been available on the market [46]. Methanol is normally a liquid at room temperature making it similar to handling conventional liquid fuel, unlike ammonia which is a gas at room temperature and requires pressurization or refrigeration to store it in liquid form. Like methanol, ammonia systems can only achieve approval in principle until IMO amends the IGF code to include ammonia as a recognised fuel source with the appropriate safety measures to use it.

From 2021 till the present time of writing *Wärtsilä* has been developing a pilot 4 stroke ammonia engine for marine use, which will be installed on an ammonia fuelled tanker to be commissioned in 2024[46]. Similarly MAN has been developing a 2 stroke ammonia fuelled engine for a bulk carrier due to be commissioned in 2024. It is foreseen that this engine will run on a combination of heavy fuel oil and ammonia. For lower loads the engine will run on a higher percentage of conventional fuel while at higher loads there will be a higher percentage of ammonia. ClassNK highlights that there are fundamental differences in the level of complexity of the fuel supply system using a four stroke or a two stroke ammonia fuelled engine. The two stroke diesel variation is a high pressure system which circulates liquefied ammonia at designated temperature and pressure. This kind of system is typically used with LPG fuel systems and is currently being developed by MAN, as shown in Figure 14.

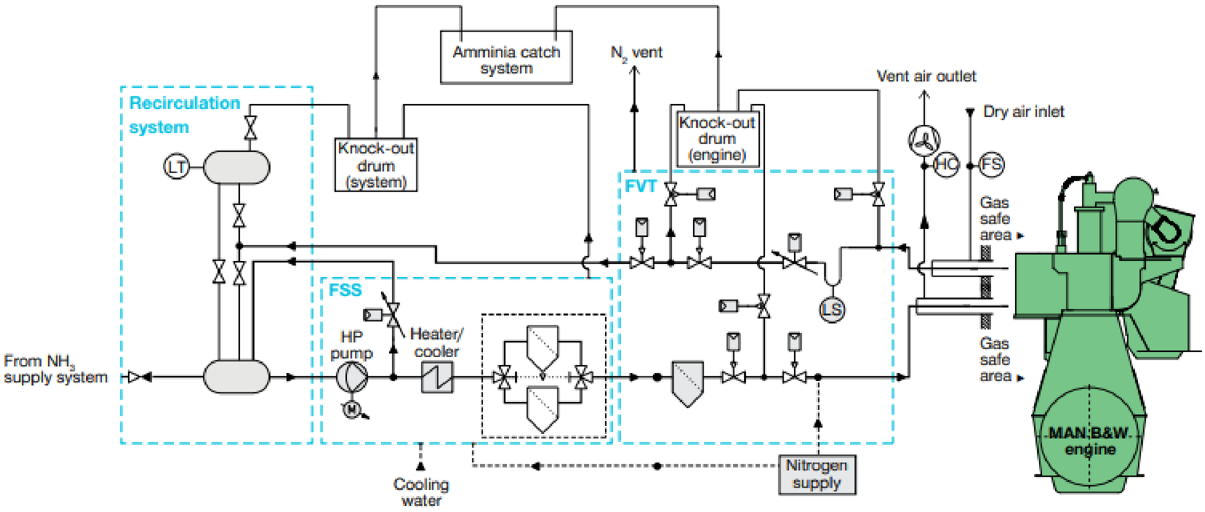


Figure 14: MAN High Pressure Dual Fuel Ammonia Supply System [36]

The system makes use of a nitrogen supply to pressurize the fuel lines every time before the engine performs a cold start. In this way the system is pre checked for leaks before toxic ammonia is allowed to flow from the storage tanks to the engines. Important notes from this design are the knock out drums[2] and ammonia catch system to eliminate toxic ammonia emissions.

The four stroke Otto cycle variation on the other hand is a low pressure system which supplies ammonia gas to the engine either at ambient or elevated temperature and pressure [31]. Similar studies are presented by the EMSA study on the potential of ammonia as fuel in shipping[23], In the study the two fuel systems were roughly modelled similar to Figure 14. A potential challenge for the low pressure Otto variation variation is that it can suffer from significant fuel slip or misfire sensitivity (also known as knocking) [23].

The diesel cycle burns the fuel in a diffusion-combustion process whereas the Otto cycle equivalent burns fuel in a pre mixed combustion working with a lean burn principle with a relatively high air-to-fuel ratio[83]. The advantage of the former is that it can be used on a wide range of gaseous fuels making it the method of choice for large 2-stroke engines[23]. MAN (Laursen,2018) have mentioned that their current ammonia engine under development (technical specifications not yet available to the public) will operate on a similar system to the ME-LGIP (Liquid Gas Injection Propane) dual fuel engine. This engine uses FBIV-P fuel injectors which receive 50 bar supply pressure and hydraulic oils to boost liquid LPG pressure to 600-700 bar (Injection pressure). The hydraulic oil is kept at 80 bar pressure using a non-return valve[36].

[2]Knock out drums are used to separate liquid and vapor streams aswell as capture residual oil from fuel consumers

To meet IMO Tier 2 requirements, engine makers expect to use 10% pilot fuel and the larger engines only 5% pilot fuel. The pilot fuel dependent on the particular engine type and operational characteristics[23]. For the short term, fossil fuels are expected to make up the most part of pilot fuels with hydrogen expected to come in as a replacement after further technological advancement in cracker technology. Table 9 below summarizes the characteristics of ammonia and methane dual fuel engines under development.

Table 9: Otto cycle vs Diesel cycle Dual Fuel Engines (EMSA,2022)[23]

	Low Pressure		**High Pressure**	
	Otto		**Diesel**	
Ignition	Pre-mixed gas/air		Direct Injection	
Pilot Fuel Req	Yes		Yes	
Fuel	Methane	Ammonia	Methane	Ammonia
Fuel Supply Pressure[Bar]	5 (4 stroke) 13-16 (2 stroke)	5-16	300	80
Injection Pressure[Bar]	Same as Supply	Same as Supply	Same as Supply	500-700
Liq Pilot % @ MCR	0.5-1.0	15-30	0.5-1.5	5-10
BMEP[3] [Bar]	17.3	17	21.0	21.0
Min load for DF mode [%]	5	30	5	15
Fuel Slip	Yes	Yes	Insignificant	Insignificant
Knock/ Misfire	Yes	Yes	No	No

In the marine industry no ammonia fuelled combustion engines currently operate in the global fleet but beginning this year the world should see the first ammonia dual fuel engines in operation, hence the importance of this study for the advancements in system design for the use of ammonia as fuel. Table 10 summarizes the current known ammonia -fueled engines under development. Notably these engines are all lower speed engines and proposed for large scale operation due to the low flame speed of ammonia [84]. Smaller scale engines rely on developments in the automotive industry at this time of writing such as the HCCI described earlier or on hydrogen dual fuel engines described next.

Table 10: Current known marine ammonia combustion engine projects[23]

	Layout	Combustion Cycle	Year of delivery
MAN B&W ME-LGIA	2-stroke slow speed	Diesel	2023
Wartsila DF	4-stroke medium speed	Otto	2024
Wartsila LG	4-stroke medium speed	Diesel	2023
Himsen	4-stroke medium speed	Diesel	2024
MAN-ES	4-stroke medium speed	Diesel	2026
WinGD X-DF A	2-stroke medium speed	Diesel	2025

[3] Brake Mean Effective Pressure

Hydrogen Dual Fuel Engines

In terms of hydrogen dual fuel engines MAN is said to be developing alternatives for use in the shipping industry post 2030. MAN already has conducted research on hydrogen operation with the HD-H2 SI (Otto) engines in 1992 and in 1997 the MS-H2-CI (Diesel) engine. These engines were used for hydrogen driven airport buses in Munich since 1999 and from 2006 MAN-busses for Berlin. MAN has also developed the D2862 LE448 hydrogen dual fuel engine for work-boats. The engine relies on an input of 5% diesel fuel and can be run on pure diesel or dual fuel. It has a rated power output of 749 kW, an engine speed of 2100 rpm , compression ratio of 19:1 and mean effective pressure of 17.7 bar. It has a rated torque of 3406 Nm and said that it is essentially a standard diesel engine without much modification to allow the use of hydrogen. An important project is the Hydrocat 48 Project which is the worlds first hydrogen powered crew transfer vessel which reduces more than 50% of its traditional fuel usage. The vessel uses two D2862 hydrogen dual fuel engines which mix hydrogen into the charge air intake. The engines are IMO Tier III compliant. A project looking into the decarbonization of the energy sector in Chile did an assessment of a hydrogen dual fuel engine in 2021 and it was noted that hydrogen needs to be supplied at 0.6-1 MPa at a temperature less than 45 ° to the engine[85].

2.6.2 Ammonia Fuel Cell Technology

Fuel Cell technology and fuel cell projects in the maritime industry have been around for over a decade with the Nemo H2 and FCS Alsterwasser projects demonstrated in 2009 and 2008 respectively. These were both hybrid power projects using PEMFC fuel cells (storing hydrogen in tanks onboard) and lead-gel or lead-acid batteries as backup[18]. The latest hot topic has been to use ammonia instead of hydrogen onboard to supply fuel cells, thus giving rise to the term Direct Ammonia Fuel Cell.

The name direct ammonia fuel cell (DAFC) as quoted in many sources[12][86][13][87]is a little misleading as it suggests that ammonia can be directly fed to the fuel cell, however, for majority of fuel cells this is not the case. There are four main types of fuel cells commonly studied for use in marine application, of which, two favor external cracking of ammonia to hydrogen before the fuel can be used directly. The four main types include Proton Exchange Membrane Fuel Cells[4] (PEMFC), Alkaline Fuel Cells (AFC), Solid Oxide Fuel Cells (SOFC) and Molten Carbonate Fuel cells (MCFC)[14][88]. Other fuel cell types less commercially developed or not applicable in this study are the Direct Methanol Fuel Cell (DMFC) and the Phosphoric Acid FC (PAFC) operating at low temperatures

[4] Also known as Polymer Electrolyte Membrane Fuel Cells

[18].In a container ship case study on use of ammonia for fuel cells the MCFC was noted as one of the most promising options for use with ammonia because of its ability to operate at high temperatures yet in recent years the SOFC has out-competed the MCFC due to reduced manufacturing costs and improvements in reliability[89].

SOFC

The most promising option in terms of direct feed ammonia is possibly the SOFC as it can operate at high temperatures (700-1000 °C) allowing for internal cracking of the ammonia within the cell thus eliminating the need for an additional cracker in the fuel supply system[13]. The excess heat can also be used in fuel reheating boosting system efficiency and eliminating the need to use expensive precious metal catalysts[88]. The SOFC uses a hard non-porous ceramic as the electrolyte which allows for reforming of fuels such as ammonia internally. SOFCs are the most tolerant to impurities in the fuel feed and can withstand direct exposure to NH3 which means that a hydrogen purifier is not necessary in the fuel supply system. Comparatively speaking, the SOFC under performs the PEMFC in terms of power density yet offers higher energy efficiency up to 60% [90]. The SOFC is also reported to have slow dynamic behavior in transient operations therefore a backup energy supply system may be needed to ensure stable operation [91]. Another concern using SOFCs are their durability, due to constant high temperature operation current SOFC technology has a lifetime of around 5 years[88]. Thermal cycling is also a point of concern for SOFC lifetime.

The SOFC market is predicted to grow at a rate of 33.9% CAGR per year from 2022 to 2027[92] which is extremely promising for further technological development. One of the few known ammonia powered fuel cell projects (ShipFC-Viking Energy) is said to be powered using SOFC technology[93]. The current problem is that there are not many commercially available systems on the market at present for marine use, however, progression has been made by IHI Corporation in Japan to release its first commercially available SOFCs which aim to produce fuel cells in the 10-100kW range for onshore and offshore application [94]. *Sunfire* also produced a 50kW SOFC unit for ThyssenKrupp Marine in 2015 but it was said to run on low sulphur diesel and not ammonia[95]. Other companies heavily invested in SOFCs primarily for onshore based application are: Bloom Energy, Mitsubishi power, Aisin Seiki, Hitachi Zosen Corporation, Ceres Power, Adelan, Adaptive Energy, Solid Power, Watt fuel cell corporation, Upstart power, AVL, Convion ltd, Kyocera, Special Power Source, ZTEK Corporation, h2e Power, ElcogenAS, Miura and many more [92][87].

PEMFC

The PEMFC is the most commercially advanced fuel cell technology which has already shown well demonstrated proof of reliability when it comes to hydrogen fuelled cars, tractors, submarines, buses, trains and many fuel cell pilot projects[21][91]. PEMFCs offer the highest energy density yet require a constant supply of pure hydrogen (99.99%). The purifier itself consumes 2% of the fuel cells capacity[91]. A recent study comparing solutions for fuel cell uses on a small ferry evaluated 40 marine fuel cell projects worldwide [93], the results showed that 28 projects used low temperature PEMFCs and 4 used High temperature PEMFCs. The remainder of the projects composed of 5 SOFC projects and 3 Molten Carbonate Fuel Cells. PEMFCs are commercially available in compact units from : Toyota, Hyundai, Ballard Power, Mitsubishi industries, Bloom Energy, Proton Motors, ThyssenKrupp Marine Systems, and many others. Particular attention to Ballard Power´s FC wave fuel cell which is a 200kW scale-able unit designed for marine use[96]. Proton Motors have also further developed their PEMFCs to include a new PM-400 model of 168 cells having a maximum power output of 49.7 kW [97]

AFC

First discovered in 1959 by Francis Thomas Bacon the alkaline fuel cell is one of the first fuel cell technologies to be commercially developed and it has been used since the 1950s in the NASA and MIR space programs. The advantages of an AFC include: low operating temperatures, low sensitivity to excess NH3 concentrations, instant operation without preheating, low cost without using a platinum catalyst and most of all high efficiencies up to 60 %. AFCs, however, suffer from CO2 exposure in the air and require CO2 scrubbing before air can be fed to the fuel cell. GenCell offers one of the only commercially available solutions to using AFCs with patented technology. Their fuel cells use a KOH liquid electrolyte which allows for a wide temperature operating range (-40°C to 45°C). They also offer a lifetime up to 15 years, far outliving a battery powered system. The currently available solutions offer 5kW systems which are extremely large and heavy, best suited for onshore use[98]. Another fuel cell company, AFC Energy, has partnered with VARD shipyard to develop a bulk carrier fuelled by ammonia using alkaline fuel cells in conjunction with ammonia crackers to propel the ship [99]. The whole system is designed to be able to fit within a 40ft shipping container to produce 600-800kW of power. Figure 15 describes the conceptual design .

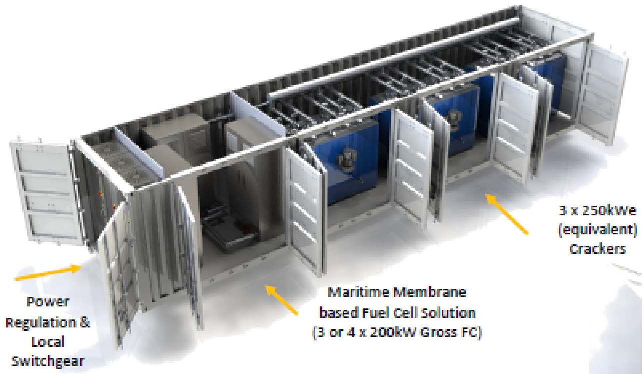


Figure 15: VARD-AFC Fuel Cell and Ammonia cracker unit [99]

2.6.3 Ammonia Cracking Technology

A key part of almost all fuel cell technology and also for the recovery of hydrogen for use after ammonia storage and transportation is cracker technology. Ammonia crackers were first developed in the 1930s, however, this was primarily for industrial scale uses instead of for small scale mobile use. The cracking process of ammonia follows Equation 14[100]:

$$2NH_3 + heat \longleftrightarrow N_2 + 3H_2 \tag{14}$$

The cracking is achieved by heating ammonia in the presence of a catalyst at high temperatures. This catalyst separates the hydrogen and nitrogen streams after the heating of ammonia at atmospheric pressure to temperatures in the range of 200-750 °C. Highest ammonia conversion rates occur from single pass conversions around temperatures above 400 °C. At temperatures higher than 773K thermal reactions occur without needing a catalyst [101]. Various reactor technologies exist and are being developed to efficiently extract hydrogen from ammonia. Some of these technologies include: microwave, catalytic, plasma, membrane, multistage continuous tubular, fixed bed, batch and flow bed reactors.

Fixed bed reactors are the most common type of of catalytic reforming reactor. They are known to suffer from poor heat and mass transfer behaviour, high temperature gradient and dust jamming. They operate at temperatures in the range of 600-900°C. Generally it is known that using Ru based catalysts improves the conversion rate of ammonia to hydrogen at lower temperatures[101].

Fluidized bed reactors have advantages for industrial scale catalytic cracking on land yet have been said to be too heavy, too big and take too long to start up to use for fueling PEM fuel cells. Thus the use of fluidised bed reactors is not recommended for use onboard for fuelling propulsion systems[101].

The most promising reactor technology for use onboard vessels for high hydrogen recovery, high ammonia conversion rates at low temperatures and low costs is catalytic membrane reactor technology. This kind of reactor uses a palladium (Pd) or hydrogen-selective silica membrane with a Cs/Ru or Ni/La-Al2O3 based catalyst [101]. Other types of membranes include micro-porous ceramic, crystalline and amorphous, dense metal and proton conducting, perovskite and non-perovskite.

A recent study on a small scale catalytic membrane reactor (Cechetto et al.,2021) achieved ammonia conversion rates above 99.998% at temperatures above 425 ° C. The study reported higher hydrogen recovery and purity under vacuum conditions yet differing feed flow rates (0.5-1 liters/min) did not significantly effect the reactor (an advantage for systems where variable flow rates are required). At higher pressures hydrogen recovery increased from 50% at 2 bar reaction pressure to 90% at 6 bar reaction pressure. Hydrogen purity dropped slightly by 0.01% from 2 to 6 bar. Ammonia conversion rate was relatively unaffected by pressure increases and was more a function of temperature. At temperatures of 400 °C ammonia conversion rates were unstable just above 80% yet at temperatures above 425 °C ammonia conversion was stable at 99.7% [102].Figure 16 demonstrates that at temperatures above 425 °C stable cracking is achieved. This is encouraging as results show that almost instantaneous high conversion is achievable given that the temperature of the ammonia reaches 425 °C. The startup time will depend on the time it takes to heat the ammonia to 425 °C.

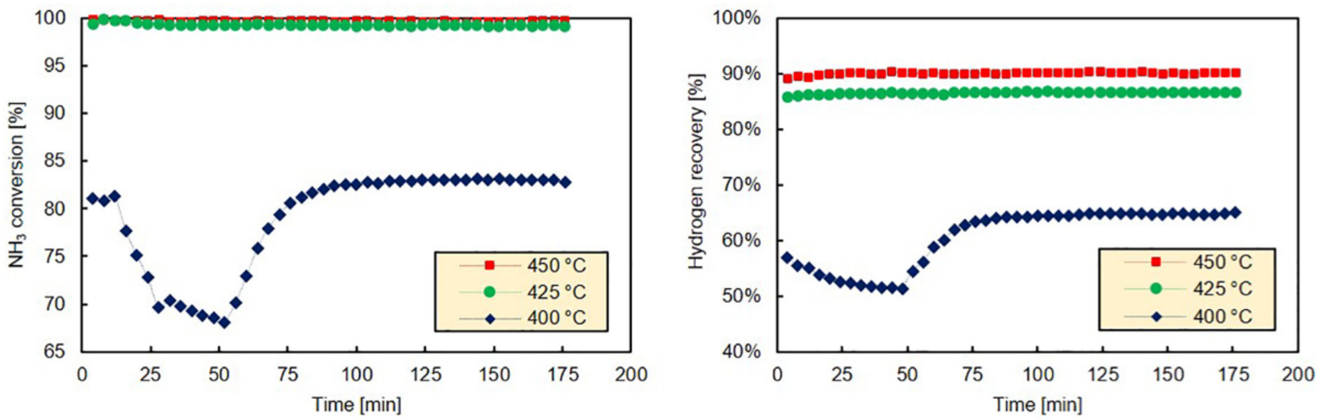


Figure 16: Ammonia catalytic membrane cracker characteristics [102]

As mentioned before AFC Energy is said to be working on small scale ammonia cracking technology for transport applications. Along with them another known startup company working to create ammonia crackers particularly for use with propulsion systems is Starfire Energy, their Prometheus fire product is currently in development [103]. Indian companies DBS Engineering Services and KCP UDHYOG offer commercially available ammonia crackers with flowrates of 10-100 m^3/hr with an operating pressure of 0.5 bar and 10 kW power requirement[104]. KCP UDHYOG says their cracker takes an estimated 10 mins of startup time until the cracker is in full operation[104] yet these specific crackers have yet to be tested for marine application.

2.7 State of the Art Brief Summary

- Anhydrous ammonia is best stored within carbon manganese steels or austenitic stainless steels.
- Nickel steels should generally be avoided due to stress corrosion cracking, ammonia is also highly corrosive to copper, zinc, mercury and cadmium.
- Ammonia is a toxic substance and generally is harmful to humans in concentrations above 30 ppm, it is also detectable at 5ppm which is helpful for early warnings of leaks.
- Anhydrous ammonia storage has been possible for over 50 years both for on land use and transportation on ships as cargo.
- IMO rules for use of ammonia as fuel do not yet exist, however, under alternative design principles ammonia can be used as fuel as long as all the IMO IGF safety requirements are met.
- Classification societies have developed interim rules for using ammonia as fuel.
- There are 3 types of independent tanks (Type A,Type B and Type C) and 1 integral tank (membrane).
- There are 3 methods of storage (Refrigerated,Semi-Refrigerated, Pressurized).
- Traditional transportation of ammonia on land has been achieved using pressurized cylinders.
- Lattice pressure vessels, adapted Moss tanks, type A SPB tanks and hexagonal membrane tanks are the latest developed methods of storing large quantities of liquefied gas with greater volume efficiency.
- Type A SPB tanks have been developed for LNG with internal baffles to prevent sloshing.
- Membrane tanks adopt hexagonal shapes to reduce sloshing yet without baffles sloshing could still be a problem at filling levels between 10 and 70%.
- A foam blanket system has been developed called ABAS to reduce sloshing effects in LNG membrane tanks, anti-sloshing ball devices have also been investigated and are commercially available.

- Vacuum insulation is an effective method for reducing boil off gas for refrigerated or semi-refrigerated tanks.
- Ammonia is difficult to combust due to its low flame speed and high auto ignition temperature.
- Ammonia fuelled internal combustion engines are under development and a few limited engines have successfully been tested such as the AmVeh vehicle.
- MAN,W*ä*rtsil*ä* and WinGD are currently developing dual fuel ammonia engines.
- There are two possible fuel supply systems for ammonia ICE engines (High pressure Diesel systems and low pressure Otto systems).
- Large scale ammonia internal combustion engines are primarily being developed to work with conventional MGO as a pilot fuel, the use of hydrogen as a pilot fuel is still under study with ammonia but proven to work with conventional fuel.
- HCCI engines can operate using Ammonia and Hydrogen.
- PEM Fuel Cells are the most common used fuel cell type offering the highest yield of electricity but require purified hydrogen.
- SOFC are the most promising type of fuel cell for direct (internal cracking) use with ammonia but offer a short lifetime compared to batteries and other fuel cell types.
- AFCs can operate at low temperatures with impure hydrogen but suffer from CO2 poisoning.
- Cracker technology for transport use is still in development but most promising are catalytic membrane reactors for high efficiency at lower temperatures.

3 Case Study

The use of ammonia as a fuel on large vessels such as VLCCs, bulk carriers and containerships has been studied from a high level perspective in multiple studies [23][21][91][5][89] but few papers document a plausible detailed case for use of ammonia on small vessels less than 5000 tons. The argument being that large vessels are more likely to be major polluters in terms of CO2 emissions and are therefore normally chosen as target vessels[5]. Another argument is that systems required to store and operate ammonia fueled vessels are too large for smaller vessels[22]. Also the lack of commercially available ammonia propulsion systems makes it difficult to conceptualise a smaller vessel as research currently suggests ammonia combustion to be inefficient compared to using other fuels[37]. A greater knowledge base on feasibility of using ammonia as fuel on small ships is required and such a knowledge base can be contributed to by such a case study.

This section outlines a case study in terms of the use of ammonia as a fuel onboard a small passenger ferry of 55 tons total displacement. What can be gained from a study of this nature is a small scale demonstration of feasibility on a detailed level and methods applicable to larger ships. It is well understood that ammonia is a toxic substance so demonstration on a passenger ferry where the risk of human contact is high shall prove reliability in the system design.

With reference to the title of this report, the ¨**integration impact**¨ of using ammonia onboard should be minimized. The identified impacts seen on ship design gained from the above literature review could be one or more of the following :

1. Cargo space loss.
2. Human health risks in terms of toxicity.
3. Potential to damage the surrounding environment.
4. Structural damage to existing ship components.
5. Fire and explosive risks.
6. Additional crew training required (Cannot be addressed by technical study).

Cargo space loss can be addressed by a study on general arrangement plans to cater for various fuel system arrangements. Dependent on each propulsion system option there are different requirements for the volume of ammonia to be stored and the feeder system from the storage tanks to the power producing unit. Using fuel cells instead of internal combustion engines demands compliance with a different set of rules and regulations. These regulations effect the way in which the system must be designed for a safe and non-toxic environment. Such rules applicable are those set out by Bureau Veritas NR671 ¨Ammonia-Fuelled Ships Tentative rules¨[20] for specific rules on storing ammonia and BV NR547 ¨Ships using Fuel Cells¨[32] for use of fuel cells onboard a vessel .

Human health risks reduction can be addressed by compliance to set regulations and guidelines. Creating a layout which includes adequate ventilation and ammonia monitoring sensors shall reduce the impact of using a toxic substance onboard as fuel.

Environmental impact can be reduced by installing catchment and drainage systems in the event of accidental leakage or system failure. These systems should be catered for in the general arrangement both reducing the risk of air and water pollution. Strict adherence to MARPOL rules should be applied.

Risk of structural damage can be minimised by using suitable materials which do not react or corrode in presence of ammonia. Such materials are mentioned in the literature review Section 2.1.2 and Section 2.2 reviews the current research into using composite tanks for ammonia storage. The scenarios described in the following section can be iterated using different materials to optimise weight and increase corrosion resistance.

The following case study aims to demonstrate how these impacts can be minimized by evaluating alternative fuel storage designs.

3.1 Definition of Scenario

The proposed scenario is to retrofit a fully electric city ferry, replacing the batteries and electrical systems with ammonia storage tanks and a corresponding ammonia propulsion system. The key importance of this study is focused on the storage of ammonia and minimizing the integration impact onboard the ship. The target ship is similar in design to the Damen Fully Electric Ferry 2306 E3 [105] which is well know by the author from previous studies. The target ship runs at an operational speed of 9 knots and runs two 57 kW electric motors. The ship was previously designed to have an autonomous range of 2.5 hours running on lithium ion batteries.

Figure 17: Damen E-Ferry 2306 E3 [105]

Table 11: Target vessel characteristics

Parameter	**Value**
Passengers	100
L [m]	23.64
B [m]	5.40
T [m]	0.85
D [m]	2
CB	0.55
Displacement [tonnes]	56
Power [kW]	164.5

3.2 Definition of Alternatives

In this section the propulsion system is first selected and described followed by the alternative tank designs which are the key focus of this work.

Propulsion Cases

The baseline vessel is a fully electric-powered ferry with a 2.5 hr operational range at max load without refuelling, therefore the ideal scenario would be to identify a solution that provides a better range than the baseline which would allow less frequent recharging or refuelling. As found in the literature review there are multiple ways to power a vessel using ammonia. The definition of the propulsion system needs to be fixed to be able to compare storage solutions, so the first step is to choose an appropriate propulsion system. The following cases are identified as plausible solutions for propulsion systems with Case 1 being defined as the base case:

- Case 1:Baseline Fully Electric propulsion system
- Case 2:NH3 based ICE propulsion system with MGO pilot

- Case 3:NH3 based ICE propulsion system with H2 pilot
- Case 4:H2 based ICE propulsion system with NH3 fuel system
- Case 5:NH3-based electric propulsion system powered by PEMFC
- Case 6:NH3 based electric propulsion system powered by SOFC

Similar plausible cases first identified and outlined in a study by Kim et al. in 2020 for a 2500 TEU container feeder ship were used as a reference, in that study the baseline was a MGO fueled diesel propulsion system[91]. Ideally, all 6 cases should be modelled with corresponding fuel tank arrangements to have a conclusive view of the best possible alternative for the ferry, however, due to brevity only the most practical case will be studied.

The decision on which case to model comes down to a decision matrix where the cases are then ranked in order of preference. Items marked with a star indicate rough estimates from literature [91][21][37]. Table 12 illustrates choices based on relevant categories, the lowest score overall is generally the preferred option. Each decision parameter was assigned a maximum value of 5 and a minimum value of 0. The more applicable the parameter to the case the higher the score would be for that case.

Table 12: Decision Matrix on Propulsion system

	Case 2	Case 3	Case 4	Case 5	Case 6
Requires fuel cell rules compliance (Hydrogen)	0	4	4	5	5
NOx Emmisions without SCR	5	4	3	1	3
CO2 Emmisions	5	0	0	0	0
Requires Cracker	0	5	5	5	0
Requires Purifier	0	0	5	5	0
Requires SCR	5	5	5	0	5
Technology Development Level[37]	3	3	2	1	3
CAPEX Cost*[21]	2	2	4	5	5
OPEX Cost*[89][45]	1	2	3	3	3
Volume requirement*[22]	2	3	4	5	5
Weight Requirement* [22]	1	3	3	5	5
System Efficiency[21]	2	3	3	5	1
Use of Fossil Fuels	5	0	0	0	0
Total	**31**	**34**	**41**	**40**	**35**

The decision matrix indicates that Case 2 would be the most favourable design case to proceed with as it scores the lowest but as a first iteration the system will be designed as fully renewable meaning that Case 3 will be modelled as it comes in a close second. In this case the type of ICE required needs to be multi-fuel such as the HCCI engine described in the literature review.

The problem with modelling this kind of system is cold starting. As seen in the literature, ammonia is not suitable for combustion without a pilot fuel at low loads. When the engines are switched off the system is designed to be free of hydrogen which is usually produced by the cracker. Three options exist to cold-start the system:

Firstly batteries onboard could be used to power the cracker, fuel heaters and pumps on startup until hydrogen is produced continuously to power the engines, thereafter utilising the shaft generators to continue powering the cracker with exhaust gases used to preheat the fuel. The shaft generators would then also feed power back to the batteries to recharge before the next cold start.

The second option is to store a small amount of MGO on board to cold start the engines until the engines could power the cracker to produce hydrogen, assuming that the engines chosen were HCCI engines or equivalent engines capable of multi-fuel operation. The amount of MGO fuel stored on board would be only enough to run for a maximum of 30 minutes of operation per day assuming one week without refueling therefore making any CO2 emissions negligible. It is predicted that the cracker should not take longer than 10 minutes to start full production of hydrogen as seen in the literature review [104]. Using this method eliminates the capability for the ship to have absolute zero emissions yet it provides a safe alternative in case the batteries lose charge or other electrical related problems arise preventing fuel supply of ammonia to the engines.

The third option is to install a small hydrogen buffer tank where enough hydrogen can be stored until the following cold start. The problem with this option is that hydrogen in gas form has a very low volumetric energy density meaning that normally a large tank would be required for non pressurized hydrogen. As mentioned in the literature review hydrogen needs to be supplied at 0.6-1 MPa and 45 °C therefore a compressor needs to be installed before the engine after the cracker.

What should be noted is that if ammonia combustion engines are developed to efficiently combust ammonia without a pilot fuel at low engine loads then Case 2 would by far be most preferable as no cracker would be required and there would be no added danger of using hydrogen onboard. The fuel supply would more closely follow the liquid fuel supply system being developed by MAN in Figure 14.

Tank Alternatives

The type of fuel tank will be chosen based on what could maximise the potential storage volume of fuel within a restricted space, taking into account safety regulations and without compromising ship design specifications such as: design draft, overall length, width and number of passengers allowed on board. Special care should also be taken not to compromise ship stability, sea-keeping performance or safety systems. The type of fuel tank chosen will determine the required safety measures,for example, Type A, Type B and membrane storage tanks will require inert gas barrier systems which take up more space whereas Type C do not. Refrigerated systems also require the additional space for 2-stage compression while fully pressurized does not. Refrigerated tanks also require insulation material as opposed to Type C tanks at ambient temperatures. Favorable to type A and membrane tanks is that usually they have higher volume efficiency in a given prismatic space. These iterations on fuel tank type are of greatest importance in this study and are what will be considered further. The following sub-cases will be detailed in the proceeding sections:

- Sub-case 3A: Type C Tank(s) with Fully Pressurized Storage.
- Sub-case 3B: Tank C Tank(s) with Semi-Pressurized Storage.
- Sub-case 3C: Type A Tank with Fully Refrigerated Storage.
- Sub-case 3D: Membrane Tank with Fully Refrigerated Storage.

Within Sub-case 3A there are also a number of different vertical or horizontal arrangements of multiple tanks which are analysed in section 3.5. The possibility of bi-lobed or tri-lobed tanks is also investigated. Type B spherical Moss tanks are generally assumed less space efficient than the cylinders in an enclosed space, therefore for this case study are excluded. Calculations would be similar as to those performed for Sub-Case 3C.

3.3 Definition of Methods

The case study is essentially two parts that work in tandem to evaluate the best alternative fuel tank arrangement. The first part addresses the integration impact in terms of a safety requirements study and how downstream fuel consumers affect the space allowance and permissible location for tanks. The second part, dependent on the first, focuses directly on the tanks and the various storage methods available. For each arrangement or tank there is an iterative procedure to follow shown in the flowchart of Figure 18. Safety requirements and fuel consumers from a Piping and Identification diagram (P & ID) analysis are input to the model at the general arrangement stage to allocate space for tanks.

P and ID diagrams evaluate what systems will be fuel consumers and how systems should be arranged to comply with ventilation and safety requirements. Similar work was done in the masters thesis by Niels De Vries in 2019 for a P and ID of an ammonia-hydrogen system and therefore can be used as a good reference[21]. This previous work did not focus on the storage solutions for the ammonia as it was assumed that the ammonia would be readily available from the cargo tank, focus was more in the propulsion system selection and risk assessment of the fuel supply system. The scope of the current thesis does not endeavor to cover the same such risk assessment in detail as the work done by De Vries was thorough and has also been performed by the European Maritime Safety Association[23]. The methods he used to calculate the fuel consumption and flow rates of an ammonia-hydrogen system are used to size equipment for the general arrangement (Calculations in Appendix E.0.1). Section 3.3.1 expands on the methodology for how the fuel consumption of the vessel is calculated and this becomes a second input to the model after the final iteration when key performance indicators need to be evaluated. Ammonia properties sourced using REFPROP software.

All sub-cases mentioned for various tank arrangements in the definition of the alternatives (3A to 3D) are modelled by P & ID diagram first and then further used to develop general arrangement plans for the placing of systems in the ferry under case study. IGC code for gas carriers and IGF code for low flashpoint fuels will be used as reference for existing applicable rules. Classification society Bureau Veritas in conjunction with classification society ABS will be used as reference for particular design specifications where IGC and IGF code do not specify enough detail. The following documents are applicable to a ammonia-hydrogen fuel system: IMO IGF code, IMO IGC code, BV NR 671 (Rules for Ammonia ships), BV NR 529 (Gas fuelled ships), BV NR 547 (Fuel Cell Vessels) for the use of an ammonia cracker and hydrogen onboard, BV NR 467 Part C (Pressure Vessel Scantling) for the tank specifications and ABS (Requirements for ammonia fuelled vessels).

Figure 18 demonstrates the complexity in finding the best alternative tank to use with three different storage methods and a range of tank types and arrangements resulting in multiple permutations for fuel tank design. There are two ways that the study can proceed, the former being that the range is maximised for the vessel and the latter being that the original range is matched. The latter of the two options is usually the way feasibility studies analyse the use of ammonia on large ships, however, taking this approach somewhat restricts the investigation of tank types that could be placed on the vessel. The goal was to maximise how much fuel could be stored onboard therefore the study went as follows:

Firstly storage method should be chosen, next the type of storage tank that is appropriate for that storage method and would best use the space available. Then the safety requirements are taken into consideration to adjust the tank size and location to fit on the ship. Following this the design loads need to be estimated to eventually find the thickness of the tanks and the material required for the tank. With the tank structure then known the insulation thickness needs to be calculated to find how much volume will be reduced for storing fuel (Type C pressurized storage alternatives are excluded as insulation is not necessary). The calculation most important in this case is the boil off gas rate and pressure build up in the tank due to heat ingress. It is generally accepted that the tank can be vented by using the boil off gas as fuel but when the fuel consumers are not running the ship must be able to sustain a pressure build up or use other mechanisms to control boil off gas. See section 3.3.2 for details of the BOG rate calculations.

The final part of the evaluation of best tank alternative methodology is taking into consideration that the ammonia fuel tanks are to be installed on a ship which requires that the ship remains stable under all load-cases. Since the original ferry was designed for inland waters the EU stability criteria for inland vessels (ES-TRIN,2019) was used to evaluate the stability of the proposed tank layout. There are six criteria for intact stability that must be passed. Upper and lower limits for the tank weight were established. The lower limit corresponding to almost empty tanks and the upper limit corresponding to the maximum design draft allowed for the ship to be used on an inland water way. The previous maximum draft was 0.85 m. See section 3.3.3 for more on intact stability.

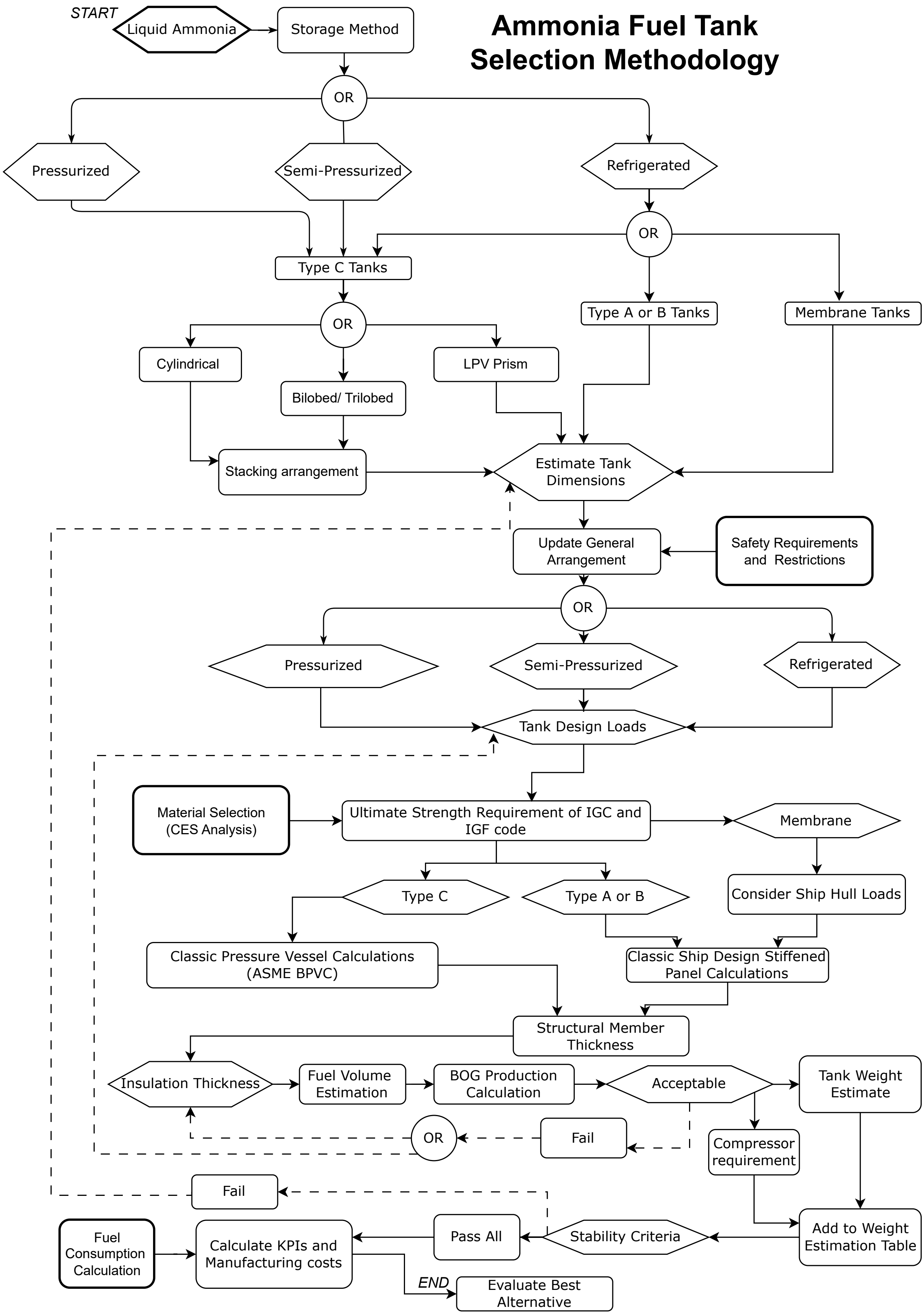


Figure 18: Flow Diagram of Methodology

3.3.1 Fuel Consumption of System

To determine the amount of ammonia fuel to fulfill the previous energy requirement of the batteries, one needs to have an idea of the propulsion system and how much power the previous ship required.

The target ship of the case study was designed as fully electric with battery capacity to run the electric motors and all hotel loads. There was a 20% additional margin to account for any unforeseen loads bringing the total power requirement to 164.5kW. The new dual fuel (HCCI) engines will need to produce power for the propulsion system and for the hotel loads if the battery capacity onboard is to be reduced. This will require shaft generators and additional power capacity of the new engines. This increase in power requirement will likely lead to an increase in weight of the engines also given that combustion engines are generally much less efficient then electric engines. Table 13 summarizes the power usage distribution for the baseline vessel against the proposed ammonia fueled vessel assuming the ammonia fueled vessel will need additional ventilation, fuel heaters and a cracker.

The ventilation system will be required to be run at all times in case of leakage. According to ABS rules [106] the ventilation system should normally extract air from ammonia containing areas at a rate of 30 air changes per hour and upon detection of a leak (150 ppm ammonia) the air supply system should ramp up to supply air at 45 air changes per hour to dilute the leakage with air. The power requirements and ventilation sizing are calculated in appendix E.0.6

A more detailed breakdown of the power requirements for the new ammonia fueled ship can be found in Appendix G.

Table 13: Power requirements onboard

	Hotel loads (Elctric Vessel) [kW]	Hotel loads (Ammonia Vessel) [kW]
Port	18.96	30.17
Emergency	12.87	14.88
Sailing	24.26	35.75
Maneuvering	33.19	44.83

As seen in Table 13 the maximum hotel load foreseen is 44.83 kW which should be covered by 2 shaft generators of 22.4 kW plus backup battery capacity of 44 kW to cold start the engines and cover auxiliary loads. These backup batteries and shaft generators will need to be routed to a switchboard and transformers onboard the ship.The calculations of the power draw for these items are found in appendix E. The proposal is to use the hot exhaust gases to preheat the ammonia before entering the cracker and the engines. The heat recovery system was modelled with COCO (COFE module) software to obtain the maximum heat transfer possible from the exhaust gases to the ammonia to be cracked. It was found that the final power requirement for the cracker furnace for continuous operation would be in the range of 9 kW additional electrical power after preheating ammonia to 233.01 °C with exhaust gases. Aspen Plus, Siemens TIA Portal, Matlab Simulink or other equivalent software could be used instead of using COFE and energy balances to simulate process flows, similarly done in the work of Henderik (2020) with Aspen Plus for an ammonia-powered ferry by SOFC [22].

Calculations for the fuel consumption and mass flow rates in the system were based on an energy balance assuming that two HCCI engines would need to be fuelled to produce 72 kW of mechanical power each. There would be 18 kW of power available at full sailing condition on each engine going to shaft generators to produce electricity and the rest used to propel the ship according to the previous propulsion requirement of 54 kW. The fuel energy content required for the engine can be calculated by taking into account the efficiency of the HCCI engines. It is assumed the engines will operate at an efficiency of 45% [81]. Thus the rate of energy delivered to the engine should be 330 kW (kJ/s). The flow rate of ammonia and hydrogen to the engine is based on a 70% ammonia to 30% hydrogen mix [21]. The flow rate in kJ/s is converted to kg/s using the lower heating value energy content of each constituent.

Once the mass flow rate before the engines is known the flow rate to and from the cracker should be calculated based on the required hydrogen output and necessary ammonia input. Equation 14 is used to determine the input of ammonia required to produce the set amount of hydrogen taking into account the efficiency and loses of the cracker. The catalytic membrane reactor of Cechetto et al will be used as reference[102].The cracker is said to recover 90% of the hydrogen. A scenario was completed to check the flowrates if the ratio of ammonia and hydrogen were to change in the engine. The final results showed that the consumption rate of ammonia at max load would be 0.0175 kg/s. The volume flow rate is dependent on the density of the fuel which changes based on storage temperature. Figure 19 illustrates the mass flow of the ammonia to and from the cracker.

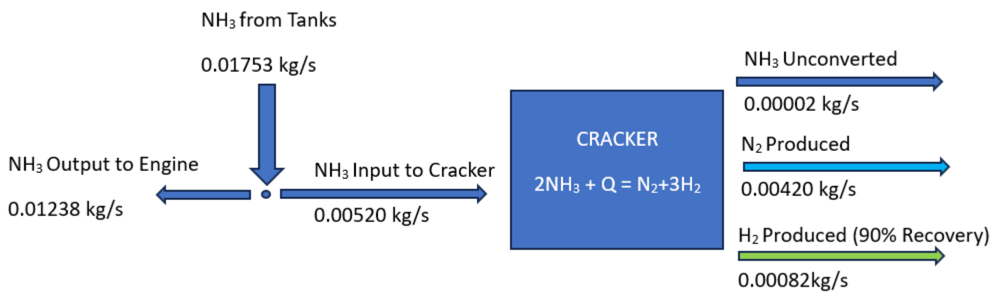


Figure 19: Cracker mass flow balance

3.3.2 Boil Off Gas Rate Method of Calculation

For refrigerated and semi-pressurized storage the calculation of boil-off gas rate can be calculated using a simplified one-dimensional steady state analysis as done in previous studies from the literature review[71][70]. Steady-state analysis allows for the calculation of the instantaneous heat ingress at a specific time, yet for the calculation of pressure build-up in the system a time-dependent model is better used.

A transient analysis tool was developed by Kalikatzarakis et al. (2022) to evaluate the pressure build-up inside a cylindrical LNG tank, the compressor flow rate and on/off control could be determined to manage the BOG [107]. A separate study modelling the natural convection in a LNG tank with ANSYS Fluent (Roh et al,2012) noted that BOG generation was strongly dependent on liquid-solid contact area and the contribution of heat transfer from the vapour region is negligible in comparison [108]. More recently BOG of LNG at different filling levels was modelled by Jo et al (2021) [109] which revealed that evaporation rate at low filling levels (around 10 %) increased rapidly over time in comparison to relatively constant rates at higher filling levels (94%). After reaching 94% filling level an inverse behaviour pattern is followed as filling percentage increased. At 10% liquid volume, the tank had a holding capacity of 3 days before reaching MAWP (Maximum allowable operating pressure) limits, at 50% the tank had a holding capacity of 20.6 days and at 94% liquid volume, the tank had a holding time of 60.2 days. Conversely bringing the tank up to 98% filling level then reduced the holding capacity down to 23.4 days.

For this thesis work, there was not enough time to fully develop a program to analyse the transient behaviour of the pressure build-up in the tank, therefore a one-dimensional steady state analysis was performed to obtain estimated values of boil-off gas rate. The assumptions of the model were as follows:

- The gas inside the annular space between the first and second barriers was nitrogen.
- The insulation type was polyurethane foam.

- The inner barrier walls were stainless steel.
- The second barrier walls were carbon steel.
- The convection between the vertical walls was modelled as a rectangular vertical enclosure (McGregor and Emery equations (1969)).
- The convection effects on the horizontal sides of the tank were negligible so pure conduction was assumed.
- The tank was modelled as a perfect rectangular prism.
- Thermal conductivity of substances was assumed constant.
- The outside air temperature was assumed 45 degrees Celsius.
- The liquid and gas temperatures of the ammonia were taken as constant at -33 degrees Celsius.
- The heat ingress was analysed as a superposition of multiple one-dimensional cases.

The boil off rate was then calculated using Equation 10 for thermal resistance, Equation 11 for heat ingress, equation 12 for boil off gas and Equation 13 for boil off rate. With the boil off rate known the compressor could be selected. In the study of Kalikatzarakis et al. (2022) a compressor with a mass flowrate of 450 kg/hr was selected to control an average boil off rate of 25 kg/hr (Compressor flowrate = 18 x boil off rate) with 4 activation times per day to control LNG boil off rate. Assuming that the insulation of the ammonia tank is reduced to imitate LNG evaporation behaviour then the compressor requirement would be calculated using the same ratio assuming a target of 4 activation times per day. Therefore the compressor selection was based off a flowrate 18 times greater than the calculated boil off rate.

What should be noted is that using this steady state analysis can give a conservative estimate of the boil off gas rate yet a transient analysis is better suited to model pressure build up and exactly how the compressor should be controlled. Especially important to semi-pressurized vessels is being able to calculate the pressure build up in the tank over time. As heat enters the tank the liquid evaporates and expands to vapor causing a notable pressure rise in the tank. If the pressure rise can be accurately modelled then a smaller compressor or no compressor at all will be necessary to control boil off gas. It is recommended for future work to use the model of Jo et al [109] to more accurately model the pressure rise in the semi-pressurized tank. This kind of model can also include the takeoff of liquid for the fuel system and how each load profile of the engine would effect the pressure in the tank.

3.3.3 Intact Stability Evaluation

The EU Classification of Inland Waterway Ships- 2006/86/EC was used to evaluate the intact stability criteria. There are 6 main criteria to be met namely:

1. Maximum Righting Lever
2. Down flooding angle
3. Area under the GZ Curve
4. Intitial GM
5. Maximum Heeling Angle
6. Residual Freeboard

Using *Maxsurf Advanced Stability* module the stability criteria were evaluated to assess the minimum lightweight required to ensure intact stability and also to assess free surface effects of tanks. The weights from the weight distribution table (appendix H) of all the equipment were input into the *Maxsurf* loadcase window. Tanks were added at their appropriate locations and then three scenarios were run for the case of no bulkheads,one longitudinal bulkhead (or two tanks) and finally for 3 longitudinal tanks. It was seen that at lower filling levels and tanks without bulkheads the free surface effects were most prevalent.

For single tanks without bulkheads it was found that the minimum lightweight would need to be at least greater than 44000 kg to overcome free surface effects. With one longitudinal division bulkhead the minimum lightweight was reduced to 41900 kgs and with 3 pressurized tanks the minimum lightweight requirement was further reduced to 41600 kgs. See appendix I for stability reports. The general criteria that was failed most often when the ship was too light was the criteria for the maximum heeling angle due to passenger crowding and wind.

The maximum displacement was set to meet a maximum design draft of 0.85m. This resulted in a displacement of 56 tons based on the shape of the hull.

3.4 System layout of alternative designs

In the P & ID diagrams found in appendix C there exist 4 boundary areas which represent areas likely to contain ammonia, these zones following recommendations by ClassNK [31]. Boundary A is represented in red illustrating a zone which always contains ammonia.

Boundary B is represented by an orange line illustrating a zone which is foreseen to contain ammonia in the event of a breach of boundary A, this area should only be entered by personal wearing appropriate safety wear covering full body and necessary breathing apparatus. Boundary C is represented in pink . In this area it is usually not anticipated for ammonia to be present, machinery which could cause a fire (Category A of IGF code) is usually located within this area. Finally Boundary D is represented in dark green and assumed to never contain ammonia, it is a safe area for crew and passengers to always be in, a breach of ammonia to boundary D should be avoided at all costs. All spaces above deck are generally considered within boundary D. In the P and ID essential systems are illustrated with special notes on the particular fuel tank advantages or disadvantages to the global system.

For all cases the advantage of having a refrigerant onboard as a fuel source was exploited to minimise integration impact. A first iteration of the P and ID diagrams used a direct cooling loop to the air conditioning unit yet after further investigation it was discovered that using an indirect cooling loop would reduce risks associated with an ammonia pipe leak. The refrigerant in the indirect air cooling loop was taken as Propylene Glycol ($C_3H_8O_2$) as it has a melting point below that of ammonia ($-59°C$) and therefore will not freeze. This indirect cooling system negates the risk of having ammonia near the passenger air supply unit. The flowrate of glycol is controlled using a pump which controls the temperature of the air in the passenger compartment. In the ammonia loop the glycol acts as a preheating source to begin the evaporation of the liquid ammonia to gas therefore reducing the heat input needed for the evaporator. The exhaust gases are then used to evaporate and super heat ammonia before entering the engines and the cracker. Three scenarios were modelled for the use of ammonia as a fuel. The first being pressurized ammonia at 1.8 MPa and 45 $°C$, In this case an expansion valve was used to reduce the pressure down from 18 bar to 6 bar which also reduced the temperature of the liquid ammonia making it suitable for refrigeration and changed the quality of the substance to be part liquid and part gas. Thereafter the same heat ex-changer sequence is followed to maintain the same equipment on the ship so that stability checks can be performed without making major changes to the lightweight of the ship other than the tanks. Appendix E.0.4 illustrates the cooling loop in COFE software.

Sub-case 3A: Type C Tank(s) with Fully Pressurized Storage

In this case important factors to consider are the operating pressures, as this system will use high pressure tanks (>18 bars). The consideration of a fully pressurized tanks reduces the need for any tank refrigeration mechanism as liquid ammonia is stored at ambient temperature. Pressure from the tanks will need to be regulated to accepted levels using valves and and an expansion devices to control the flow before integrating with the rest of the system.

The ambient temperature means that the liquid ammonia will require less heating to reach inlet conditions for the fuel reformer and engine than with refrigerated storage. With type C pressurized storage there is no need for inert gas within the fuel tank hold space which is an advantage in terms of space saving. Inert gas (Nitrogen is permitted) can be held in smaller quantities onboard only for purging fuel lines. The hold space should be maintained at a pressure below atmospheric pressure to ensure good ventilation in case of tank rupture, this recovered air should be analysed for ammonia content at the ventilation outlet from each room. In case of ammonia leak, contaminated air should be directed first to the vent mast until the master valve is switched off to the fuel lines. After shut off a diversion to the SCR for scrubbing in relatively low concentrations (30 ppm) to make the air suitable for a worker to enter the room and fix the leak. ABS requirements for ammonia fueled vessels [106] specifies in Table 1, page 55 the procedures of the gas detection and ship system response matrix for monitoring and safety systems onboard the ship.

The hold space surrounding the fuel tanks should be gas tight in case of a tank rupture with only entrance and exits through air ventilation systems for cleaning the room of ammonia. Clean air needs to be readily available to dilute ammonia concentrations and prevent fire and explosion risk. A water mist system should be activated in case of a leak which necessitates a fire pump to transfer fresh water to the fuel storage room or fuel preparation room. Drip trays should be installed to capture runoff and pump it to a special bilge tank which is able to detect ammonia concentrations. If concentration of ammonia is high, the bilge tank may not be released into open water and must be treated onshore.

Sub-case 3B: Tank C with Semi-Pressurized Storage

In this case Type C tanks are used, much like sub-case 3A a secondary barrier is not required due to the high design requirements of this type of tank to withstand extreme temperatures and pressures. As mentioned before this arrangement will require single stage refrigeration which requires a compressor and an condenser located in a room outside the fuel tank area. Due to the nature of refrigerated ammonia the area surrounding the tanks will need to be normally filled with inert gas or dry air to prevent explosion risks in the event of a tank leakage. The room needs to be gas tight but no requirement for cofferdams either side of the fuel storage room due to the use of Type C tanks.

Sub-case 3C: Tank A Tank with Fully Refrigerated Storage

Type A and membrane tanks have some of the most lengthy requirements. In between the second barrier and the first it is required to have inert gas. Cofferdams are also required either side of the fuel storage room separating tanks from machinery by at least 900 mm, a requirement normally fulfilled by the secondary barrier of the tank. Noted in IGF 5.3.5 when the fuel containment system requires a partial or complete second barrier the fuel storage space shall be segregated from the sea by a double bottom and the ship shall also have a longitudinal bulkhead.

Sub-case 3D: Membrane Tank with Fully Refrigerated Storage

Much of the requirements for type A tanks apply for membrane tanks yet in this case the tank is supported by built in bulkheads and stiffening systems instead of being self supported. The tank takes the shape of the hold space and can fill the space up to the safety limits for collision and grounding. On a smaller vessel the benefits of such a tank are therefore more limited as if a double bottom was previously not required a double bottom will have to support the membrane. In general the P and ID would look similar for both cases 3C and 3D therefore only one drawing is presented in appendix C.

3.5 General Arrangement of Alternative Designs

The next step is to determine the volume and mass of the components of each system and place them onboard the existing ship. Components are sourced from systems currently commercially available and in the case of the dual fuel engines, where commercial ammonia engines are not yet available, the mass and dimensions of such systems are assumed similar to conventional slow-speed MGO or LPG fuelled engines (analysis in appendix E.0.2). The aim is to outperform the baseline case (in terms of KPIs later mentioned in Section 3.6) while maintaining design specifications of the original ship. The scope of this work does not include designing the propulsion system in detail so the purpose of having the system in the general arrangement is to have a high level view of available volume and mass allowance for fuel storage systems.

The size of the fuel reformer has to be estimated analytically based on the quantities of ammonia required to be converted to feed the dual fuel engine. The calculation is shown in appendix E.0.5.

The main assumption in the system integration procedure is that in order to retain design draft the overall displacement of the ship must remain constant or as close to it as possible. It is therefore assumed that the case 1 baseline displacement must not be exceeded . This section is where it becomes apparent which fuel tank solution would be most appropriate for the space available and if there is allowance for heavy fuel tanks or if lightweight fuel tanks need to be investigated. Worst case scenario is that deck space is lost to allow for more machinery or tank space which would reduce the amount of passengers or cargo allowed onboard. Appendix H details how the weight distribution changes for each case.

Baseline: Fully Electric Powered Ferry

The base case comprises of an electrical system with heavy batteries and inverters. The areas in the boat which carry this heavy equipment have been designed with extra strong stiffening systems and a level deck for installation. These areas are foreseen to be the best areas to install new heavy tanks or dual fuel engines.

Sub-case 3A: Type C Tank(s) with Fully Pressurized Storage

The first step to generating the general arrangement is to evaluate the rules and find the amount of space available and the position on the ship where fuel tanks could be stored. In this case the class rules of BV NR 529 for Gas Fuelled vessels (2022),IGF code (2016), IGC Code (2014) and ASTM VIII div 1 (2019) standards for boilers and pressure vessels were used as reference in order to correctly model the pressure tanks for the correct design pressure. Tanks are designed based off the saturation pressure (1.7 MPa) at an ambient condition of 45°C according to rules (IGF 6.4.9.3.3.1.1).

Most important design restrictions about fuel tank placement arise from accessibility (IGC Chapter 3.5.3) and safety against collision and grounding from IGF (Chapter 5.3), some of the restrictions are:

- Minimum clearance of 450mm between curved tank surface and deck stiffener or 600 mm between flat walled tank surface if a human is required to pass the tank(IGC 3.5.3.5.1).
- There should be a minimum distance of 50 mm between two flat surfaces for visual inspection but a minimum of 380 mm for areas that a person may be able to pass through (IGC 3.5.3.5.2 and IGC 3.5.3.5.3).
- The distance between a cargo tank sump and inner bottom without a suction well shall not be less than 50 mm (IGC 3.5.3.5.5).

- Fuel tanks should be located a minimum of B/5 from the side of the ship at the level of the summer load line (B is maximum breadth) (IGF 5.3.3).
- Gas fuel tanks for passenger ships should be located a minimum of B/10 (no less than 0.8 m) from shell plating or aft terminal of the ship.(IGF 5.3.3.4.1).
- Fuel tanks must be a minimum distance of B/15 from the moulded line of the bottom plating at the center-line (IGF 5.3.5).
- Fuel tank lines containing ammonia must be located a minimum distance of 800 mm from the sides of the ship (IGF 5.7.1).
- Fuel piping shall not be led directly through accomodation spaces (IGF 5.7.2).
- The fuel tanks shall be abaft a transverse plane 0.08L from the FPP for passenger ships.

Applicable regulations for the fuel preparation room and airlocks are:

- Fuel preparation spaces should be located on the open deck unless the room complies with the requirements for a tank connection space (IGF 5.8).
- Airlocks should be provided between hazardous and non-hazardous spaces (IGF 5.11.1).
- When a fuel preparation room is located below deck as far as practicable possible an independent access from the open deck should be provided otherwise via an airlock (IGF 5.11.2).
- Airlocks should have two gastight doors spaced at least 1.5m apart but not more than 2.5m apart. Door height should not be less than 300mm (IGF 5.12.1).
- Airlocks shall have a deck area not less than 1.5 m^3 and may not be used as a storage space (IGF 5.12.4).

Applicable regulations for the fuel containment room are:

- When storing Natural Gas in liquid form the Maximum Allowable Relief Valve Setting (MARVS) is 1 MPa (IGF 6.3.1) [5].
- The Maximum Allowable Working Pressure (MAWP) is 90% of the MARVS.
- Tanks below deck must be gas tight towards adjacent spaces (IGF 6.3.3) and Tank connections not on an open deck must be enclosed by gastight connection spaces (IGF 6.3.4).

[5] This regulation is most concerning when needing to store ammonia above 1 MPa in fully pressurized ,ambient temperature conditions

- Pipe connections to fuel storage tank shall be mounted above the highest liquid level in the tanks except for Type C storage (IGF 6.3.5).
- If piping is located below the liquid level of the tank it must be protected by a secondary barrier (IGF 6.3.9).
- Secondary barriers should fulfil their functions up to 30 degrees of heel (IGF 6.4.4.6).
- For single fuel installations the fuel system should be arranged with full redundancy and segregation so that fuel storage shall be divided between two or more tanks and located in separate compartments (IGF 9.3.1 and IGF 9.3.2).

It is also necessary to fit drip trays and a separate bilge tank system to collect any potential aqueous ammonia solution.These drip trays must be able to detect concentrations of ammonia above 30 ppm.

According to IGF code sec 6.7.2.7 [110] the vent mast outlet must be a minimum distance of 6m or B/3 (whatever is greater) above the weather deck to safely release ammonia. Additionally the outlet of the pressure relief valves from the tanks must be at least 10m from any air intake ,opening to accommodation space, service and control spaces or other non-hazardous area (if the pressure relief valves are directed to the vent mast this applies to the vent mast outlet). Similarly according to IGC code section 8.2.11.1 pressure release valve outlet must be no less than 25m or B (whichever is less) from the above mentioned areas, however, for ships less than 90m in length smaller distances may be permissible based on gas dispersion analysis. The lowest permissible distance is therefore B which is a radius of 5.2 m from the outlet of the vent mast.

The tank arrangements are designed with the intention of maximising the volume of fuel stored. From the mentioned restrictions above; a general box shape of the allowable tank placement for a tank with curved surfaces has dimensions L= 3000 mm ,B = 2500 mm and H= 1000 mm. Figures 20,21 and 22 illustrate the space allowance limiting box. Figure 22 illustrates how the sloped gas tight ceiling also must be accounted for which holds the ventilation ducts The fuel system redundancy regulation of needing to use two compartments for the fuel tank can be bypassed by using a single type C tank or having a backup fuel source such as that of the reserve fuel tank located below the fuel deck shown in 20. See appendix D for full drawings of the general arrangements.

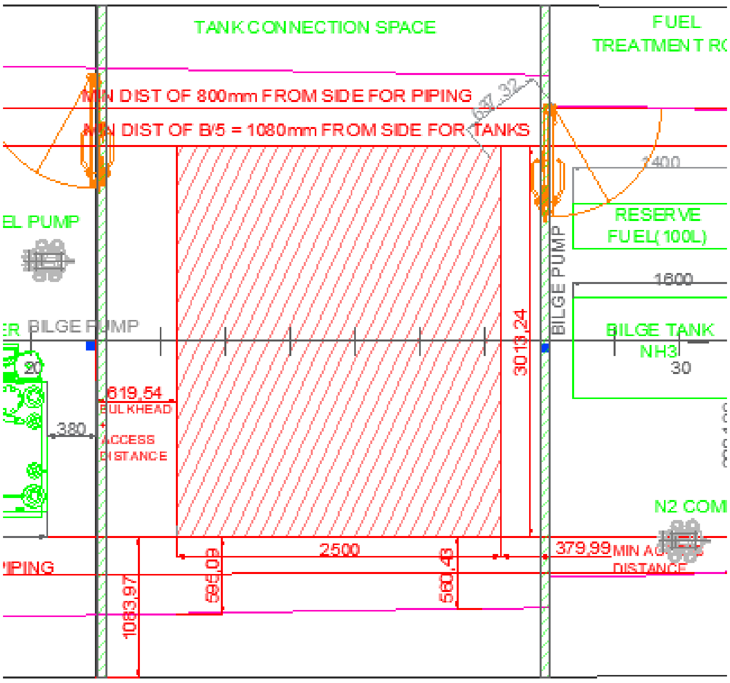


Figure 20: Space allocation on ferry for tanks defined by rules (Top View)

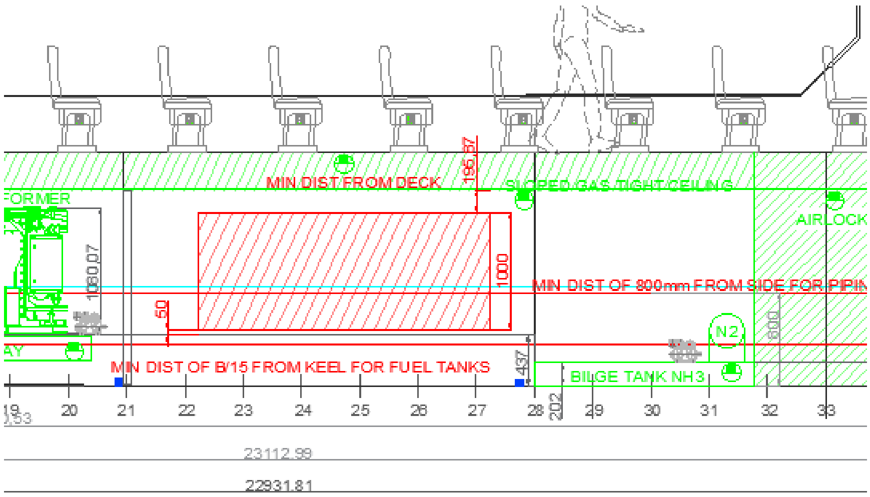


Figure 21: Space allocation on ferry for tanks defined by rules (Front View)

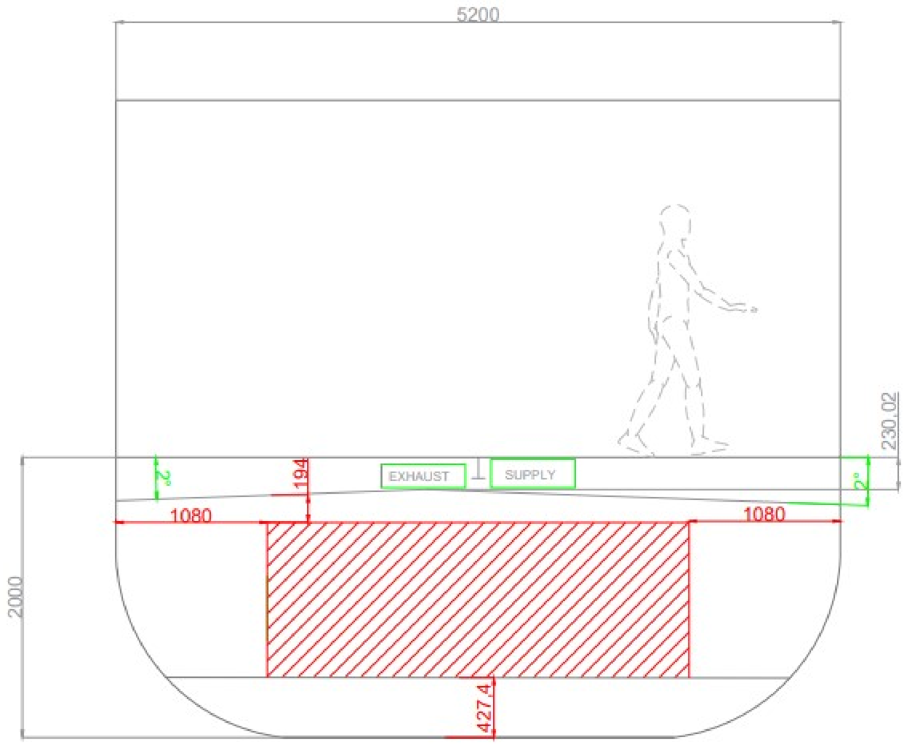
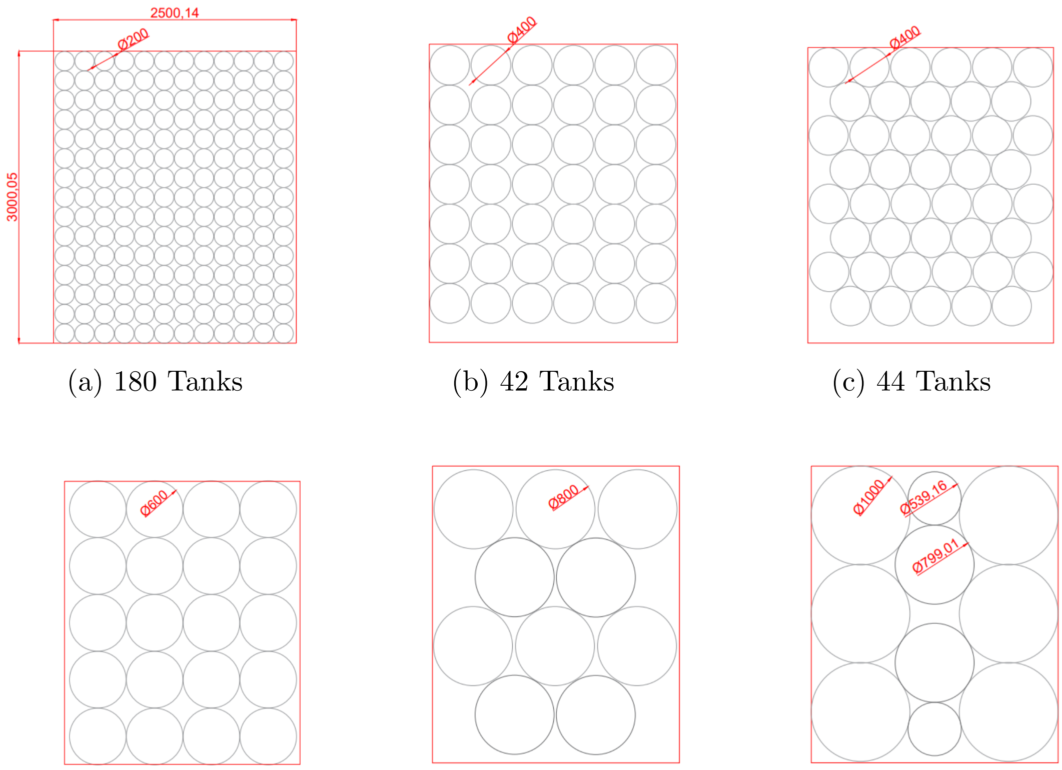


Figure 22: Space allocation on ferry for tanks defined by rules (Side View)

3.5.1 Vertical Type C Cylindrical Tank Arrangements

In terms of tank shape, there are a few possible alternatives for the type C tank. Usual shapes are cylindrical, spherical, bi-lobed or tri-lobed but recently there is also the possibility of having flat-walled prism shapes with rounded corners or rounded walls provided the tank is internally supported by a lattice structure. For the first iteration of general arrangement the possibility of using vertically arranged cylinders was investigated to find solutions that maximised the usage of the space available in Figure 20 and Figure 22.

The comparison of multiple stacking arrangements for various diameter cylinders vertically positioned was completed to study which combination of cylinder radius with the number of tanks maximises the available storage volume. A thing to consider is that the more tanks in the arrangement the more piping required and the higher the risk of accidental leakage. Therefore designs with fewer tanks are preferred. Also worth noting is the larger the cylinder the larger the required thickness to maintain design pressure. On smaller tanks there is a limitation by IGC 4.23.2 that the the minimum thickness of a pressure vessels type C tank may not be less than 5mm for carbon-manganese steels or if not applicable to fuel tanks then BV suggests no less than 3+ D/1500 mm. The seven different vertical stacking arrangements are shown in Figure 23 below followed by the evaluation of the spacial efficiency of each arrangement in Section 4. Ellipsoidal heads were assumed on both top and bottom ends of the tanks to incorporate 3D effects of volume loss.



(a) 180 Tanks (b) 42 Tanks (c) 44 Tanks

(d) 20 Tanks (e) 10 Tanks (f) 10 Tanks

Figure 23: Vertical Cylinder Arrangements

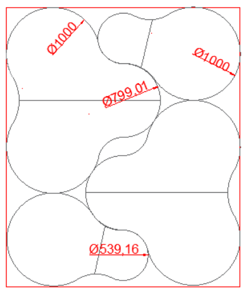


Figure 24: Vertical Arrangement (g) Bilobes and Trilobes

3.5.2 Horizontal Type C Cylindrical Tank Arrangements

The next type of arrangement for the space is to use horizontal tanks. The most simple design which allows for the largest diameter cylinders was chosen to minimize the amount of tanks required onboard. Special attention was paid here to maximize the space saving potential at the end heads where a perfectly flat head is not possible due to stress concentrations. The options are hemispherical, ellipsoidal, conical or torispherical. The minimum distance of the smaller dimension of the ellipse shape is 0.2 x Diameter of the cylinder by the rules (BV NR 467). The dimensions of a cylindrical shape that would maximise the use of available space are detailed in Table 14 and calculation found in appendix F. The tanks were designed to be orientated longitudinally along the ship and equally spaced across the hull to give an even weight distribution and reduce free surface effects in roll motion, consequently horizontal tanks in an arrangement transverse to the centre line were not investigated. The table below summarizes the tank dimensions based on the available storage space of 7.5 m^3

Table 14: Type C Horizontal Pressure Vessel Dimensions

Type C Pressurized Tanks	
Quantity of Tanks	3
Length of cylinder [m]	2.12
Outer Diameter [m]	1
Head type	Ellipsoidal
End cap ellipse height [m]	0.20
Total length of tank [m]	2.52

The calculations for the the tank sizing for type C cylindrical vessels are shown in appendix F where IGF code makes reference to ASME BPVC code and states that pressure vessels should be approved by the administration. Figure 25 shows the side view of the spacing around the tanks.

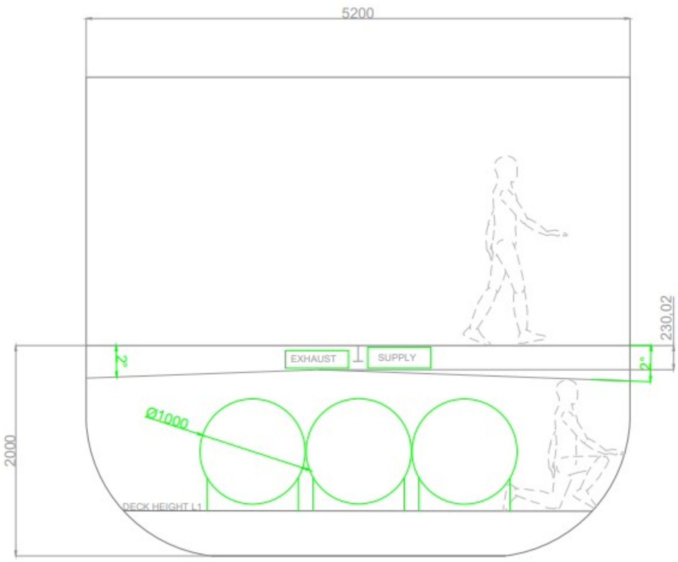


Figure 25: Horizontal cylinders side view of the arrangement

3.5.3 Lattice Pressure Vessel Arrangements

The final type C pressurized arrangement investigated was to look at lattice pressure vessels. Lattice pressure vessels can almost fully use the available space with the only space lost to rounded tank corners and the volume of internal lattice material. If membrane tanks are taken as using the full space then the volume efficiency of a rounded wall lattice tank is said to be in the range of 82% to 90% [68]. Figure 26 illustrates the shape of a round walled LPV as provided by CTO of Lattice Technology.

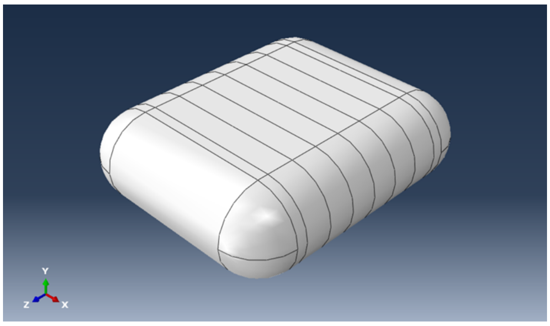


Figure 26: LPV design as provided by Lattice Technology

Strength calculations for lattice pressure vessels were not specifically calculated due to the specialised patented design which was not publicly accessible therefore a quotation was requested from the CTO of Lattice Technology for an overall tank weight and volume to estimate the material needed and space taken up by the tank. The quotation was given based on a round walled LPV tank designed for a pressure of 1.8 MPa. The article on LPVs by Bergen (Bergen,2017) [68] or more specifically the data in figures 7 and 8 was used to then scale volume and mass to the other types of LPVs taking into account that previous weight of tanks in Figure 8 was based on a design pressure of 0.3 MPa therefore a greater amount of tank material would be expected for a tank designed for 1.8 MPa.

Sub-case 3B: Tank C with Semi-Pressurized Storage

As with case 3A the most important design restrictions about fuel tank placement arise from accessibility (IGC Chapter 3.5.3) and safety against collision and grounding from IGF (Chapter 5.3). Alternative to this placing arrangement a probabilistic method proposed by SOLAS regulation II-1/7-1.1.1.1. can be followed to place tanks using an F_{CN}.

The general arrangement for semi-pressurized storage can be regarded as similar to pressurized storage with the extra requirement of a single stage compressor and a condenser to re-liquefy boil off gas. This equipment should be stored in a separate room to the tank connection space and insulated against fire as well as ventilated in case of ammonia leakage. A liquid return line is an additional requirement for these kind of tanks. For redundancy and safety all tanks should have their own compressor and condenser which means that having more than one tank could be space demanding and hence another reason why a lattice pressure vessel could be a promising option where pressure build up is designed for.

Sub-case 3C: Type A Tank with Fully Refrigerated Storage

The type A tank is a rectangular prism shaped tank often ending in a trapezium shaped top wall to prevent sloshing effects. This chamfer at the top can reduce the gross volume of the tank by some small margin which is incorporated into the calculation of the tank in appendix F. The type A tank is usually filled with refrigerated ammonia and usually requires a 2 stage compression system to re-liquefy vapors. This two stage system uses an inter cooler tank, a recovery tank , two compressors and a condenser. This equipment is to be arranged in the fuel treatment room 2 of the general arrangement in appendix D. Figure 27 shows the typical shape of a type A or membrane tank that would fit within the boundaries for prevention of tank rupture on collision.

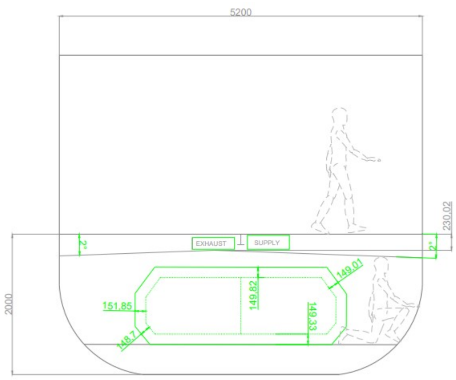


Figure 27: Side view of a Type A or Membrane tank

For more detailed general arrangements see appendix D.
Sub-case 3D: Membrane Tank with Fully Refrigerated Storage

The membrane tank general arrangement is in many ways the same as the general arrangement for the type A tank. This is mainly attributed to the small size of the passenger ferry in which the collision and grounding limits for fuel tank placement fall far inside the hold space which greatly reduce the benefits of using this type of tank arrangement. Usually on larger ships the collision distance can be measured to the inner barrier which would exclude the insulation layer from taking up fuel volume yet due to the small size of the tank the insulation layer will take up a significant portion of the fuel volume. Other than fuel tank placement the two stage refrigeration system described for the type A tank will also take up additional storage volumes that may be needed for cargo. The membrane tank itself is hexagonal shaped to reduce sloshing effects, the difference between Figure 27 and a single membrane tank would be the longitudinal bulkhead. In the case of the membrane tank a longitudinal bulkhead would not be installed as it would cause a heat bridge from the outside shell of the ship to the inner refrigerated ammonia.

3.6 Definition of Key Performance Indicators

The Key Performance indicators (KPI´s) that can be evaluated are:

- Mass of the tanks
- Volume of Fuel
- Volume efficiency
- Material Ratio
- Mass of Fuel to Mass of Tank Ratio
- Approximate capital cost of the system (EUR),

The mass of the tanks will first be calculated taking into consideration the material that should be used to achieve the required strength. The volume of fuel is simply the gross tank volume minus all the material. The material ratio η and volume efficiency ξ are defined as follows:

$$\xi = V_{tank}/V_{prism} \tag{15}$$

Where V_{tank} represents the gross volume of the tank and V_{prism} represents the volume of an ideal rectangular prism fitting the exact space available.

$$\eta = V_{material}/V_{stored} \tag{16}$$

Where $V_{material}$ represents the actual volume of material used for making the tank. V_{stored} represents the volume of space available to store fluid in the tank. For solutions of multiple tanks the gross volume of all the tanks was be used.

The KPIs are first evaluated taking into consideration a fixed space allowance to fit the tanks. What is not considered at first is if the mass of the tanks needs to be increased or decreased to meet the stability criteria. Also not first considered is the insulation requirement of the tanks to maintain refrigerated or semi-pressurized conditions.

After insulation requirements are taken into account the mass of fuel to mass of tank ratio can be calculated which remains fixed for scaling the tank. The tanks are scaled on this ratio to adjust weight to meet the stability criteria for the following analysis.

Following the boil off calculation and stability analysis the actual tank size was calculated and KPIs were re-evaluated to yield a second set of results. With the second set of results the range of the vessel could be calculated and finally the capital costs of manufacturing such tanks.

4 Results and Discussions

This section summarizes results obtained by evaluating 17 different tank alternatives described earlier. As mentioned there are four phases of analyses which help to identify the most favorable solutions.

4.1 Tank Analysis 1: No Insulation and No Stability Check

In this analysis the tanks are considered standalone tanks without insulation. This kind of analysis is often documented in feasibility studies for ammonia as fuel. It is included here to illustrate how the choice of tank can change after taking into consideration insulation requirements.

Table 15: Key Performance Indicators Without Insulation and Stability Checks

Tank Type	No.of Tanks	Gross Vol (m3)	Max Vol Fuel (m3)	Material Ratio (η)	Volume Efficiency (ξ)	Mass of tank(s) (kg)
Type C P (Vert Cyl-A)	180	5.50	5.0	11%	73%	4195
Type C P (Vert Cyl-B)	42	4.98	4.8	4%	66%	1502
Type C P (Vert Cyl-C)	44	5.22	5.0	4%	70%	1574
Type C P (Vert Cyl-D)	20	5.18	5.0	4%	69%	1537
Type C P (Vert Cyl-E)	10	4.47	4.3	4%	60%	1302
Type C P (Vert Cyl-F)	10	5.37	5.2	4%	72%	1558
Type C P (Vert Cyl-G)	4	5.33	5.1	4%	71%	1807
Type C P (Horizontal Cyl)	3	5.65	5.3	7%	75%	2794
Type C P (LPV FW tank)	1	6.77	6.0	13%	94%	6167
Type C P (LPV RC tank)	1	6.41	5.8	11%	89%	4933
Type C P (LPV RW tank)	1	6.25	5.8	8%	83%	3700
Type C SP (Horizontal Cyl)	3	5.65	5.5	3%	78%	1503
Type C SP (LPV FW tank)	1	6.77	6.4	5%	94%	2716
Type C SP (LPV RC tank)	1	6.41	6.1	4%	89%	2173
Type C SP (LPV RW tank)	1	6.25	6.0	3%	83%	1630
Type A Refrigerated Tank	1	7.28	6.4	14%	97%	2748
Membrane Refrigerated Tank	1	7.28	7.0	3%	97%	1800

Most favorable alternatives obtain a high maximum volume of fuel with low material ratio and a high volume efficiency. In Table 15 it is first notable that the membrane tank far outperforms the other alternatives achieving the most desirable values across all three KPIs mentioned. Typically the lower the material ratio the lower the material costs for the tanks. Out of all the fully pressurized options the best performer in terms of material ratio are the vertical cylinders yet when it comes to volume of fuel and volume efficiency the horizontal alternative or lattice pressure vessels are preferred.

All vertical arrangements have a volume efficiency between 60 % and 73%. To break this barrier, and make more use of the space available, an array of horizontal cylinders or a lattice pressure vessel should be used. Surprising at first glance is the high material ratio of the *Vertical Cylinder Arrangement A* in comparison to other vertical tank arrangements but this can be attributed to the requirement of IGC code to have a minimum wall thickness of each individual tank above 5 mm.

The most lightweight options for pressurized storage are vertically arranged tanks yet the decrease in maximum volume of fuel for some arrangements may make horizontal cylinders more attractive. It should be realized that with additional tanks the amount of piping will also increase leading to higher risks of leaks at connection points and a greater amount of specialised valves required. Notable also is that bunkering would be much faster and safer if fewer tanks are used.

Table 16 takes into consideration different fuel densities and maximum allowable filling level, in this case the refrigerated storage alternatives far outperform the pressurized storage alternatives. For example the membrane tank can store up to 60% more fuel by mass than the best pressurized solution.

Table 16: Mass of Fuel and Fuel to Tank Mass Ratio

Type of Fuel Tank	Max Fill Level (%)	Fuel Density (kg/m3)	Mass of Fuel (kg)	Mass Fuel / Mass Tank
Type C P (Vert Cyl-A)	85%	570	2407	0.57
Type C P (Vert Cyl-B)	85%	570	2323	1.55
Type C P (Vert Cyl-C)	85%	570	2434	1.55
Type C P (Vert Cyl-D)	85%	570	2418	1.57
Type C P (Vert Cyl-E)	85%	570	2085	1.60
Type C P (Vert Cyl-F)	85%	570	2507	1.61
Type C P (Vert Cyl-G)	85%	570	2470	1.37
Type C P (Horizontal Cyl)	85%	570	2567	0.92
Type C P (LPV FW tank)	85%	570	2902	0.47
Type C P (LPV RC tank)	85%	570	2803	0.57
Type C P (LPV RW tank)	85%	570	2802	0.76
Type C SP (Horizontal Cyl)	90%	625	3071	2.04
Type C SP (LPV FW tank)	90%	625	3614	1.33
Type C SP (LPV RC tank)	90%	625	3450	1.59
Type C SP (LPV RW tank)	90%	625	3400	2.09
Type A Refrigerated Tank	95%	693	4199	1.53
Membrane Refrigerated Tank	95%	693	4640	2.58

4.2 Tank Analysis 2: Insulation Requirement

The following analysis takes into account a more realistic scenario where insulation is assumed to occupy a part of the volume previously only allocated for refrigerated and semi-pressurized fuel. The KPIs are re-evaluated and assessed after boil off rates for semi-pressurized and refrigerated tanks were estimated. The target was to maintain the boil off rate below 0,08 kg/hr to keep the size of the boil off gas refrigeration compressor or compressors as small as possible. With this target it was calculated that insulation of 70 mm polyurethane foam would be able to sustain the desired boil off rate. It should be noted for further study that the pressure build up inside the semi-pressurized tanks should be modelled to predict if the tanks could go without any re-liquefaction for the time between refueling. Another point worth noting is that if vacuum insulation is used the boil off rates could be even lower. Generally it was accepted that having a BOG management system onboard was safer for a more robust solution in case the ship was left unable to reach a port to depressurize tanks without venting to the atmosphere. Table 17 illustrates the KPIs after taking into account the insulation requirement.

Table 17: Key Performance Indicators with Insulation

Tank Type	No.of Tanks	Gross Vol (m3)	Insulation Vol (m3)	Max Vol Fuel (m3)	Material Ratio (η)	Volume Efficiency (ξ)	Mass of tank(s) (kg)
Type C P (Vert Cyl-A)	180	5.50	0	5.0	11%	73%	4195
Type C P (Vert Cyl-B)	42	4.98	0	4.8	4%	66%	1502
Type C P (Vert Cyl-C)	44	5.22	0	5.0	4%	70%	1574
Type C P (Vert Cyl-D)	20	5.18	0	5.0	4%	69%	1537
Type C P (Vert Cyl-E)	10	4.47	0	4.3	4%	60%	1302
Type C P (Vert Cyl-F)	10	5.37	0	5.2	4%	72%	1558
Type C P (Vert Cyl-G)	4	5.33	0	5.1	4%	71%	1807
Type C P (Horizontal Cyl)	3	5.65	0	5.3	7%	75%	2794
Type C P (LPV FW tank)	1	6.77	0	6.0	13%	94%	6167
Type C P (LPV RC tank)	1	6.41	0	5.8	11%	89%	4933
Type C P (LPV RW tank)	1	6.25	0	5.8	8%	83%	3700
Type C SP (Horizontal Cyl)	3	5.65	1.51	4.0	43%	78%	1684
Type C SP (LPV FW tank)	1	6.77	1.05	5.4	26%	94%	2842
Type C SP (LPV RC tank)	1	6.41	0.99	5.1	25%	89%	2292
Type C SP (LPV RW tank)	1	6.25	0.97	5.1	23%	83%	1746
Type A Refrigerated Tank	1	7.275	1.51	4.9	49%	97%	2929
Membrane Refrigerated Tank	1	7.275	3.216	3.8	90%	97%	2186

In this case the pressurized lattice pressure vessels achieve the highest maximum volumes of fuel yet when it comes to the mass of fuel (Table 18) the type A refrigerated tank achieves the maximum. It is evident for a small tank size the membrane tank requires a relatively large percentage of insulation, thus it can be seen as a reason why membrane tanks are generally used on large cargo tanks and have not yet been considered for small scale fuel tank application. Important to note is also the superior sloshing mitigation in lattice pressure vessels, type A SPB tanks and cylinders in comparison to membrane tanks.

Table 18: Mass of Fuel and Fuel to Tank Mass Ratio after insulation addition

Type of Fuel Tank	Max Fill Level (%)	Fuel Density (kg/m3)	Mass of Fuel (kg)	Mass Fuel / Mass Tank
Type C P (Vert Cyl-A)	85%	570	2407	0.57
Type C P (Vert Cyl-B)	85%	570	2323	1.55
Type C P (Vert Cyl-C)	85%	570	2434	1.55
Type C P (Vert Cyl-D)	85%	570	2418	1.57
Type C P (Vert Cyl-E)	85%	570	2085	1.60
Type C P (Vert Cyl-F)	85%	570	2507	1.61
Type C P (Vert Cyl-G)	85%	570	2470	1.37
Type C P (Horizontal Cyl)	85%	570	2567	0.92
Type C P (LPV FW tank)	85%	570	2902	0.47
Type C P (LPV RC tank)	85%	570	2803	0.57
Type C P (LPV RW tank)	85%	570	2802	0.76
Type C SP (Horizontal Cyl)	90%	625	2223	1.32
Type C SP (LPV FW tank)	90%	625	3024	1.06
Type C SP (LPV RC tank)	90%	625	2891	1.26
Type C SP (LPV RW tank)	90%	625	2855	1.63
Type A Refrigerated Tank	95%	693	3205	1.09
Membrane Refrigerated Tank	95%	693	2523	1.15

The mass of fuel to the mass of the tank ratio is important for lightweight transport applications, generally speaking the higher the ratio the more favorable. This is where the vertical cylinder arrangements find their biggest positive claiming on average the highest mass of fuel per unit of tank mass after semi-pressurized round walled pressure vessels.

4.3 Tank Analysis 3: Intact Stability Criteria

The third analysis evaluates the tank arrangements for use on a ship which can be affected by free surface moments. Tank weight plays a more important role in maintaining stability and draft restrictions for smaller ships such as the passenger ferry under study. It is important to realise that certain tanks cannot be used standalone on a vessel without necessary boil-off gas management systems. In this third analysis the auxiliary systems are accounted for and stability checks are performed to determine the maximum and minimum permissible tank weight. The various refrigeration auxiliary equipment mass and estimated costs are provided in Table 19.

Table 19: Auxiliary Items

Auxiliary Component	**Mass (kg)**	**Estimated Cost (EUR)**
BOG Compressor	390	2000 [111]
BOG Condenser	100	2000
Condenser Pump	20	400
Inter-cooler Tank	100	8000
Recovery Tank	100	8000
Pressure Relief Valve	0.5	184.92
Expansion Valve	0.5	239.90

Table 20: Maximum and Minimum Tank System Weight To Meet Stability Criteria

Limit	**Weight (kg)**
Maximum Total Weight of Tank System	6893
Minimum Tank(s) Weight [No Bulkhead]	3015.58
Minimum Tank(s) Weight [1 Bulkhead]	915
Minimum Tank(s) Weight [2 or more Bulkheads]	616

The maximum and minimum allowable tank weights calculated using the weight distribution table of appendix H and verified for stability using Max-surf advanced stability module are shown in Table 20. If the total weight of the system was exceeded the tank was scaled down to remove tank weight. Overall this resulted in a reduction of fuel volume which is then accounted for in Table 22. The column for the mass of the system in Table 22 is based on the fuel system including auxiliary weights.

Table 21: Mass Adjustment after stability criteria check

Type of Fuel Tank	Mass of System (kg)	Add or Remove Weight	Adjustment (kg)	New Mass Of Fuel and Tank (kg)	New Mass of Tank (kg)
Type C P (Vert Cyl-A)	6692	Permissable	0	6602	4195
Type C P (Vert Cyl-B)	3847	Permissable	0	3826	1502
Type C P (Vert Cyl-C)	4030	Permissable	0	4008	1574
Type C P (Vert Cyl-D)	3965	Permissable	0	3955	1537
Type C P (Vert Cyl-E)	3393	Permissable	0	3387	1302
Type C P (Vert Cyl-F)	4071	Permissable	0	4066	1558
Type C P (Vert Cyl-G)	4280	Permissable	0	4278	1807
Type C P (Horizontal Cyl)	5363	Permissable	0	5361	2794
Type C P (LPV FW tank)	9070	Remove	-2292	6777	4608
Type C P (LPV RC tank)	7738	Remove	-960	6777	4321
Type C P (LPV RW tank)	6503	Permissable	0	6502	3700
Type C SP (Horizontal Cyl)	5297	Permissable	0	3906	1684
Type C SP (LPV FW tank)	6476	Permissable	0	5866	2842
Type C SP (LPV RC tank)	5793	Permissable	0	5183	2292
Type C SP (LPV RW tank)	5211	Permissable	0	4601	1746
Type A Refrigerated Tank	7235	Remove	-457	5676	2711
Membrane Refrigerated Tank	5810	Permissable	0	4708	2186

Table 22 illustrates how the mass of the fuel onboard the ship would influence the range of the vessel. The fuel consumption rate was calculated as 0.0175 kg/s in section 3.3.1 using a HCCI engine burning ammonia and hydrogen. The columns for the range of the vessel are conservative in that they assume the ship runs continuously at full load condition. The days without refuel column assumes a 10 hour working day in which the ferry is run at full load.

Table 22: KPIs based on fuel consumption

Type of Fuel Tank	New Mass of Fuel (kg)	Range of Vessel (hrs)	Days without refuel	Range (km)
Type C P (Vert Cyl-A)	2407	38.1	3.8	636
Type C P (Vert Cyl-B)	2323	36.8	3.7	614
Type C P (Vert Cyl-C)	2434	38.6	3.9	643
Type C P (Vert Cyl-D)	2418	38.3	3.8	638
Type C P (Vert Cyl-E)	2085	33.0	3.3	551
Type C P (Vert Cyl-F)	2507	39.7	4.0	662
Type C P (Vert Cyl-G)	2470	39.1	3.9	652
Type C P (Horizontal Cyl)	2567	40.7	4.1	678
Type C P (LPV FW tank)	2169	34.4	3.4	573
Type C P (LPV RC tank)	2455	38.9	3.9	648
Type C P (LPV RW tank)	2802	44.4	4.4	740
Type C SP (Horizontal Cyl)	2223	35.2	3.5	587
Type C SP (LPV FW tank)	3024	47.9	4.8	798
Type C SP (LPV RC tank)	2891	45.8	4.6	763
Type C SP (LPV RW tank)	2855	45.2	4.5	754
Type A Refrigerated Tank	2966	47.0	4.7	783
Membrane Refrigerated Tank	2523	40.0	4.0	666

The semi-pressurized flat walled lattice pressure vessel offers the the greatest mass of stored ammonia onboard after scaling down the Type A refrigerated tank to meet the max draft requirements of 0.85m. The best alternative for maximum fuel mass in the pressurized storage alternatives was the round walled lattice pressure vessel as the overall weight of the tank did not need to be reduced.

In comparison to the original baseline case where the lithium ion battery powered vessel was designed for 2.5 hrs of continuous operation at maximum load the new ammonia solution offers up to 47.9 hours using semi-pressurized lattice pressure vessels. This significant improvement means that either the ferry can do longer voyages or refuel less frequently. Assuming the ferry would be used on average for 10 hours a day the ferry could go without refueling for a total of 4.8 days. It is preferable to have a solution which allows for less frequent bunkering as refueling has often been found to be the time when most accidents with ammonia leakage occur as shown in the study on ammonia nurse tanks [57]. It is recommended to refuel the vessel when there are no passengers or crew onboard the vessel. Most accidents noted in the literature occurred just hours after refueling therefore it is further recommended that a safety window of time after refueling before passengers are allowed onboard the vessel should pass to ensure that the system has no leaks and that the tanks are not over-pressurized.

4.4 Material Selection

The tank material selection is based on criteria set out in IGC rules for strength requirement and stress corrosion cracking resistance. Together with this the requirements for material composition of low temperature carbon-manganese steels of IGF rules table 7.2. CES EduPack 2016 software was used to determine a set of suitable materials that could be used for tank material.

Table 23: Alloying element composition limits[110]

Element Constituent	**Percentage**
C	0.16 max
Mn	0.7-1.6
Si	0.1-0.5
S	0.025 max
P	0.025 max
Ni	0.8 max
Cr	0.25 max
Mo	0.08 max
Cu	0.35 max
Nb	0.05 max
V	0.1 max

It was identified that low temperature carbon-manganese steels require low carbon and higher manganese content to improve weld-ability. Other alloying elements improve brittle behaviour at low temperatures. These alloying limits generally disqualify many carbon steels from being used as a low temperature tank material. Since ammonia is corrosive to nickel and copper, all materials with a high content of these materials are disqualified. The constraint used in CES EduPack which eliminates these materials is setting an acceptable tolerance to strong alkalis.

Table 24: Strength constraints of the material

Material Property	Value
Minimum Yield Strength (MPa)	355
Maximum Yield Strength (Mpa)	440
Minimum Tensile Strength (Mpa)	410
Tolerance to Strong Alkalis	Acceptable

The final list of suitable materials is listed in Table 25. Carbon Manganese steels are listed first followed by austenitic stainless steels. The first six steels on the list from the CES database can also be classified as carbon manganese steels. The ASTM designated steels are those used in industry for manufacturing ammonia nurse tanks. The ship building steel was recommended by the CTO of Lattice Technology as a possible alternative due to its yield strength characteristics and corrosion resistance. Noted in IGF code is that carbon-manganese steels with yield strength exceeding 410 MPa should also undergo post weld heat treatment to prevent brittle behaviour occurring in the heat effected zones (HAZ).

Table 25: List of Applicable Steels that meet standard requirements

Steel Type	**Designation**	**Treatment**
Dual phase steel (CES database)	YS350	Cold Rolled
Dual phase steel (CES database)	YS400	Hot Rolled
High Strength low alloy steel (CES database)	YS300	Cold Rolled
High Strength low alloy steel (CES database)	YS350	Cold Rolled
High Strength low alloy steel (CES database)	YS355	Hot Rolled
High Strength low alloy steel (CES database)	YS420	Cold Rolled
Carbon-Manganese (Shipbuilding steel)	DNV FH36	Solution Anneal
Carbon-Manganese (Pressure vessel steel)	ASTM A285	Solution Anneal
Carbon-Manganese (Pressure vessel steel)	ASTM A455	Solution Anneal
Carbon-Manganese (Pressure vessel steel)	ASTM A516 grade 70	Solution Anneal
Stainless Steel (Austenitic)	AISI 302	HT grade B
Stainless Steel (Austenitic)	AISI 304	1/8 hard
Stainless Steel (Austenitic)	AISI 304L	Solution Anneal
Stainless Steel (Austenitic)	AISI 316	Solution Anneal
Stainless Steel (Austenitic)	AISI 316L	Solution Anneal
Stainless Steel (Austenitic)	AISI 321	Solution Anneal
Stainless Steel (Austenitic)	AISI 347	Solution Anneal

The design objectives aimed to be minimized were: thermal conductivity, density and price. The lower the thermal conductivity, the better insulator the material will be and the lower the density the lighter the tanks will be. Generally since all the materials are steel the density is quite similar between the alternatives (in the range of 7850-8000 kg/m^3). The austenitic stainless steel 300 series is generally 7 times the price of the carbon steel as shown in Figure 28. The benefits of 300 series stainless steel are the greater tolerance to ammonia stress corrosion cracking.

Carbon steels are generally less resistant to corrosion yet high strength low alloy steel such as that used by the ship building industry is evidently a plausible material to use with ammonia but paying careful attention to oxygen and water content of the stored ammonia. A method of reducing oxygen content in stored ammonia suggested by the invention in section 2.5.2 on a previously patented anhydrous ammonia storage tank.

Austenitic stainless steel has a thermal conductivity of 15 W/m.C whereas carbon steel is much higher at 50 W/m.C making stainless steel a better insulator. It should be noted that in practice this difference is negligible when compared to insulators such as polyurethane foam. The stainless steel with the highest yield strength was SS304 with 1/8 hard treatment. Therefore it is suggested to use such a material for stainless steel tank types, a current cost evaluation table in the following sub section confirms that in Europe SS304 is generally the cheaper alternative over SS316. Generally SS304 would provide a robust maintenance free tank shell with little susceptibility to corrosion and the ability to deform without leakage in the event of a collision or grounding.

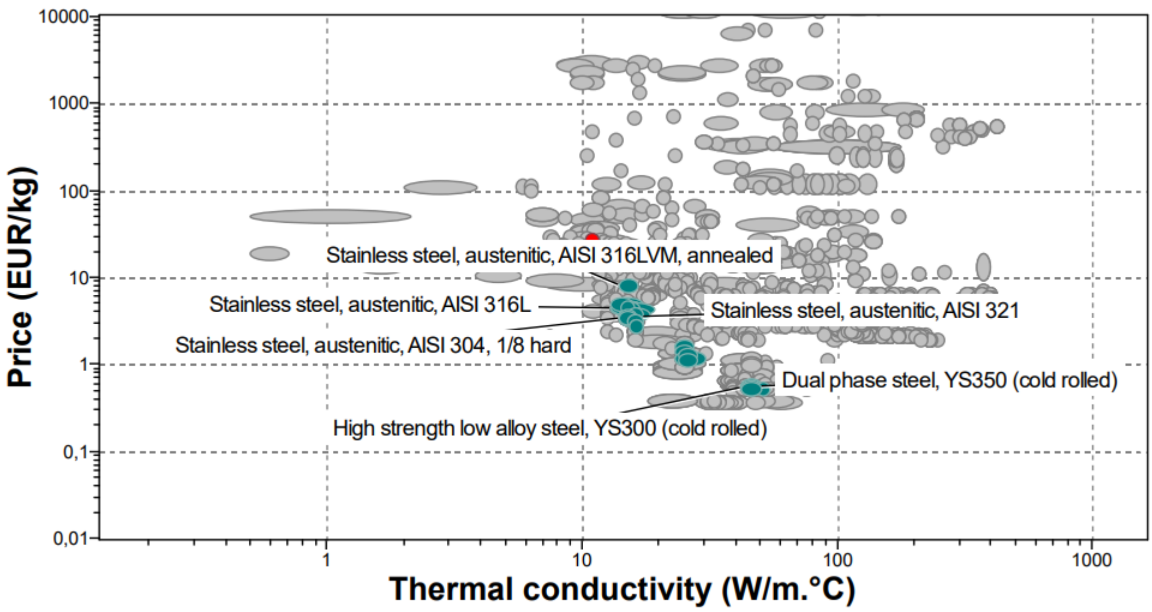


Figure 28: CES (2016) Evaluation of Materials in Terms of Cost and Thermal Conductivity

The use of aluminium for tanks was also evaluated. Generally it is accepted that dry ammonia (containing no water content) is not corrosive to aluminium, however, an aqueous solution or a solution with just trace amounts of water will react and corrode aluminum surfaces. Coupled with this aluminum will generally require 3 times the thickness of material to achieve the same strength as carbon steel, therefore, in terms of space saving and higher fuel volumes using carbon steel is a better alternative. The main advantage of aluminium is its weight saving capability and low temperature resistance, yet on a ship weight saving may not be as important as say in the aerospace industry. Aluminium is generally also a more expensive material than steel so if steel can be used it is preferable for cost saving. For LNG storage, temperatures need to be below -165 $°C$ before aluminium is considered for a tank material[110].

Finally the use of polymers and glass/ carbon fibre reinforced tanks were looked at. Polyethylene such as HDPE or XLPE is known to be chemically resistant to anhydrous ammonia and used as tank material to store a number of chemicals yet it does not have a high yield strength so cannot handle pressurized ammonia on its own. As mentioned in section 2.2 the use of composites for ammonia storage is a recent topic of study therefore future studies will dictate if it is a suitable material, however, it can be said that with current costs of carbon fiber material (CFRP) it will not be a competitive alternative for ship ammonia storage tanks being over ten times the cost of using stainless steel. The recent implosion of the Titan submarine has also brought global bad publicity for the use of composite materials for pressurized tanks demonstrating that small defects in composite shells can lead to catastrophic consequences [112].

4.5 Analysis 4: Cost Estimation of Material and Manufacturing

The cost of manufacturing the tanks can be estimated with a knowledge of manufacturing processes. The type of welding procedures, the forming processes, the weld time, forming time, the labor requirement, electricity cost, electricity consumption and the consumables requirement all play a part in accurately estimating manufacturing costs. Coupled with the labor requirement is the labor rate which is heavily region dependent, it is possible to have vastly different quotes from companies in different countries. Therefore for a good comparison of manufacturing costs the labor rate has to be fixed to a chosen country. Likewise electricity costs to run machinery should also be fixed across different tank manufacturing techniques.

Cutting Edge Industrial Sales [113] offers a guide on welding costs. The guide recommends that labour and overhead costs account for approximately 85% of the total welding cost. Power costs usually account for less than 1% and the remaining 14% is related to the specific weld process. The source compares four welding techniques in terms of total cost per unit length (1 \$/ft = 2.95 EUR/m). The source evaluated 20 welding variables such as electrode diameter,Wire feed speeds, travel speeds and operator costs amongst others. The four welding techniques which were compared were Gas metal Arc Welding (GMAW),Flux Core Arc Welding (FCAW),Shielded Metal Arc Welding (SMAW) and Submerged Arc Welding (SAW). To find out more about the costs in Europe an industrial expert was contacted to gain a better cost of operation. The costs for each process were tabulated as follows:

Table 26: Welding Price Estimate based on type

Welding Type	**Price (EUR/m)**
SMAW Manual	6.33 [113]
GMAW Semi-Auto	4.44 [113]
FCAW Semi-Auto	4.10 [113]
GTAW	16.67
SAW	4.76
k-TIG	2.38

The cost of GTAW is exceptionally high mainly due to the extremely slow weld speed required (0.05 m/min). A new technique known as keyhole TIG welding is able to weld stainless steels at a speed equal to or exceeding SAW welding with deep penetration. This kind of welding technique has been used on stainless steel pressure vessels up to 16mm thickness[114]. This would by far be the best method for cylindrical pressure vessels built from stainless steel whereas other tanks would likely use traditional methods with automated welding procedures. The industrial expert was also asked to approximate costs for forming methods which included labor costs and consumables. The costs shown in Table 27

Table 27: Forming Methods Cost

Forming Method	Cost (EUR/hr)
Hydraulic Press Forming	60
Rolling	60
Plasma Cutting	60
Milling	45
Post Weld Heat treatment	40
Painting	25

Finally an overall manufacturing cost per kg of tank material was estimated by the industrial expert for three tank types. These costs were not applied directly but instead were used as a scaling factor to translate estimated manufacturing costs closer to realistic costs.

Table 28: Tank types and costs

Forming Method	Cost (EUR/kg)
Cylindrical Pressure Vessel	16
Lattice Pressure Vessel	12
Type A or B Tank	14.5

Generally the pressurized tanks are more expensive as there are various testing procedures necessary to ensure safety along with high precision welding techniques. The lattice pressure vessels come in at a relatively low cost due to the use of thin shells and basic welding techniques. The Type A tank is similar to that of the lattice pressure vessel yet slightly thicker material is used on the shell which necessitates more costly welding procedures. Membrane tanks are expected to have manufacturing costs somewhere in the range of Type A (assuming the cost of constructing the secondary barrier or ship double bottom).

The material cost on the other hand is more easily estimated. It is also region specific and market dependent so it may vary from month to month or even day to day but generally looking at market trends an average value can be found. MEPS international was consulted for relatively recent steel prices from January 2023 [115].

Table 29: Material Alternatives for tank manufacture

Material	Density (kg/m3)	(EUR/kg)
CFRP (prepreg)	1750	50 [116]
Carbon-Manganese Steel	7930	0.959 [115]
SS304L	7930	3.825 [115]
SS316	7980	5.731 [115]

With these costs and densities it can be extrapolated based on the calculated volume of material required for the tank how much the tank would cost in terms of material. Generally it can be concluded from the literature study that pressurized tanks are made from carbon manganese steel, type A tanks are a made of a stainless steel inner layer and carbon steel outer layer, membrane tanks are made from a carbon steel outer shell and a stainless steel membrane.

SS304L was chosen as the most appropriate austenitic stainless steel from the material analysis so it will be used for the cost estimation of stainless steel parts. For a fair comparison and in the interest of passenger safety, to avoid stress corrosion cracking, SS304L austenitic stainless steel was chosen to construct the primary barrier of all tanks. Evidently using carbon manganese steel would be cheaper for cylindrical tanks, therefore, it is suggested for use with large ships where the risk of human harm is lower than on a small passenger ferry.

Table 30 describes the welding and forming costs estimated for the various tank types. It was assumed that the primary manufacturing processes that would differentiate different tanks would be costs associated with welding and costs for rolling and press forming. It was assumed for smaller cylindrical tanks that less metal work forming time would be required (1 hour for rolling small cylinders versus 4 hours for rolling larger cylinders). Similar assumptions were made for the pressing time required for making curved or rounded steel plates.

The welding costs were estimated by first quantifying which weld techniques would be used for each tank design. The use of k-tig welding was exploited for cylindrical stainless steel pressure vessels while standard GMAW automated welding was assumed for the default welding type. For the membrane tank it was assumed that for specialised thin material membrane welds GTAW welds may be necessary in 20% of the design.

The weld lengths were estimated as the typical weld seams seen on a pressure vessel (1 longitudinal and 2 to weld the cylinder heads to the cylinder. An estimated 20% extra weld length was assumed for all designs, It should be noted that for lattice pressure vessels the weld length estimation is quite difficult to estimate since the interior lattice structural shape of stiffeners is not known. To define this kind of structure a separate FEM analysis and additional calculations would be necessary. The assumption was that at least 10 x 8 x 3 stiffeners would be needed to makeup the internal structure and with such stiffeners 2 fillet welds along the seams of such stiffeners. The costs per unit meter of weld from Table 26 are assumed for a full weld including multiple passes.

Table 30: Estimated Manufacturing Costs For Various Tanks

	Welding Type	Weld Length (m)	Welding Cost (EUR)	Rolling Time (hrs)	Hydraulic Pressing (hrs)	Forming Cost (EUR)
Type C P (Vert Cyl-A)	k-tig (80%) , GMAW (20%)	487	1,360.80	180	360	32,400.00
Type C P (Vert Cyl-B)	k-tig (80%) , GMAW (20%)	177	494.34	50	168	13,104.00
Type C P (Vert Cyl-C)	k-tig (80%) , GMAW (20%)	186	517.88	53	176	13,728.00
Type C P (Vert Cyl-D)	k-tig (80%) , GMAW (20%)	114	319.60	40	120	9,600.00
Type C P (Vert Cyl-E)	k-tig (80%) , GMAW (20%)	72	201.90	20	60	4,800.00
Type C P (Vert Cyl-F)	k-tig (80%) , GMAW (20%)	87	244.00	20	60	4,800.00
Type C P (Vert Cyl-G)	k-tig (80%) , GMAW (20%)	109	304.99	20	60	4,800.00
Type C P (Horizontal Cyl)	k-tig (80%) , GMAW (20%)	29	80.90	12	18	1,800.00
Type C P (LPV FW tank)	k-tig(20%) GMAW(80%)	599	2,410.67	4	12	960.00
Type C P (LPV RC tank)	k-tig(20%) GMAW(80%)	479	1,928.54	12	18	1,800.00
Type C P (LPV RW tank)	k-tig(20%) GMAW(80%)	359	1,446.40	18	20	2,280.00
Type C SP (Horizontal Cyl)	k-tig (80%) , GMAW (20%)	29	80.90	12	18	1,800.00
Type C SP (LPV FW tank)	k-tig(20%) GMAW(80%)	599	2,410.67	4	12	960.00
Type C SP (LPV RC tank)	k-tig(20%) GMAW(80%)	479	1,928.54	12	18	1,800.00
Type C SP (LPV RW tank)	k-tig(20%) GMAW(80%)	359	1,446.40	18	20	2,280.00
Type A Refrigerated Tank	GMAW(100%)	926	4,108.62	0	0	-
Membrane Refrigerated Tank	GMAW(80%), GTAW (20%)	463	3,187.45	0	48	2,880.00

After contacting various suppliers of pressure vessels it was clear that the overall cost of manufacturing was too low. The main reasons being the other manufacturing costs such as for heat treatment, cutting, milling, painting and inspections/ testing were not included. As a result a correction factor needed to be implemented to make the manufacturing costs more realistic. The correction factor was derived by using the estimates of the overall manufacturing costs of Table 28. The manufacturing cost of making the horizontal cylinders was originally quoted for, therefore, to produce a correction factor that could be applied in general to all the pressure vessels the base case was taken for the horizontal cylinders. The manufacturing cost found using the overall manufacturing cost figure (16 EUR/kg multiplied by material cost) was divided by the manufacturing cost found in in Table 30. This gave a correction factor of 24. There after all pressurized tanks were multiplied by this factor to make the manufacturing cost ration more realistic. The refrigerated and semi-pressurized lattice pressure vessels used a smaller correction factor as Table 28 indicated relatively lower costs. The higher correction factor for pressure vessels can be attributed to additional safety tests and standards that have to be met to meet pressure vessel code requirements.

Table 31: Estimated manufacturing costs adjustment

	Total Manufacturing Cost (EUR)	Correction Factor	Adjusted Total Manufacturing Cost (EUR)
Type C P (Vert Cyl-A)	33,760.80	24	810,259.26
Type C P (Vert Cyl-B)	13,598.34	24	326,360.06
Type C P (Vert Cyl-C)	14,245.88	24	341,901.02
Type C P (Vert Cyl-D)	9,919.60	24	238,070.30
Type C P (Vert Cyl-E)	5,001.90	24	120,045.52
Type C P (Vert Cyl-F)	5,044.00	24	121,055.90
Type C P (Vert Cyl-G)	5,104.99	24	122,519.87
Type C P (Horizontal Cyl)	1,880.90	24	44,708.41
Type C P (LPV FW tank)	3,370.67	24	80,896.08
Type C P (LPV RC tank)	3,728.54	24	89,484.86
Type C P (LPV RW tank)	3,726.40	24	89,433.65
Type C SP (Horizontal Cyl)	1,880.90	18	33,856.27
Type C SP (LPV FW tank)	3,370.67	18	60,672.06
Type C SP (LPV RC tank)	3,728.54	18	67,113.65
Type C SP (LPV RW tank)	3,726.40	18	67,075.24
Type A Refrigerated Tank	4,108.62	21	86,281.03
Membrane Refrigerated Tank	6,067.45	21	127,416.41

After the manufacturing costs had been adjusted there remained the addition of the material costs and the cost of necessary auxiliary systems for BOG management in refrigerated or semi-pressurized cases. There was also an auxiliary cost for the amount of specialised pressure relief valves which would need to be added to the pressure tanks, the more tanks used in the arrangement the higher the auxiliary cost. Table 32 summarizes the final overall CAPEX cost for each tank alternative. The first alternative was a distinct outlier and therefore purposefully was not shaded so it was easier to see the comparisons with other tank arrangements.

Table 32: CAPEX estimate for Various Tanks

	Material Cost (EUR)	Adjusted Manufacturing Cost (EUR)	Manufacturing Cost ratio	Auxiliaries Cost (EUR)	Overall CAPEX Cost (EUR)
Type C P (Vert Cyl-A)	16,044.12	810,259.26	0.98	33,526.04	843,785.29
Type C P (Vert Cyl-B)	5,746.24	326,360.06	0.98	8,006.66	334,366.72
Type C P (Vert Cyl-C)	6,019.87	341,901.02	0.98	8,376.51	350,277.52
Type C P (Vert Cyl-D)	5,879.84	238,070.30	0.98	3,938.36	242,008.66
Type C P (Vert Cyl-E)	4,980.44	120,045.52	0.96	2,089.13	122,134.65
Type C P (Vert Cyl-F)	5,961.18	121,055.90	0.95	2,089.13	123,145.02
Type C P (Vert Cyl-G)	6,913.25	122,519.87	0.95	979.59	123,499.46
Type C P (Horizontal Cyl)	10,688.11	44,708.41	0.81	794.66	45,503.08
Type C P (LPV FW tank)	17,625.33	80,896.08	0.82	424.82	81,320.90
Type C P (LPV RC tank)	16,528.62	89,484.86	0.84	424.82	89,909.68
Type C P (LPV RW tank)	14,152.50	89,433.65	0.86	424.82	89,858.46
Type C SP (Horizontal Cyl)	8,694.90	33,856.27	0.80	17,194.66	51,050.94
Type C SP (LPV FW tank)	4,295.24	60,672.06	0.93	13,194.66	73,866.72
Type C SP (LPV RC tank)	10,253.57	67,113.65	0.87	13,194.66	80,308.31
Type C SP (LPV RW tank)	11,455.73	67,075.24	0.85	13,194.66	80,269.90
Type A Refrigerated Tank	8,430.51	86,281.03	0.91	23,064.71	109,345.74
Membrane Refrigerated Tank	7,970.26	127,416.41	0.94	23,064.71	150,481.12

From Table 32 the cheapest alternative is evidently the pressurized horizontal cylinders, following this is the semi-pressurized horizontal cylinders. Notably the cost of using multiple cylinders to maximise volume efficiency quickly becomes extremely expensive with the more cylinders required. Therefore cylinders may only be cheaper when there are less of them. For large ships with a high volume of fuel required other alternatives will likely be more cost competitive, such as the lattice pressure vessels, membrane tank and type A refrigerated tank. In terms of r

4.6 Summary of Case Study Results

Seventeen alternatives were compared in the case study. In the first analysis it was clear that the membrane tank far out competed the other alternatives in terms of fuel mass, volume efficiency and material ratio. There was no question that as a standalone tank this would be the most favorable option, however, one cannot only look at the tank without considering insulation, boil off gas management systems and sloshing effects.

In the second analysis, including the insulation requirement, the refrigerated storage options lost a large amount of storage volume. Yet due to a higher allowable filling percentage and higher density at lower temperature the storage capacity in terms of mass was still competitive. It was found that the flat walled lattice pressure vessel was most preferable for the maximum amount of fuel by mass.

For lightweight transportation applications majority of the vertical pressurized cylindrical arrangements offer significant mass of fuel per unit of tank mass. The only superior tank arrangement for lightweight application being the semi-pressurized round walled tank yet this arrangement would require additional boil off gas management systems which would add to the system weight.

Consequently a third analysis was necessary to quantify how the auxiliary systems would add to the fuel system weight and what the limits were for how much additional weight was permissible on a small passenger ferry. This analysis was performed using *Maxsurf Advanced stability* module to find the limiting fuel system weights to ensure stability, meet the maximum draft and avoid tank free surface effects. Despite the reduction in tank weight necessary for refrigerated type A tanks and type C LPV flat walled vessels the mass of fuel that was able to be stored still remained the greatest. Thus it was concluded that for highest fuel storage capacity by mass the flat walled LPV would be best.

A decision matrix approach was required to finally select the best alternative. The decision matrix was created using a weighted table with three independent performance indicators. The three decision making parameters are based on mass of fuel (range) , CAPEX (cost) and additional storage space for cargo. Table 33 illustrates the weightings chosen for a typical passenger vessel where cost is first priority followed by range and then followed by additional storage volumes. In the case of a container vessel or bulk carrier the additional storage volumes may have a higher weighting. Likewise for a ship doing longer sea journeys the mass of fuel would be more important. For military applications the cost may not be an important factor, therefore there is a case and point for weighting alternatives differently, yet for this case study priority was on cost.

Table 33: Weight of Decision Making Parameters

	Weight of KPI (/10)
Mass of Fuel (Range)	3
CAPEX Cost	5
Additional Storage Volumes	2

With these weightings the decision matrix of Table 34 concludes the study. In the first three columns the alternatives are ranked in reverse order to award the highest score to the best alternative and the lowest to the worst for each category. The category score is then multiplied by the weighting factor from Table 33 for each category. The total score is then added up out of 10 to determine the best performer overall.

Table 34: Decision Matrix of Best Alternatives

	Mass of Fuel	**CAPEX Cost**	**Additional Storage**	**Overall Score (/10)**	**Rank Overall**
Type C P (Vert Cyl-A)	5	1	3	3.18	17
Type C P (Vert Cyl-B)	4	3	3	3.59	16
Type C P (Vert Cyl-C)	7	2	3	3.82	15
Type C P (Vert Cyl-D)	6	4	3	4.24	13
Type C P (Vert Cyl-E)	1	8	3	4.53	12
Type C P (Vert Cyl-F)	10	7	3	5.82	10
Type C P (Vert Cyl-G)	9	6	3	5.35	11
Type C P (Horizontal Cyl)	12	17	3	9.12	1
Type C P (LPV FW tank)	2	12	3	5.88	9
Type C P (LPV RC tank)	8	10	3	6.35	7
Type C P (LPV RW tank)	13	11	3	7.53	5
Type C SP (Horizontal Cyl)	3	16	2	6.57	6
Type C SP (LPV FW tank)	17	15	2	8.75	2
Type C SP (LPV RC tank)	15	13	2	7.80	4
Type C SP (LPV RW tank)	14	14	2	7.92	3
Type A Refrigerated Tank	16	9	1	6.14	8
Membrane Refrigerated Tank	11	5	1	4.08	14

It is clear that the pressurized horizontal cylinders would be the best alternative for the small passenger ferry due to the lower CAPEX cost in comparison to other types of storage. Coupled with this, the alternative does not require boil off gas management systems so additional storage space will be available. The tanks are suggested to be manufactured using AISI 304L stainless steel to prevent stress corrosion cracking and sustain a long tank lifetime. Keyhole TIG (k-TIG) welding is suggested to vastly decrease manufacturing costs for mass production and improve weld quality with deep penetration. An additional benefit of these pressurized cylindrical tanks is that they could be used for drop in alternative fuels such as propane. Propane is required to be stored under relatively similar conditions to ammonia, therefore if there was a shortage of ammonia at any point propane could be a viable substitute.

4.7 Applications To Other Ship Types

It should be noted that for different countries, different tank sizes and different ships the costs for manufacturing, material and auxiliary equipment could vary substantially. What should be considered for larger ships is that the insulation of refrigerated and semi-pressurized solutions will take up less of the gross tank volume therefore pushing the storage capacity closer to 60% more than pressurized storage as demonstrated in Table 16. Larger ships with the pre-requisite requirement for a double bottom will further benefit from membrane tanks as an additional structure will not be required to support the membrane which would greatly reduce the cost of the tank.

For ships with short sea voyages for many days, the use of pressurized storage with cylindrical tanks will become more limited as cylindrical tanks become heavier and less volume efficient as the required shell thickness increases. It is therefore suggested that if additional storage space is required and there is a large fuel capacity requirement that pressurized round walled lattice pressure vessels are used. For further range it is suggested to then consider semi-pressurized lattice storage vessels ,membrane tanks or type A tanks.

For other ship types of similar size designed for inland waterways the case study could be more directly applied. The weighting parameters can simply be adjusted to suit the preferences of the ship owner.

The EMSA study on the potential of ammonia as a marine fuel (2022)[23] focused on a VLCC, a container ship and a RoPax . In all cases it was evaluated by risk assessment that pressurized or semi-pressurized storage should be kept on deck to minimize contamination of adjacent spaces and also avoid explosion risk if gas were to accumulate in a closed space. Therefore if the ship has the space on deck to offer it would be beneficial to eliminate those risks. Having fuel tanks on deck, however,also has its own set of risks such as exposure to dropped objects. On these large vessels such as container ships and bulk carriers there are always cranes operating above deck, the risk of damaging a tank or more likely fuel lines is very high which then forces the placement of the tanks closer to accommodation spaces which is not preferable for human safety.

In the case study of the city ferry in this work there was no alternative to place tanks on the deck without vastly changing the layout and design of the baseline vessel. For larger ships there may be more spaces available therefore further iterations with a HAZID assessment would necessary to compare risks of installing tanks at different locations. These kind of analyses are on a case by case basis and cannot be generalised for all ship types.

4.8 Key Findings Of This Study

- Pressurized storage can lead to space saving by reducing BOG management systems.
- Vacuum insulation technology in combination with lattice pressure vessels could

reduce the need for BOG management systems for refrigerated storage systems.

- Lattice pressure vessels offer the opportunity to use pressurized storage in custom geometries.
- Lattice pressure vessels can offer pressurized storage for large vessels.
- Type A tanks and semi-pressurized FW LPVs offer the highest storage capacity of fuel by mass for a small ship.
- For larger ships a membrane tank could be more feasible as the volume of the insulation would be much lower than the volume of the tank unlike with a small ferry boat.
- Including an onboard cracker is beneficial not only to produce hydrogen as a pilot fuel but also to be a nitrogen inert gas generator.
- Vent mast positioning,ventilation and leak prevention is essential to prevent passengers from noticing very small concentrations of ammonia.
- Ammonia can be used as a refrigerant onboard the ship, best used in combination with refrigerated or semi-pressurized storage where there is a return line to the tank, otherwise with pressurized storage a buffer tank will be necessary.
- Using ammonia as fuel in combination with hydrogen for a zero emission ship requires multiple fuel heating and cooling systems.
- The use of ammonia with a conventional hydrocarbon pilot fuel is likely a cheaper alternative in terms of fuel system complexity and heating/ cooling requirements.
- Stress corrosion cracking is especially present in carbon steels used to store ammonia, therefore care must be taken to frequently inspect tanks made of such materials.

4.9 Reflection on regulatory development

Maritime regulations and class rules for use of ammonia as fuel are in constant development, the most recent being announced by classification societies as of 2022. The International Maritime Organization (IMO) has agreed meeting dates for years to come to reconvene and discuss regulatory gaps found by pioneering members in the shipbuilding industry. Meetings such as those mentioned in section 2.3.1 will serve to develop regulations on ammonia storage and other low-flashpoint fuels such as hydrogen. The goal is to have an amended IGF code in service by January 2028, still some years from now but in the meantime the classification societies will strive to bridge the regulatory gap offering approval in principal to new designs. In addition to this the IMO has agreed that at the

CCC 9 sub-commitee meeting ,happening in September 2023, a draft interim guidelines for use of hydrogen as fuel and an interim guidelines for safety of ships using ammonia as fuel will be developed or finalised.

As more and more pilot projects with ammonia fueled vessels are coming up, more is learnt about issues surrounding safety and how to store ammonia for a wide range of vessels. As mentioned before the current regulations primarily focus on large ships as smaller ships are said to possibly use hydrogen or remain on fossil fuels to maintain higher engine speeds. But for the market of slow moving city ferries and other slow moving vessels under high load there could be potential for the use of ammonia as fuel onboard. The case study justifies the need for more specific rules that not only specify rules for large ships but also focus on smaller ships in the range of 15m to 100m capable of maintaining safe storage conditions for ammonia.

IGF code section 2.3 of alternative design allows for the use of alternative low flashpoint fuels as long as the level of safety is to the same level as stated for the case of using liquefied natural gas. This clause allows shipbuilders and class societies to use IGF code to design ammonia fuelled vessels under the condition that an equivalent level of safety can be shown as for natural gas. The documentation was created in 2017 and since then many ships have been built to use alternative fuels which is helping to further develop regulations for vessels fueled on ammonia in the future.

4.9.1 EU ETS and Carbon Taxation

In present times the price of using ammonia as a fuel is not competitive in comparison to conventional fossil fuel based alternatives as proven in other studies[89]. Nevertheless the need still remains to reduce carbon dioxide emissions to reduce global warming. The method used to reduce reliance on fossil fuels and drive change to renewable alternatives is for governments to introduce incentives and taxation.

In a general sense the drive to use ammonia as a fuel for the shipping industry stems from governmental regulations. Within the Fit for 55 package the European Commission has introduced the FuelEU maritime proposal. The proposed regulation introduces increasingly stringent limits on carbon intensity of the energy used by vessels from 2025, which should oblige them to use alternative fuels. It applies to commercial vessels of 5 000 gross tonnes and above, regardless of their flag with the only exclusion being fishing ships[6].

Globally countries either use a fixed carbon tax price or a credit system (Price is not capped) to reduce CO2 emissions. The latter of the two methods is implemented by the

EU and has a greater focus on meeting the emission cut targets. In this situation price rate fluctuates throughout the year and as seen in Figure 29 can result in higher costs for ship owners to pay if they are likely to be carbon emitters. The graph below represents the average price of emitting carbon from the year 2015 to 2022. Data taken from the world bank pricing carbon dashboard [117].

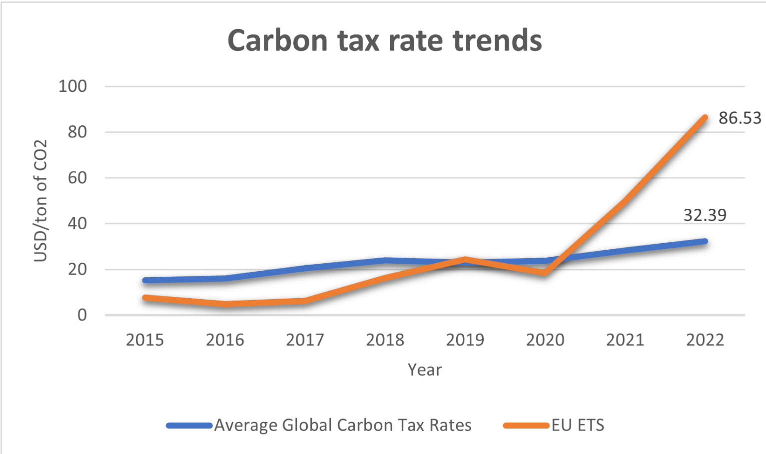


Figure 29: Global Carbon tax prices

A study made into the feasibility of using ammonia as fuel for a SOFC driven containership by Wu et al in 2022 [89] revealed in great detail how carbon tax rates and the EU ETS could effect feasibility of ammonia fuel projects for ships. In the article, forecasts on fuel prices and taxation were made up until the year 2050. Notably the cost of green ammonia was reduced and the cost of HFO also decreased. The cost of LNG and blue ammonia were set to steadily increase. Noted from the study was that, based on cumulative cost, the use of an ammonia SOFC with blue ammonia or the use of an ammonia ICE with blue ammonia would be cheaper than using a HFO ICE by the late 2040s. So for a ship built in 2030 for a 20 year life span the use of ammonia as a fuel has the potential to be cheaper and emit far less carbon dioxide than a HFO ICE. Until that time the current outlook is that ammonia fuelled vessels will not be cost competitive primarily due to a high fuel cost.

4.10 Reflection on regulatory gaps

The IMO International Code for Low-Flashpoint fuels (IGF) is the best reference the shipping industry has to base decisions on storing ammonia as a fuel, however, this code focuses primarily on natural gas and to some extent limited methods of storage that were available in the commercial market pre 2017. The distinction between different methods of storage such as pressurized, semi-pressurized and refrigerated is not outlined as is done in class society rules. Therefore there is a possibility to misinterpret regulations that only pertain to one storage mechanism as being applicable to all.

An example being potential conflict of IGF rule 6.4.9.3.3.1.2 and IGF 6.3.1. ,The former mentioning that pressure tanks at ambient temperature must be designed to guage vapour pressure of the liquefied gas fuel at a temperature of 45°. The later mentioning the Maximum Allowable Relief Valve Setting (MARVS) for natural gas is 1 MPa. This would mean that, under alternative design principles, pressurized storage for ammonia (Gauge vapour pressure of 1.8 MPa) would not be permitted. Clearly pressurized storage at atmospheric temperature is allowed as outlined in BV NR 671 so a slight adjustment to the code may be necessary to clear up any conflicts which may be unintended.

The closest regulation in IGF code to using pressurized storage for ammonia is IGF 6.6 (Regulations for compressed natural gas (CNG)). In this section it mentions that CNG storage is generally not accepted in enclosed spaces unless approval from administration. The pressure required for storing CNG at ambient temperature is in the range of 24 MPa whereas the pressure for storing ammonia at ambient temperature is around 1.8 MPa. IGF code defines high pressure as exceeding 1 MPa, however, the gap between the pressure required to store ammonia at ambient temperature versus the pressure to store CNG is evidently large, therefore an adjustment to the regulation should make it clear if ammonia can be stored in fully pressurized condition below deck in enclosed spaces. It is particularly important for ferry and RoRo vessels where open deck space is often used by passengers so having pressurized tanks on the open deck is either not an option or limited to smaller sized tanks.

There is a big question raised about how to safely use hydrogen onboard vessels. In particular the IGF does not mention anything about how to safely use fuel cells or ammonia cracking technologies which could be needed to convert ammonia fuel or other alternative low flash-point fuels into propulsion power. Reference for fuel cells is based on classification society rules which is based off recommendations and not necessarily set rules. A new IMO IGF code that is all encompassing and includes the use of hydrogen as fuel or rules for onsite production of hydrogen onboard a vessel would be helpful.

Besides that modern developments on lattice pressure vessel tank types are not quite well documented in IMO IGF or IGC code. Mostly Type C tanks are considered to only have rounded surfaces of cylindrical or spherical shape. It is not made clear that type C tanks may indeed have flat surfaces if an internal lattice structure is used.

It is still unclear as to whether IGC rules apply at all for ammonia fueled ships, however, if they do not, then regulations such as (IGC 3.5.3.5.1) minimum clearance of 450 mm between curved tank surfaces and deck stiffener or 600 mm for flat walled tank inspections may be relaxed as to allow greater usage of available volumes onboard ships. Alternatively these rules in IGC pertaining to accessibility of spaces around fuel tanks should also be adapted and integrated where appropriate to IGF code.

The use of ammonia as a fuel onboard a ship often assumes that ammonia gas is lighter than air and will easily dissipate into the atmosphere if there is an emergency leak situation. Not well considered in class rules or regulation is that ammonia, unlike other low flash-point fuels, has a very high water solubility which means that in humid environments (often over water) the gas will not dissipate quickly into the atmosphere but instead form a cloud of toxic gas on low lying flat surfaces. The placement of the vent mast therefore should be carefully placed not only to a distance high enough from the weather deck but also in a location which allows ammonium liquid (formed via the contact of water vapour) to cascade down to a marked exclusion zone on deck or purposefully aimed overboard. Ammonium released to the environment in small concentrations should not be a great environmental concern, especially in emergency situations where human life should be prioritized.

Ammonia-absorption systems that completely eliminate the possibility of discharges into the atmosphere during normal operation and emergencies should be more well developed. Current systems offered on the commercial market are mainly developed to scrub exhaust emissions from burning conventional fuels. Often urea is required to be stored onboard to dose the SCR and convert nitrogen oxides, however, ammonia itself can be used instead of urea if it is already stored onboard as seen in the catalytic reduction equations of section 2.1.3. A system of this nature needs to be able to recapture ammonia and send it back to a buffer tank.

5 Conclusions

This work investigated the integration impact of installing ammonia as a fuel source on a ship and alternative storage arrangements. The primary objectives were to establish limitations and complexities of storing ammonia on manned platforms, to minimise the integration impact of the energy vector and to maximise energy density on-board per unit volume. Integration impact was measured in terms of cargo space loss, human health risks, potential to damage the surrounding environment, structural damage to existing ship components and fire or explosive risks.

Through extensive research and completing a case study of retrofitting an electric ferry to be fuelled by ammonia, it can be said that all objectives were fulfilled. The state-of-the-art review consolidated the knowledge of multiple authors and answered the question of integration impact, looking not only at how to store ammonia but also how it would be used on a ship. It was discovered that ammonia is a highly toxic substance that needs special storage conditions to limit integration impact onboard ships. Human safety is one of the biggest concerns with using ammonia as a fuel and current regulations, namely the IMO IGF code, do not address the dangers of using ammonia as a fuel. Isolation and prevention mechanisms should be installed to mitigate ammonia leakage into crew spaces and also into the environment. In the case study, system diagrams documented how ammonia could be installed on a vessel using three different methods of storage (refrigerated, semi-pressurized and pressurized). General arrangement drawings including key components of an ammonia-fuelled ship showed what equipment should be considered for its use with a HCCI dual fuel engine and how much space would be available for tanks. Finally, an evaluation of seventeen different storage tank arrangements found that pressurized horizontal cylinders would be the best alternative for the small passenger ferry under study. This solution was found to: limit human health risks using reliable type C pressure vessels, reduce cargo space loss as there would be no boil-off gas management system necessary and provide the most cost-effective solution. The secondary objective was met by addressing the impact of using different materials for tank construction. The material analysis showed that stainless steel AISI 304L would be the best material to reduce structural damage and prevent stress corrosion cracking.

Section 2.6, *Ship Fuel System Integration*, described how ammonia could be used onboard a vessel and helped to quantify the integration impact of the energy vector. Diesel cycle two-stroke and four-stroke engines were found to be the predominant ammonia engine types being developed by engine manufacturers. The fuel integration system is in many ways simple in that liquid ammonia is taken directly to the engine under high pressure and injected at high pressure in a similar fashion to liquid propane-fueled engines.

Therefore pressurized storage would not require as many auxiliary energy consumers to heat and cool the fuel to the desired temperature and pressure. If a diesel dual fuel engine was to be used the most convenient storage mechanism would likely be semi-pressurized as the substance is already at pressure and higher temperature suitable to be injected at the engine. Pressurized storage would also be suitable but is less favoured due to safety concerns and the fact that usually more fuel by mass can be stored in a semi-pressurized state.

Although there are many benefits to using diesel cycle ammonia engines, they were not chosen for the case study (section 3) as they rely on a fossil fuel pilot which does not fit the net zero carbon emissions target for 2050. The alternative was to use low-pressure Otto cycle ammonia engines with a hydrogen pilot fuel which requires a pre-mixed gas mixture. Consequently, fuel heaters and compressors were required to maintain supply conditions. Using the gas from pressurized storage requires an expansion process to reduce the pressure to a usable pressure as storage pressure of 1.8 MPa would be very high and ammonia would be in liquid form instead of gas. This expansion is coupled with simultaneous cooling which then may require additional fuel heating systems or can be used in a regenerative cooling loop. A heat consumption analysis revealed that with Otto cycle engines the heat required to take fuel to engine supply conditions would differ between storage conditions. In a semi-pressurized state and refrigerated state up to 4 kW and 6 kW more heat would be required than pressurized fuel respectively.

In general, a benefit of using ammonia as a fuel onboard a vessel (reducing integration impact) is that it can be used directly or indirectly as a refrigerant. For many ship types such as passenger vessels requiring air conditioning, vessels with heavy machinery requiring cooling, or cargo carriers requiring refrigerated cargo holds this is a big benefit. Such refrigeration loops were drawn into the system diagrams for the retrofitted electric passenger ferry under study.

Drawing the ship system using P & ID diagrams allowed easy identification of areas of the target ship where ammonia would likely be a health risk. Integration impact was minimized by modelling the ventilation system which would prevent contamination of the passenger compartment. Each room was fitted with necessary ammonia and hydrogen sensors which would be able to detect leakages and immediately shut off the primary fuel supply and flood the room with excess air to prevent the ammonia from reaching flammability concentration limits. Airlocks and A60 fire insulation were used on walls between areas of fuel and areas of potential ignition sources.

The appropriate selective catalytic reduction device was sourced and modelled in the general arrangements to ensure that the environmental impact of burning ammonia in an internal combustion engine was minimised. The use of ammonia to convert nitrogen oxides to nitrogen was seen as a key benefit of having ammonia stored onboard. Bilge tank systems were also installed to capture any aqueous ammonia but also could be used to capture excess anhydrous ammonia unburnt after combustion.

The case study, section 3, evaluated a number of different storage options from cylindrical tank arrangements to fixed membrane tanks. A series of analyses were completed covering the influence of insulation material and auxiliary boil-off gas management systems, the requirement to meet stability criteria and the cost of manufacturing the tanks. The overall conclusion of the case study was that horizontal cylindrical pressurized tanks would be the best alternative for the vessel under study based on a multi-criteria decision analysis. This result would likely not be the same for all ship types as the thickness of type C cylindrical tanks increases exponentially with size meaning that multiple cylinders would need to be used for larger ships. Important findings were that the cost of manufacturing the tanks will increase exponentially if multiple cylindrical tanks are used. Multiple cylindrical tanks would also require more pipe connections and more expensive valves therefore increasing the risk of leakages and creating a large amount of maintenance which, if neglected, could lead to catastrophic consequences.

It was concluded that if the quantity of ammonia fuel to be stored was very large then membrane tanks with refrigerated storage should be used instead of cylindrical tanks. However, sloshing effects should be mitigated by installing baffle systems such as ball baffles or an ABAS floating blanket. The case study also revealed that for smaller membrane tanks the volume of insulation required around the tank may compromise the amount of available storage space but insulation thickness does not scale linearly with tank size meaning that bigger tanks should not have this issue of insulation taking up large volumes.

For the majority of ships that do not require very large quantities of fuel yet still complete long sea journeys a new technology known as lattice pressure vessels (LPVs) would be the best alternative. LPVs allow for the construction of a single robust type C pressurized tank which maximises storage volume and reduces the need for boil-off gas treatment systems. This technology can be cost-competitive and offers a storage solution for custom-shaped prismatic tanks. The major positive for the technology in comparison to membrane tanks is that sloshing effects are eliminated.

A material selection analysis achieved the final objective of evaluating which materials would be suitable to store anhydrous ammonia. Research showed that carbon manganese steels could be used for tanks provided that necessary control mechanisms are put in place to limit oxygen content in the fuel tanks to prevent stress corrosion cracking. In general, a safer alternative less prone to corrosion would be to use austenitic stainless steel such as AISI 304L. In the case study, this was the material of choice in light of passenger safety and durability.

A final discussion on the regulatory framework from the IMO and classification societies revealed concerns with existing rules and current development towards new rules. At this point the IMO IGF code does not include ammonia as a fuel yet with alternative design principles and class rules the design of a safe ammonia-fueled ship is possible.

5.1 Future work

Future work that could be done includes:

- Detailed tank designs in terms of finite element analysis
- Fluid-structure interactions on tank walls or non-linear failure analysis.
- Model boil-off gas more accurately to predict the pressure build-up in the tank to better determine insulation as done for LNG carriers by Jo et al.[109].

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References

[1] D. J. Starkey. "History of world trade since 1450." (2023), [Online]. Available: https://www.encyclopedia.com. (accessed: 23.03.2023).

[2] GTC. "How a steel box changed the world: A brief history of shipping." (2023), [Online]. Available: https://www.globaltrainingcenter.com/how-a-steel-box-changed-the-world-a-brief-history-of-shipping.. (accessed: 23.03.2023).

[3] U. Nations, *Paris agreement international treaty*, COP21, Dec. 2015.

[4] IMO. "Rules on ship carbon intensity and rating system enter into force." (), [Online]. Available: https://www.imo.org/en/MediaCentre/PressBriefings/pages/CII-and-EEXI-entry-into-force.aspx. (accessed: 23.03.2023).

[5] B. Bonnet-Cantalloube, M. Espitlier-Noel, P. Ferrari de Carvalho, J. Foneseca, and G. Pawelec, "Clean ammonia in the future energy system," Hydrogen Europe, Tech. Rep., Mar. 2023.

[6] E. Parliament. "Fuel eu maritime - sustainable maritime fuels." (), [Online]. Available: https://www.europarl.europa.eu/legislative-train/theme-a-european-green-deal/file-fuel-eu-maritime. (accessed: 26.03.2023).

[7] DNV. "Eu ets -emissions trading system." (), [Online]. Available: https://www.dnv.com/maritime/insights/topics/eu-emissions-trading-system. (accessed: 23.03.2023).

[8] C. Tornatore, L. Marchitto, P. Sabia, and M. De Joannon, "Ammonia as green fuel in internal combustion engines: State-of-the-art and future perspectives," vol. 8, 2022. DOI: https://doi.org/10.3389/fmech.2022.944201.

[9] Zamfirescu and Dincer, "Using ammonia as a sustainable fuel," *Journal of Power Resources*, vol. 8, 2008. DOI: 10.1016/j.jpowsour.2008.02.097.

[10] G. M. Forum. "The shipping industry's fuel choices on the path to net zero." (2023), [Online]. Available: https://www.globalmaritimeforum.org/content/2023/04/the-shipping-industrys-fuel-choices-on-the-path-to-net-zero_final.pdf. (accessed: 13.06.2023).

[11] A. M. Fuels. "Fuel properties of ammonia." (), [Online]. Available: https://www.iea-amf.org/content/fuel_information/ammonia. (accessed: 03.02.2023).

[12] A. et al, "Low-temperature direct ammonia fuel cells: Recent developments and remaining challenges," *Current opinion in electrochemistry*, 2020. DOI: http://dx.doi.org/10.1016/j.coelec.2020.03.021.

[13] O. Siddiqui and I. Dincer, "A review and comparative assessment of direct ammonia fuel cells," *Current opinion in electrochemistry*, vol. 5, pp. 568–578, 2018. DOI: https://doi.org/10.1016/j.tsep.2018.02.011.

[14] A. Valera-Medina, H. Xiao, M. Owen-Jones, W. David, and P. Bowen, "Ammonia for power," *Progress in Energy and Combustion Science*, vol. 69, pp. 63–102, 2018. DOI: https://doi.org/10.1016/j.pecs.2018.07.001.

[15] b. Abhinav Yadava and B. Jeong, "Safety evaluation of using ammonia as marine fuel by analysing gas dispersion in a ship engine room using cfd," *Journal of international maritime safety,environmental affairs and shipping*, vol. 6, no. 2-3, pp. 99–116, 2022. DOI: https://doi.org/10.1080/25725084.2022.2083295.

[16] T. Europe. "Toyota fuel cell technology." (), [Online]. Available: https://www.toyota-europe.com/brands-and-services/toyota-fuel-cell/fuel-cell-technology. (accessed: 14.04.2023).

[17] Hyundai. "Hyundai fuel cell technology." (), [Online]. Available: https://www.hyundai.com/eu/mobility-and-innovation/hydrogen-energy/hydrogen.html. (accessed: 14.04.2023).

[18] S. Chakraborty, T. Koroglu, A. Dzielendziak, and K. Yang, "Evaluation of smart eco-friendly public transport options in coastal cities: Towards a green future for the city of southampton," *LRF Collegium 2013 Series*, vol. 2, 2013.

[19] M. T. Forum, "Preliminary discussion report on use of ammonia as fuel for ships," Maritime Technologies Forum, Tech. Rep., 2022.

[20] B. Veritas, *Ammonia-Fuelled Ships Tentative Rules NR671*. Bureau Veritas, 2022.

[21] N. de Vries, "Safe and effective application of ammonia as a marine fuel," Masters thesis, TU Delft Mechanical, Maritime and Materials Engineering, Jun. 2019.

[22] H. L, "Solid oxide fuel cell propulsion system fueled by ammonia," Masters thesis, TU Delft, Mar. 2020.

[23] Laursen, Barcarolo, Patel, *et al.*, "Potential use of ammonia in shipping," European Maritime Safety Agency, Tech. Rep., 2022. [Online]. Available: https://www.emsa.europa.eu/publications/reports/item/4833-potential-of-ammonia-as-fuel-in-shipping.

[24] F. Y Al-Aboosi, M. M El-Halwagi, M. Moore, and R. B Nielsen, "Renewable ammonia as an alternative fuel for the shipping industry," *Current opinion in chemical engineering 2021*, 2021. DOI: https://doi.org/10.1016/j.coche.2021.100670.

[25] J. L. Viviente. "Advanced materials and reactors for energy storage through ammonia." (), [Online]. Available: https://www.innovationnewsnetwork.com/advanced-materials-reactors-energy-storage-through-ammonia/29289/. (accessed: 15.03.2023).

[26] S. Naqvi. "What is ammonia and how does it impact me?" (), [Online]. Available: https://www.blueland.com/articles/what-is-ammonia-and-how-does-it-impact-me. (accessed: 05.02.2023).

[27] G. H2. "Hydrogen a to z series: D for density." (), [Online]. Available: https://genh2hydrogen.com/blog/defining-hydrogen-from-a-to-z-3/#:~:text=Although%20Hydrogen%20has%20the%20highest,(9%20MJ%2FL).. (accessed: 10.07.2023).

[28] A. B. of Shipping, "Sustainability whitepaper: Ammonia as marine fuel," ABS, Tech. Rep., Oct. 2020.

[29] E. toolbox. "Gas carrier types." (), [Online]. Available: https://www.engineeringtoolbox.com/ammonia-gas-liquid-equilibrium-condition-properties-temperature-pressure-boiling-curve-d_2013.html. (accessed: 26.04.2023).

[30] I. M. Organization, *International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk*. IMO, 2014.

[31] C. society, "Part c: Guidlines for safety of ships using ammonia as fuel," ClassNK, ClassNK Technical Journal No.5, 2022.

[32] B. Veritas, *Ships using Fuel Cells NR547*. Bureau Veritas, 2022.

[33] F. Europe. "Guidance for transporting ammonia by rail." (2007), [Online]. Available: https://www.fertilizerseurope.com/wp-content/uploads/2019/08/Guidance_for_transporting_ammonia_in_rail_4.pdf. (accessed: 20.04.2023).

[34] U. of East Anglia. "Nitrous oxide emissions pose an increasing climate threat, study finds." (2020), [Online]. Available: https://phys.org/news/2020-10-nitrous-oxide-emissions-pose-climate.html. (accessed: 02.03.2023).

[35] D. Erdemir and I. Dincer, "A perspective on the use of ammonia as a clean fuel: Challenges and solutions," *International Journal of Energy Research*, vol. 45, pp. 4827–4834, 2020.

[36] G. Trauthwein. "Man es:moving forward on ammonia engines." (), [Online]. Available: https://www.marinelink.com/news/man-es-moving-forward-ammonia-engines-502260. (accessed: 01.03.2023).

[37] J. Cardoso, V. Silva, R. Rocha, M. Hall, M. Costa, and D. Eusebio, "Ammonia as an energy vector:current and future prospects for low-carbon fuel applications in internal combustion engines," *Journal of Cleaner Production*, vol. 296, no. 126462, pp. 0959–6526, 2021. DOI: https://doi.org/10.1016/j.jclepro.2021.126562.

[38] C. Lhuillier, P. Brequigny, F. Contino, and C. Rousselle, "Performance and emissions of an ammonia-fueled si engine with hydrogen enrichment," *SAE Technical Paper*, 2019. DOI: https://doi.org/10.4271/2019-24-0137.

[39] N. Y. D. of Health. "The facts about ammonia." (), [Online]. Available: https://www.health.ny.gov/environmental/emergency/chemical_terrorism/ammonia_tech.. (accessed: 05.02.2023).

[40] ECHA. "Ammonia info card." (), [Online]. Available: https://echa.europa.eu/substance-information/-/substanceinfo/100.028.760. (accessed: 02.03.2023).

[41] CDC. "Ammonia solution, ammonia, anhydrous: Lung damaging agent." (), [Online]. Available: https://echa.europa.eu/substance-information/-/substanceinfo/100.028.760. (accessed: 02.03.2023).

[42] Christensen and Soensen, "Metal ammine complexes for hydrogen storage," *Journal of Material Chemistry*, vol. 38, 2005. DOI: https://doi.org/10.1039/b511589b.

[43] Z. Ksenia, "Ammonia bunkering of passenger vessel-concept quantitive risk assessment," DNV AS Energy Systems Safety Risk Management, Report, 2021.

[44] M. Hine, *Safety aspects about handling ammonia*, Presentation for Mission Hydrogen Webinar, Apr. 2023.

[45] O. Herbinet, P. Bartocci, and G. Dana, "On the use of ammonia as a fuel – a perspective," *Fuel Communications*, vol. 11, no. 100064, 2022. DOI: https://doi.org/10.1016/j.jfueco.2022.100064.

[46] Wartsila, *Wartsila presentation*, Madrid Conference for Renewable Fuel Engines, Mar. 2023.

[47] S. Furstenberg Stott. "Quantifying the environmental impacts of ammonia at sea." (), [Online]. Available: https://www.ammoniaenergy.org/articles/quantifying-the-environmental-impacts-of-ammonia-at-sea/. (accessed: 26.03.2023).

[48] S. Parkar. "Aea webinar: Ammonia at sea." (), [Online]. Available: https://www.ammoniaenergy.org/wp-content/uploads/2023/02/Maritime-Insights-speaker-slides-Feb-2023.pdf. (accessed: 26.03.2023).

[49] F. D. of Agriculture Website. "Aquarium water quality: Nitrogen cycle." (), [Online]. Available: https://www.fdacs.gov/Consumer-Resources/Recreation-and-Leisure/Aquarium-Fish/Aquarium-Water-Quality-Nitrogen-Cycle. (accessed: 27.03.2023).

[50] C. news. "Toxic algae blamed for 300 tons of dead fish in oder river on german-polish border." (2022), [Online]. Available: https://www.cbsnews.com/news/germany-poland-oder-river-fish-deaths-off-toxic-algae-man-made. (accessed: 25.05.2023).

[51] Franklin and Edwards, "Ammonia toxicity and adaptive response in marine fishes," 2019. DOI: https://www.researchgate.net/publication/328118885_Ammonia_toxicity_and_adaptive_response_in_marine_fishes_-_A_review.

[52] G. D. Boardman, "Toxicity of ammonia to three marine fish and three marine invertebrates," *Environmental Toxicology*, vol. 19, no. 2, pp. 134–142, 2004. DOI: https://doi.org/10.1002/tox.20006.

[53] C. S. Facts. "Ammonia - chemical safety." (), [Online]. Available: https://www.chemicalsafetyfacts.org. (accessed: 03.02.2023).

[54] V. Pattabathula, R. Nayak, and D. Timbres. "The safety and integrity of ammonia storage tanks." (2014), [Online]. Available: https://ammoniaknowhow.com/ammonia-storage-tanks/. (accessed: 26.04.2023).

[55] U. D. of Transportation, *Letter to anhydrous ammonia owners*, Nurse Tank Safety Advisory, Dec. 2008.

[56] J. Deere. "2410 c nutrient applicator." (), [Online]. Available: https://www.deere.ca/en/products/application-equipment/2410c-nutrient-applicator/. (accessed: 21.06.2023).

[57] A. Russell, S. Chumbley, and A. Becker, "Anhydrous ammonia nurse tank safety," *Material Science and Engineering*, 2014. DOI: https://lib.dr.iastate.edu/mse_books.

[58] M. Gallucci. "Why the shipping industry is betting big on ammonia." (), [Online]. Available: https://spectrum.ieee.org/why-the-shipping-industry-is-betting-big-on-ammonia. (accessed: 27.03.2023).

[59] R. et al, "Finite element analysis of liquefied ammonia tank for mobility vehicles employing polymers and composites," 2020. DOI: http://dx.doi.org/10.3390/en13205312.

[60] T. Brown. "The amveh-an ammonia fueled car from south korea." (), [Online]. Available: https://nh3fuel.wordpress.com/2013/06/20/the-amveh-an-ammonia-fueled-car-from-south-korea/. (accessed: 17.04.2023).

[61] A. G. Association. "International fuel gas code." (), [Online]. Available: https://county.mckenziecounty.net/usrfiles/2018_International_Fuel_Gas_Code.pdf. (accessed: 14.04.2023).

[62] N. line, *Nyk line presentation*, IMO Synopsium 2021, Feb. 2021.

[63] Amogy. "Amogy demonstrates first ammonia-powered, zero-emissions tractor." (2022), [Online]. Available: https://amogy.co/amogy-demonstrates-first-ammonia-powered-zero-emissions-tractor/. (accessed: 20.04.2023).

[64] Gaztransport and Technigaz. "Mark iii systems." (), [Online]. Available: https://gtt.fr/technologies/markiii-systems. (accessed: 13.06.2023).

[65] A. Valera-Medina and R. Banares-Alcantara, *Techno-economic Challenges of Green Ammonia as an energy vector*. Oxford, United Kingdom: Elsevier, 2020.

[66] Wartsila. "Gas carrier types." (2023), [Online]. Available: https://www.wartsila.com/encyclopedia/term/gas-carrier-types. (accessed: 20.04.2023).

[67] D. N. Veritas, *DNV RU SHIP: Rules and standards for Ships*. DNV, 2022.

[68] P. Bergen and D. Chang, "A scalable and prismatic pressure vessel for transport and storage of natural gas," *VII International Conference on Computational Methods in Marine Engineering*, 2017.

[69] L. Technology. "S. korea: Lattice technology gets lloyds register approval." (2022), [Online]. Available: https://lattice-technology.com/s-korea-lattice-technology-gets-lloyds-register-approval/. (accessed: 02.06.2023).

[70] J. Kim, J. Jeong, and D. Chang, "Optimal shape and boil-off gas generation of fuel tank for lng fueled tugboat," *Journal of Ocean Engineering*, vol. 34, no. 1, pp. 19–25, 2020. DOI: https://doi.org/10.26748/KSOE.2019.103.

[71] M. Al-Breiki and Y. Bicer, "Technical assessment of liquefied natural gas, ammonia and methanol for overseas energy transport based on energy and exergy analyses," *International Journal of Hydrogen Energy*, vol. 45, pp. 34 927–34 937, 2020. DOI: https://doi.org/10.1016/j.ijhydene.2020.04.181.

[72] K. Machaj, J. Kupeck, Z. Malecha, *et al.*, "Ammonia as a potential marine fuel: A review," *Energy Strategy Reviews*, vol. 44, 2013. DOI: https://doi.org/10.1016/j.esr.2022.100926.

[73] D. Lee, M. Ha, S. Kim, and S. Shin, "Research of design challenges and new technologies for floating lng," *International Journal of Naval Architechture and Ocean Engineering*, no. 6, pp. 307–322, 2014. DOI: http://dx.doi.org/10.2478/IJNAOE-2013-0181.

[74] T. LNG. "Torgy lng containment systems." (), [Online]. Available: http://www.golng.eu/files/upload/Torgy_LNG_Client_Presentation_V6_Homepage_presentation.pdf. (accessed: 21.06.2023).

[75] Y. Ju and S. Wu, "Numerical study of the boil-off gas (bog) generation characteristics in a type c independent liquefied natural gas (lng) tank under sloshing excitation," *Energy*, no. 223, pp. 0360–5442, 2021. DOI: https://doi.org/10.1016/j.energy.2021.120001.

[76] Y. Ju and S. Wu, "Numerical study on pressure variation of marine liquefied natural gas (lng) fuel tanks under sinusoidal sloshing excitation," *IOP Conf. Series: Materials Science and Engineering*, no. 1240, 2022. DOI: doi:10.1088/1757-899X/1240/1/012032.

[77] B. Veritas. "With its "nh3 ready" membrane tanks and their alternative "1 barg" tank pressure rating recognized by bureau veritas, gtt technology offers increased flexibility to shipowners." (2021), [Online]. Available: https://marine-offshore.bureauveritas.com/newsroom/gtt-nh3-ready-membrane-tanks-and-alternative-1-barg-recognized-bureau-veritas. (accessed: 13.06.2023).

[78] T. Gavory and P. de Seze, "Sloshing in membrane lng carriers and its consequences from a designer´s perspective," *Marketing department GTT*, 2009.

[79] J. Sygal and Janardhanan, "Numerical study on the effect of ball baffles in reducing sloshing loads in ship tanks," *International Journal of Structural Mechanics and Finite Elements*, vol. 4, no. 1, 2018.

[80] Mazda. "Next generation skyactive x gasoline engine." (2019), [Online]. Available: https://www2.mazda.com/en/next-generation/technology/. (accessed: 25.05.2023).

[81] M. Pochet, I. Truedsson, F. Foucher, H. Jeanmart, and F. Contino, "Ammonia-hydrogen blends in homogeneous-charge compression-ignition engine," *SAE International*, no. 0087, 2017. DOI: doi:10.4271/2017-24-0087.

[82] M. ES, *Man es presentation*, Madrid Conference for Renewable Fuel Engines, Mar. 2023.

[83] WinGD. "Wingd x-df a." (), [Online]. Available: https://www.wingd.com/en/technology-innovation/engine-technology/x-df-dual-fuel-design/. (accessed: 10.07.2023).

[84] E. Ali Brohi, "Ammonia as fuel for internal combustion engines," Masters thesis, Chalmers university of Technology, Gothenburg,Sweeden, 2014.

[85] GIZ, P. Turner, T. Jasper, and F. Fukunaga, *Hydrogen internal combustion engine dual fuel assessment*, Chilean Ministry of Energy, September 2021.

[86] M. Sonker, S. Tiwary, N. Shreyash, *et al.*, "Ammonia as an alternative fuel for vehicular applications: Paving the way for adsorbed ammonia and direct ammonia fuel cells," *Journal of Cleaner Production*, vol. 376, 2022. DOI: https://doi.org/10.1016/j.jclepro.2022.133960.

[87] S. Rathore, D. Biswas Sand Fini, A. Kulkarni, and S. Giddey, "Direct ammonia solid-oxide fuel cells: A review of progress and prospects," *International Journal of Hydrogen Energy*, vol. 46, pp. 35 365–35 384, 2021. DOI: https://doi.org/10.1016/j.ijhydene.2021.08.092.

[88] O. of Energy Efficiency and R. Energy. "Comparison of fuel cell technologies." (2023), [Online]. Available: https://www.energy.gov/eere/fuelcells/comparison-fuel-cell-technologies. (accessed: 20.04.2023).

[89] S. Wu, B. Miao, and S. Chan, "Feasibility assessment of a container ship applying ammonia cracker-integrated solid oxide fuel cell technology," *International journal of Hydrogen Energy*, vol. 47, pp. 27 166–27 176, 2022. DOI: https://doi.org/10.1016/j.ijhydene.2022.06.068.

[90] "Innovation outlook:renewable ammonia," Irena and Ammonia Energy Association, Tech. Rep., 2022.

[91] K. K, R. G, K. W, and C. K, "A preliminary study on an alternative ship propulsion system fueled by ammonia: Environmental and economic assessments," *Journal of Marine Science and Engineering*, vol. 8, no. 183, 2020. DOI: doi:10.3390/jmse8030183.

[92] Markets and Markets. "Solid oxide fuel cell markets." (), [Online]. Available: https://www.marketsandmarkets.com/Market-Reports/solid-oxide-fuel-cell-market-39365796.html#. (accessed: 18.04.2023).

[93] C. Dall'Armi, D. Micheli, and R. Taccani, "Comparison of different plant layouts and fuel storage solutions for fuel cells utilization on a small ferry," *International Journal of Hydrogen Energy*, vol. 46, no. 26, pp. 13 878–13 897, 2021. DOI: https://doi.org/10.1016/j.ijhydene.2021.02.138.

[94] J. C. Daily. "Ihi speeds up development of several ammonia-based technologies." (), [Online]. Available: https://www.japanchemicaldaily.com/2019/10/07/ihi-speeds-up-development-of-several-ammonia-based-technologies/. (accessed: 18.04.2023).

[95] Sunfire. "Sunfire supplies thssenkrupp marine systems with 50kw sofc." (2015), [Online]. Available: https://www.sunfire.de/en/news/detail/sunfire-supplies-thyssenkrupp-marine-systems-with-50-kw-sofc-20. (accessed: 20.04.2023).

[96] B. Power. "Fcwave marine module." (), [Online]. Available: https://www.ballard.com/fuel-cell-solutions/fuel-cell-power-products/marine-modules. (accessed: 18.04.2023).

[97] P. Motors. "Pm-400 fact sheet." (2023), [Online]. Available: https://www.proton-motor.de/produkte/brennstoffzellen-stacks/. (accessed: 20.04.2023).

[98] GenCell. "Gencell technologies." (), [Online]. Available: https://www.gencellenergy.com/our-products/gencell-a5/. (accessed: 18.04.2023).

[99] A. Energy. "Afc agm 2021." (2021), [Online]. Available: https://www.afcenergy.com/media/h2rjqfwx/afc-energy-agm-presentation-7-april-2022-_final.pdf. (accessed: 25.05.2023).

[100] Topsoe. "H2retake process." (2023), [Online]. Available: https://www.topsoe.com/our-resources/knowledge/our-products/process-licensing/h2retake-process. (accessed: 01.05.2023).

[101] H. Yousefi Rizi and D. Shin, "Green hydrogen production technologies from ammonia cracking," *Energies*, vol. 15, no. 8246, 2022. DOI: https://doi.org/10.3390/en15218246.

[102] V. Cechetto, L. Di Felice, J. Medrano, C. Makhloufi, J. Zuniga, and F. Gallucci, "H2 production via ammonia decomposition in a catalytic membrane reactor," *Fuel Processing Technology*, vol. 216, no. 106772, 2021. DOI: https://doi.org/10.1016/j.fuproc.2021.106772.

[103] S. Energy. "Starfire energy prometheus fire." (), [Online]. Available: https://starfireenergy.com/products/. (accessed: 13.06.2023).

[104] K. UDHYOG. "Ammonia cracking units." (), [Online]. Available: https://kcpudhyog.co.in/ammonia-cracking-unit-ultimate-solution-of-hydrogen-generation/. (accessed: 13.06.2023).

[105] Damen. "Ferry 2306-e3 fully electric ferry." (2023), [Online]. Available: https://www.damen.com/vessels/ferries/city-ferries/ferry-2306-e3-full-electric. (accessed: 20.04.2023).

[106] A. bureau of Shipping, *Requirements for Ammonia Fueled vessels*. ABS, 2022.

[107] M. Kalikatzarakis, G. Theotokatos, A. Coraddu, P. S. c, and S. Y. Wong, "Model based analysis of the boil-off gas management and control for lng fuelled vessels," *Journal of Energy*, no. 123872, 2022. DOI: https://doi.org/10.1016/j.energy.2022.123872.

[108] S. Roh and G. Son, "Numerical study if natural convection in a liquefied natural gas tank," *J Mech Sci Technol*, vol. 26, pp. 3133–3140, 2012. DOI: https://doi.org/10.1007/s12206-012-0820-x.

[109] Y. Jo, K. Shin, and S. Hwang, "Development of dynamic simulation model of lng tank and its operational strategy," *Energy*, vol. 223, no. 120060, pp. 0360–5442, 2021. DOI: https://doi.org/10.1016/j.energy.2021.120060.

[110] I. M. Organization, *International Code of Safety for Ships Using Gases or Other Low-Flashpoint Fuels*. IMO, 2016.

[111] Cowellmac. "Lpg industrial compressor." (), [Online]. Available: https://www.alibaba.com/product-detail/LPG-industrial-compressor-oil-free-propane_60636120288.html. (accessed: 10.07.2023).

[112] U. news today. "Pressure vessel cost estimations." (2023), [Online]. Available: https://eu.usatoday.com/story/news/nation/2023/06/22/titan-sub-would-be-vulnerable-to-enormous-pressure-near-ocean-floor/70348277007/. (accessed: 27.07.2023).

[113] C. E. I. Sales. "Welding costs." (), [Online]. Available: https://ceisales.ca/. (accessed: 10.07.2023).

[114] K. P. Automation. "Keyhole tig welding." (), [Online]. Available: https://www.keyplant.com/k-tig-keyhole-welding/. (accessed: 10.07.2023).

[115] M. International. "European steel prices." (), [Online]. Available: https://mepsinternational.com/gb/en/products/europe-steel-prices. (accessed: 10.07.2023).

[116] E. Abbate, M. Mirpourian, C. Brondi, A. Ballarino, and G. Copani, "Environmental and economic assessment of repairable carbon-fiber-reinforced polymers in circular economy perspective," 2022. DOI: 10.3390/ma15092986.

[117] W. Bank. "Carbon pricing dashboard." (), [Online]. Available: https://carbonpricingdashboar worldbank.org/. (accessed: 015.03.2023).

[118] BluNox. "Thermal properties of ammonia." (), [Online]. Available: https://www.blunox.com/home.html. (accessed: 21.06.2023).

[119] A. Laval. "Alfanova 14 heat exchanger." (), [Online]. Available: https://assets.alfalaval.com/documents/p3837d4b0/alfa-laval-product-leaflet-en-che00045.pdf. (accessed: 26.03.2023).

[120] Elektor. "Maf medium pressure fans." (), [Online]. Available: https://www.elektror.com/en/product/maf/. (accessed: 10.07.2023).

[121] B. Veritas. "Rules for the classification of inland navigation vessels, part b." (), [Online]. Available: https://marine-offshore.bureauveritas.com/nr217-rules-classification-inland-navigation-vessels. (accessed: 10.07.2023).

6 APPENDIX

A Phase Diagrams of Ammonia Refrigeration

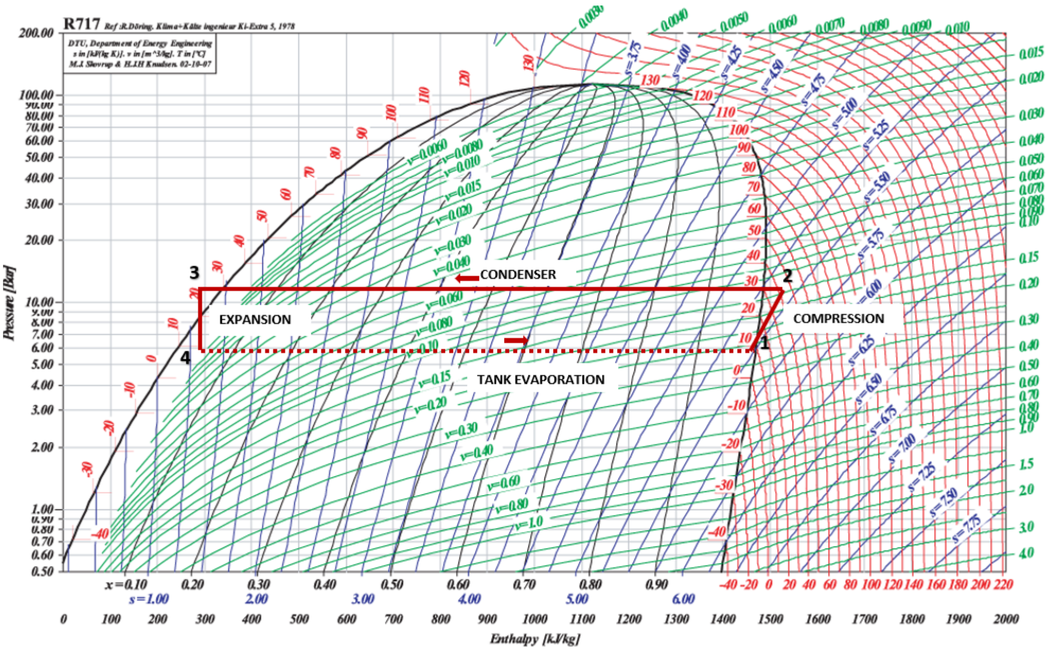

Figure 30: Single Stage Refrigeration of Ammonia

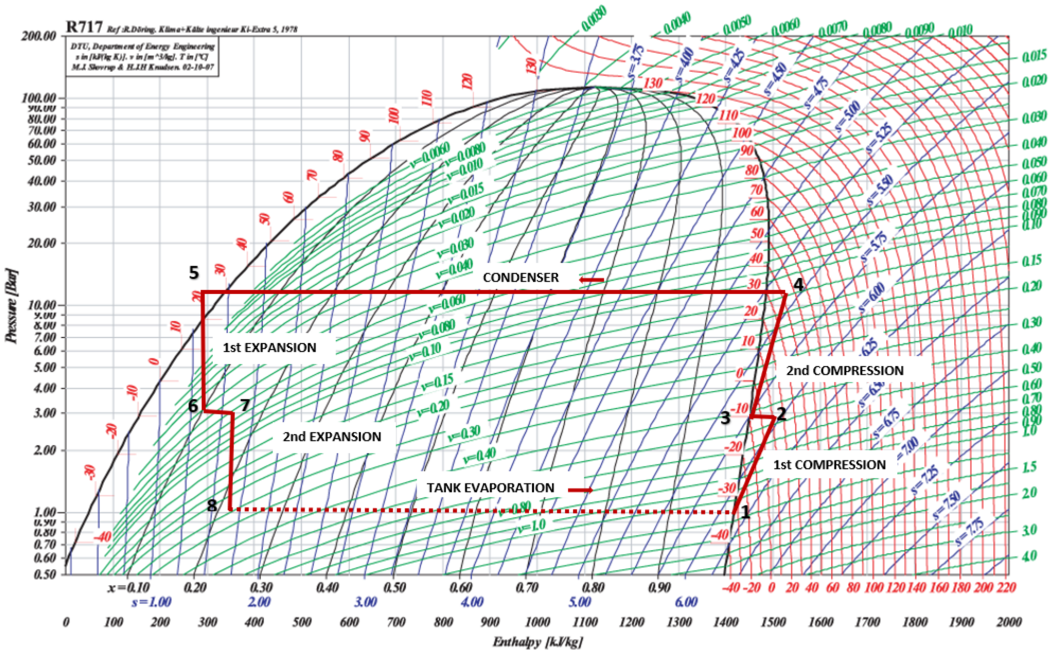

Figure 31: 2 Stage Refrigeration of Ammonia

B Patents

B.1 Lattice Pressure Vessel

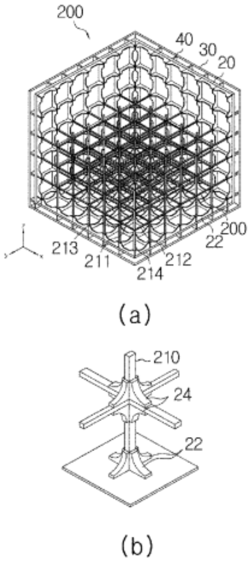

Figure 32: Lattice tank structure overall (US Patent 10429008 B2,2019).

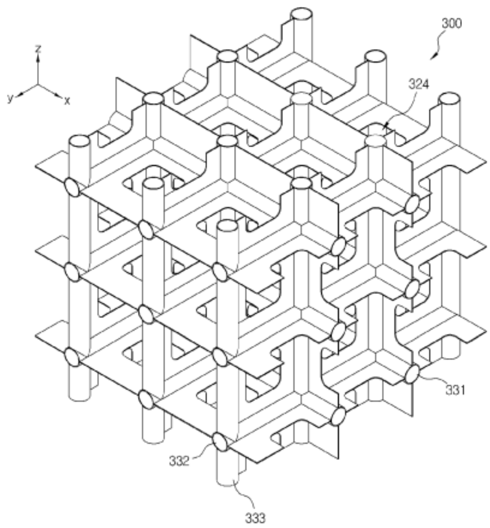

Figure 33: Lattice tank structure detailed (US Patent 10429008 B2,2019).

B.2 Anhydrous Ammonia Tank

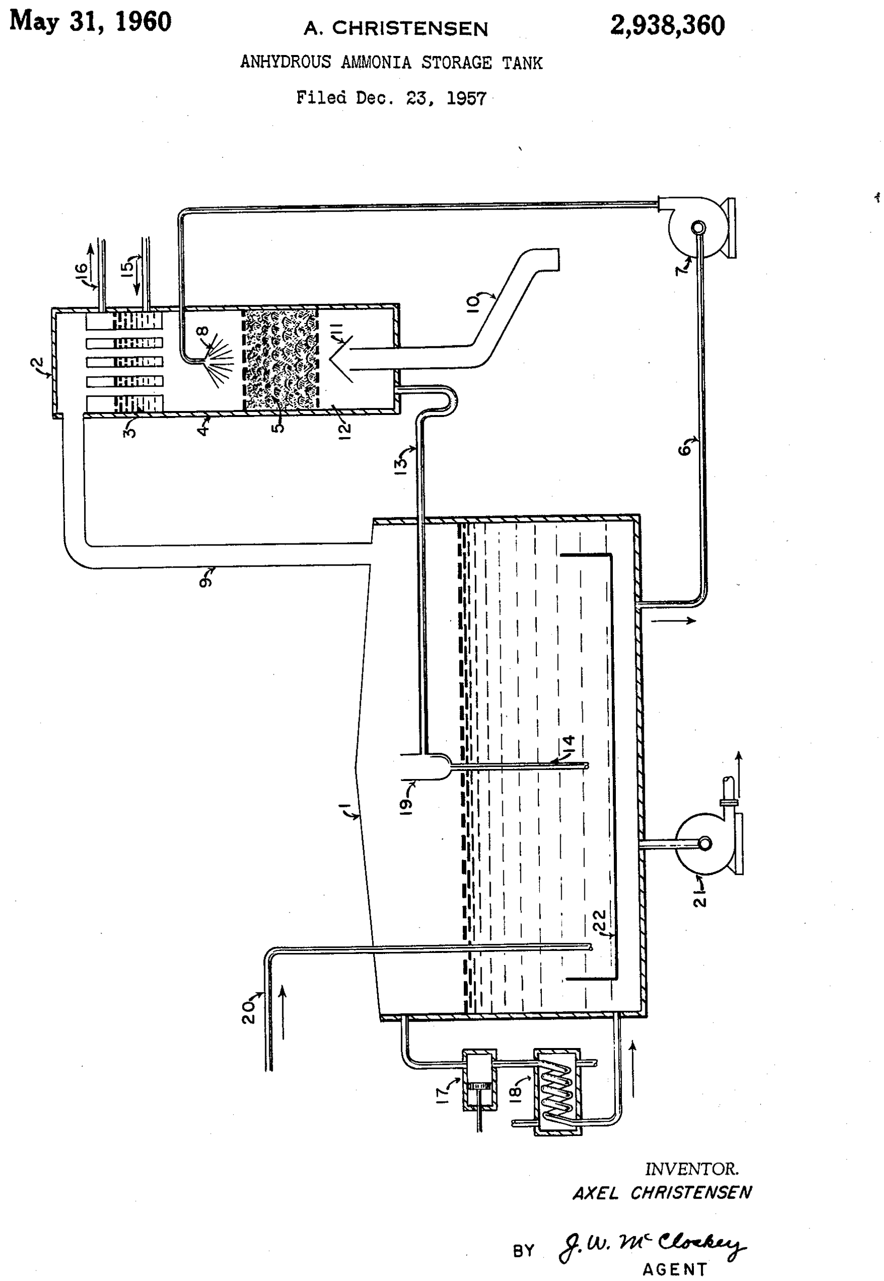

Figure 34: Anhydrous Ammonia Tank by Christensen (Christensen,1957)

C Piping and Identification Drawings

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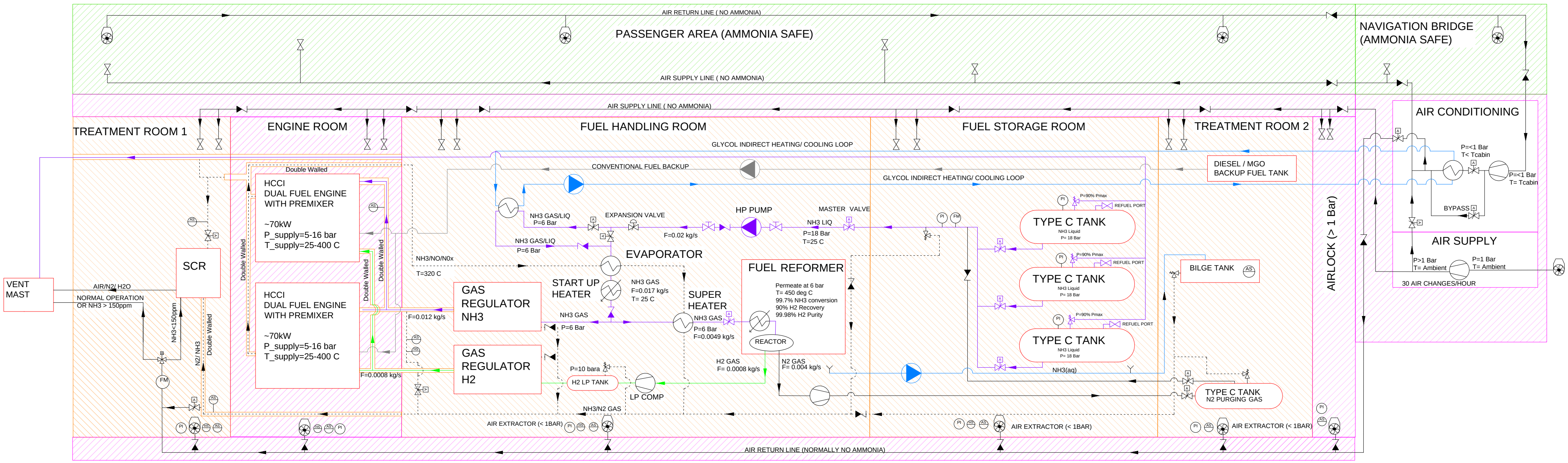


NOTES :

Type C tank can be used for other gaseous fuels if fully pressurized

Type C tank fully pressurized has no requirement for 900mm cofferdam to machinery room (Fuel room space is sufficient)

Type C tank will require only single barrier

Type C tanks pressurized may only be filled to 85% capacity (PRV setting 10 bar)

Type C tank will have lower storage capacity than Type B but may weigh less

Type C tanks fully pressurized have no inert or dry air requirement

Type C tank is assumed not to need recompression when fully pressurized

Type C tank tank pipe connections are not required to be above the highest liquid level

Ammonia can only be used as a refrigerant for the air conditioning system while the engines are running

- NH3 LINE
- H2 LINE
- AIR / INERT GAS LINE
- DRAIN LINE
- GAS LINE TO SCRUBBER
- CONVENTIONAL FUEL

- BOUNDARY A
- BOUNDARY B
- BOUNDARY C
- BOUNDARY D

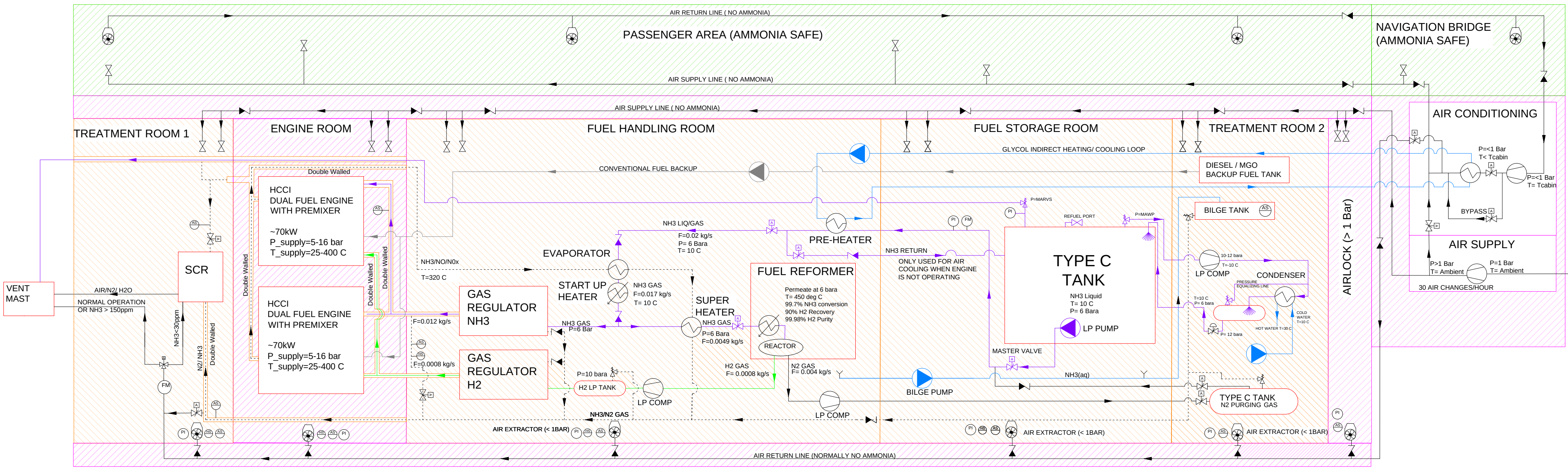
PI	PRESSURE INDICATOR
TI	TEMPERATURE INDICATOR
FM	FLOW METER
	HEATER
	COMPRESSOR
	CENTRIFUGAL PUMP
	SPRAY BALL
	DRAIN
	BUTTERFLY VALVE
	HEAT EXCHANGER
	PNEUMATIC BUTTERFLY VALVE
	3 WAY VALVE
	NON RETURN VALVE
	GATE VALVE

	HAND OPERATED VALVE
	PRESSURE RELIEF VALVE
	REMOTE OPERATED PNEUMATIC VALVE
	DIAPHRAGM VALVE
AS	AMMONIA SENSOR
	FLOW DIRECTION
HS	HYDROGEN SENSOR
	CONTROL VALVE
	EXTRACTOR FAN

REV	DATE	REVISIONS	BY

POLITÉCNICA — LIÈGE université — Navantia SEANERGIES

TITLE:	CASE 3A: TYPE C PRESSURIZED TANKS	NAME:	PIPING & IDENTIFICATION DIAGRAM
DATE.	25 JULY 2023		
DRAWN BY:	D.R.O'CONNOR	SCALE	NTS
DRW NO.	PID001	SHEET	1 OF 1



NOTES :

Semi-Pressurized Storage utilizes lower pressure lines than fully pressurized therefore safer

Type C tank will require only single barrier

Type C tank is assumed to need single stage compression when semi pressurized

Type C tanks at lower pressure than fully pressurized can be filled to higher filling levels with BOG treatment system

Type C semi-pressurized tank will have higher storage capacity than Type C pressurized but less than Type A and Membrane

Type C tanks semi pressurized have inert or dry air requirement around tank

Type C tank will have less BOG than Type A therefore less compressor work

The higher pressure reduces the need for additional compressors for the ammonia line before the engine

NH3 LINE
H2 LINE
AIR / INERT GAS LINE
DRAIN LINE
GAS LINE TO SCRUBBER
CONVENTIONAL FUEL

BOUNDARY A
BOUNDARY B
BOUNDARY C
BOUNDARY D

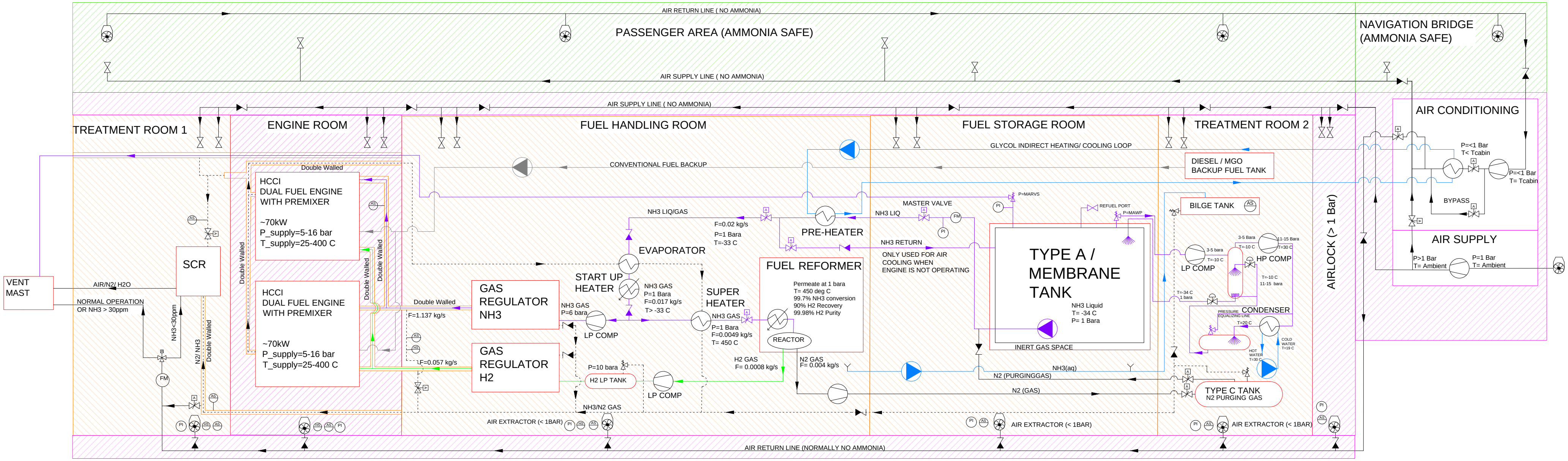
PRESSURE INDICATOR
TEMPERATURE INDICATOR
FLOW METER
HEATER
COMPRESSOR
CENTRIFUGAL PUMP
SPRAY BALL
DRAIN
BUTTERFLY VALVE
HEAT EXCHANGER
PNEUMATIC BUTTERFLY VALVE
3 WAY VALVE
NON RETURN VALVE
GATE VALVE

HAND OPERATED VALVE
PRESSURE RELIEF VALVE
REMOTE OPERATED PNEUMATIC VALVE
DIAPHRAGM VALVE
AMMONIA SENSOR
FLOW DIRECTION
HYDROGEN SENSOR
CONTROL VALVE
EXTRACTOR FAN

REV	DATE	REVISIONS	BY

POLITÉCNICA | LIÈGE université | Navantia SEANERGIES

TITLE: CASE 3B: TYPE C SEMI-PRESSURIZED		NAME: PIPING & IDENTIFICATION DIAGRAM	
DATE.	25 JULY 2023		
DRAWN BY:	D.R.O'CONNOR	SCALE	NTS
DRW NO.	PID002	SHEET	1 OF 1



NOTES :

Type A tank fully pressurized has a requirement for 900mm cofferdam to machinery room

Type A tank will require secondary barrier

Type A tank is assumed to require 2 stage compression

Type A tanks at or near atmospheric pressure can be filled to 95%

Type A semi-refrigerated tank will have higher storage capacity than Type C pressurized or semi-pressurized

Type A tank will require a secondary barrier (with extra insulation)

Type A tank will require inert gas within 2nd barrier (nitrogen is permissible)

- NH3 LINE
- H2 LINE
- AIR / INERT GAS LINE
- DRAIN LINE
- GAS LINE TO SCRUBBER
- CONVENTIONAL FUEL

- BOUNDARY A
- BOUNDARY B
- BOUNDARY C
- BOUNDARY D

PI	PRESSURE INDICATOR
TI	TEMPERATURE INDICATOR
FM	FLOW METER
	HEATER
	COMPRESSOR
	CENTRIFUGAL PUMP
	SPRAY BALL
	DRAIN
	BUTTERFLY VALVE
	HEAT EXCHANGER
	PNEUMATIC BUTTERFLY VALVE
	3 WAY VALVE
	NON RETURN VALVE
	GATE VALVE

	HAND OPERATED VALVE
	PRESSURE RELIEF VALVE
	REMOTE OPERATED PNEUMATIC VALVE
	DIAPHRAGM VALVE
	AMMONIA SENSOR
	FLOW DIRECTION
HS	HYDROGEN SENSOR
	CONTROL VALVE
	EXTRACTOR FAN

REV	DATE	REVISIONS	BY

POLITÉCNICA · LIÈGE université · Navantia SEANERGIES

TITLE: TYPE A OR MEMBRANE REFRIGERATED		NAME: PIPING & IDENTIFICATION DIAGRAM	
DATE.	25 JULY 2023		
DRAWN BY:	D.R.O'CONNOR	SCALE	NTS
DRW NO.	PID003	SHEET	1 OF 1

D General Arrangement Drawings

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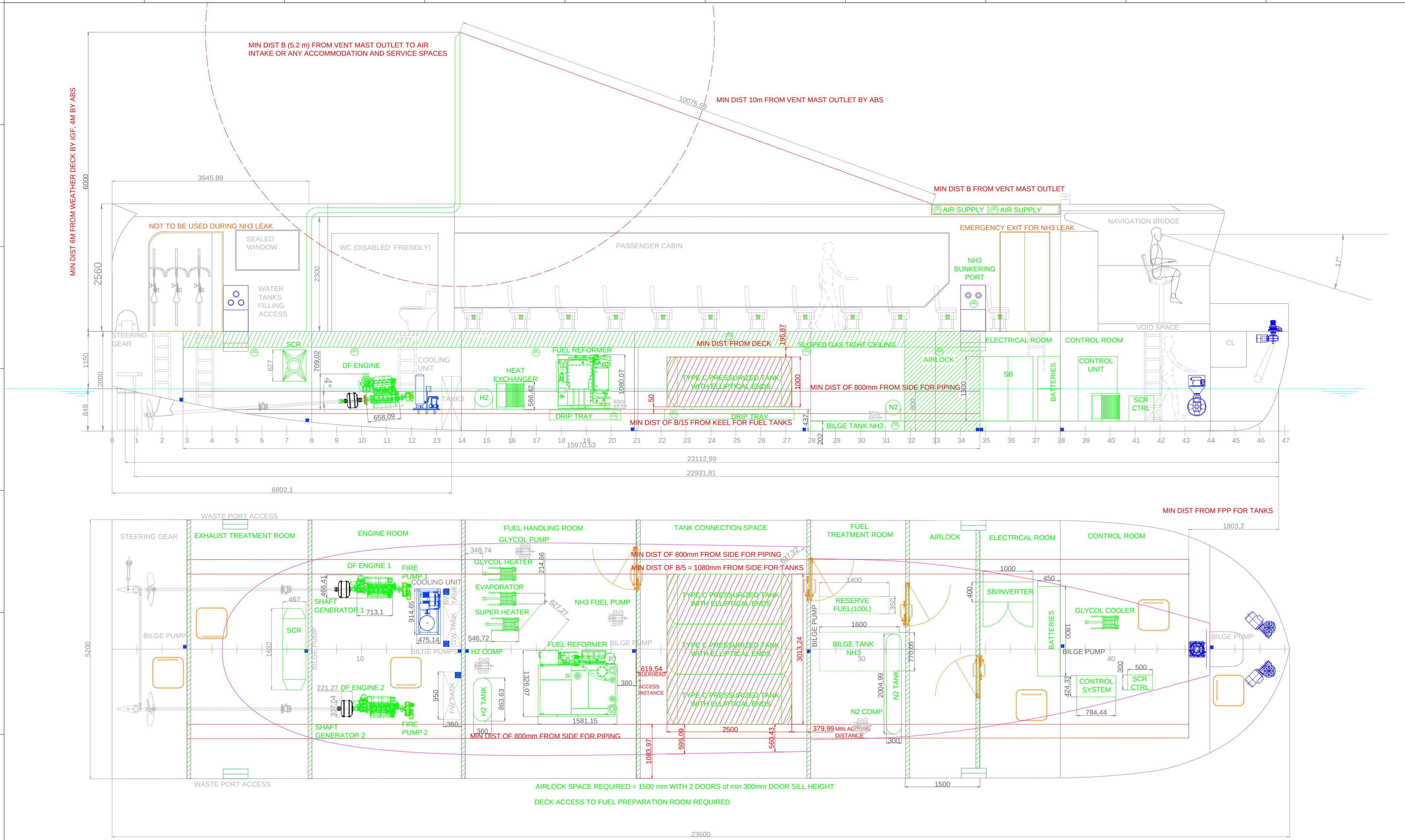
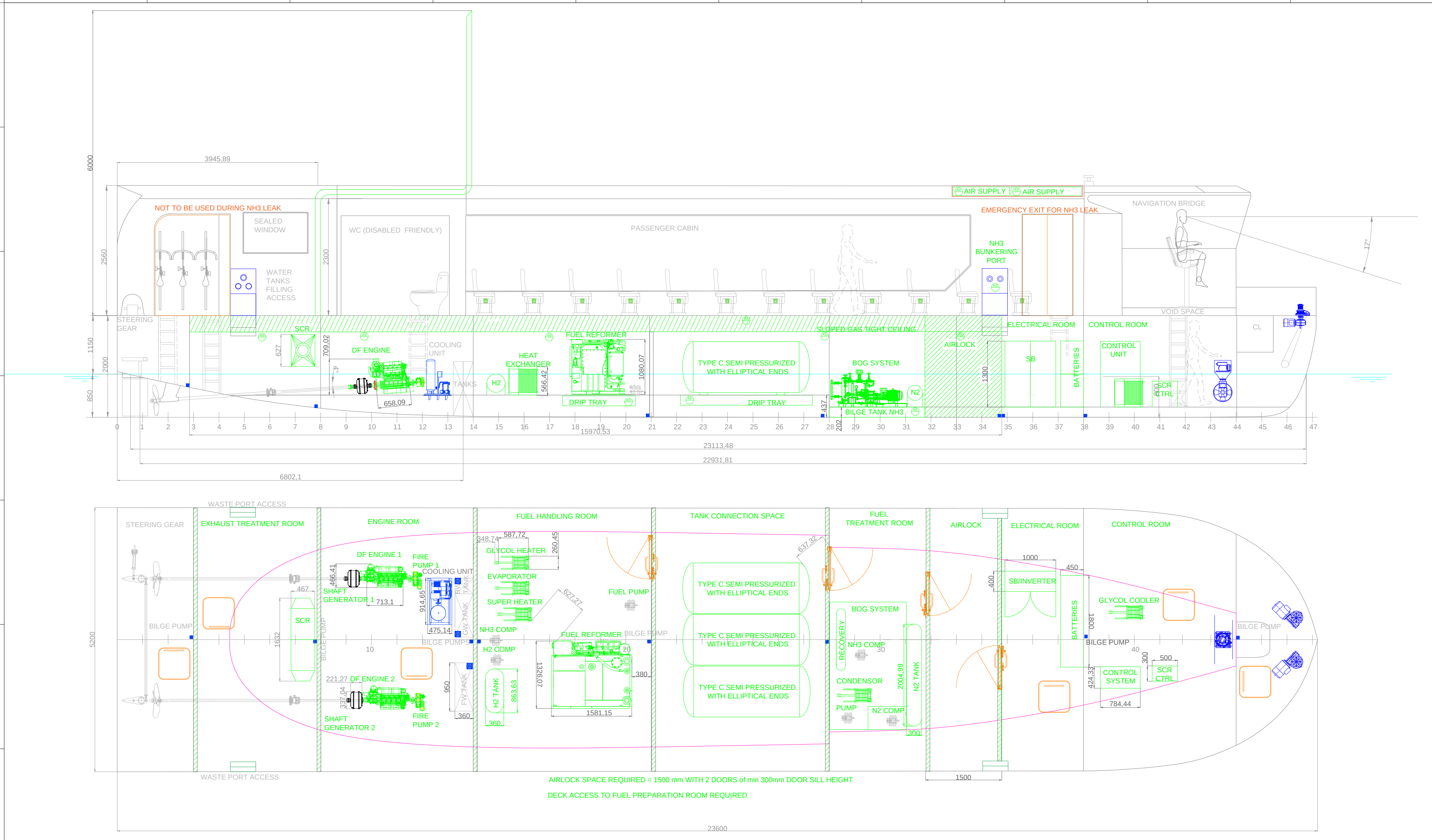


LEGEND: NEW NH3 EQUIPMENT (GREEN)
EXISTING ELECTRICAL EQUIPMENT (BLUE)
ACCESS DOORS/ HATCHES (ORANGE)
DECK LINE (PINK)
DRAFT LINE (CYAN)
REGULATION DISTANCE (RED)

REV	DATE	REVISIONS	BY

POLITÉCNICA | LIÈGE université | Navantia SEANERGIES

TITLE: CASE 3A: TYPE C PRESSURIZED TANKS		TYPE: GENERAL ARRANGEMENT DRAWING	
DATE.	30 JULY 2023		
DRAWN BY:	DR. O'CONNOR	SCALE	1:100
DRW NO.	GA002	SHEET	1 OF 1

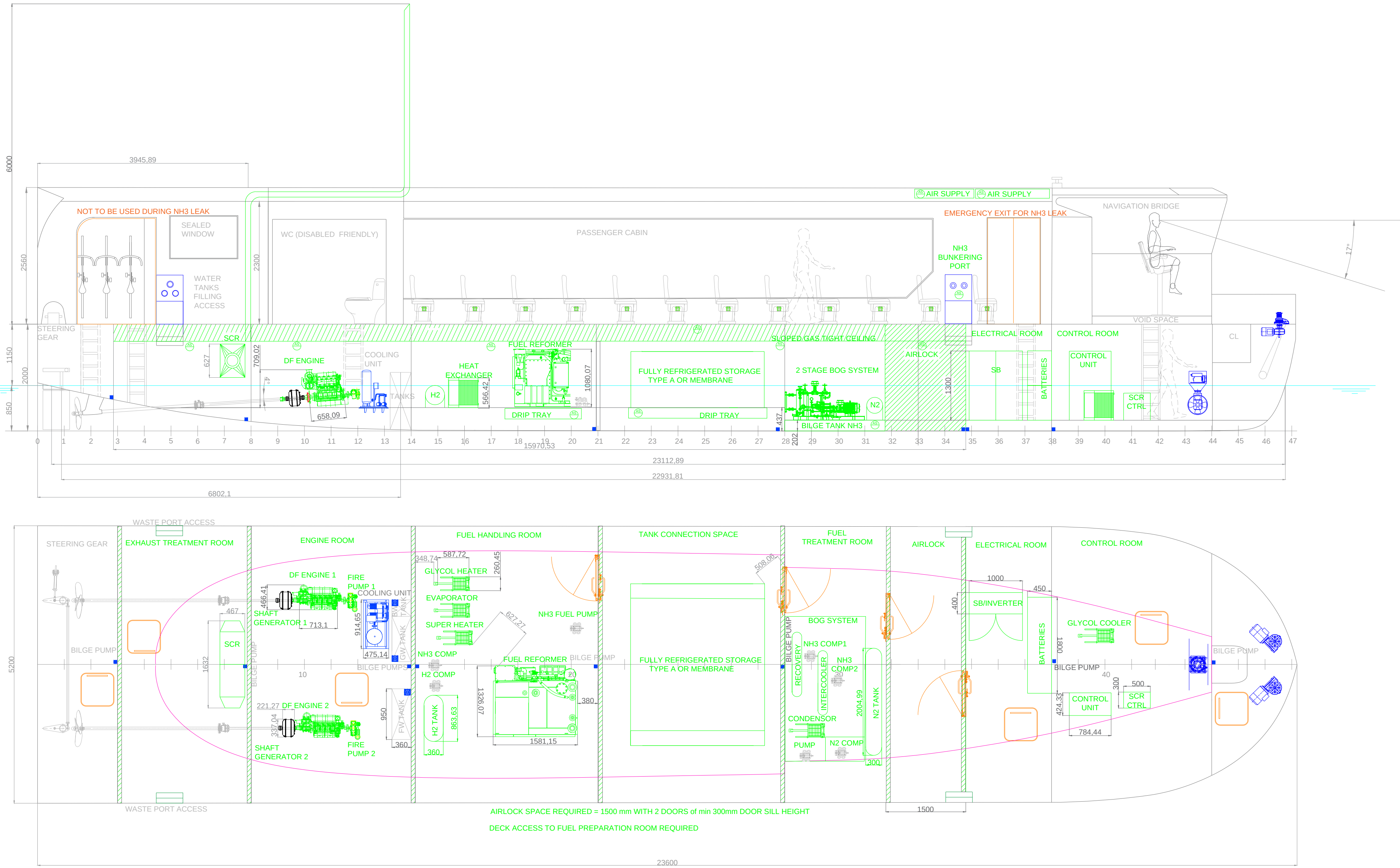


LEGEND:
NEW NH3 EQUIPMENT (GREEN)
EXISTING ELECTRICAL EQUIPMENT (BLUE)
ACCESS DOORS/ HATCHES (ORANGE)
DECK LINE (PINK)
DRAFT LINE (CYAN)

REV	DATE	REVISIONS	BY

POLITÉCNICA · LIÈGE université · Navantia SEANERGIES

TITLE:	CASE 3B: TYPE C SEMI PRESSURIZED TANKS	TYPE:	GENERAL ARRANGEMENT DRAWING
DATE:	30 JULY 2023		
DRAWN BY:	DR. O'CONNOR	SCALE	1:100
DRW NO.	GA002	SHEET	1 OF 1



LEGEND:
NEW NH3 EQUIPMENT (GREEN)
EXISTING ELECTRICAL EQUIPMENT (BLUE)
ACCESS DOORS/ HATCHES (ORANGE)
DECK LINE (PINK)
DRAFT LINE (CYAN)

REV	DATE	REVISIONS	BY

POLITÉCNICA — LIÈGE université — Navantia SEANERGIES

TITLE: CASE 3C & 3D: REFRIGERATED TANKS		TYPE: GENERAL ARRANGEMENT DRAWING	
DATE.	30 JULY 2023		
DRAWN BY:	DR. O'CONNOR	SCALE	1:100
DRW NO.	GA002	SHEET	1 OF 1

E Calculations for Machinery Sizing

E.0.1 Overall system Flow Rates

Table 35: Energy and mass balances to and from the engine

Engine Requirements	
Power output of engines (kW)	148.089
New Engine Efficiency	0.450
Power requirement in Fuel for Engine (kW)	329.086
Energy Balance of Flows to Engine	
Mol % of ammonia	0.700
Mol% of hydrogen	0.300
Ammonia LHV (kJ/kg)	18603
Hydrogen LHV (kJ/kg)	119957
Ammonia flow (kJ/s) Assuming 70% Ammonia	230.360
Hydrogen flow (kJ/s) Assuming 30% Hydrogen	98.726
Ammonia mass flow into engine(kg/s)	0.012
Hydrogen mass flow into engine(kg/s)	0.001
Mass of NH3 (g/mol)	17.031
Mass of N2(g/mol)	28.013
Mass of O2 (g/mol)	31.999
Mass Balance of Flows to Engine	
Excess Air ratio	2
Air mass stoichiometric with excess air (kg Air/kg Ammonia)	11.865
Air mass stoichiometric with excess air (kg Air/kg H2)	144.053
Air mass flow (kg of air/s)	0.265
Total mass flow (kg/s)	0.279
Exhaust Outlet Flow	
Exhaust gas flow (kg/s)	0.279
Mass flow of excess air (kg/s)	0.133

The composition and temperature of the gases are provided in Table 36.

Table 36: Composition of Exhaust Gases

Composition of exhaust gas	
Air (47%)	0.470
Water Vapor (12%)	0.120
Nitrogen (41%)	0.410
Specific Heats	
Air (kg/kg.K)	1.075
Water Vapour(kg/kg.K)	2.080
Nitrogen (kg/kg.K)	1.098
Combine Exhaust gas (kg/kg.K)	1.208
Temperatures	
Exhaust gas heat availability (kW)-30 % assumption	98.726
Start Temp fuel and air (K)	298
Temperature change (delta K)	293.253
Temperature of exhaust gas (K)	591.253
Temperature of exhaust gas (C)	318.253

Equation 14 was used to perform a mass balance of products and reactants results of which are shown in Table 37. In this way the flow of ammonia to the cracker could be estimated.

Table 37: Mass Balance of Ammonia Cracking

	Molar Mass (kg/mol)	Mols	Mass distribution	kg substance /kg hydrogen	Mass flow rates (kg/s)
NH3 -Reactant	0.0170305	2	0.034061	5.63	0.0052
N2-Product	0.0280134	1	0.0280134	4.63	0.0042
H2-Product	0.002016	3	0.006048	1.00	0.0009

Further consideration was made that the cracker of Cechetto et al[102] had a 90% hydrogen recovery rate and a 99.7% conversion efficiency. The fuel consumption of the vessel is described in Table 38.

Table 38: Final Fuel consumption of the vessel

Outlet conditions	Flow (kg/s)
Direct Ammonia flow to engines	0.01238
Direct hydrogen flow to engines	0.00082
Ammonia flow to cracker	0.00515
Input required	
Ammonia liquid flow from tanks (kg/s)	0.01753

E.0.2 Engines

The engine was sized for dimensions and weight based off existing HCCI , Diesel and dual fuel engines. It is predicted that an HCCI engine used for combusting ammonia and hydrogen should be similar to those used by automobiles with diesel and gasoline. A number of engines were compared to evaluate characteristic power density of different fueled engines. The proposed HCCI engine is estimated to have a power density similar to a Diesel engine yet lower due to the use of ammonia. Based on this estimated power density a weight was estimated for a 70kW engine around 175kg.

Table 39: Comparison of Different Engine types

Engine Selection	**Fuel Type**	**Power (kW)**	**Weight (kG)**	**Power Density (kW/kg)**	**Length (m)**	**Width (m)**	**Height (m)**
GM 2.2L HCCI	Diesel	110	175	0.63	0.7	0.6	0.7
Toyota 2L HCCI	Gasoline	125	150	0.83	0.6	0.6	0.7
Mazda SKYACTIVE-X HCCI	Gasoline	132	104	1.27	0.4	0.4	0.6
Scania DI13 092M (1200 rpm)	Diesel	320	1180	0.27	1.5	1.0	1.2
ABC BeHydro 6DZD/I	H2	1000	10620	0.09	na	na	na
ABC BeHydro 8DZD/I	H2	1335	13905	0.10	na	na	na
ABC BeHydro 12DZD/I	H2	2000	18000	0.11	na	na	na
ABC BeHydro 16DZD/I	H2	2670	21750	0.12	na	na	na
WinGD X62DF	NG/Diesel	14310	377000	0.04	9.8	4.2	11.9
Proposed engine (HCCI)	**H2/ NH3**	**70**	**175**	**0.4**	**0.7**	**0.5**	**0.7**

E.0.3 Selective Catalytic Reduction System

The SCR to be used in the general arrangement was sized according to systems available from BlueNox [118]. Generally their systems cater for larger ships ,however, for the conceptual design a regression analysis was used to size the dimensions of the converter. The length is represented by characteristic dimension A. The width represented by dimension B and height represented by dimension C. The regression analysis curves are shown below comparing the power of the required system with the dimension required. This is repeated for the weight of the required system which is taken at an estimated 140 kW for both the engines. It is assumed one SCR will handle the exhaust flow of both engines.

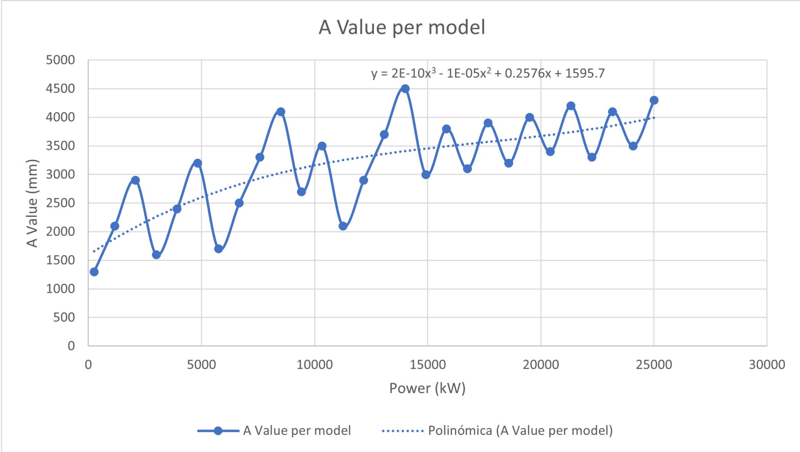


Figure 35: Power vs dimension A of SCR

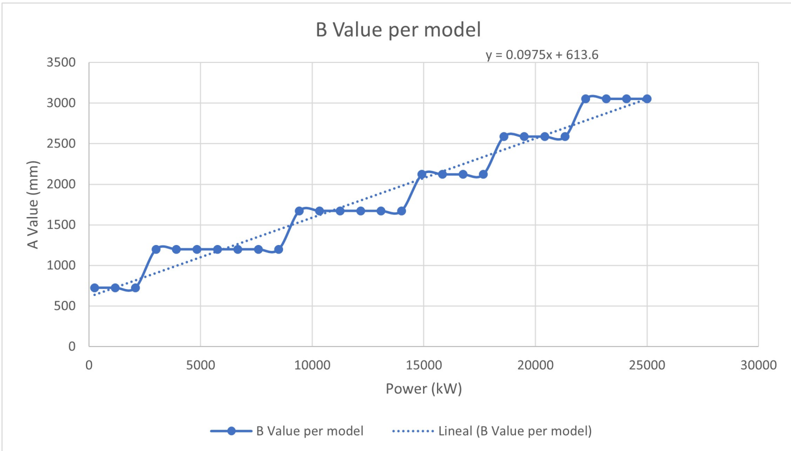


Figure 36: Power vs dimension B of SCR

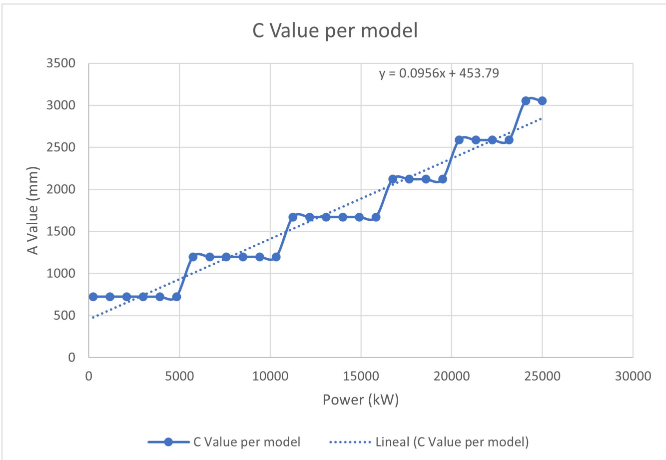


Figure 37: Power vs dimension C of SCR

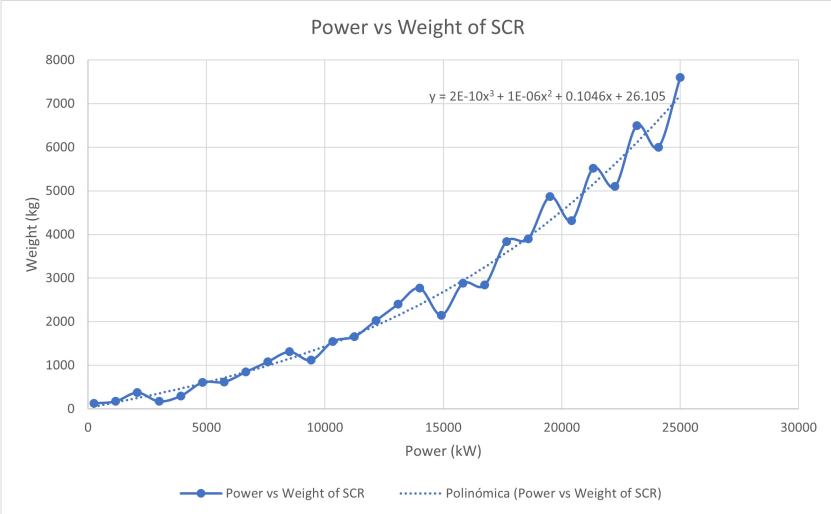


Figure 38: Power vs Weight of SCR

Final dimensions are as follows taken from the regression analysis including the dimensions of the control box required. The supplier did not supply the power requirement for the control system but it is assumed to be small in comparison to other electrical systems:

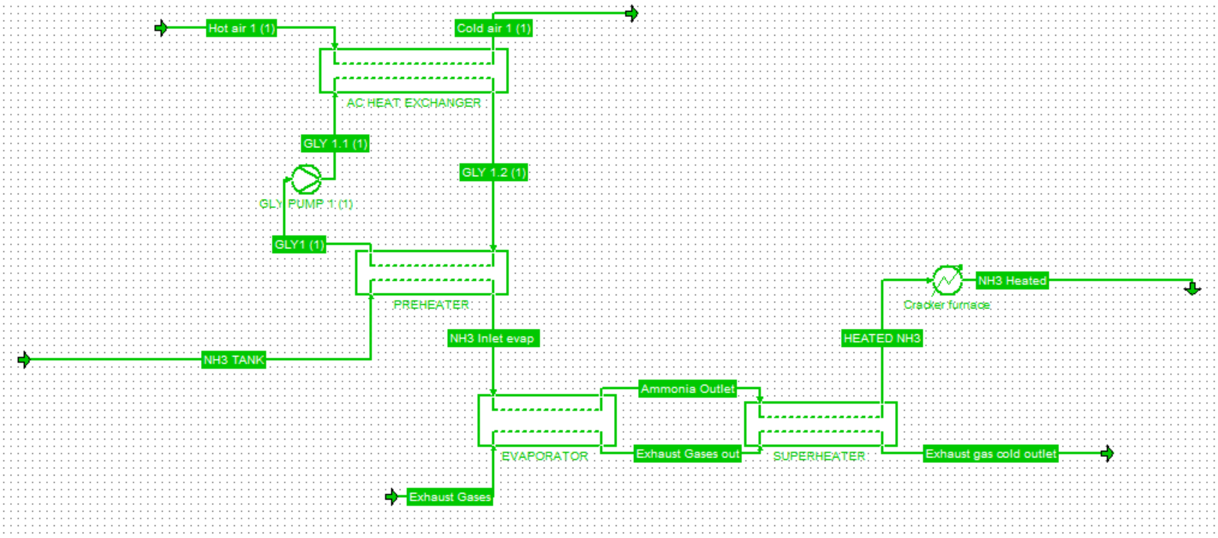
Table 40: SCR and SCR Control System Dimensions

	SCR Unit	Control System
Power [kW]	140.0	Unknown
Weight [kg]	40.8	Unknown
A (length)[mm]	1631.6	500
B (width) [mm]	627.3	300
C (height) [mm]	467.2	500
Vol of System [m^3]	0.5	0.075

E.0.4 Heat Ex-changers

The heat ex-changers were modelled with COFE free software to have a clear picture of the thermodynamics of the problem. The equation solver was set to Peng Robinson for estimating properties of real gases. The air was taken as having a flowrate of 0.255 kg/s and a temperature of 45 $°C$ at ambient pressure. Each fuel condition (Pressurized, Semi-Pressurized and Refrigerated) were modeled to identify the different heating and cooling requirements for each alternative. An example layout from the COFE software is given in the Figure 39. The heat ex-changer settings were all set to counter current flow to maximise heat transfer (despite the diagram appearing to be parallel flow).

Figure 39: Example layout of the heat ex-changers in COFE software model



The heating requirements were calculated with an energy balance and verified using the COFE software.The difference in heating requirements for each fuel condition are illustrated in the Table 41.

Table 41: Heat inputs for cracker

	Refrigerated	Semi	Pressurized
Heat Consumption From Evaporation			
Ammonia heat of evaporation (kJ/kg)	1393	1290	1070
Exhaust heat consumption from vaporizing (kW)	24.4	22.6	18.8
Heat consumption from Cracking			
Input cracker ammonia flow (kg/s)	0.0052	0.0052	0.0052
Start temperature (K)	239	263	283
End temperature (K)	723	723	723
Temperature change (K)	484	460	440
Averaged temperature (K)	481	493	503
Cp -Specific heat of ammonia gas (kJ/kg.K)	2.44	2.50	2.52
Heat consumption to meet cracking temperature (kW)	6.08	5.92	5.71
Heat demand (enthalpy of formation)kJ/kg	2695	2695	2695
Heat required for reaction (kW)	13.88	13.88	13.88
Total heat consumption from cracking (kW)	19.96	19.80	19.59
Evaporation and Cracking (kW)	44.39	42.42	38.35

The size of the heat ex-changers in terms of weight and volume requires knowledge of the type of heat ex-changer that would be suitable for the job. Generally the most cost effective type is a gasket separated plate heat ex-changer, however, for applications with large temperature differences and higher pressure fluids a fully welded plate heat ex-changer is suggested by manufacturers. Another alternative to the plate heat ex-changer is a welded plate and shell heat ex-changer which better handles high temperature differences and pressures yet at a higher cost and greater weight. In the interest of cost and weight saving the welded plate heat ex-changer type was chosen for all heat ex-changers yet in a more detailed analysis the applicability of this type of heat ex-changer should be further investigated with suppliers for particular conditions.

The number of plates to use for the heat ex-changer depends on the amount of heat transfer required and the overall heat transfer coefficient. The heat transfer rate is shown in Table 41 and the overall heat transfer coefficient was assumed to be 175 W/m^2K between the exhaust gas and the liquid ammonia. The dimensions, weight and plate characteristics are based off a ALfaNova 14/HP 14 fusion bonded stainless steel heat exchanger from Alpha Laval [119]. The frame of the heat exchanger holding the plates was estimated at a mass of 15 kgs with the number of plates dependent on the heat transfer scenario. An example calculation for the mass of the evaporator is given in Table 42. The temperature differences dT1 and dT2 are calculated ad follows for Counter current and for parallel flow heat ex-changers. In light of weight and cost saving the counter current option was selected.
The equations for the counter current heat ex-changer are as follows:

$$dT1 = T_{hot_{in}} - T_{cold_{in}} \tag{17}$$

$$dT2 = T_{hot_{out}} - T_{cold_{out}} \tag{18}$$

The equation for the log-mean temperature difference is given as:

$$dTm = (dT1 - dT2)/ln(dT1/dT2) \tag{19}$$

Table 42: Weight Estimation of Heat Ex-changer

Heat ex-changer Evaporator	
Q (W)	24400
U (W/m2K)	175
COUNTER CURRENT	
dT1 (K)	86.62
dT2 (K)	226.95
dTm (K)	145.69
Area required (m^2)	0.957
n (number of plates)	8
Plate area (m^2)	0.11988
Estimated plate thickness (m)	0.001
Stainless steel density (kg/m3)	8000
Weight of plates (kg)	7.65
Weight of supports	15
Total Weight of HE (kg)	23
Height (m)	0.555
Length	0.32
Width	0.216
PARALLEL	
dT1	310.65
dT2	2.92
dTm	65.93
Area required (m^2)	2.115
n (number of plates)	18
Plate area (m^2)	0.12
Total Weight of HE (kg)	31.92
Height (m)	0.555
Length	0.37
Width	0.22

E.0.5 Cracker

Figure 40: Cracker from Cechetto et al. [102]

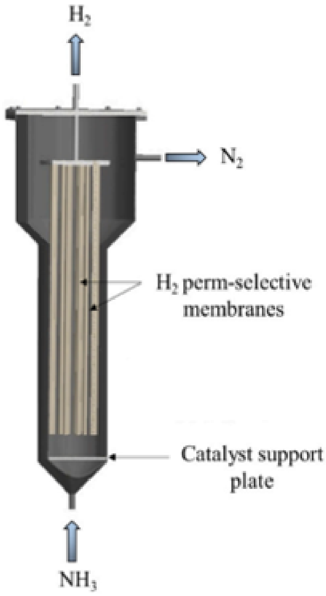


Table 43: Cracker from Cechetto et al. size estimate [102]

Reactor Body	
Density of SS310 shell (kg/m3)	8000.00000
Inner diameter (m)	0.04500
Thickness of reactor (estimate)	0.00300
Inlet plate diameter (m)	0.04500
Top plate diameter (m)	0.08000
Top length (m)	0.15000
Bottom length (m)	0.35000
Top section volume (m3)	0.00011
Bottom section volume (m3)	0.00015
Top plate volume (m3)	0.00002
Bottom plate volume with cone (m3)	0.00001
Overall volume of the reactor shell (m3)	0.00029
Reactor shell weight (kg)	2.32745
Catalyst	
Catalyst weight Ru (kg)	0.25000
Active Length (m)	0.28000
Alumina density	3950.00000
Alumina pellets volume estimate	0.00066
Alumina pellets weight (kg)	2.62084
Membrane	
Pd based membrane desnsity (kg/m3)	12023.00000
Palladium membrane max volume	0.00031
Palladium membrane weight (kg)	3.74786
Total	
Total weight of the Reactor (kg)	8.94616
Total weight + Allowance for other bolts and welding (kg)	10.00000
Scaling and Final Estimate	
Required mass flow rates (L/min)	1.22000
Designed for flow rate(L/min)	0.50000
Number of Reactor units Required to fulfill flow requirement	2.44000
Approximate weight of new reactor (assume 3 reactor units required) (kg)	30.00000
Furnace surrounding the reactor weight (for 3 x 24kW standard electric heaters) (kg)	39.00000
Additional insulation (kgs)	10.00000
Cage support and gas tight box surrounding the reformer (kgs)	20.00000
Total weight of the reformer unit (kg)	99.00000

E.0.6 Ventilation

The ventilation system layout is determined by first calculating the required flow rate of air to each compartment to maintain 30 air changes per hour. For this calculation the volume of each compartment was calculated. In total there were 6 spaces that needed active ventilation. Five of the six spaces requiring negative pressure and the airlock space requiring positive pressure. The pressure difference in the room can be controlled by varying the supply and exhaust ducting. The total flowrate required from the air intake fan is given in the final column of Table 44

Table 44: Ventilation System flowrates required for each space

	Airlock	Bunkering	Fuel	Fuel Prep	Machinery	SCR	Total
Pressure type	Pos	Neg	Neg	Neg	Neg	Neg	
Area (m^2)	7.5	7.5	6.4	6.4	6.4	4.7	38.8
Length (m)	1.5	1.9	3.5	3.5	3.0	2.5	15.9
Vol (m3)	11.2	14.0	22.3	22.3	19.1	11.8	100.8
Flowrate (m^3/s)	0.1	0.1	0.2	0.2	0.2	0.1	0.8
Flowrate (m^3/min)	5.6	7.0	11.2	11.2	9.6	5.9	50.4
Flowrate (m^3/hr)	335.7	420.7	669.9	669.9	574.2	354.6	3025.0

The compartment space available for ventilation ducts was evaluated by looking at the AutoCad drawing. It was determined to maintain adequate kneeling space so that the duct sizing could not exceed the dimensions in Figure 41.

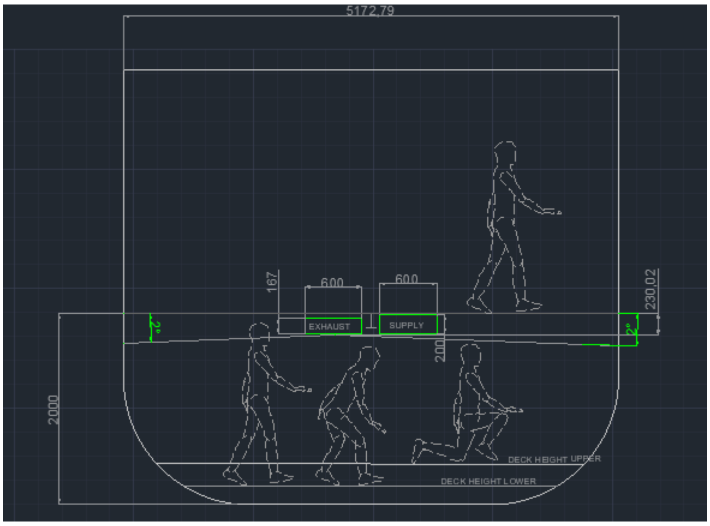


Figure 41: Ventilation Duct Layout

The supply air requirement was such that in the case of a leak the rooms would be flooded with clean air at a rate of 45 air changes per hour to dilute the concentration of ammonia gas. Under normal operation all the rooms except the airlock would be under negative pressure. It is also necessary to estimate the size of the blower so that the pressure losses and suction pressure are known.

The required pressure difference over the supply fan to maintain a negative pressure in the ammonia rooms and overcome the pipe losses in the system can be calculated using Bernoulli's steady flow energy equation. The steady flow energy equation can be written as follows taking into account pipe losses for the system,the supply fan requirement and the pressure difference over the room:

$$P_1 + 0.5\rho(v_1)^2 + \rho g h_1 + P_{fan} - P_{room} - P_{losses} = P_2 + 0.5\rho(v_2)^2 + \rho g h_2 \tag{20}$$

The static pressure at the inlet P1 and at the outlet are assumed to be atmospheric pressure and therefore cancel. The pressure losses due to pipe bends and friction are calculated as follows:

$$P_{losses} = (f \cdot \frac{L}{D} + \sum K) \cdot 0.5\rho(v_2)^2 \tag{21}$$

Where L is the total length of the ducting taken as 19.74m and D is the hydraulic diameter of the pipe taken as 0.3 m based off previous calculation for the flow rates required. The friction factor f is calculated using the Colebrook friction factor equation where the surface roughness ϵ is taken as 0.15. Minor losses are calculated assuming 12 inlets conditions (k1 =0.5) and at least 2 90 degree bends (k2=0.5). The summary of values obtained for the pipe losses are provided in Table 45.

Table 45: Pipe losses summary

Air Density (kg/m3)	1.29
Velocity (m/s)	6.69
Diameter(m)	0.31
Re	183103.73
Roughness	0.15
f	0.08
Length of pipe (m)	19.74
Major Pipe loses (Pa)	146.97
Minor pipe loses (Pa)	236.99
Total Pipe loses (Pa)	383.97

Finally based on Equation 20 P_{fan} was calculated as 415 Pa assuming P_{room} was a negative pressure difference of 15 Pa. Based on the calculated requirements for fan pressure and flow rate an appropriate supply fan was sourced. The chosen fan had the following specifications:

Table 46: Elecktor MAF fan specifications [120]

Max capacity (m3/h)	3400
Total pressure difference (Pa)	2000
Power Requirement (kW)	0.25
Weight (kg)	1.5

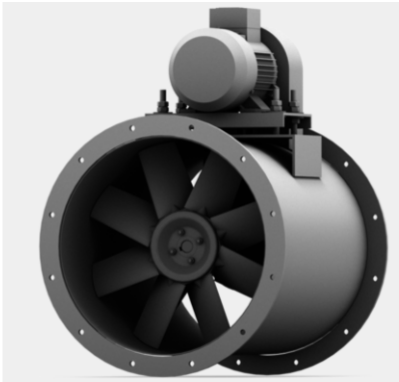


Figure 42: Axial flow fan from Elektror [120]

F Calculations for Tank Sizing

The tanks are designed in compliance with IGC regulations, ASTM VIII div 1 rules and recommendations from BV NR 467 Pt C,Ch1,Sec 3 (Classification of steel ships).

F.0.1 Calculations for Fully pressurized NH3 Tank

IGF code (6.4.9.3.3.1) states that in all cases the design pressure should not be less than the maximum allowable relief valve setting and for liquefied gas fuel tanks where there is no temperature control and where the pressure of a liquefied gas fuel is dictated only by the ambient temperature ,the design pressure shall not be less than the guage vapour pressure of the liquefied gas fuel at a temperature of 45° except if a lower temperature is accepted by administration.Therefore pressurized tanks would need to be designed based off the saturation pressure (1.8 MPa) at an ambient condition of 45°C based off current IGF regulation.

An exception is IGF 6.4.9.3.3.1.2 where the ships voyage is of restricted duration, in which case P0 may be calculated based on the actual pressure rise during the voyage and account may be taken of any thermal insulation of the tank.

So that means pressurized tanks would need to be designed based off the saturation pressure (1.8 MPa) at an ambient condition of 45°C for unlimited voyages and in the case that the duration of the voyage is known the actual pressure rise during the voyage should be known.

The maximum Internal pressure is a function of static and dynamic pressure (Peq=P0+Pgd) Where P0 is the design fluid pressure and Pgd is the pressure generated by dynamic loads and accelerations.

$$P_{gd} = \alpha \cdot Z \cdot (\frac{\rho}{(1.02 * 10^5)})(MPa) \tag{22}$$

α represents the acceleration in the direction resulting from gravity and dynamic loads.Z is the largest fluid height (m) above the point where the pressure is to be determined measured from the tank shell in the direction. In the case of an inland waters ferry the accelerations are negligible meaning the main contribution to the design pressure is indeed the saturation pressure of ammonia at ambient air temperature which is at least 1.8 MPa and the gravitational.

Assuming 3 horizontal pressure tanks the end cap types available are hemispherical, ellipsoidal,conical or torispherical. The option which enables the greatest volume saving is an ellipsoidal shaped head. For such a tank the smaller axis of the ellipse has a required height from the end of the cylinder of h = 0.2D where D is the diameter of the cylinder according to NR 467 ,Pt C ,Chapter 1 ,Section 3, 2.4.

Using a diameter 1m the distance h must be greater than 0.2 m. Once the end cap dimensions were known it was possible to calculate the tank volume and how much fuel could be held within the tank. Thus the external load on the tank due to the reaction supports could be calculated.

According to IGC section 4.23.2 the minimum thickness of a pressure vessels type C tank may not be less than 5mm for carbon-manganese steels, 3mm for austenitic steels and 7mm for aluminium alloys. BV 467 also states that for pressure vessels the minimum thickness should always be at-least : t = 3 + D/1500 mm = 3.67 mm.

The ultimate strength criteria that were used for the calculation of the required thickness were as noted in Table 47. See section 6.4.12 of IGF code for more details [110]. Re is the specified minimum yield strength and Rm is the specified minimum tensile yield stress at room temperature, both given by IGC code for ammonia (Section 17.12 of IGC code [30]). Parameter A depends on the type of steel used. IGF 6.4.15.3.3.1 states that for nickel and carbon manganese steels parameter A has a value of 3, for austenitic stainless steel a value of 3.5 and for aluminium alloys a value of 4. The value of B is set as 1.5 for all 3 metal groups just mentioned. The parameter f is defined as the lesser of Rm/A or Re/B.

Table 47: Table of Ultimate Strength Parameters

A	3.5
B	1.5
Rm (MPa)	410
Re (MPa)	355
f (the lesser of Rm/A or Re/B) (MPa)	117

Parameter f is essentially the maximum allowable stress calculated as 117 MPa for austenitic steels and 137 MPa for carbon-manganese steels. Generally meaning that carbon manganese steels will not require the same thickness as stainless steels. The allowable stress for type C independent tanks should not exceed any of the following limits:

$\sigma_m \leq f$
$\sigma_L \leq 1.5f$
$\sigma_b \leq 1.5f$
$\sigma_L + \sigma_b \leq 1.5f$
$\sigma_m + \sigma_b \leq 1.5f$
$\sigma_m + \sigma_b + \sigma_g \leq 3f$
$\sigma_L + \sigma_b + \sigma_g \leq 3f$

Where:

σ_m is the equivalent primary general membrane stress
σ_L is the equivalent primary local membrane stress
σ_b is the equivalent primary bending stress
σ_g is the equivalent secondary stress

Most important for cylindrical vessels was the first criteria which is that the equivalent primary general membrane stress should not exceed the allowable stress. To calculate the primary general membrane stress thin walled assumptions were used and then the Von Mises equivalent stress was taken as the primary general membrane stress.

The hoop stress is calculated for thin walled pressure vessels by:

$$\sigma_{hoop} = \frac{P \cdot r_i}{t} \tag{23}$$

The longitudinal and shear stress is calculated by:

$$\sigma_{shear} = \sigma_{Longitudinal} = \frac{P \cdot r_i}{2t} \tag{24}$$

Where t (m) is the thickness of the shell ,r_i is the internal radius of the cylinder and P is the internal design pressure. Then the Von Mises overall primary membrane stress is calculated as:

$$\sigma_m = \sqrt{\sigma_{hoop}^2 + \sigma_{Longitudinal}^2 - \sigma_{hoop} * \sigma_{Longitudinal} + 3 * \sigma_{shear}^2} \tag{25}$$

The required tank thickness for the pressurized horizontal cylinder made of austenitic steel was given as 17 mm. This kind of estimation gives a large factor of safety as maintaining an allowable stress of 117 MPa is far below the yield stress of 355 MPa. To validate these figures a quotation from a reputable pressure vessel and boiler manufacturer in Spain was sourced and they suggested using 15mm 304L steel. Therefore the thickness of 17mm was by far sufficient.

As mentioned in the report the calculations for the LPVs were based off a quotation for a round walled tank from *Lattice Technology*. In the quotation the overall weight was given for the specified design pressure.

F.0.2 Calculations for Semi-Pressurized NH3 Tank

IGF code does not specifically mention requirements for semi-pressurized tanks, however, it is generally accepted that semi-pressurized tanks will be designed to the specifications of a normal type C tank. Anticipated operating pressures are usually in the range of 0.85 MPa or 0.85 MPa (see Table 5). For the analysis of the semi-pressurized vessels the same formulas from appendix F.0.1 were used. Since the design pressure was lower it was no surprise that required thicknesses of structural members were a lot less than the fully pressurized alternative. In comparison to the required tank thickness for the fully pressurized horizontal cylinder made of austenitic steel, the semi-pressurized alternative only required a shell thickness of 6 mm to keep below the maximum allowable stress.

F.0.3 Calculations for Refrigerated Type A NH3 Tank

Tanks are designed based off 0.07 MPa design pressure or 0.7 bar as specified in IGF 6.4.15.1.1.1. The tank designed for here is a type A tank which requires a full secondary barrier. Regulation 6.4.15.1.3.1 of IGF code [110] states that nominal membrane stresses for primary and secondary members (stiffeners, web frames, stringers, girders), when calculated by classical analysis procedures, shall not exceed the lower Rm/2.66 or Re/1.33.

In the case for an ammonia fuel tank the values of Rm and Re are set as 410 MPa and 355 MPa respectively. Thus the maximum allowable stress is set as 154 MPa.

Using classical beam theory the dimensions of each of the stiffeners to make up a tank of the required size were calculated. In all cases simplified T shaped stiffeners were chosen. An analysis was performed for bending and shear stress in the stiffeners, frames and plate. Each stiffener was analysed at significant points such as in the web, in the flange and in the plate to web junction, The stiffener and frame dimensions which ensured the stress was below the allowable limit were calculated as in 48.

Table 48: Dimensions for stiffeners and frames for Type A tank

	T Girders Frame Trans(mm)	Bulb Stiffs Top Long (mm)
Frame Spacing	833	833
Stiffener Spacing	300	300
Width of the effective plate	90	300
Thickness of the plate	6	6
Thickness of the web	4	4
Height of the Web	50	50
Width of the Flange	20	20
Thickness of the Flange	6	6

With these dimensions a Von Mises analysis was performed to simultaneously analyse the stress due to frame bending , the stress due to stiffener bending and the stress due to plate bending. For this analysis the primary stress due to the hull hogging and sagging was set as zero as the tank is fully independent and said to be secured by simple rolling supports therefore the ships bending loads were assumed to not effect the required strength for the tank. In all locations the maximum stress that was achieved was 147 MPa which is within the limits.

F.0.4 Calculations for Membrane NH3 Tank

According to IGF 6.4.15.4.1.4 the design pressure (internal pressure) for a membrane tank should not exceed 0.025 MPa in normal circumstances, however the maximum allowable with special insulation mechanisms is 0.07 MPa. For this study the latter was assumed as the probable case.

IGF 6.4.15.4.1.6 states that the thickness of the membranes shall normally not exceed 10 mm. This membrane is fully supported by an outer shell of stiffened panels therefore the calculation for the strength of the tank is rather for the supporting structure and not for the membrane.Typical membranes from GTT are said to be just over 1mm thick therefore do not contribute significantly to the tank strength. The supporting structure is joint to the ship hull, therefore deformations of the hull directly effect the membrane tank. Such loads can be assumed from the maximum loads specified by BV rules for inland navigation vessels[121]. The maximum bending moment given by BV NR217 section 2.1.6 is calculated by 26:

$$M_H = 0,2L^2 \cdot B^1,48 \cdot D^0,172(1,265 - CB) \tag{26}$$

Where L is the length of the vessel, B is the breadth of the vessel, D is the Depth of the vessel and CB is the block coefficient. the ship inertia at the mid-ship cross section was calculated from previous studies as I = 0.0574 mm^4. The distance from the neutral axis to the keel for the maximum bending moment was calculated as 1.08m. Therefor given that the maximum bending moment was calculated as 1020 kN.m using the target ship parameters the maximum primary bending stress contributing to tank design would be 19 MPa.

The same Von Mises analysis was performed as for the Type A refrigerated tank to find that the maximum stress was always below the 154 MPa allowable limit. In all locations the maximum stress that was achieved was 147 MPa. The size of the stiffeners and frames was not required to be changed. It was assumed the same shape of tank would be used for the type A tank as for the membrane tank.

F.0.5 Calculations for Hydrogen Buffer Tank

The hydrogen buffer tank size was evaluated for an assumed 5 minute startup time and as small as possible tank. To achieve this hydrogen should be compressed however within safe limits. Gas pressurized above 1MPa is classified as high pressure and not suitable for piping by IGF code [110] therefore it is inherently safer to temporarily store hydrogen at 1 MPa instead of 2 MPa. To allow for a smaller tank size the hydrogen needs to be at or near atmospheric temperature after cracking which can be achieved by running the incoming cold ammonia past the outgoing hot hydrogen stream within the cracker module, Table 49 illustrates the different densities of hydrogen at various temperatures and pressures. The weight is estimated for a typical cylindrical pressure vessel designed for 1 MPa using the same method as the cylindrical ammonia pressure vessels.

Table 49: Hydrogen Buffer Tank Calculation

Hydrogen Tank Sizing	
Startup time assumed (min)	5.00
Flow rate of hydrogen required to engine during operation (kg/hr)	3.29
Time required to run hydrogen before cracker startup (hr)	0.08
Tank fuel mass required (kg)	0.27
H2 Density 0.1 MPa and T=723K (kg/m3)	0.03
H2 Density 1 MPa and T=723K (kg/m3)	0.33
H2 Density 2 MPa and T=723K (kg/m3)	0.67
H2 Density 1 MPa and T=300 K (kg/m3)	0.80
Tank Volume (No Compressor and T=723K) (m3)	8.18
Tank Volume(1 MPa compressor and T=723K) (m3)	0.82
Tank Volume (2 MPa compressor and T=723K) (m3)	0.41
Tank Volume (1 MPa compressor and 300K) (m3)	0.34
Dimensions for Tank of 1 MPa and 300K	
Diameter (m)	0.36
Length (m)	0.85
Thickness (m)	0.008
Weight (kg)	59.88

F.0.6 Calculations for Nitrogen Buffer Tank

The nitrogen buffer tank was designed to hold enough nitrogen to purge the fuel lines after the end of each day of operation. The volume of nitrogen to be stored would be the same amount of gas that would be required to completely full the fuel lines. Here an overall length of fuel line was estimated as 30 m and the averaged pipe diameter was taken as 32 mm. A short study of nitrogen density at various temperatures and pressures revealed the need for a nitrogen compressor to take the nitrogen up to 1 MPa to save as much space onboard the vessel as possible. Much like the hydrogen tank the weight was estimated for a typical cylindrical pressure vessel designed for 1 MPa pressure. Table 50 illustrates the parameters used for estimating the nitrogen tank.

Table 50: Nitrogen Purging tank calculation

Nitrogen Tank Sizing	
Length of pipes to be purged (m)	30.00
Diameter of pipes (average) (m)	0.03
Volume of Nitrogen required (m3)	0.96
Flow rate of Nitrogen from cracker(kg/hr)	15.25
N2 Required Density 6 bar and T=300 K (kg/m3)	6.74
Mass of Nitrogen required (assume at 6 bar pressure) (kg)	6.47
N2 Density 1 MPa and T=300K (kg/m3)	11.25
N2 Density 1.2 MPa and T=300K (kg/m3)	13.50
N2 Desity 2 MPa and T=300 K (kg/m3)	22.52
N2 Volume 1 MPa and T=300K (m3)	0.58
N2 Volume 1.2 MPa and T=300K (m3)	0.48
N2 Volume 2.0 MPa and T=300 K (m3)	0.29
Dimensions for a Tank of 1 MPa and 300K	
Diameter (m)	0.30
Length (m)	2.00
Thickness (m)	0.008
Weight of tank (kg)	118.45

G Power Requirements

The items highlighted in green are new systems added to the previous electric vessel which will have an additional power requirement.

Equipments	Unit power [W]	Qty.	Total Power [W]	Sailing		Manuvering		Port		Emergency	
				Utility coeff	Power [W]	Utility coeff	Power [W]	Utility coeff	Power [W]	Utility coeff	Power [W]
Extraction Fans Ventilation	50	7	350	1	350	1	350	1	350	1	350
NH3 fuel pump	300	3	900	0.3	270	0.3	270	0.3	270	0.2	180
Fuel Pump backup	200	1	200	0.1	20	0.1	20	0.1	20	0.9	180
Cracker unit	9000	1	9000	0.9	8100	0.9	8100	0.9	8100	0	0
Bow Thruster	10800	1	10800	0.5	5400	0.95	10260	0	0	0	0
Steering Motor	5000	1	5000	0.3	1500	0.8	4000	0	0	0.5	2500
Door Actuators	150	7	1050	0.5	525	0.5	525	0.8	840	0.9	945
Liquid Cooling (Motors)	5000	1	5000	0.95	4750	0.95	4750	0.5	2500	0.9	4500
Windlass (Model V8 - Lewmar)	4000	2	8000	0	0	0	0	0.9	7200	0.8	0
Bilge Pumps	200	10	2000	0.7	1400	0.8	1600	0.5	1000	0.8	1600
Fresh Water System	700	1	700	0.8	560	0.8	560	0	0	0.3	210
Grey and Black Water System	750	2	1500	0	0	0	0	0.9	1350	0	0
Air Conditioning (Machinery Space)	2000	1	2000	0.9	1800	0.9	1800	0.4	800	0	0
Air Conditioning (Passenger Space)	1750	2	3500	0.9	3150	0.9	3150	0.4	1400	0	0
Internal Communication	80.00	6	480	0.9	432	0.9	432	0.9	432	0.9	432
External Communication	300.00	2	600	0.9	540	0.9	540	0.5	300	0.9	540
Radar (Raymarine Cyclone)	195.00	1	195	0.95	185.25	0.95	185.25	0.5	97.5	0.9	175.5
Indoor Cameras (Raymarine CAM50)	3.3	6	19.8	0.95	18.81	0.95	18.81	0.5	9.9	0.9	17.82
Outdoor Cameras (CAM220 IP Marine Camera)	3.3	6	19.8	1	19.8	1	19.8	1	19.8	0.9	17.82
LED Lights (Passenger Compartments, Captain Cabin and Captain Room)	18	24	432	0.75	324	0.75	324	0.2	86.4	0.5	216
LED Lights (Battery Compartment, Engine room, Switchboard Room)	36	12	432	0.5	216	0.5	216	0.5	216	0.5	216
Navigation Lights	55	5	275	0.5	137.5	0.5	137.5	0.2	55	0.8	220
Multipurpose Display	96	1	96	1	96	1	96	1	96	1	96
	40687		52549.6		29794.36		37354.36		25142.6		12396.14

H Weight distribution

The weight distribution tables highlight elements which have been changed or added since the previous electric ferry layout. The items highlighted in green are new systems onboard while those highlighted in orange are existing items which are adjusted to to new requirements of the ship, The input of the weight of the tanks remains as a yellow block illustrating the input needed dependent on tank type. For this reason the LCG ,TCG and KG should not be taken as accurate to any solution in these tables. The lightweights of fixed systems are shown in Table 51 and Table 52.

Table 51: Lightweight Distribution on Ferry Before Adding NH3 Tanks Part 1

	ELECTRIC BOAT	NH3 BOAT	ELECTRIC BOAT				NH3 BOAT		
	[kg]	[kg]	X	Y	Z		X	Y	Z
STRUCTURE									
HULL STRUCTURE	20,350	20,350	11.532	0	1.2		11.532	0	1.2
GASTIGHT CEILING BELOW DECK (0.6 mm thick)	0	400					10.136	0	1.631
BULKHEAD FOR AIRLOCK	0	414					16.5	0	0.985
WELDING	2,035	2,035	11.532	0	1.2		11.532	0	1.2
SUPERSTRUCTURE STRUCTURE	4,828	4,828	11.72	0	3.507		11.72	0	3.507
SEATINGS AND VARIOUS	2,784	2,784	11.633	0	2.43		11.633	0	2.43
PAINTS - INSULATION - LININGS - FLOORS									
PAINTING	160	160	11.532	0	1.2		11.532	0	1.2
INSULATION FIRE, SOUND \THERMIC	780	780	11.72	0	3.507		11.72	0	3.507
FLOOR LININGS	763	763	11.36	0	2		11.36	0	2
FLOORS	1,442	1,442	12.456	0	0.437		12.456	0	0.437
HULLS /DECK EQUIPMENT									
ANCHORS	250	250	22.727	0	0.971		22.727	0	0.971
ANCHORS CHAINS	272	272	22.727	0	1.1		22.727	0	1.1
WINDLASS	72	72	23.335	0	2.107		23.335	0	2.107
RUDDER	81	81	0.5	0	0.365		0.5	0	0.365
RUDDER STOCK	33	33	0.5	0	0.8		0.5	0	0.8
STEERING GEAR	4	4	0.5	-1.7	1.5		0.5	-1.7	1.5
RESCUE \SECURITY SYSTEMS	69	69	11.633	0	2.336		11.633	0	2.336
BATHROOMS	29	29	6.656	0	2.43		6.656	0	2.43
ELECTRIC MOTORS FOR DOORS	63	63	78.183	0.724	4.146		78.183	0.724	4.146
GLASS (DOORS)	195	195	12.074	0.125	3		12.074	0.125	3
GLASS (WINDOWS \ROOF)	601	601	11.629	0	3.76		11.629	0	3.76
GLASS WHEELHOUSE	102	102	21.49	0	3.85		21.49	0	3.85
LADDERS	186	186	11	0	1.215		11	0	1.215
HANDRAILS	148	148	11.633	0	2.5		11.633	0	2.5
CAPACITIES \CIRCUITS									
NH3 TANKS	0						12.42	0	0.99
HYDROGEN TANK	0	60					7.42	1.02	0.67
NITROGEN TANK	0	118					15.67	0.67	0.47
NH3 LIQUID SYSTEMS (PIPES+PUMPS + LIQUID)	0	20					12.42	0	0.99
DIESEL SYSTEMS (PUMP AND PIPES +LIQUID)	0	15					10.5	-0.883	0.351
BACKUP DIESEL TANK	0	25					14.85	-0.883	0.351
VENT MAST	0	17					3.945	0	5.365
BILGE TANK (NH3)	0	82					15	0	0.1
BILGE SYSTEM (PUMP AND PIPES)	0	16					11.7	0	0.2
BILGE SYSTEM (PUMP AND PIPES)	16	16	11.7	0	0.2		11.7	0	0.2
FIRE FIGHTING SYSTEM (PUMP AND PIPES)	243	243	6.368	0	1.491		6.368	0	1.491
FRESH WATER SYSTEM (PUMP AND PIPES)	8	8	5.88	0.57	0.5		5.88	0.57	0.5
GRAY WATER SYSTEM (PUMP AND PIPES)	28	28	6.37	-0.47	0.5		6.37	-0.47	0.5
BLACK WATER SYSTEM (PUMP AND PIPES)	28	28	6.37	-1.05	0.5		6.37	-1.05	0.5
FRESH WATER LIQUIDS IN SYSTEMS	24	24	5.5	0	1.57		5.5	0	1.57
GREY WATER IN SYSTEMS	30	30	5.5	0	1.57		5.5	0	1.57
BLACK WATER IN SYSTEMS	34	34	5.5	0	1.57		5.5	0	1.57

Table 52: Lightweight Distribution on Ferry Before Adding NH3 Tanks Part 2

HVAC									
AC FOR PASSENGERS /BATTERY (SPLIT UNIT)	0	100					16.989	0	4.38
GLYCOL AIR COOLER		16					19.57	-0.54	0.48
AIR SUPPLY -PASSENGER CABIN /ELECTRICS	170	170	6.24	0	4.38		16.989	0	4.38
LIQUID COOLING FOR MOTORS	95	95	6.445	-0.586	0.786		6.445	-0.586	0.786
AIR SUPPLY -BELOW DECK NH3 COMPARTMENTS	85	85	4.93	0	4.38		18.266	0	4.38
AIR RETURN EXTRACTION FANS BELOW DECK	0	30					8	0	1.75
DUCTING FOR SUPPLY AND RETURN AIR	0	425					10	0	1.75
ELECTRICITY/ FUEL SUPPLY									
PRIMARY BATTERY PACKS	6,100	1,669	12.36	0	0.88		18.773	-0.162	0.701
MAIN SWITCHBOARDS	1,000	0	14.515	0	1.08				
AUXILIARY SWITCHBOARD	300	300	17	0.589	1.08		17.974	-0.879	0.849
MAIN INVERTERS	1,100	0	8.906	0.139	1.08				
AUXILIARY INVERTER	220	220	8.91	-0.782	1.08		17.974	-0.879	0.849
VARIABLE FREQUENCY DRIVE (VFD)	136	0	14.513	0	1.08				
EVAPORATOR	0	23					8.053	-1.3	0.717
SUPER HEATER	0	18					8.053	-0.8	0.717
GLYCOL NH3 HEATER	0	16					8.053	-1.8	0.717
AMMONIA CRACKER	0	100					9.329	0.862	1.086
SCR CONTROL UNIT	0	40					19.394	-0.438	0.505
SCR SCRUBBER	0	41					3.652	0	1.316
SHAFT GENERATORS	0	200					4.69	0	0.693
PROPULSION									
MOTORS	288	350	5.785	0	0.693		5.344	0	0.693
SHAFT LINE, PROPS AND PROP BRACKET	242	242	2.044	0	0.446		2.044	0	0.446
BOW THRUSTER	65	65	21.35	0	0.755		21.35	0	0.755
TRANSMISSION \ELECTRONIC EQUIPMENT									
RADAR	29	29	19.1	0	4.64		19.1	0	4.64
OTHER SYSTEMS	34	34	14	0	3.5		14	0	3.5

LIGHTWEIGHT	**Δ BEFORE**	**Δ AFTER**	LCG	TCG	KG		LCG	TCG	KG
TOTALS	45,522	41,093	11.809	0.002792	1.56861133		12.116	-0.01621	1.645367

The dead weight and total weight table is shown in Table 53 with the maximum allowance for the vessels design draft shown at the bottom.

Table 53: Dead-weight Distribution on Ferry Before Adding NH3 Tanks

DEADWEIGHTS	**WEIGHT BEFORE**	**WEIGHT AFTER**	**POSITION BEFORE**				**POSITION AFTER**		
	[kg]	**[kg]**	**X**	**Y**	**Z**		**X**	**Y**	**Z**
CREW AND LUGGAGE	200	200	11.633	0	2.5		11.633	0	2.5
MOBILE LOADS (BIKES AND WHEELCHAIRS)	364	364	2.148	0	3		2.148	0	3
PASSENGERS (100)	7500	7500	11.633	0	3		11.633	0	3
FRESH WATER (98%)	341	341	6.82	0.92	0.634		6.82	0.92	0.634
GREY WATER (98%)	327	327	6.82	-0.668	0.634		6.82	-0.668	0.634
BLACK WATER (98%)	118	118	6.82	-1.287	0.633		6.82	-1.287	0.633
HYDROGEN FOR BUFFER TANK	0	0.27					7.42	1.02	0.67
NITROGEN FOR PURGING TANK	0	6.47					15.67	0.67	0.47
NH3 LIQUID FUEL	0						12.42	0	0.99
DIESEL LIQUID FUEL	0	83					14.85	-0.883	0.351
NH3 BILGE TANK LIQUIDS	0	0							
TOTAL DEADWEIGHT	8850	8939.74	10.8154228	-0.006393	2.77855435		10.84495	-0.0145	2.753941
	Δ BEFORE	**Δ AFTER**	**LCG**	**TCG**	**KG**		**LCG**	**TCG**	**KG**
TOTAL WEIGHT (LIGHT + DEAD)	54,372	50,032	11.648	0.001	1.766		10.94	-0.01	1.70

I Stability Criteria

Table 54: Stability Report for Single Tank without Bulkheads

	Code	Criteria	Value	Units	Actual	Statu	Margi n
1	**Intact**	**3.(a)ii: Value of max. GZ**				**Pass**	
2		*in the range from the greater of*					
3		spec. heel angle	0,0	deg	0,0		
4		angle of equilibrium	-0,5	deg			
5		*to the lesser of*					
6		angle of max. GZ	34,5	deg			
7		first downflooding angle	30,0	deg	30,0		
8		**shall be greater than (>)**	**0,200**	**m**	**0,685**	**Pass**	+242,
9		*Intermediate values*					
10		angle at which this GZ occurs		deg	30,0		
11							
12	**Intact**	**3.(b): Angle of downflooding**				**Pass**	
13		**shall not be less than (>=)**	**15,0**	**deg**	**30,0**	**Pass**	+100,
14							
15	**Intact**	**3.(c): GZ area between limits**				**Pass**	
16		*from the greater of*					
17		spec. heel angle	0,0	deg	0,0		
18		*to the lesser of*					
19		angle of max. GZ	34,5	deg			
20		first downflooding angle	30,0	deg	30,0		
21		lower heel angle	15,0	deg			
22		required GZ area at lower heel angle	4,0107	m.deg			
23		higher heel angle	30,0	deg			
24		required GZ area at higher heel angle	3,1513	m.deg			
25		**shall not be less than (>=)**	**3,1513**	**m.deg**	**13,176**	**Pass**	+318,
26							
27	**Intact**	**3.(d): Initial GMt**				**Pass**	
28		spec. heel angle	0,0	deg			
29		**shall be greater than (>)**	**0,150**	**m**	**2,064**	**Pass**	+127
30							
31	**Intact**	**3.(e): Angle of equilibrium - multiple heeling**				**Pass**	
32		*Pass. crowding arm = nPass M / disp. D cos^n(*					
33		number of passengers: nPass =	100				
34		passenger mass: M =	0,075	tonne			
35		distance from centre line: D =	2,000	m			
36		cosine power: n =	0				
37		*Turn arm: a v^2 / (R g) h cos^n(phi)*					
38		constant: a =	0,45				
39		vessel speed: v =	9,000	kn			
40		turn radius, R, as percentage of Lwl	510,00	%			
41		h = KG - mean draft / 2	1,277	m			
42		cosine power: n =	0				
43		*Wind arm: a P A (h - H) / (g disp.) cos^n(phi)*					
44		constant: a =	1				
45		wind pressure: P =	250,0	Pa			
46		area centroid height (from zero point): h =	1,880	m			
47		total area: A =	86,960	m^2			
48		H = mean draft / 2	0,363	m			
49		cosine power: n =	0				
50		**Criteria: Angle of equilibrium due to the follo**				**Pass**	
51		**Hpc + Ht**	**12,0**	**deg**	**9,7**	**Pass**	+19,1
52		**Hpc + Hw**	**12,0**	**deg**	**12,0**	**Pass**	+0,07
53		*Intermediate values*					
54		Pass. crowding heel arm amplitude (Hpc)		m	0,337		
55		Turning heel arm amplitude (Ht)		m	0,011		
56		Wind heeling heel arm amplitude (Hw)		m	0,075		
57							
58	**Passenger Vessel**	**1.(a): Min. freeboard at equilibrium**				**Not**	
59		the min. freeboard of the	Margin				
60		**shall not be less than (>=)**	**0,200**	**m**		**Not**	
61							

Results View - Criteria

Table 55: Stability Report for Single Tank with 2 Longitudinal Bulkheads

	Code	Criteria	Value	Units	Actual	Statu	Margi n
1	**Intact**	**3.(a)ii: Value of max. GZ**				**Pass**	
2		*in the range from the greater of*					
3		spec. heel angle	0,0	deg	0,0		
4		angle of equilibrium	-0,5	deg			
5		*to the lesser of*					
6		angle of max. GZ	35,5	deg			
7		first downflooding angle	30,8	deg	30,8		
8		**shall be greater than (>)**	**0,200**	**m**	**0,728**	**Pass**	+264,
9		*Intermediate values*					
10		angle at which this GZ occurs		deg	30,8		
11							
12	**Intact**	**3.(b): Angle of downflooding**				**Pass**	
13		**shall not be less than (>=)**	**15,0**	**deg**	**30,8**	**Pass**	+105,
14							
15	**Intact**	**3.(c): GZ area between limits**				**Pass**	
16		*from the greater of*					
17		spec. heel angle	0,0	deg	0,0		
18		*to the lesser of*					
19		angle of max. GZ	35,5	deg			
20		first downflooding angle	30,8	deg	30,8		
21		lower heel angle	15,0	deg			
22		required GZ area at lower heel angle	4,0107	m.deg			
23		higher heel angle	30,0	deg			
24		required GZ area at higher heel angle	3,1513	m.deg			
25		**shall not be less than (>=)**	**3,1513**	**m.deg**	**14,436**	**Pass**	+358,
26							
27	**Intact**	**3.(d): Initial GMt**				**Pass**	
28		spec. heel angle	0,0	deg			
29		**shall be greater than (>)**	**0,150**	**m**	**2,181**	**Pass**	+135
30							
31	**Intact**	**3.(e): Angle of equilibrium - multiple heeling**				**Pass**	
32		*Pass. crowding arm = nPass M / disp. D cos^n(*					
33		number of passengers: nPass =	100				
34		passenger mass: M =	0,075	tonne			
35		distance from centre line: D =	2,000	m			
36		cosine power: n =	0				
37		*Turn arm: a v^2 / (R g) h cos^n(phi)*					
38		constant: a =	0,45				
39		vessel speed: v =	9,000	kn			
40		turn radius, R, as percentage of Lwl	510,00	%			
41		h = KG - mean draft / 2	1,288	m			
42		cosine power: n =	0				
43		*Wind arm: a P A (h - H) / (g disp.) cos^n(phi)*					
44		constant: a =	1				
45		wind pressure: P =	250,0	Pa			
46		area centroid height (from zero point): h =	1,880	m			
47		total area: A =	86,960	m^2			
48		H = mean draft / 2	0,352	m			
49		cosine power: n =	0				
50		**Criteria: Angle of equilibrium due to the follo**				**Pass**	
51		**Hpc + Ht**	**12,0**	**deg**	**9,7**	**Pass**	+19,4
52		**Hpc + Hw**	**12,0**	**deg**	**12,0**	**Pass**	+0,13
53		*Intermediate values*					
54		Pass. crowding heel arm amplitude (Hpc)		m	0,353		
55		Turning heel arm amplitude (Ht)		m	0,011		
56		Wind heeling heel arm amplitude (Hw)		m	0,080		
57							
58	**Passenger Vessel**	**1.(a): Min. freeboard at equilibrium**				**Not**	
59		the min. freeboard of the	Margin				
60		**shall not be less than (>=)**	**0,200**	**m**		**Not**	
61							

Results View - Criteria

Table 56: Stability Report for Single Tank with 3 or more Longitudinal Bulkheads

	Code	Criteria	Value	Units	Actual	Statu	Margin
1	**Intact**	**3.(a)ii: Value of max. GZ**				**Pass**	
2		*in the range from the greater of*					
3		spec. heel angle	0,0	deg	0,0		
4		angle of equilibrium	-0,5	deg			
5		*to the lesser of*					
6		angle of max. GZ	35,5	deg			
7		first downflooding angle	31,0	deg	31,0		
8		**shall be greater than (>)**	**0,200**	**m**	**0,736**	**Pass**	+268,
9		*Intermediate values*					
10		angle at which this GZ occurs		deg	31,0		
11							
12	**Intact**	**3.(b): Angle of downflooding**				**Pass**	
13		**shall not be less than (>=)**	**15,0**	**deg**	**31,0**	**Pass**	+106,
14							
15	**Intact**	**3.(c): GZ area between limits**				**Pass**	
16		*from the greater of*					
17		spec. heel angle	0,0	deg	0,0		
18		*to the lesser of*					
19		angle of max. GZ	35,5	deg			
20		first downflooding angle	31,0	deg	31,0		
21		lower heel angle	15,0	deg			
22		required GZ area at lower heel angle	4,0107	m.deg			
23		higher heel angle	30,0	deg			
24		required GZ area at higher heel angle	3,1513	m.deg			
25		**shall not be less than (>=)**	**3,1513**	**m.deg**	**14,680**	**Pass**	+365,
26							
27	**Intact**	**3.(d): Initial GMt**				**Pass**	
28		spec. heel angle	0,0	deg			
29		**shall be greater than (>)**	**0,150**	**m**	**2,203**	**Pass**	+136
30							
31	**Intact**	**3.(e): Angle of equilibrium - multiple heeling**				**Pass**	
32		*Pass. crowding arm = nPass M / disp. D cos^n(*					
33		number of passengers: nPass =	100				
34		passenger mass: M =	0,075	tonne			
35		distance from centre line: D =	2,000	m			
36		cosine power: n =	0				
37		*Turn arm: a v^2 / (R g) h cos^n(phi)*					
38		constant: a =	0,45				
39		vessel speed: v =	9,000	kn			
40		turn radius, R, as percentage of Lwl	510,00	%			
41		h = KG - mean draft / 2	1,292	m			
42		cosine power: n =	0				
43		*Wind arm: a P A (h - H) / (g disp.) cos^n(phi)*					
44		constant: a =	1				
45		wind pressure: P =	250,0	Pa			
46		area centroid height (from zero point): h =	1,880	m			
47		total area: A =	86,960	m^2			
48		H = mean draft / 2	0,350	m			
49		cosine power: n =	0				
50		**Criteria: Angle of equilibrium due to the follo**				**Pass**	
51		**Hpc + Ht**	**12,0**	**deg**	**9,7**	**Pass**	+19,5
52		**Hpc + Hw**	**12,0**	**deg**	**12,0**	**Pass**	+0,13
53		*Intermediate values*					
54		Pass. crowding heel arm amplitude (Hpc)		m	0,356		
55		Turning heel arm amplitude (Ht)		m	0,011		
56		Wind heeling heel arm amplitude (Hw)		m	0,081		
57							
58	**Passenger Vessel**	**1.(a): Min. freeboard at equilibrium**				**Not**	
59		the min. freeboard of the	Margin				
60		**shall not be less than (>=)**	**0,200**	**m**		**Not**	
61							

Results View - Criteria

J Definitions by the Rules

The definition of a **hazardous zone** is given by IGF section 12.5 as the following as quoted:

12.5 Hazardous area zones
12.5.1 Hazardous area zone 0
This zone includes, but is not limited to the interiors of fuel tanks, any pipework for pressure-relief or other venting systems for fuel tanks, pipes and equipment containing fuel.
12.5.2 Hazardous area zone 1
This zone includes, but is not limited to:

.1 tank connection spaces, fuel storage hold spaces and inter barrier spaces;
.2 fuel preparation room arranged with ventilation according to 13.6;
.3 areas on open deck, or semi-enclosed spaces on deck, within 3 m of any fuel tank outlet,footnote gas or vapour outlet, bunker manifold valve, other fuel valve, fuel pipe flange, fuel preparation room ventilation outlets and fuel tank openings for pressure release provided to permit the flow of small volumes of gas or vapour mixtures caused by thermal variation;
.4 areas on open deck or semi-enclosed spaces on deck, within 1.5 m of fuel preparation room entrances, fuel preparation room ventilation inlets and other openings into zone 1 spaces;
.5 areas on the open deck within spillage coamings surrounding gas bunker manifold valves and 3 m beyond these, up to a height of 2.4 m above the deck;
.6 enclosed or semi-enclosed spaces in which pipes containing fuel are located, e.g. ducts around fuel pipes, semi-enclosed bunkering stations;
.7 the ESD-protected machinery space is considered a non-hazardous area during normal operation, but will require equipment required to operate following detection of gas leakage to be certified as suitable for zone 1;
.8 a space protected by an airlock is considered as non-hazardous area during normal operation, but will require equipment required to operate following loss of differential pressure between the protected space and the hazardous area to be certified as suitable for zone 1;
.9 except for type C tanks, an area within 2.4 m of the outer surface of a fuel containment system where such surface is exposed to the weather.

12.5.3 Hazardous area zone 2
12.5.3.1 This zone includes, but is not limited to areas within 1.5 m surrounding open or semi-enclosed spaces of zone 1.
12.5.3.2 Space containing bolted hatch to tank connection space.