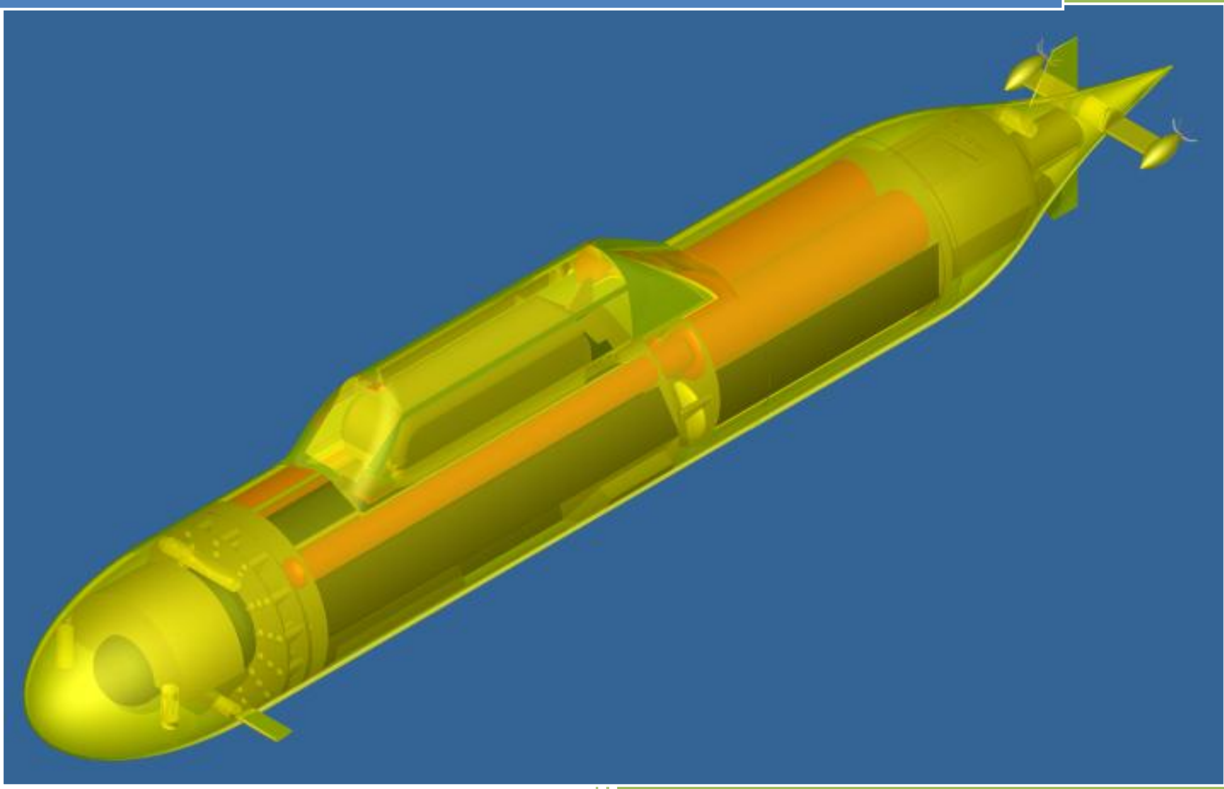
2011

Concept Design of a Commercial Submarine



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10.06.2011

Preface

This report presents the work done by Henrik Carlberg on his master's thesis. It is a continuation of the project thesis performed in the ninth semester at NTNU. The project thesis was a literature study aimed at establishing the types of intervention tasks which would be sensible to perform with a manned submarine. The project thesis also looked at the different types of tooling used in intervention tasks and possible air-independent propulsion systems. The master's thesis is a vessel design study aimed at producing a concept design of an arctic submerged intervention vessel.

The work has been spread out evenly through the semester, though with three distinct phases. The initial phase consisted of a literature study of submarine design books in order to establish the areas where special care is needed when designing a submarine. This was followed by a long period where the size and performance of the various vessel systems and components were determined. The third stage was the development of the 3D model and actual design of several vessel systems.

There were several factors that made the work challenging. Most literature on submarine design is for instance on the design of warships, not civilian vessels. It was also difficult to find realistic sizes and weights of different equipment pieces and machinery based on the calculated performance requirements. Last but not least the performance of most submarines is shrouded in secrecy. This is not unexpected as they are military vessels, of which the exact capabilities always are classified. This left me with few comparison vessels for a comparative performance analysis.

The usefulness of an accurate 3D model of the vessel also became apparent during the process, and was in, agreement with my academic supervisor, afforded more attention and detail than originally intended.

I would like to thank my academic supervisor, Professor Maurice F. White, for his guidance and aid with information gathering. I would also like to thank Jan Erik Faugstadmo and Bjørn Jalving from Kongsberg for their forthcomingness and information on navigation systems.

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10.06.2011, Trondheim

Summary

Oil and gas production in the Arctic poses several new challenges that require new solutions. One such is the use of manned submarines for light intervention tasks. The submarine is completely independent of the surface conditions while adequately submerged, which is their main advantage in the Arctic. This report presents the initial design of an intervention submarine intended for the Shtokman gas condensate field.

The vessel is able to perform structural inspection with an ROV and replace smaller subsea components. The vessel is intended for two week missions to the Shtokman field and is designed for operation at depths up to 537 metres. It carries an array of positioning systems originally developed for the military and offshore industry in order to safely transit within, to and from the field. The vessel is completely independent from the surface and other vessels, and do not need specially adapted infrastructure at the field in order to perform the intended tasks.

The primary power plant is based on the proton exchange membrane (PEM) fuel cells used in the German Type 212 submarines, while the secondary power source is a large battery rack. The battery rack is large enough to enable the vessel to try to perform repairs on-site before an emergency return on battery power if the primary power source is disabled. The primary power plant is fuelled by pure hydrogen and oxygen. The fuel is stored as cryogenic liquids outside the pressure hull. The key performance characteristics and main dimensions of the vessel are presented in table 1.

Table 1 Vessel performance and size summary

Mission length	14 [days]
Crew size	15 [persons]
Power plant	4x Siemens 120 kW PEM fuel cells
Transit speed	6,2 [knots]
Flank speed	8,4 [knots]
Length overall	71,3 [m]
Maximum height	12,7 [m]
Outer hull diameter	9,2 [m]
Maximum payload weight	20 Te
Cargo hold storage area size (LxW)	3x3 [m]
Crane maximum rated load	43 [Te]
Maximum ROV size	2,6x1,5x1,8 [m]
Maximum head current speed on DP	3,4 [m/s]
Maximum beam current speed on DP	3,1 [m/s]
Maximum operational depth	537 [m]
Hull crush depth	1075 [m]

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List of Abbreviations

ABS- American Bureau of Shipping
AIP- Air-Independent Propulsion
AUV- Autonomous Underwater Vehicle
CB- Centre of Buoyancy
CG- Centre of Gravity
CCO- Component Change-Out Tool
COTS- Commercial off the Shelf
DAF- Dynamic Amplification Factor
DC-Direct Current
DEFSTAN- Defence Standard
DP-Dynamic Positioning
DSRV- Deep Sea Rescue Vehicle
FPU- Floating Production Unit
FSE- Free Surface Effect
GPS- Global Positioning System
HP- High Pressure
Hp- Horse Power
HVAC-Heating, Ventilation and Air-Conditioning
IMR- Inspection, Maintenance & Repair
INS- Inertial Navigation System
LBL- Long Base Line
LH_2- Liquid Hydrogen
LO_2- Liquid Oxygen
LP- Low Pressure
MBT- Main Ballast Tank
MESMA- Module d'Energie Sous-Marin Autonome
MRT-Module Replacement tool
PEM-Proton Exchange Membrane
PH-Pressure Hull
PLEM- Pipeline End Module
RDT- Rim Driven Thruster
ROB- Reserve of Buoyancy
ROV- Remotely Operated Vehicle
SBL- Short Base Line
SCM-Subsea Control Module
SFC- Specific Fuel Consumption
SNAME-Society of Naval Architects and Marine Engineers
T&C- Trim & Compensation
USBL- Ultra Short Base Line

List of Symbols

A-Area

A_{33}-added mass coefficient in heave

B-Breadth

BM- Distance between the metacentre and the centre of buoyancy

c-flow speed

C_D-Drag coefficient

$C_{friction}$-friction coefficient

C_P-heat capacity

D,d- Diameter, cabin length

E-Young's modulus

F-force, load

g-gravitational acceleration,

GB- Distance between the centre of buoyancy and the centre of gravity

$\overline{gg'}$-single tank contribution to total FSE

$\overline{GG'}$-total FSE

GM- Distance between the centre of gravity and the metacentre.

H, h- Height

Δh_{form}- Heat of formation

I,i- Electric current, moment of section

k-heat transfer coefficient

K_f-friction to form resistance coefficient

K_P- Propulsion factor

L-Length

L_{HV}- Lower heat value

M, m -Molar mass, bending moment, mass

$\dot{m}$- Mass flow

N,n-"number of" (moles, persons etc.)

P,p- Power, pressure

$\dot{Q}$-heat flow

$\dot{q}$-specific heat flow

R, r- Resistance force, radius

R_N- Reynold's number

S- Surface area

T,t- Temperature, thrust force. thickness, time, depth

Te- Metric tonne

U-Speed, electric voltage

V- Volume, speed

$\dot{V}$- Volume flow

W,w- width, weight,

$\dot{w}$-specific work per time unit

$\dot{W}$- work per time unit, i.e. power.

W-work, heat capacity of the mass flow

Greek letters

∇-Volumetric displacement
Δ-Displacement, differential
η-efficiencies, dimensionless factors
ρ-density
κ- Isentropic exponent
π- 3,14..., Pressure ratio
ε_v- Heat exchanger efficiency
λ- piping friction coefficient
σ-stress
δ-beam deflection
α-trim angle, half apex angle
ν-kinematic viscocity

1 Introduction

The oil and gas industry has turned its attention to the Arctic and the expected undiscovered hydrocarbon deposits there. Operating in the Arctic region will be much more challenging than in the North Sea with lower temperatures and generally harsher conditions. The summer months are best suited for offshore operations conducted by surface vessels. This particular vessel is designed with use on the Shtokman field in mind. The Shtokman field was chosen as it is still in the planning phase, thus simplifying an implementation of submarines in the field maintenance strategy.

Submarines are completely independent of the surface conditions and can therefore operate all year. Various potential offshore tasks were examined in my project thesis. These were evaluated both for practical and economic applicability in a submarine. It was determined that a submarine is best suited for inspection and the replacement of smaller components. A submarine was chosen over an unmanned autonomous underwater vehicle (AUV) because of the great distances involved. A similar thesis was performed by Silje Bordvik(1), her vehicle being an AUV intended for the Snøhvit field. This vehicle had a range of only 145 km, while the Shtokman field is located 550 km offshore.

The object of this thesis was to produce a concept design of an intervention submarine taking the following eight design aspects into account:

- General arrangement and overall hull and system design
- Speed, power and endurance analysis (including propulsion system selection)
- Weight, buoyancy and stability
- General structural design and strength
- Mooring/station keeping. Design of dynamic positioning and anchoring systems.
- Definition, development, and mechanical design of specialized mission work systems. (Such as intervention tooling, pipeline monitoring and repair, handling of ROV & AUV's, remote operated tooling etc.)
- Life support, health, safety and environmental issues.
- Transport to/from surface, emergency evacuation, communications.

While all aspects have been taken into account some have been more deeply explored than others, particularly the electric load and endurance analyses have been given much attention, as have the general structural design of the pressure hull. The cargo handling system and ROV handling systems have also been given a fair deal of attention. A 3D model of the complete vessel was created in Autodesk Inventor in order to develop the general arrangement, hull shape and as a means to perform accurate measurements of equipment locations for the stability and trim calculations. It was also instrument in developing the specialized systems such as the crane and airlock. While very useful, creating an accurate model is rather time consuming. The development of a 3D model was therefore, in agreement with my academic supervisor, given more attention than originally intended.

The increased attention on the modeling was at the expense of smaller systems like life support systems, electronics and communication systems. Similarly the ROV tooling has not been given much attention as these and the ROV can be swapped depending on the mission type. The common denominator is that these systems are more or less commercial off the shelf products (COTS) which do not require much more than power and space margins until the detailed design phase.

2 Initial Design

2.1 Preliminary Project Thesis Findings

As it is an unproven concept the financial risk for investors will be high. Emphasis is therefore put on keeping the initial investment costs as low as possible in order to retain economic realism. The vessel size is a major contributor to the overall cost. Any new product needs to find a niche of its own or replace an existing product.

The findings of the project thesis were that a submarine would be best suited for inspection tasks and replacement of smaller modules and components on the Shtokman field. That conclusion was based on the following considerations:

- The main advantage of a submarine is the independence of the surface conditions, enabling it to operate all year in any weather condition. This is provided that surfacing of the vessel is unnecessary.
- Submarines are more expensive owing to higher quality demands and have less space for cargo than surface vessels of similar displacement.
- The size of the vessel is proportional to the payload volume. A rule of thumb in the initial phase states that the payload volumes, i.e. cargo hold, an airlock for the remotely operated vehicle (ROV) and ROV garage, is roughly 30% of the pressure hull volume. Transportation and installation of large and heavy protection structures and equipment modules will therefore require a large vessel.
- A submarine designed for inspection, maintenance and repair (IMR) can service Shtokman throughout both the lifetime of the vessel and the field. A submarine designed for deployment of large modules and structures should however be redeployed to a field in development once Shtokman is completed rather than begin conducting IMR operations. This is in order to give a better return on the investment in and usage of the larger and more capable vessel. This also adds a degree of uncertainty regarding the future employment of the vessel.
- Subsea installations are designed such that no major maintenance other than inspection is to be carried out for several years, which means that cleaning and inspection will be the dominant tasks.

2.2 Shtokman Field and Operational Pattern

2.2.1 General Field Information

The Shtokman field is a gas condensate field located far north in the Barents Sea. It was discovered by the research vessel *Professor Shtokman*, from which the field name is derived, in 1988. The proposed field design is displayed in Figure 1. Some general information about the field is listed below (2):

- *Field discovered in 1988.*
- *Located 550 km from shore.*
- *Ïnitial geological reserves estimated at 3.9 trillion cu.m.of gas and 56 million tonnes of gas concentrate.*
- *Sea depth is 340 m.*
- *Wave height is up to 27 m.*
- *Annual temperature range from -50˚C to +33˚C.*
- *Presence of icebergs weighing up to 4 million tonnes.*

While the sea depth is no more than 340 metres the vessel is to be able to dive to at least 500 metres. This is done to partly to achieve flexibility in the design and to allow the vessel to service fields with greater depths, but also to demonstrate the potential depths a submarine can reach.

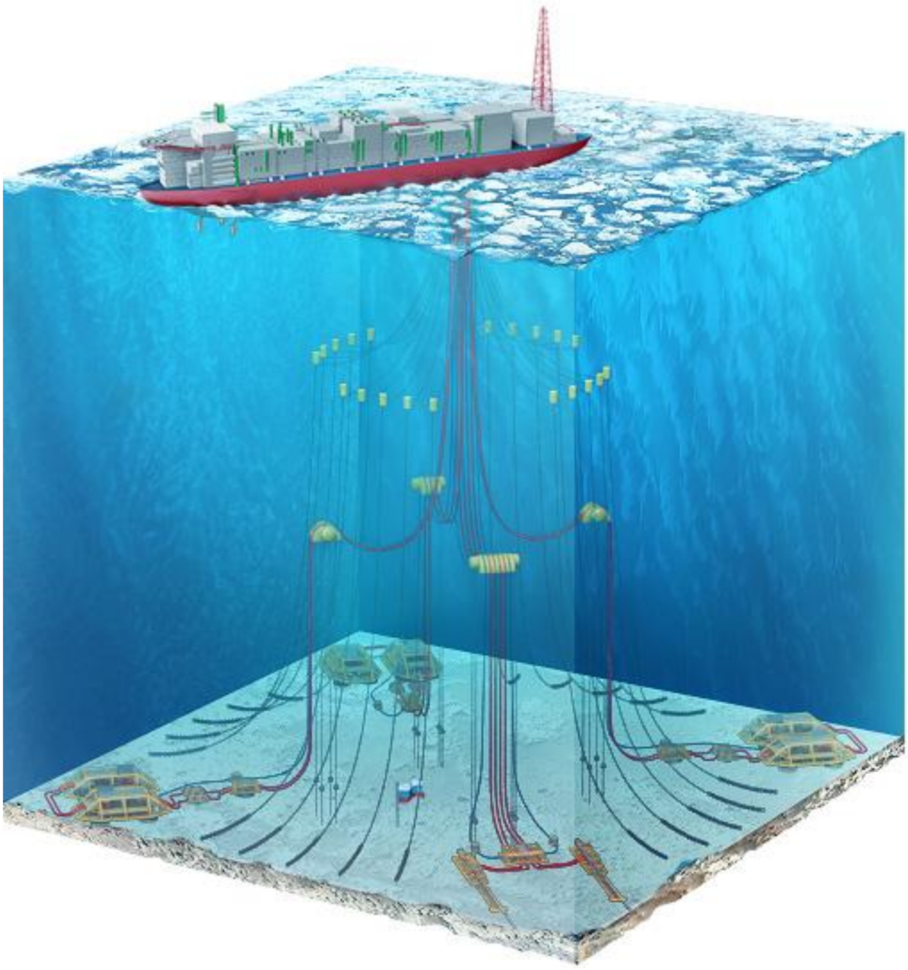

Figure 1 Shtokman floating production unit (FPU) and subsea system concept drawing(2)

2.2.2 Operational Pattern

A summary of the time require for the complete inspection of all subsea components was developed in the project thesis (3). With 14 hour work days the vessel will be perform inspection for 87 days and 12 hours. Including 72 hours for the transit to and from the field this is equivalent to almost 8 two-week missions. The vessel is also assumed to spend one week in port for at least one week between missions for maintenance, keeping the vessel busy with inspection missions for a total of 24 weeks. The task analysis from the project thesis can be found in Table 2. It is deemed likely that the vessel will be put to work the entire year and avoid long periods of time off-hire considering that tasks like inspecting the FPU hull, replacement of defect components and assisting surface vessels with tie-ins and module installations during the summer months are not included in this estimate.

Table 2 Task analysis (3)

Task	Duration	Comments
Structural inspection of well templates	10 hours each, 60 hours for all six	Assumes a general and close inspection, with one hour spent on cleaning each structure
Structural inspection of well manifolds	9 hours each, 27 hours for all three	
Structural inspection of pipepline end modules (PLEM) and pipeline manifold	19 hours each, 30 hours for all three	
Total time spent on structural inspection	117 hours (4 days and 21 hours)	
Umbilical inspection	56 hours 40 minutes each, 283 hours and 20 minutes for all five	Assumes one umbilical to each well template pair and two umbilicals to the export pipeline. The total length is approximated as twice the water depth, 680 metres. Only the top 100 metres need extensive cleaning. Inspection is assumed to take 5 minutes per metre, cleaning 15 minutes per metre
Umbilical cleaning	25 hours each, 125 hours for all five	
Riser inspection	56 hours 40 minutes each,566 hours and 40 minutes for all ten	Assumes two risers to each well template pair and four risers to the export pipeline. The total length is approximated as twice the water depth, 680 metres. Only the top 100 metres need extensive cleaning. Inspection is assumed to take 5 minutes per metre, cleaning 15 minutes per metre
Riser cleaning	25 hours each, 250 hours for all ten	
Total time spent on umbilical and riser cleaning & inspection	**1225 hours**	

2.2.3 Crew

The crew is assumed to have the following composition:

- One ship master in overall command of the vessel
- One executive officer assisting the ship master
- One chief engineer in charge of the machinery and general maintenance.
- Three ROV pilots in charge of ROV piloting and maintenance. These should have varying degrees of experience in order to facilitate experience transfer between the pilots.
- Four multi-disciplinary sailors capable of cooking, performing equipment repairs, standing watch etc.
- Four mission specific crewmembers, for instance specialist engineers.

On such a small vessel it is paramount that the crew is cross-trained so that they can assist in other tasks and help each other. Maintaining separate command and engineering crews would result in a larger vessel than strictly necessary due to the extra berths. So-called “hot-bunking” is not regarded as an option because this will be a civilian vessel with a set daily routine and not a military vessel which must be more or less ready for action at all times. The crew is so small and living in such cramped quarters that the organization structure should be relatively flat in order to avoid a build-up of grievances within the crew. All of this is intended to reduce the wear on the crew so that people will want to be career civilian submariners, not merely adventurers who want a different experience for a few years. In the long term this should yield a steady stream of qualified personnel.

2.3 Design requirements

The space required in the cargo hold is estimated by the size required to fit two choke bridge modules. The reason for being able to fit two modules is that the hold must accommodate the faulty module as well as the replacement module during the replacement operation. A margin is added to allow for slightly larger equipment pieces. The three week patrol length was chosen so that the vessel is to be able to perform a significant amount of work during a single deployment, but the large total fuel consumption of a three week mission necessitated a reduction of the mission length to two weeks. The transit time to the field requires a somewhat high speed of 8,2 knots, however the initial estimates of the required propulsion power indicate that this is not an unreasonable speed. The usual day is expected to consist of a 14 hour work period followed by a 10 hour rest and maintenance period. Table 3 summarizes the basic vessel operational requirements.

Table 3 Basic operational requirements

Cargo hold dimensions (LxWxH)	6x3x4 [m]
Cargo hold storage area (LxW)	3x3 [m]
Maximum crew size	15 (11 regular and 4 spare berths)
Mission length	14 days
Distance to field	550 [km]
Transit time	36 [hours]
Transit speed	8,2 [knots]

2.4 Hull Configuration

There are three main types of submarine hull configurations, namely single hull, double hull and multiple hulls. In single hull configurations the pressure hull forms the outer hull, with the main ballast tanks (MBTs) placed fore and aft of the pressure hull. Double hull vessels have an outer hull which protects the pressure hull, streamlines the vessel shape and provides space for equipment and tankage between the pressure hull and outer hull. In multiple hull configurations two or more pressure hulls are contained within one outer hull. The main advantage of the single hull configuration is an increased speed potential as the hull diameter and surface area is reduced. The pros and cons of the different configurations are listed in Table 4.

Table 4 Hull Concepts

Vessel type	Pros	Cons
Single hull	• Lower resistance than a double hull vessel of comparable displacement. • Good dynamic stability while submerged.	• Poor stability while surfaced due to small waterline area • Almost everything has to be placed within the pressure hull where space is at a premium
Double hull	• Better surface stability due to larger surfaced waterline area • The space between hulls can be utilized by machinery that can sustain full water pressure, MBTs and other tankage. • Higher buoyancy reserve than a single hull vessel.	• Minimum distances between the inner and outer hull due to manufacturing constraints can lead to an unreasonably large external hull volume in submarines with a small pressure hull (less than 500 tonnes pressure hull displacement).
Multiple-hull	• The vessel will have better surfaced stability if the hulls are arranged catamaran-style. • The hulls can be autonomous from each other and provide safe zones in case one hull is breached.	• Using several pressure hulls will require more construction material than a single pressure hull with the same displacement. • Dividing the displacement between two hulls also reduces their size. Small hulls can be more difficult both to build and maintain which increases overall cost.

2.5 Cargo hold and Cargo Handling

The submarine is not to carry large modules, the intention is to carry extra equipment such as specialized tools and smaller components such as subsea control modules (SCMs) and choke bridge modules. There are two main types of crane that can be used to deploy the payload; a boom crane or a gantry crane.

2.5.1 Boom Crane

When using a boom crane some sort of docking station will be necessary to counteract the heeling moment generated by the load at the end of the boom. This docking station can however also be used to supply the vessel with power and perhaps other consumables such as fuel. Once docked cargo can be offloaded much like cargo is offloaded from a truck. The docking station would have to be fairly close to the target area, as even the largest cranes cannot reach much further than 50 metres.

2.5.2 Gantry Crane

When using a gantry crane the stability issue is taken care of by handling the load so that it does not create a heeling moment, or if so, no more than can be handled by the vessel. The main strength of this approach is that no specialized infrastructure is needed. An acoustic dynamic positioning system must be used when positioning the vessel. Acoustic positioning systems can attain a high degree of accuracy (10 cm), so it is merely a question of current speed and thruster power. An integrated gantry crane also requires hatches in the cargo hold floor, otherwise the system would not be able to lower the cargo. This space is not available for cargo storage during transit. The use of a gantry crane also reduces the possible rigging height for cargo, so that only modules and components that can be deployed by a component change-out tool (CCO) or a module replacement tool (MRT) can be replaced. The pros and cons are listed in Table 5.

Table 5 Crane concept evaluation

Crane Type	Pros	Cons
Boom crane	• A high degree of accuracy when delivering the payload. • The docking station can be used to supply power, replenish the batteries, fuel and other consumables. • The cargo hold can be more efficiently used as there is no need for hatches in the floor as with a gantry crane.	• A docking station must be present near each module and will likely be a considerable investment. • There are dangers inherent in manoeuvring a rather large submarine close to subsea structures.
Gantry crane	• No need to invest much in extra infrastructure subsea as the vessel will hover over the target area using DP. • There is no need to manoeuvre the vessel in close proximity to any subsea structure.	• Increases the space used by the cargo hold as floor hatches have to be used when deploying payload. This space is unlikely to be available for storage during transit. • Station keeping will only be possible up to a certain current speed.

3 Initial Estimates

The first step in the design process once the mission has been established is to estimate some key characteristics such as displacement, resistance and ballast tank volume. A rule of thumb from the book by Burcher & Rydill (4) states that the payload volume is roughly 30% of the volume inside the pressure hull. Although their book deals with military submarines, one must take into account that the torpedo and missile launch and storage facilities are not all that different in principle from the ROV launch facility and cargo hold found in this concept. The 30% assumption is therefore regarded as valid. The payload volume is taken as the cargo hold volume and the volume of the airlock and ROV maintenance compartment. A 20% margin is included in the payload volume estimates. On this assumption an estimate of the sizing of the MBTs, trim tanks, displacement, propulsion power and various other things can be made. These estimates are crucial when establishing the general shape and layout of the vessel. The formulas from Burcher & Rydill (4) and the results are summarized in Table 6.

In most calculation I have deemed it prudent to err on the side of caution. While it is not desirable to have too much power or space available, any surplus is relatively easy to shave off. Not having enough power warrants a larger and heavier power plant, which again likely requires a larger vessel which again requires more power, and so a vicious circle begins. All calculations are therefore relatively conservative at this stage.

The utility factor compensates for residual air or water in the ballast tanks, as they cannot be counted on to be completely emptied or filled. It is kept as recommended by Burcher & Rydill.

The buoyancy reserve outside the pressure hull is kept high because of the relatively small size of the vessel as well as to ensure enough buoyancy in case of hull breach.

The free flood margin ensures that the free flooding volumes that will be present in the bow, at the stern, around thrusters etc are accounted for.

The propulsion factor is a somewhat mysterious factor introduced by Burcher & Rydill. It is said to be 20 for an ideally shaped submarine, but that if the vessel shape is expected to differ much from the ideal shape the propulsion factor can be slightly adjusted. However, as the definition of “slightly” is not stated, 20 is used in this case. In the end the propulsion power required seems quite low, but one must not forget that most of the resistance of a surface vessel is wave-resistance; a resistance component a submerged submarine is not subject to. Additionally the speed is relatively low at a mere 8,2 knots. A more thorough resistance calculation is found in section 8.

Table 6 Initial estimates as per the Burcher & Rydill method

Parameter	Value	Formula/Symbol
Payload volume	134,4 [m³]	$Payload\ Vol = 1{,}2 \cdot (V_{hold} + V_{airlock} + V_{workshop})$
Payload/pressure hull volume fraction	0,3 [-]	-
Internal pressure hull volume	448,1 [m³]	$PH\ Vol_{int} = \frac{Payload\ Vol}{0{,}3}$
External pressure hull volume	515,36 [m³]	$PH\ Vol_{ext} = PH\ Vol_{int} \cdot 1{,}15$
Utility factor	0,95 [-]	$\eta_{utility}$
Stores consumption per capita	2,5 [kg]	S_{con}
Weight of stores	787,5 [kg]	$W_{stores} = S_{con} \cdot 21\ days \cdot 15\ crewmembers$
Trim & compensating tank volume	5,41 [m³]	$T\&\ C\ Vol = \left(\frac{PH\ Vol_{int} \cdot (\rho_{max} - \rho_{min})}{\rho_{sw}} + \frac{W_{stores}}{\rho_{sw}}\right) \cdot \frac{1}{\eta_{utility}}$
Submerged displacement	528,14 [Te]	$\Delta_{\mathrm{submerged}} = \mathrm{PH\ VOL_{ext}} \cdot \rho_{sw}$
Reserve of buoyancy	0,25 [-]	ROB
Main ballast tank volume	135,59 [m³]	$MBT\ Vol = \frac{PH\ Vol \cdot ROB}{\eta_{utility}}$
Free flood margin	1,25 [-]	η_{FF}
Form volume	813,56 [m³]	$Form\ Vol = (PH\ Vol + MBT\ Vol) \cdot \eta_{FF}$
Propulsion factor	20	K_P
Hull and propulsor efficiency	0,6 [-]	$\eta_0 \cdot \eta_H$
Transmission efficiency	0,98 [-]	η_S
Propulsion power	96,4 [kW]	$P_{eff} = K_p \cdot (Form\ Vol)^{0{,}64} \cdot U_{transit}^{2{,}9}$
Propulsion motor power	164 [kW]	$P_{motor\ initial} = \frac{P_{eff}}{\eta_0 \cdot \eta_H \cdot \eta_S}$
Stationkeeping	197 [kW]	$P_{station} = 1{,}2 \cdot P_{motor\ initial}$
Hotel load	61 [kW]	$P_{hotel} = 0{,}075 \cdot PH\ Vol$

Burcher & Rydill recommends hull and propulsor efficiencies of 0,6 for a twin propeller vessel and 0,75 for a single propeller vessel. From my experience[1] these values seem reasonable. Azimuting thrusters or azipods are likely to be used in this concept. Such pods can be streamlined in order to achieve high propulsor efficiency, and their position is away from the hull, limiting the flow disturbance from the hull. The combined efficiency is however assumed to be 0,6 in order to ensure conservatism. Further, the power required for station keeping was estimated by adding a 20% margin to the propulsion power.

The hotel load estimate covers heating, ventilation and air-conditioning (HVAC) systems as well as auxiliary machinery; it is however unclear whether the coefficient yields an answer in kW or W. Due to the low figure kW is assumed. As it is very low verification is needed in the form of an electric load analysis.

3.1 Pressure Hull Sizing

From an iterative process with the approximate internal pressure hull volume is found to be close to 500m^3. This figure will be used to establish the approximate length of the cylindrical part of the main pressure hull (see Figure 3). A single deck configuration has been chosen as more would demand a very large diameter compared to the length, while a no deck configuration leads to a very narrow vessel. This is problematic considering the width required by the ROV workshop and cargo hold. The two different configurations are demonstrated in Figure 2.

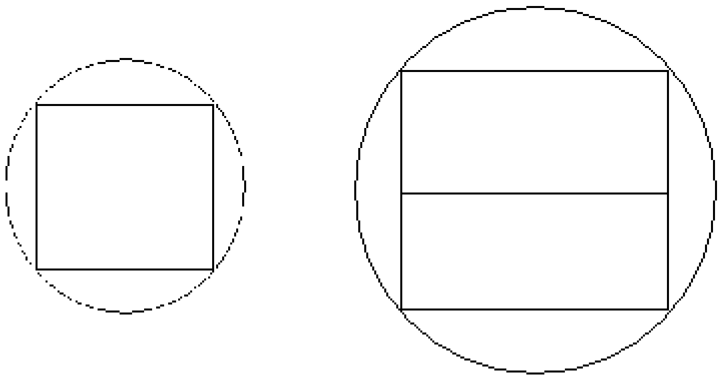

Figure 2 No deck and single deck configurations

The requirement is of course that each deck must be high enough to fit a grown man and then some. The deck is in this case 1,9 metres high and with a deck thickness of 0,1 metres. 0,3 metres of the radius is assumed used by stiffeners, insulation, piping and wiring etc. A space margin was added to ensure a more or less reasonable deck area with full deck height. After some iterations with different space margins an internal pressure hull diameter of 5 metres was deemed reasonable. The pressure hull sizing calculations are summarized in Table 7.

[1] Experience from courses at NTNU with supposedly realistic examples.

Table 7 Pressure hull sizing

Parameter	Value	Formula/Symbol
Internal pressure hull volume	500 [m^3]	PH_{iVOL}
No. Of decks	2 [-]	n_{deck}
Internal deck height	1,9 [m]	h_{deck}
Deck thickness	0,1 [m]	t_{deck}
Stiffeners, insulation and piping	0,25 [m]	h_{stiff}
Space margin	0,6 [m]	h_{marg}
Pressure hull inner diameter	5 [m]	$d_{iPH} = 2h_{stiff+}n_{deck}h_{deck} + t_{deck} + h_{marg}$
Pressure hull length (cylindrical section only)	25 [m]	$L_{PH} = \frac{4 \cdot PH_{VOL}}{\pi \cdot d_{iPH}{}^2}$
Pressure hull interior surface area	439,25 [m^2]	$A_{iPH} = \pi\left(\frac{d_{iPH}{}^2}{2} + L_{PH}d_{iPH}\right)$
Width of deck with full height	2,41 [m]	$W_{fdeck} \approx \sqrt{\left(\frac{d_{iPH} - h_{stiff} - h_{marg}}{2}\right)^2 - h_{deck}{}^2}$
Deck area with full height	120,5 [m^2]	$A_{fdeck} \approx L_{PH} \cdot W_{fdeck}$

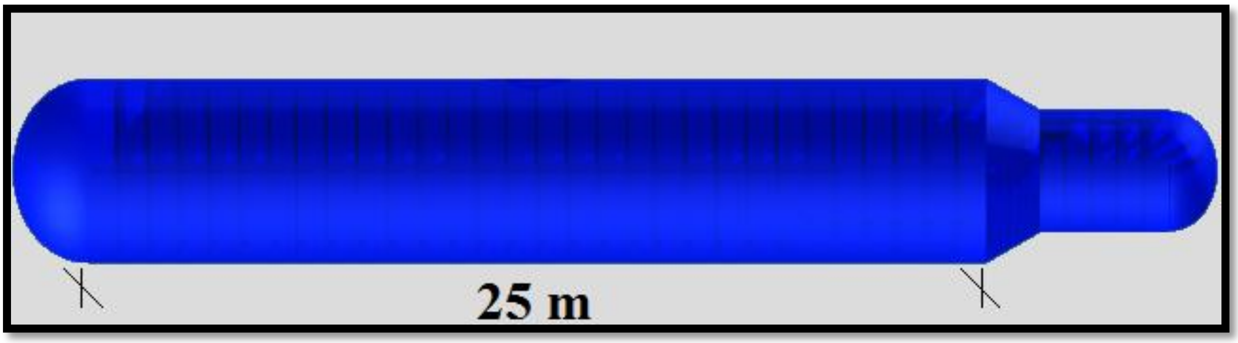

Figure 3 Pressure hull cylindrical section length

3.2 Power Plant

In the project thesis several air independent propulsion systems were investigated, namely fuel cells, closed cycle diesel engines, the MESMA steam turbine and stirling engines. The end goal is to make this vessel completely independent of the surface, hence an all-AIP power plant is envisioned.

3.2.1 AIP Technologies

The different power plant alternatives were investigated in the preliminary project (3), but a brief summary of the systems will be given in this section.

3.2.1.1 MESMA(5)

The MESMA steam turbine system is large, heavy and inefficient. The relatively high power required during operation of this vessel means that more than one MESMA (Module d'Energie Sous-Marin Autonome) module would be required. One such module is about 8 metres long, weighs 305 Te and delivers 200 kW. The MESMA system is deemed unfit for this vessel as it is too heavy compared to the power generated.

3.2.1.2 Stirling Engine (6)

The stirling engine is a proven design, however the models currently in use in AIP systems are fairly small (60-75 kW). Even though the stirling engine is compact the vessel would still require several engines, each with a separate generator. Hence the stirling engine is also deemed unfit.

3.2.1.3 Fuel Cell

The fuel cell is very compact, has a DC output that can be fed directly to the batteries and is an up-and-coming technology where improvements are expected. Current fuel cells have a rated efficiency at peak load of 59% (ref. Appendix VI: Siemens PEM Fuel Cell Information Leaflet), with the part load efficiencies being even higher. The main challenge with a fuel cell is how to store the hydrogen. Hydrogen gas has a very low density, while liquid hydrogen requires very low temperatures. A common solution is to use methanol or high grade diesel oil. These are denser than hydrogen gas and are easier to store and handle than liquid hydrogen. They do however also require a reformer in order to separate the hydrogen from the carbon.

A bi-product of this is CO_2 which has to be captured and disposed of. Adding a reformer will also draw power, lowering the overall efficiency of the power plant.

3.2.1.4 Closed Cycle Diesel Engine

The diesel engine is a tried and tested technology, and no groundbreaking advances are expected. There is no difference between a closed cycle diesel engine and a normal diesel engine, the key in achieving air-independence is as always to store oxygen in the vessel and use this oxygen while submerged. The oxygen is diluted with an inert gas before combustion, and after combustion the inert gas is recycled while the CO_2 is released to the sea. The efficiency at optimal load (about 85% MCR) cannot be expected to be much higher than 50% in a diesel engine, while the part load efficiency is lower(7).

3.2.2 Chosen System

The Siemens 120 kW proton exchange membrane fuel cell is chosen as it is the only AIP system that can produce the required amount of power without drastically increasing the size of the vessel. It is also used in the AIP system of the German Type 212 and Type 214 attack submarines, so some experience is assumed to have been gained and improvements on the original design are assumed to be implemented. There is however little that is available on the performance characteristics of the fuel cell. *Appendix VI: Siemens PEM Fuel Cell Information Leaflet* contains all of the information publicly available on the Siemens fuel cells. In this chapter the fuel cell will therefore be “reverse engineered” in order to estimate key characteristics such as fuel consumption. The results of the reverse engineering can be found in Table 8.

In the publicly available material the only clue to the fuel consumption of the fuel cell is that the efficiency at peak load is 58% of the lower heat value of hydrogen, in other words the formation enthalpy of water vapour. This follows from the basic reaction equation.

$$H_2 + \frac{1}{2}O_2 \rightarrow H_2O_{(g)} + 242kJ/mol$$

The lower heat value can be calculated from the formation enthalpy and molar mass.

$$L_{hv} = \frac{\Delta h_{form}}{M}$$

The H_2 consumption can then be calculated by rearranging the equation for the fuel cell power output. The O_2 consumption can be found through evaluation of the reaction equation, the molar mass and mole ratio of O_2 and H_2.

$$\dot{m}_{H_2} = \frac{P_{FC}}{\eta_{FC} \cdot L_{hv\,H_2}}, \dot{m}_{O_2} = \dot{m}_{H_2} \cdot \frac{n_{O_2}}{n_{H_2}} \cdot \frac{M_{O_2}}{M_{H_2}}$$

The heat generated by the fuel cells must be the remaining energy released by combining H_2 and O_2, and if not removed by the cooling system the temperature would rise at a steady pace. All other potential losses are neglected.

$$L_{hv} = \dot{q}_{FC} + \dot{w}_{FC}$$

$$\Longrightarrow \dot{q}_{FC} = L_{hv} - \dot{w}_{FC} = \frac{\dot{w}_{FC}}{\eta_{FC}} - \dot{w}_{FC} = \dot{w}_{FC}\left(\frac{1}{\eta_{FC}} - 1\right)$$

$$\Rightarrow \frac{\dot{Q}_{FC}}{\dot{W}_{FC}} = \frac{(1 - \eta_{FC})}{\eta_{FC}}$$

Table 8 Fuel cell calculation summary

Water vapor formation enthalpy	242 [kJ/mol]
O_2 molecular mass	32 [g/mol]
H_2 molecular mass	2 [g/mol]
Liquid O_2 (LO_2) density	1141 [kg/m^3]
Liquid H_2 (LH_2) density	70 [kg/m^3]
H_2 L_{HV}	121000 [kJ/kg]
Siemens PEM FC efficiency of H_2 L_{HV} at rated load	0,58 [-]
Rated load	120 [kW]
H_2 consumption per FC at 120kW	0,002 [kg/s]
H_2 SFC	0,051 [kg/kWh]
O_2 consumption per FC at 120 kW	0,014 [kg/s]
O_2 SFC	0,41 [kg/kWh]
H_2 heat of vaporization	0,904 (452) [kJ/mol (kJ/kg)]
Required heat to vaporize LH_2 for fuel per kW load	6,4 [W/kW]
O_2 heat of vaporization	6,820 (213,125) [kJ/mol (kJ/kg)]
Required heat to vaporize LO_2 for fuel per kW load	24,29 [W/kW]
Amount of water produced per FC at 120 kW	0,015 [kg/s]

A simplified analysis of the heat generated by the fuel cells reveal that even with significant losses the heat from the fuel cells should be enough to maintain the air temperature and evaporate the fuel. As shown in Table 9 the fuel cells generate more than the required amount of heat just covering the hotel load.

Table 9 Thermal power production summary

FC Operating temperature	80˚C
Heat flow per kW electric power	0,7 [kW heat/kW work]
Maximum heat flow	336 [kW]
Heat flow from "hotel load"	64,4 [kW]
Compartment heating & fuel re-gasification heat (ref. Table 28)	22,2 [kW]

The hydrogen and oxygen is to be stored as cryogenic liquids. The liquid is to be kept at a constant pressure by collecting the boil-off gas, using it as fuel. This eliminates the need for a fuel reformer, but the extreme temperatures require high-quality insulation. Please see chapter 4.5.3 for the final fuel tank capacity.

3.3 ROV and Cargo

3.3.1 Cargo Handling

The cargo hold will be approximately 6x3x4 metres (LxWxH), while the hatch sizes will only be 3x3 metres. When assessing the power required for the lifts one must take lifting speed, weight, added mass and drag effects into account. Since the vessel is submerged the total lifting height will not be that great, allowing for low lifting speeds. In these calculations a lifting speed of 0,25 m/s is assumed. Further, the maximum submerged weight of the load is set to 10 Te. A bar approximation is used for the load when calculating the added mass and drag forces. The "load bar" is assumed to be 3x3x4 metres. When calculating the drag the projected surface area is used with a drag coefficient of 0,5. Table 10 summarizes the calculations. Figure 4 displays the intended cargo hold on- and offloading hatch layout.

Table 10 Cargo handling summary

Characteristic	Value	Formula/Symbol
Gravitational acceleration	9,81 [m/s^2]	g
Lifting speed	0,25 [m/s]	V_{lift}
Component submerged weight	10 [Te]	m_{load}
Load bar width	3 [m]	W_{load}
Load bar length	3 [m]	L_{load}
Added mass	32,8 [Te]	$A_{33\,load} = A_{33\,load(2D)} \cdot L_{load} = 1{,}51 \cdot \pi \rho_{sw} \left(\frac{W_{load}}{2}\right)^2 \cdot L_{load}$
Drag forces	147 [kg]	$F_{drag} = \frac{1}{2} \cdot C_D \cdot \rho_{sw} \cdot W_{load} \cdot L_{load} \cdot V_{lift}^2$
Required crane lifting power	105 [kW]	$P_{lift} = \left(F_{drag} + A_{33\,load} \cdot g\right) V_{lift}$

The 2D added mass coefficient was obtained from the hydrodynamics reader used in TMR 4247(8).

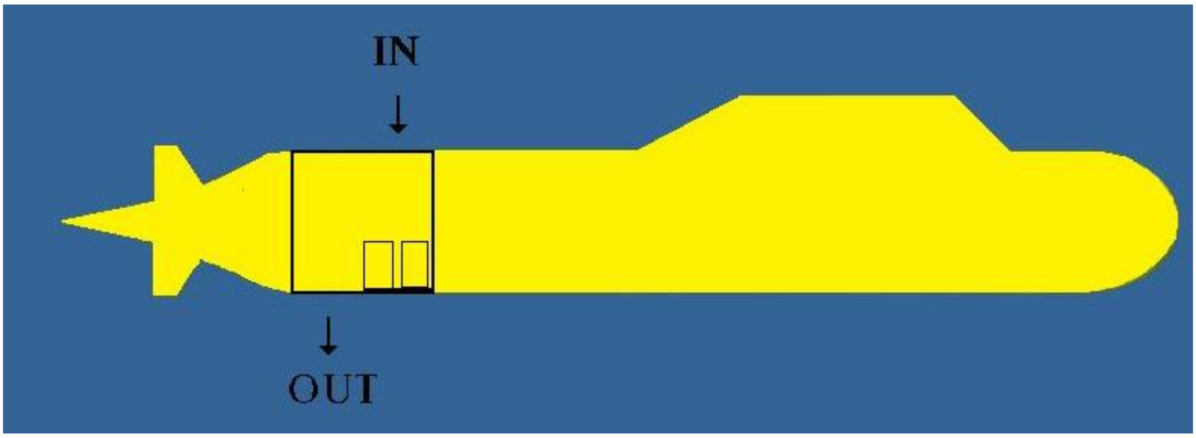

Figure 4 Cargo hold hatch layout

3.3.2 ROV

The vessel must be able to operate an ROV both to inspect structures and risers and to replace faulty equipment. For this the vessel will need an airlock and a workshop large enough to accommodate and service an ROV. In the preliminary project work it was determined that the tool with the highest power demand is the torque tool. These can require as much as 8 kW. As for the ROV itself, there are many different types of ROVs, most of which are custom tailored. The maximum size of the ROV is set to units similar to the Magnum Plus heavy work class ROV developed by Oceaneering (9). This is mainly due to the large power required by the other heavy work class ROVs offered by Oceaneering (330 Hp, about 50% more than the estimated power required for propulsion of the submarine itself).

The submarine is to have a workshop used to service the ROV before and during the deployment. This is also where the standard tooling package is located. The ROV may carry a tool basket to perform some tool changes without returning to the submarine, but it cannot exceed the "maximum box" size of the Magnum Plus. If larger tools are required these must be stored in and deployed from the cargo hold. The minimum required space for the workshop has been estimated by assuming a minimum clearance of 2 metres on each side and half a metre clearance to the deck and ceiling. In the volume estimate a 20% margin is added to compensate for internal stiffeners.

The airlock must be treated like a separate hull section able to withstand the external pressure. It must also contain a framework which will aid the launch and recovery of the ROV. To accommodate this 60 cm is added to the minimum internal diameter. Lengthwise the lock is set to be one metre longer than the design ROV. This is simply to add some flexibility in terms of ROV size and handling. This margin is also necessary to fit the internal airlock door and umbilical reel. The airlock, ROV and workshop requirements are summarized in Table 11.

Table 11 Size requirements related to the ROV

Attribute		
ROV dimensions	2,6x1,5x1,8 [m]	L_{ROV} x W_{ROV} x H_{ROV}
ROV power requirement	170 [Hp (126 kW [2])]	
Workshop minimum size	4,6x3,5x2,8 [m]	L_{WORK} x W_{WORK} x H_{WORK}
Workshop volume	54,1 [m^3]	$V_{work} = 1{,}2 \cdot (L_{work} \cdot W_{work} \cdot H_{work})$
Airlock minimum diameter	2,7 [m]	$D_{lock,min} = \sqrt{W_{ROV}{}^2 + H_{ROV}{}^2}$
Airlock dimensions	3,6x3,3 [m]	L_{lock}xD_{lock}
Airlock section volume	30,3 [m^3]	$V_{lock} = \frac{D_{lock}{}^2}{4} \cdot \pi \cdot L_{lock}$

3.3.3 Electric Load Estimate by Comparison

The first rough electric load analysis was based on a comparative analysis of somewhat similar vessels (10)(11)(12). The main problem was of course that most submarines are military submarines, details of which are hard to come by. The most accessible material is student design reports from Virginia Tech. Several of these originate from the annual Lisnyk design contest held by the Society for Naval Architects and Marine Engineers (SNAME), and are therefore believed to be fairly realistic. By comparing the different reports a best guess estimate of the power required for HVAC, seawater and freshwater systems, ship control and auxiliary machinery was made. The end result was somewhat larger than the estimate for the hotel load given by Burcher & Rydill, however the difference was only on the order of 20 kW. The power required for propulsion is of course dependent on hull size and shape, the Burcher & Rydill estimates were therefore used rather than the comparison vessels.

This is done to enable the vessel to remain in position even in strong currents. The power required for ROV and crane operation was calculated in the previous chapter. From this information the power needed in a few rudimentary load profiles can be established. These are used to evaluate how many fuel cells are needed. The load profiles are the listed below, and a summary is given in Table 12:

I. Normal transit. Expected duration is 72 hours, 36 hours each way.
II. The maximum normal load, i.e. the load expected when inspecting structures with an ROV. Expected duration is 14 hours each day while at the field
III. Peak load, i.e. the load expected when installing a module in strong current. The duration will be no more than the time required to lift and lower the modules. This is estimated to take no more than 4 hours.
IV. Maintenance load, i.e. the load expected in the rest and maintenance period. The vessel is to carry enough fuel to keep the fuel cells running at capacity during the entire maintenance period. The surplus power can be used to recharge batteries, compress air etc. The time set aside for this heavy duty work is no more than 5 hours, resulting in a 5 hour period well suited for crew rest.

[2] Assuming 1Hp=0,746 kW

Table 12 Initial load profiles

Category	Power [kW]	I	II	III	IV
HVAC	25	X	X	X	X
Seawater systems	2	X	X	X	X
Freshwater systems	20	X	X	X	X
Ship control	20	X	X	X	X
Auxiliary machinery	25	X	X	X	X
ROV	119		X	X	
ROV tool	8		X	X	
Crane	105			X	
Propulsion power	164	X			
Stationkeeping	197		X	X	X
Total load [kW]		**256**	**416**	**521**	**289**

The vessel will need a considerable battery rack in order to cope with emergency loads, for instance a return from the field solely on battery power. Norwegian regulations (13) also state that the vessel must be habitable for no less than 48 hours in case of a grounding/sinking. With that in mind it is obvious that the fuel cells do not need to provide the power necessary for all load profiles. Initially three fuel cells were proposed as this would be sufficient in most load conditions. However the more detailed load analysis revealed that the three fuel cells would not be able to replenish the energy expended by the batteries in the time allotted. I therefore propose 4 fuel cells. This reduces the strain on the batteries in normal conditions and the energy expended by the batteries while increasing the power available for battery recharging. The expected peak load and available fuel cell power are compared in Table 13.

Table 13 Initial maximum load vs available power

Number of 120 kW fuel cells	4 [-]
Overall available fuel cell power	480 [kW]
Maximum load	521 [kW]
Peak load handled by batteries	41 [kW]

With an estimate of the required power and number of fuel cells a more detailed description of the functional requirements of the auxiliary machinery can be made by means of the Siemens fuel cell information brochure (Appendix VI: Siemens PEM Fuel Cell Information Leaflet).

The initial estimates provide the basis for a more detailed design and sizing of the vessel. At this point it is necessary to verify and establish how accurate the initial estimates were. The first order of business is a more comprehensive electric load analysis and generation of an equipment list. The next order of business will be to ensure that the pressure hull is strong enough to operate at the required depth. The hull will be dimensioned according to the American Bureau of Shipping's (ABS) rules for manned submersibles. At this stage a 3d-model of the vessel can be created in conjunction with the preparation of the arrangement drawings. This will ensure that the vessel indeed will fit together. The model will also be used in the resistance calculations.

4 Electric Load Analysis

While a statistical electric load analysis by means of comparison vessels can be quite accurate, it requires a significant database in order to evaluate the deviations and trend lines. In this case the statistical material is not only very thin, it is solely comprised of design studies rather than completed vessels. It is therefore important to verify the power demanded by the on-board equipment. The power demand of any equipment piece is dependent on its capabilities and performance characteristics. The power required by a pump is for instance strongly dependent on flow rate, pressure difference and efficiency. This electric load analysis will therefore in large part be performance based. As the vessel geometry is not determined the early propulsion estimates will still be used.

4.1 Life Support Systems

4.1.1 Freshwater Production System

The one thing a human cannot do without is a daily supply of water. The vessel must therefore either carry a supply of freshwater large enough to cater for the entire trip or be able to produce freshwater from seawater. The daily consumption was estimated from consumption rates given in Kormilitsin(14). The consumption rates are given in Table 14.The ABS rules state similar consumption rates. The freshwater tank is to hold one cubic metre of water, as this is enough for over four days of normal consumption. The emergency return condition requires a three day transit with power usage kept to a minimum. The vessel must therefore replenish freshwater each day in order to carry enough for an emergence return.

Table 14 Daily water consumption (14)

Drinking grade freshwater	6 [litres/person]
Other fresh water (sanitary applications etc)	9 [litres/person]
Dirty water/waste water	65 [litres/person]
Crew size	15 [persons]
Daily drinking water production	90 [litres]
Daily other fresh water production	135 [litres]
Daily waste water production	975 [litres]
Human waste treatment plant	3 [m^3]

It is obvious from the daily water consumption that having some sort of water purification system on board greatly reduces the required tankage for freshwater. The question is whether to use evaporate seawater or use a reverse osmosis system where a high pressure pump forces seawater through a membrane to purify it. The main issue with an evaporation plant is that they usually are quite extensive (Figure 5) and require a considerable amount of heat. A vacuum evaporator could be used if the vessel was much larger and needed a larger supply of water. A reverse osmosis system seems more suitable because the high external hydrostatic pressure reduces the work required by the pump. Such systems are also quite compact.

I suggest using a COTS product like Sea Recovery's Coral Sea model 2800. This unit is able to produce 442 litres per hour, which is by far enough to cover the freshwater demand. This allows the vessel to replenish the water supply quickly. The main advantage of this is that the potentially noisy high pressure pump will not be running for long periods of time.

The power required for the entire unit has been calculated from the circuit breaker requirements for voltage and current listed in the product specification (15).

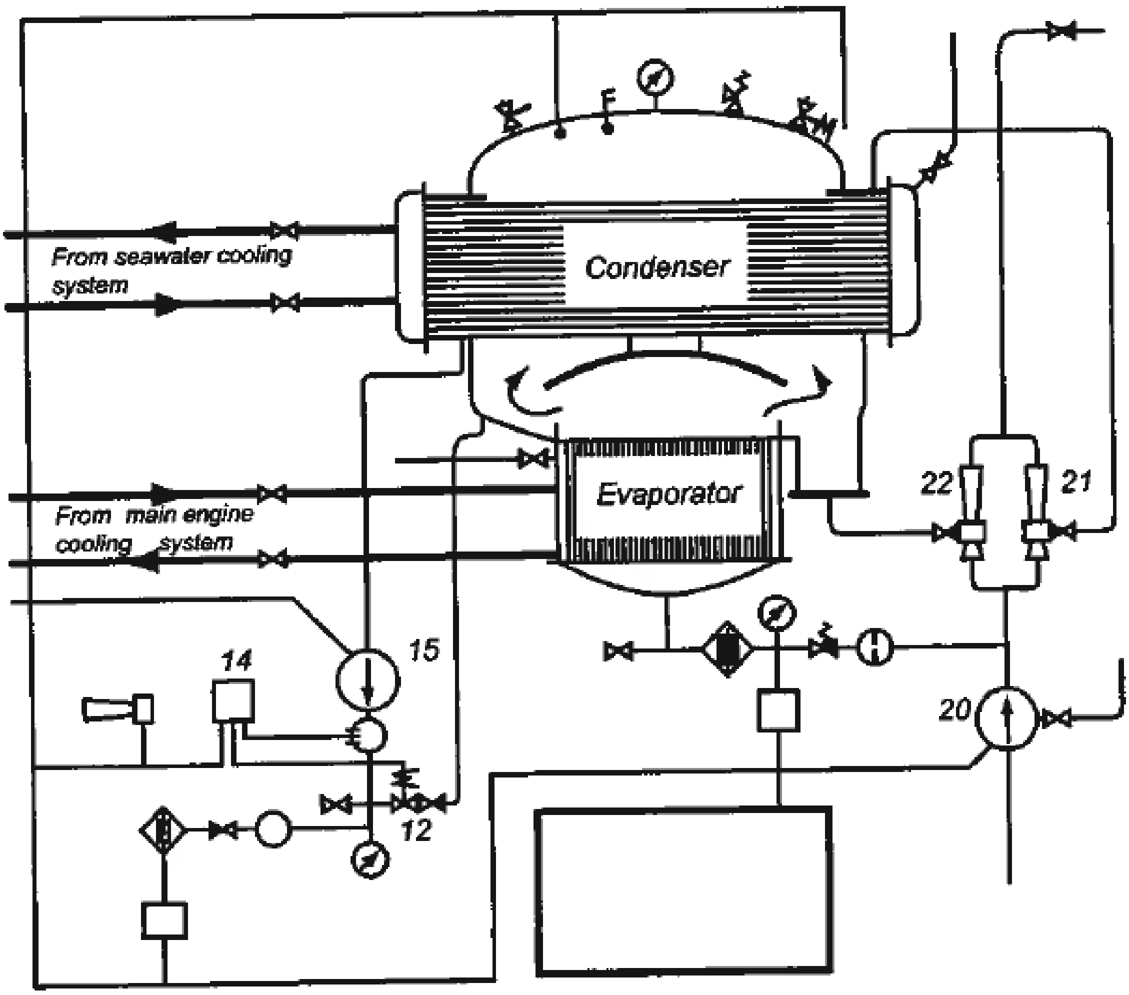

Figure 5 Vacuum evaporator principle sketch (16)

A similar system will purify the human waste produce each day so that it can be discharged overboard without risking environmental damage. Such a plant is assumed to require about 3 m^3 (14), however no further information is given on the inner workings of this system. The main function is to negate toxins, hence the purification process is assumed to rely on chemical treatment with a low power demand per unit purified waste. This, coupled with the fairly low average flow of waste (approx. 0,68 litres/min), is why the purification plant is neglected in the load analysis. The freshwater production system and waste storage plant characteristics are listed in Table 15.

Table 15 Freshwater production summary

Parameter	Value	Unit/formula
Daily water need	225	[litres]
Design demand for daily water production	300	[litres]
Water production margin	1,33	[-]
Freshwater storage	1 (0,7 cold and 0,3 hot)	[m^3]
Waste storage	4	[m^3]
Minimum pressure for the rev. Osmosis machine	56,5	[bar]
Minimum pump head	56,5	[bar]
Maximum power required for overall system	15kW	$P = U \cdot I$
Production capacity	442	[litres/hour]
Time to replenish	40,7	[minutes]

4.1.2 Household Appliances

In addition to producing water, the crew must be able to prepare food. One solution is to use freeze-dried meals similar to military field rations. These meals are rich on energy and nutrients and hardly require any kitchen appliances (some varieties even only require cold water to be added, using an exothermal reaction). The main issue with such rations is that a real variety of taste and texture is difficult to achieve, so one quickly becomes tired of the meals. As a former infantryman the author can certainly attest to the long-term detrimental effect to morale of field rations. In order to avoid issues with crew morale the galley is to be fully equipped with all modern kitchen appliances.

There is a similar issue regarding personal hygiene and the ability to wash clothes. Even though it is not strictly necessary, the ability to feel clean and change into fresh clothes can be quite the morale boost. Showers will therefore be available as well as a washing machine and tumble dryer.

With a relatively high level of comfort recruitment of crew should be easier, not to mention that a happy and well rested crew is likely to be less prone to error. A review of various household appliances (17)indicates that 10 kW should be enough to supply a fully equipped galley as well as a tumble dryer and a washing machine.

4.1.3 Ventilation

Very little is specified in the submarine design literature other than that HVAC system design is taken care of by specialist engineers, so all one must do is to save a margin of space for them. The HVAC system design done in this report is therefore basic.

The ventilation system is considered to be so light and evenly distributed that many parts of it (tubing for instance) need not be accounted with regards to weight and the centre of gravity. The power required by pumps, fans and compressors can however be estimated by the required flow rate and pump work. The equation used to estimate the power is the standard pump equation. For simplicity CO_2 compressor was also regarded as a pump.

$$P = \frac{\dot{V} \cdot \Delta p}{\eta}$$

The air conditioning system must be able to replace the O_2 consumed by the crew as well as filter out the CO_2 and water vapour produced by the crew. The O_2 consumption rate and CO_2 production rate is based on the standard person defined in the ABS rules for classing underwater vehicles, systems and hyperbaric facilities. These are listed in Table 16.

Table 16 Selected ABS standard person characteristics

Item	Quantity	Unit (per person)
O_2 consumption	0,038	[kg/hour] at 1 atm
CO_2 production	0,0523	[kg/hour] at 1 atm
Water vapour produced	1,81	[kg/day]
Heat production (sensible, not latent)	73,61	[W]

The ventilation system is assumed to be a fairly low pressure system where the object is to achieve circulation (flow) rather than pressure. I have therefore assumed that the circulation pump only will need to overcome flow losses which are assumed to be at 0,1 bar at this point. A further simplification used is that the entire circulation and O_2-replacement/CO_2-removal system consists of a large pump, a CO_2 scrubber, a CO_2-compressor (with a storage tank) and a separate O_2 source. To allow for lower efficiencies due to use of several smaller fans the fan efficiency has been set to η=0.6, and a 30% margin has been added to the overall power requirement. The water vapour is removed using a condenser. A CO_2 scrubbing system used in a previous intervention submarine study (10) is displayed in Figure 6.

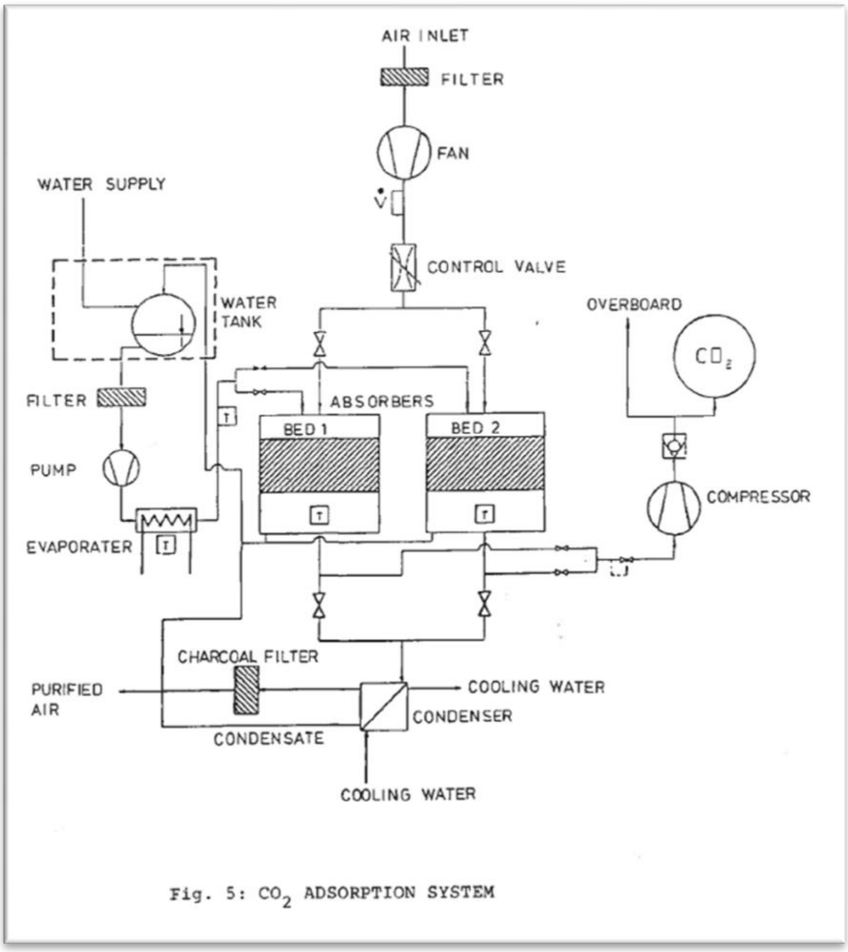

Figure 6 CO2 scrubber system example(10)

The CO_2-production is higher than the O_2-consumption, while the partial pressure of O_2 is significantly higher than that of CO_2. Remembering the universal gas law, the mass flow in the ventilation system must therefore be governed by the CO_2-removal requirement. Using the ideal gas law the following expression for volume flow can be derived:

$$\dot{V}_{air} = \frac{\dot{m}_{CO_2} RT}{M_{CO_2} P_{CO_2}}$$

$\dot{V}$ is the total air volume flow, R the universal gas constant, T is the temperature, M_{CO_2} is the CO_2 molar mass and P_{CO_2} is the CO_2 partial pressure. With the volume flow established the power required by the circulation and O_2-replacement/CO_2-removal system can be found. The scrubbed CO_2 is assumed to be stored in canisters under a maximum pressure of 200 bar. The results are listed in Table 17.

Table 17 Air quality & ventilation

Parameter	Value	Symbol/formula
Number of crewmembers	15 persons	N_{crew}
CO_2 density at 0˚C	1,98 [kg/m^3]	ρ_{CO_2}
Air density at	1,20 [kg/m^3]	ρ_{air}
O_2 atmospheric partial pressure	0,21 [atm]	P_{O_2}
CO_2 atmospheric partial pressure	0,04 [atm]	P_{CO_2}
O_2 consumption per hour	0,57 [kg/hour]	$\dot{m}_{O_2}$
Total O_2 consumption per mission	191,5 [kg]	$m_{total\ O_2}$
CO_2 production per hour	0,79 [kg/hour]	$\dot{m}_{CO_2}$
Total CO_2 produced per mission	263,6 [kg]	$m_{total\ CO_2}$
Pressure loss in pipes	0,01 [MPa]	$\Delta p_{pipe\ loss}$
CO_2 container maximum pressure	20 [MPa]	$P_{CO2\ storage}$
Total air volume flow due to CO_2 (T=20˚C)	4,3x10^{-3} [m^3/s]	$\dot{V}_{air}$
Total air mass flow due to CO_2 (T=20˚C)	35,4x10^{-3} [kg/s]	$\dot{m}_{air}$
Pump efficiency η	0,6 [-]	η_{pump}
CO_2 scrubber supply pump/ventilation flow fan power	70,9 [W]	$P_{fan} = \frac{\dot{V}_{air} \cdot \Delta p_{\text{pipe loss}}}{\eta_{pump}}$
CO_2 compressor power	3674,2 [W]	$P_{comp} = \frac{\dot{m}_{CO_2} \cdot \Delta p_{\text{pipe loss}}}{\rho_{CO_2} \cdot \eta_{pump}}$
Approximate ventilation system power (with 30% margin)	**4,9 [kW]**	$\boldsymbol{P_{HVAC} = 1,3 \cdot \left(P_{fan} + P_{comp}\right)}$

4.1.4 Heating

To estimate the power necessary to keep the temperature in the vessel at the comfortable 20˚C a simplified thin wall heat flow analysis has been used. The pressure hull is also considered to be in full contact with the ambient water. Since there will be other components stored outside the pressure hull as well as an outer streamlining hull this is not true, however the result of the analysis can be considered a worst case. The heat produced by the crew is at this point considered marginal and is neglected. Further the heat transfer coefficient has been estimated from the estimation values presented in the reader in TMR 4222 (16). The suggestions listed indicate that the air-water bulkhead heat transfer coefficient is roughly twice the air-air coefficient. The heat transfer coefficient for an insulated air-air bulkhead is listed as 1 W/m^{2}K, therefore the air-water coefficient is assumed to be 2 W/m^{2}K. The formulas and calculations are summarized in Table 18.

Table 18 Compartment heating

Parameter	Value	Formula/Symbol
Pressure hull approximate surface area	439,25 [m^2]	$A_{iPH} = \pi\left(\frac{d_{iPH}{}^2}{2} + L_{PH} d_{iPH}\right)$
Hull inside temperature	20˚C	T_i
Ambient water temperature	3˚C	T_o
Temperature difference	-17˚C	ΔT
Heat transfer coefficient	2 [W/m^2 K]	k
Heat lost	-14,9 [kW]	$\dot{Q}_{hl} = k \cdot A_{iPH} \cdot \Delta T$
Heat margin	0,3 [-]	η_{hm}
Required heat	19,4 [kW]	$\dot{Q}_{hreq} = -(1+\eta_{hm}) \cdot \dot{Q}_{hl}$

The heat required for heating will either have to be produced by electric ovens/heat pumps or by using the heat in the fuel cell coolant. The fuel cells operate at 80˚C and with an efficiency of 59% when producing 120 kW. As the remaining 41% of the fuel energy therefore must be converted into heat the coolant should be more than able to provide the 19,4 kW necessary. (ref. section 3.2.2)

4.2 Auxiliary Machinery

4.2.1 High Pressure Air and Ballast Handling Systems

High pressure air is a crucial and valuable resource aboard a submarine. The chief usages according to a British Defence Standard (18) are:

- Emptying of MBTs, both during normal operation and in the event of an emergency.
- Compartment blow systems. Such systems are used to pressurize compartments in order to minimize flooding.
- Emergency breathing system.

Most of the minor air consumers are systems for water tank pressurization, human waste disposal, emergency rudder controls etc.

Another possible considerable source of “consumption” is leakage. The Defence Standard 314 (DEFSTAN 314) states that leakages are not expected to cause great losses of air from high pressure (HP) systems. Low pressure (LP) systems where leaks are more difficult to detect are more susceptible to considerable losses. DEFSTAN 314 also states that a loss of 250 m^3 per day is reasonable from the LP system in a nuclear submarine. As far as I have been able to determine this submarine will not have an extensive LP system in addition to be significantly smaller than the British nuclear submarines (Vanguard class displacement: 15680 long tonnes, Astute class displacement: 7400 metric tonnes, Resolution class displacement: 8400 long tonnes, Trafalgar class displacement: 5208 long tonnes) . A daily leak of 20 m^3 is therefore (somewhat arbitrarily) assumed.

At this point the layout of the vessel is not available, hence flow losses are difficult to calculate. A generic flow loss of 0,1 bar is assumed for the airlock drain pump, trim transfer pump and the main deballasting pump.

4.2.2 Airlock Drain Pump

On this vessel emergency breathing is handled by pure oxygen replenishment and CO_2-scrubbing, eliminating the need for storage of breathable air. It does however have an additional possibly significant consumer; airlock cycling at 500 metres. At that depth one cubic metre of air is compressed down to a fiftieth if the compression is isotherm and compressibility is disregarded. This will lead to a high HP air demand if the airlock is to be emptied by HP air only. I therefore suggest that air and water can be transferred between the airlock and the forward trim tank by means of piping and a high pressure water pump. When the airlock is filled with water the air will be run to the trim tank while the water will be run from the trim tank. This way cycling the airlock will not have much impact on the overall state of trim. Preferably this contraption would conserve all the air, however it is prudent to allow for a non-perfect design at this stage. The retained air ratio is therefore set to 0,9.

The airlock drain pump can also function as a bilge pump, removing any water that may leak and collect inside the pressure hull. The drain pump characteristics are summarized in Table 19.

Table 19 Airlock drain pump summary

Parameter	Value	Symbol/formula
Airlock volume	30,3 [m^3]	V_{lock}
Retained air ratio	0,9 [-]	$RAR = \frac{m_{a\ retained}}{m_{a\ tot}}$
Weight of air in lock at 1 bar	36,1 [kg]	$m_{air} = \frac{V_{lock}}{\rho_{air}}$
Weight of air to be replenished	7,2 [kg]	$m_{air\ rep.} = m_{air} \cdot RAR$
Max operating depth	500 [m]	$T_{operating}$
Desired time to empty lock	2400,0 [s]	t_{drain}
Required volumetric flow rate	0,0126 [m^3/s]	$\dot{V}_{drain} = \frac{V_{lock}}{t_{drain}}$
Pump efficiency	0,6 [-]	η_{pump}
Flow loss	0,1 [MPa]	$\Delta p_{pipe\ loss}$
Hydrostatic pressure	50,3 [bar]	$p_{static} = \rho_{water} \cdot g \cdot T_{operating}$
Airlock drain pump power	**106 [kW]**	$\mathbf{P_{drain\ pump}} = \frac{\dot{\mathbf{V}}_{\mathbf{water}} \cdot \left(\mathbf{p_{static}} + \Delta \mathbf{p_{pipe\ loss}}\right)}{\boldsymbol{\eta}_{\mathbf{pump}}}$

4.2.3 Trim Ballast Transfer Pump

A pump dedicated to transferring water between the trim tanks is needed to control and adjust the trim continuously. Flow losses of 0,1 MPa were assumed. As the position of the trim tanks is defined later in the process, the length of the piping is difficult to estimate. A loss of 0,1 bar is regarded as a reasonable first estimate. The specifications are listed in Table 20.

Table 20 Ballast transfer pump specifications

Parameter	Value	Symbol/formula
Estimated trim tank volume	25 [m^3]	$V_{trim\ tanks}$
Transfer time for all trim ballast	50 [s]	$T_{transfer}$
Pump efficiency	0,6[-]	η_{pump}
Flow losses	0,1 [MPa]	$\Delta p_{pipe\ loss}$
Transfer flow rate	0,5 [m^3/s]	$\dot{V}_{transfer\ pump} = \frac{V_{trim\ tanks}}{t_{transfer}}$
Transfer pump power	**8,3 [kW]**	$\mathbf{P_{transfer\ pump} = \frac{\dot{V}_{transfer\ pump} \cdot \Delta p_{pipe\ loss}}{\eta_{pump}}}$

4.2.4 Main Deballasting Pump

The submarine must also be able to deballast some water at depth. This will typically be done prior to retrieving a defect module. At this point the extra weight must either be accounted for by vertical thrust by the DP system or deballasting/trimming. The vessel must also be able to take on and dispose of ballast while changing depth because of changes in buoyancy due to elastic deformation (compression) of the hull caused by the external pressure. The trim tanks are at this point assumed to be hard tanks, that is, they are subject to differential pressure. A later change to soft tanks, which operate without a differential pressure, would then only mean a reduced energy need as there required pump head is greatly reduced. The on-/offloading process will require much higher flow rates than the minute adjustments for hull compression. The deballasting pump is therefore dimensioned according to the on-/offloading condition. The weight difference is assumed to be completely compensated for by ballast.

The main deballasting pump can like the airlock drain pump also function as a bilge pump. The vessel therefore has two bilge pumps, one at either end of the pressure hull. The pump specifications are listed in Table 21.

Table 21 Trim deballasting pump specifications

Parameter	Value	Symbol/formula
Max lifting weight	10 [Te]	$W_{payloas}$
Water volume	10 [m^3]	$V_{payload\ ballast} = \frac{W_{payload}}{\rho_{water}}$
Maximum deballasting time	975,6 [s]	$T_{deballasting}$
Transfer flow rate	0,01 [m^3/s]	$\dot{V}_{deballasting} = \frac{V_{payload\ ballast}}{t_{deballasting}}$
Flow losses	0,1 [MPa]	$\Delta p_{pipe\ loss}$
Hydrostatic pressure	50,3 [bar]	p_{static}
Pump efficiency	0,6 [-]	-
Deballasting pump power	**84 [kW]**	$\mathbf{P_{deballasting\ pump}} = \frac{\dot{\mathbf{V}}_{\mathbf{deballasting}} \cdot (\mathbf{p_{static}} + \mathbf{\Delta p_{pipe\ loss}})}{\boldsymbol{\eta}_{\mathbf{pump}}}$

4.2.5 High Pressure Air Requirements for Deballasting the Main Ballast Tanks

The biggest single consumer of HP air is without doubt the de-ballasting of the main ballast tanks. During a normal surfacing 50% of the ballast water is to be forced out using HP air. The remainder of the ballast water is expelled by a low pressure blower once surfaced. This technique is commonly used by contemporary military submarines, and offers a greatly reduced use of HP air. The blow is assumed to be done at a depth of 10 metres.

The vessel is normally not to surface except when departing the home base. This is mainly due to the simple fact that surfacing is not required for the tasks the vessel is to perform; however it has the added benefit of reducing the compressed air consumption. Even though there isn't set a requirement on how many times the vessel is to be able to surface before refilling the compressed air, the vessel carries enough compressed air to perform a 50% blow almost six times without compromising the ability to perform an emergency blow.

The emergency blow requires that the HP air system can deliver enough air at the maximum operating depth of 500 metres (not to be confused with the hull crush depth) to displace 20% of the ballast. Due to the high compression of air at 500 metres this is only intended to start an initial rise as a complete blow of the ballast tanks would require about nine tonnes of compressed air. As the depth decreases the air will expand and eventually completely empty the ballast tanks.

The HP air system is to be kept at a maximum pressure of 277 bar. During normal operation the pressure is not to drop under 200 bar, as the amount of air stored between 200 bar and 51,3 bar is the minimum amount of air required for an emergency blow. It should be noted that the vessel should not under any circumstance come even close to this limit at it is able to surface almost six times before the pressure drops below 200 bar.

The low pressure systems will be supplied by a separate HP air system. This system is not to supply the deballasting process and is therefore not subject to the 200 bar limit. By separating the two systems the low pressure system can be made much more compact. An added benefit by separating the HP system from the LP system is that a leak in the low pressure system cannot compromise the ability to perform an emergency blow. The deballasting HP air requirements are summarized in Table 22. Air was assumed to behave like an ideal gas, allowing the use of the ideal gas law.

Table 22 High pressure air summary

Parameter	Value	Symbol/formula
Main Ballast Tanks volume	136 [m^3]	V_{MBT}
Hydrostatic pressure at 500 metres	50,3 [bar]	p_{static}
Standard storage pressure	277 [bar]	$p_{hp\ storage}$
Minimum emergency blow pressure	200 [bar]	$p_{emerg.\ min}$
Removed ballast per blow	50 %	η_{blow}
Air density at standard blow depth (2 bar)	2,4 [kg/ m^3]	$\rho_{blow\ air}$
Mass of air needed per surfacing	163,2 [kg]	$m_{surfacing\ air}$
Storage cylinder volume	0,258 [m^3]	$V_{cylinder}$
Mass of air per bottle for normal surfacing (from 277 bar to 200 bar)	25,4 [kg]	$m_{norm.\ surfacing}$
Mass of air per bottle used in everyday consumption (from 277 to 1 bar)	91,2 [kg]	$m_{low\ pressure\ supply}$
Mass of air per bottle for emergency ascent (from 200 bar to 50,3 bar)	49,3 [kg]	$m_{air\ emerg.\ surfacing}$
Ballast expelled by emergency blow (20% emptied at 500m)	27,9 [Te]	$m_{emerg.\ ballast}$
Mass of air needed for one emergency blow	1780[kg]	$m_{emerg.\ air}$
Number of ordinary MBT blows	5,8 [-]	$\frac{m_{surfacing\ air}}{n_{installed} \cdot m_{norm.surfacing}}$
Number of bottles needed for emergency ascent	36,1 [-]	$n_{minimum}$
Number of surfacing bottles installed	37 [-]	$n_{installed}$

4.2.6 Low Pressure Ballast Tank Blower

The low pressure blower is assumed to work as a pump/fan rather than a compressor due to the low pressure difference. As before the pump efficiency is assumed to be low. The vessel will have a fair amount of power available as it is not designed to perform any operations while surfaced. The vessel can therefore complete the deballasting process quickly. A two bar overpressure is assumed to be adequate for an efficient blow. The LP blower specifications are listed in Table 23.

Table 23 Low pressure main ballast tanks blower

Parameter	Value	Symbol/formula
Blower pressure	2 [bar]	$p_{LP\ blower}$
Required blow time	180 [s]	$t_{LP\ blow}$
Volume to be emptied	68,0 [m^3]	$V_{LP\ blow} = \eta_{blow} \cdot V_{MBT}$
Blower efficiency	0,6 [-]	η_{blower}
Flow rate	0,4 [m^3/s]	$\dot{V}_{LP\ blow} = \frac{V_{LP\ blow}}{t_{LP\ blow}}$
Blower power	**125,9 [kW]**	$\mathbf{P_{LP\ blower}} = \frac{\dot{\mathbf{V}}_{\mathbf{LP\ blow}} \cdot \mathbf{p_{LP\ blower}}}{\boldsymbol{\eta}_{\mathbf{blower}}}$

4.2.7 High Pressure Air Compressor

Now that the air consumption from the main HP air consumers has been established, the requirements for the HP air compressor can be formulated. The air lost from leaks, daily tank pressurizing consumption and atmospheric analyzer are scaled down from the DEFSTAN (18) estimates for a nuclear submarine. HP air replenishment requires some time at periscope depth, exposing the vessel to the surface conditions. Keeping in mind that the main advantage of this vessel is the independence from the surface conditions, then minimizing the time spent at periscope depth compressing air is important. This is why the ballast handling is done by electric pumps rather than compressed air. The vessel is therefore also not to surface during normal operation. The air consumption is summarized in Table 24. Again air is regarded as an ideal gas.

Table 24 Air consumption summary

Air Consumption (volumes assume P=1 bar)	Value
Lost air from LP leaks	20 [m^3/day]
Sanitary tank pressurizing	6 [m^3/day]
Freshwater tank pressurizing	12 [m^3/day]
Atmospheric analyser	0,3 [m^3/day]
Airlock cycling	3,03 [m^3/day]
Total consumed air per day w/o airlock cycling	38,3 [m^3/day]
Mass of air consumed w/o airlock cycling	45,5 [m^3/day]
Total consumed air per day w/ airlock cycling	41,3 [m^3/day]
Mass of air consumed w/ airlock cycling	49,1 [kg/day]
Amount of HP bottles consumed per day w/o airlock cycling	0,5 [bottles]
Amount of HP bottles consumed per day w/ airlock cycling	0,54 [bottles]

The vessel is to be able to return to base on batteries only at a reduced speed of 4 knots. To conserve power during such an emergency return the vessel should not have to replenish the HP air. The vessel must therefore at all times have enough HP air stored for a three day transit. I propose carrying enough air to last seven days, with replenishment of the consumed air every fourth day. With this operating profile only four HP air storage bottles are needed for the LP air consumption. To ensure enough air the daily consumption with airlock cycling is assumed. During the design of the HP storage bottle rack it was decided that increasing the storage capacity to 42 bottles was preferred to leaving one bottle slot unused. The location outside the pressure hull and storage rack configuration can be seen on Figure 7. The HP air storage is summarized in Table 25.

Table 25 Pressurized air storage requirements

Category	Number of required HP storage bottles
LP system	4
HP system	37
Total number of HP bottles required	41
Number of installed HP bottles	**42**

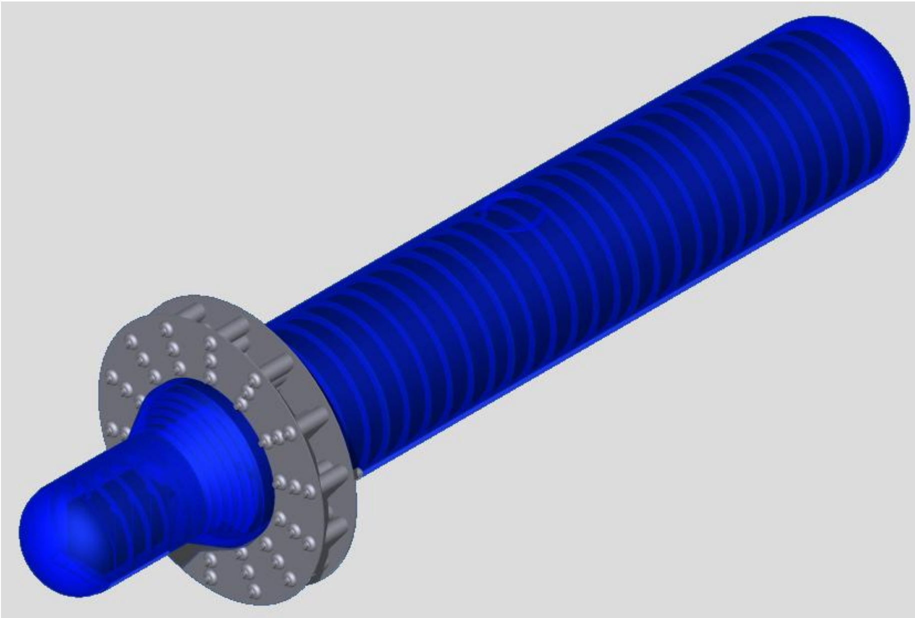

Figure 7 HP air bottle storage rack model

The compressor power has been estimated by basic formulas based on changes in enthalpy. The constant pressure heat capacity of air is assumed to be constant. The formulas and results are summarized in Table 26.

Table 26 Air compressor specifications

Compressor power	Value	Formula/symbol
HP storage bottles for LP consumption	4 [bottles]	n_{LP}
Time available for HP air replenishment	3 [hours]	$t_{HP\ rep.}$
Four day air consumption	167,6 [kg]	$m_{air\ cons.}$
Compressor isentropic efficiency	0,95 [-]	η_{is}
Required pressure ratio	277 [-]	π
Isentropic exponent	1,4 [-]	κ
Air heat capacity	1 [kJ/kg K]	$C_{p\ air}$
Compressor inlet temperature	273˚K	T_{in}
Compressor exit temperature	1419˚K	$T_{out} = \frac{T_{in}\left(\pi^{\frac{\kappa-1}{\kappa}} - 1\right)}{\eta_{is}} + T_{in}$
Compressor mechanical efficiency	0,95 [-]	η_m
Compressor power	**21,95 kW**	$\boldsymbol{P_{comp} = \frac{m_{air\ cons.}}{t_{HP\ rep.} \cdot \eta_m} \cdot C_{p\ air}(T_{out} - T_{in})}$

4.2.8 Fuel Pumps

The fuel pumps must be able to supply all of the fuel cells while running at peak capacity. The fuel storage tanks are intended to be of the constant pressure type where the boil-off gas is siphoned off and used as fuel, however this is disregarded in these calculations. The hydrogen and oxygen is assumed to remain liquid while being pumped. Re-gasification is handled by a small heat exchanger between the pumps and the fuel cells. Flow losses are expected to be low due to the low flow rate and are therefore neglected. As earlier the standard pump power equation was used to calculate the required power, Table 27 summarizes the fuel pump specifications.

Table 27 LO_2 & LH_2 pump specifications

Pump efficiency	0,6 [-]
LH_2 max flow	0,007 [kg/s]
LH_2 max volumetric flow	0,0001 [m^3/s]
Required delivery pressure	2,300 [bar]
LH_2 pump power	0,037 [kW]
LO_2 max flow	0,055 [kg/s]
LO_2 max volumetric flow	0,00005 [m^3/s]
Required delivery pressure	2,600 [bar]
LO_2 pump power	0,021 [kW]
Total fuel pump power	**0,058 [kW]**

4.2.9 Heat exchangers

It was shown in chapter 3.2.2 that the fuel cells produce 0,7 kW heat per every kW of electric power. It is also stated that the operating temperature of the fuel cells is 80˚C. The coolant flow rates have been estimated by choosing temperature ranges and coolant fluids, while the sizes of the heat-exchangers have been estimated from plots over varying heat exchanger efficiencies found in the reader in the course TMR 4222 Machinery Basic Course (ref. Appendix I: Heat Exchanger Efficiency Plots).

4.2.9.1 Flow Rates

The fuel re-gasification and compartment heating circuits are all to use fresh water. This is to reduce corrosion and eliminate the risk of sediments in the high temperature cooling circuits (16). Both of these systems use heat from the fuel cell coolant. The remaining heat in the fuel cell coolant is removed in the primary seawater heat exchanger. The heat exchanger system is sized to handle all of the fuel cells at full capacity, however the heat required for fuel re-gasification and compartment heating will be available even at part load.

The maximum and minimum temperatures of all circuits must be taken into account when setting the desired temperature ranges. The fuel cell operates at 80˚C, hence the coolant cannot exceed this temperature. Similarly rather large flow rates are required to cool water lower than 40˚C to 50˚C in a water-water heat exchanger due to the low achievable temperature differences. The fuel cell coolant is therefore set to maximum and minimum temperatures of 75˚C and 50˚C respectively.

The water used for compartment heating is to be maximum 50˚C and minimum 40˚C. This is to avoid burn hazards related to the heating radiators and reduce the overall fire potential. The amount of heat required to evaporate the fuel is so miniscule that the fuel line is to be run through the compartment heating heat exchanger. The evaporation heat required is simply added to the HVAC heat requirement from chapter 3.1.1.4.

Some of the potable water from the reverse osmosis plant is heated to 70˚C by circulating it through a small heat exchanger on the hot water tank. By doing this the hot water tank heat element electric load is decreased as it no longer has to heat the potable water from 3˚C t 70˚C, indeed if desired it can be kept off and only be used as a reserve in case fuel cell power is lost. The flow rate is very low as the heat exchanger is only to be able to cycle the 300 litre tank in two hours.

All heat exchangers are to reduce the fuel cell coolant to 50˚C. The fuel cell coolant is the hot medium in all heat exchangers. The fuel cell coolant heat flow is the heat generated by the fuel cell, and the seawater heat flow is the residual heat in the fuel cell coolant not utilized by shipboard systems. The heat flow in the remaining heat exchangers is the heat required to achieve the desired temperature increases. The heat exchanger flow rates are summarized in Table 28.

Table 28 Engine cooling system flow specifications

Parameter	Fuel Cell Coolant	Compartment Heating & Fuel Re-gasification	Potable Hotwater	Seawater
Heat flow [kW]	347,6	22,2	11,6	325,4
T_{min}	50˚C	45˚C	3˚C	3˚C
T_{max}	75˚C	50˚C	70˚C	10˚C
C_P [kJ/kg K]	4,2	4,2	4,2	4,2
Required hot circuit flow rate [kg/s]	3,3	0,2	0,1	3,1
Required cold circuit flow rate [kg/s]	-	1,1	0,04	11,2

4.2.9.2 Heat Exchanger Sizes

There are two formulas for heat exchanger efficiency, which one to use is dependent on the product of the mass flow and heat capacity of the fluids in the heat exchanger. As usual efficiency is the actual heat transferred divided by the maximum theoretical heat transfer.

$$\varepsilon_v = \frac{\dot{Q}}{\dot{Q}_{Max}}$$

$$\dot{Q}_{Max} = W \cdot \Delta T = \dot{m} \cdot c_P \cdot (T_{max} - T_{min})$$

The amount of heat transferred in a heat exchanger with a set temperature difference must be restricted by the fluid with the smallest *W;* if the fluid with the largest *W* is subjected to the largest temperature difference the other fluid would have to suffer a larger temperature difference than the maximum temperature difference in order to receive/transmit the heat. It is this paradox that necessitates the use of two formulas. In these calculations the hot and cold flows will be subscripted 1 and 2 respectively. The two efficiency formulas can then be expressed as following.

$$\varepsilon_v = \frac{\dot{m}_1 \cdot c_{P_1} \cdot (T_{1\,in} - T_{1\,out})}{\dot{m}_1 \cdot c_{P_1} \cdot (T_{1\,in} - T_{2\,in})} = \frac{T_{1\,in} - T_{1\,out}}{T_{1\,in} - T_{2\,in}}$$

$$\varepsilon_v = \frac{\dot{m}_2 \cdot c_{P_2} \cdot (T_{2\,in} - T_{2\,out})}{\dot{m}_2 \cdot c_{P_2} \cdot (T_{1\,in} - T_{2\,in})} = \frac{T_{2\,in} - T_{2\,out}}{T_{1\,in} - T_{2\,in}}$$

The overall heat transfer coefficient *k* is the final piece in the puzzle. It is dependent on the material properties of the pipe, incrustation and flow speed. The flow speed should not be less than approximately 0,8 m/s in seawater systems to avoid attachment of marine organisms. Increasing the flow speed increases the transfer coefficient, however at a certain threshold speed the level of corrosion and erosion becomes unacceptable. Experience has shown that little is gained by increasing the flow speed to more than 2 m/s. This is also within the tolerable corrosion/erosion limits of most piping materials (16). 2 m/s is therefore set as the flow speed limit for all piping in the vessel. The heat transfer coefficient *k* is assumed to be 500 W/m^{2}K, a conservative but reasonable estimate based on Figure 8 from the reader in machinery basic course.

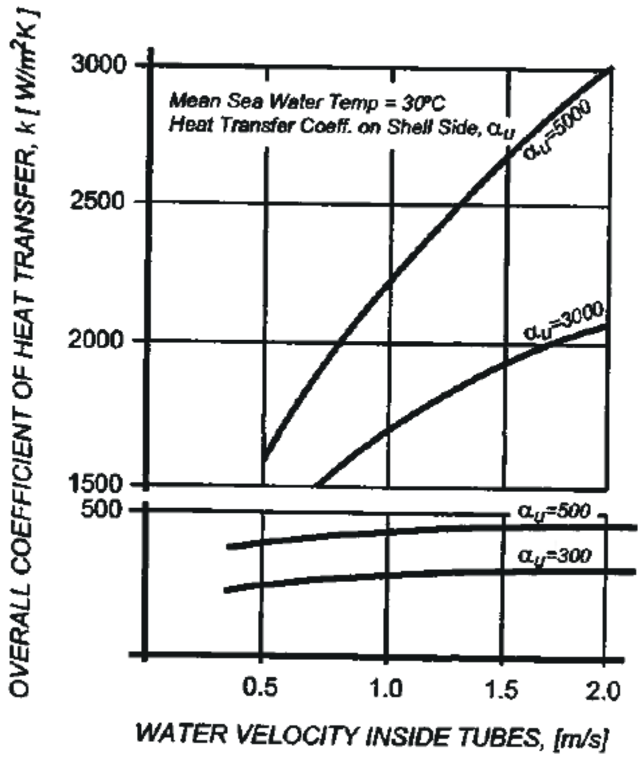

Figure 8 Heat transfer coefficient diagram (16)

The heat exchanger sizes were estimated by means of the ideal cross-flow graph in appendix xx. By calculating the efficiencies of the heat exchangers the NTU ($NTU = \frac{k \cdot A}{W_1}$) value could be read from the graph. The required heat exchange surface area could then easily be calculated as both the heat transfer coefficients and W_1-values are known. The heat exchanger sizes are summarized in Table 29.

Table 29 Heat exchanger specifications

Parameter	Fuel cell coolant/seawater heat exchanger	Drinking water heat exchanger	Compartment heating/ fuel re-gasification heat exchanger
Heat exchanger efficiency [-]	0,347	0,931	0,167
W_1 [kW/K]	13	0,47	0,89 kW/K
W_2 [kW/K]	46,5	0,17	4,435 kW/K
W_{min}/W_{max} [-]	0,28	0,373	0,200
NTU [-]	0,5	3	0,25
Coefficient of heat transfer [W/m^{2}K]	500	500	500
Interior surface area [m^2]	13	2,8	0,443
Heat exchanger main dimensions (LxHxW [m])	1,5x1,5x0,5	Integrated in the water storage unit	0,25x0,42x0,3

The relatively small heat exchange area required for potable water heating allows integration of the heat exchanger and water tanks in a compact water storage unit. The remaining two heat exchangers are assumed to be plate heat exchangers. The main seawater heat exchanger assumed to be divided into six layers and the compartment heating heat exchanger into five layers. This is done in order to estimate the external dimensions while still achieving the necessary surface area. The width of the heat exchangers and layer thickness is quite large in order to compensate for the crudeness of these initial calculations.

4.2.10 Fuel Cell Coolant Circulation Pump& Seawater Pump

These pumps are required for coolant circulation and seawater circulation through their respective circuits. The main task is to maintain flow and compensating for flow losses. The flow losses are dependent on the piping length, diameter, internal roughness and flow speed. The frictional pressure loss can be expressed as following (19):

$$\Delta p = \lambda \cdot \rho \cdot \frac{L}{d} \cdot \frac{c^2}{2}$$

λ is a friction coefficient which depends on the Reynolds number, pipe smoothness, diameter and material of the pipe and flow in question. A completely smooth pipe with a Reynolds number of 10^4 has a friction coefficient of 0,04 (19). This decreases with increasing Reynolds numbers. λ is therefore set to 0,04 for conservatism. L is the length of the pipe, d is the diameter and c is the flow speed. The flow speed was restricted to 2 m/s to avoid erosion (16). The piping length is somewhat arbitrarily chosen based on the size of the fuel cells and pressure hull diameter. As usual the basic pump power formula has been used to estimate the required power. The results are listed in Table 30.

Table 30 Engine cooling system pump specifications

Parameter	Coolant circulation pump	Seawater Pump
Volumetric flow [m^3/s]	0,0033	0,011
Pipe cross sectional area [m^2]	0,0017	0,005
Pipe diameter [m]	0,0461	0,0833
Piping length [m]	20	10
Internal pipe resistance loss [Pa]	34705,9	9607,9
Pump efficiency [-]	0,6	0,6
Pump power [W]	193,01	174,30

4.2.11 Hydraulic Pump & Accumulator

The main hydraulic systems on board are typically used in hatch opening mechanisms, valve control mechanisms and as emergency back-up actuators for electrical actuators such as the ones used by the vessel control surfaces. What they all have in common is that their power requirements are quite difficult to estimate at this point. An example of this is the power required by a control surface actuator. This depends largely on the torque required, which in turn depends on control surface area, angle in relation to the flow and the flow speed. This requires a fairly time consuming analysis, as only the possible flow speed is known at this point. Eventually a hydraulic load analysis should be performed, but for now the vessel is only fitted with a single hydraulic actuator and pump similar to the ones used in the Virginia Tech reports. The suggested configuration is capable of handling power-intensive tasks for a short duration of time, however such a demand is not expected in normal operation. The pump power equation was used to calculate the maximum available power. The characteristics of the hydraulic power components are summarized in Table 31 and Table 32.

Table 31 Hydraulic accumulator specifications

Max pressure	160 [bar]
Total accumulator volume	2 [m^3]
Sustained flow time	10 [minutes]
Sustainable flow rate	0,2 [m^3/min]
Total hydraulic system efficiency	0,8 [-]
Maximum power available	42,7 [kW]
Stored energy	7,1 [kWh]

Table 32 Hydraulic pump specifications

Time to accumulator replenishment	30 [minutes]
Required flow rate	0,067 [m^3/min]
Pump efficiency	0,8 [-]
Hydraulic pump power	22,2 [kW]

4.3 Control and Information Systems

4.3.1 Navigation & Positioning Systems

The vessel must be able to safely navigate to and from the field as well as on the field. This is both to avoid collisions as well as accurate positioning of cargo. The vessel will therefore require accurate positioning systems and collision avoidance sonars. Sonar was developed to detect submerged vessels during the Second World War and have been further refined during the Cold War to such a level that vessels can be detected several kilometres away. Collision avoidance sonar only needs to detect the obstacle a few hundred metres distant to allow the vessel to perform evasive manoeuvres. Such a system is used on the HUGIN AUVs (20). Subsection 11/25 in the ABS "Rules for Building and Classing Underwater Vehicles" requires that a manned submersible are to be equipped with the following equipment (21):

i. *At least one compass or gyro*
ii. *An obstacle avoidance system such as sonar*
iii. *Where low-light operations are expected, appropriate lighting is to be provided*
iv. *Means for determining distance from the seabed*
v. *Two independent means of measuring the depth of the unit. If both means are electrical, then at least one must be operable upon loss of the main source of power*
vi. *Means to indicate heel and trim, as applicable*
vii. *Locating devices as per Subsection 11/29*

Subsection 11/29 states that the vessel must be equipped with a surface and a subsurface emergency locating device.

In conjunction with the increased use of subsea solutions in the offshore industry acoustic positioning systems have been developed that offer accuracy of up to ±10 cm. (22). The most common systems use sets of underwater transponders with a known position to calculate the vessel position. The accuracy depends on the number of transponders and transponder spacing. The terms Long Base Line (LBL), Short Base Line (SBL) and Ultra Short Base Line (USBL) all refer to the spacing between transponders. Transponders can be placed along the route to assist in course keeping as well as on structures and pipes on the field to help with in-field navigation and installation of components. Transponders will probably have to be installed on the field regardless, as they are currently used by surface vessels to maintain position control both for the DP system and an ROV.

Long-distance navigational aids such as inertial navigation systems (INS) have been developed for military submarines some time ago. They are equally important for a military vessel which needs to be able to patrol a designated area as well as detect and engage other vessels. Inertial navigation systems use accelerometers, gyros as well as compass heading and speed measurements to keep track of the vessel position. This method alone is however somewhat inaccurate over time, hence positional references are needed to verify the position and calibrate the INS at certain intervals.

Jalving, Gade, Hagen and Vestgård (23)present the different approaches used to augment the INS's of the HUGIN 1000 AUV. The HUGIN INS can process correctional input from GPS, underwater transponders and underwater terrain recognition. The latter requires that bathymetry of the relevant area is available and that the vessel carries a bathymetric sensor, for instance a multi-beam echo sounder. GPS requires that the vessel ascends to "periscope" depth and raises an antenna in order to get a position fix. Figure 9 is a visualization of the HUGIN 1000 AUV INS system structure.

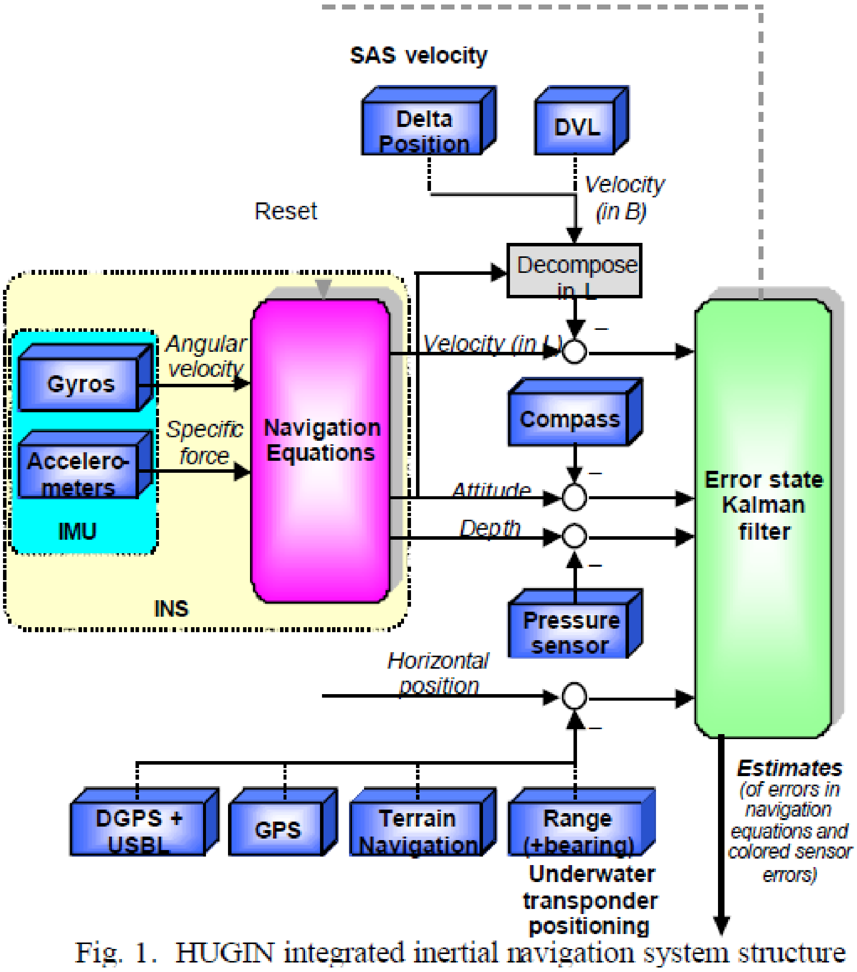

Figure 9 HUGIN INS structure (23)

As this system is adequate for an AUV it should definitely be adequate for a manned vessel where it is possible to perform manual control calculations as well as manual overrides in case of unexpected situations. The HUGIN navigation and obstacle avoidance systems are fully compliant with points i through iv. Compliance with the remaining points should not present any difficulties, although visual navigation is only expected while surfaced, limiting the need for external lighting. Further the equipment is fairly lightweight and compact, after all it is carried by a relatively small AUV. There should not be any difficulties integrating the sensors in the external hull and the rest of the equipment with the other control room electronics. The sensors are considered too lightweight to be considered in further weight and stability calculations. The overall power requirements will be presented in the next section.

4.3.2 Control, Computers & Monitors

The vessel is to feature a fully computerized control system similar to those used on modern surface vessels. There are to be two main control rooms with distinct purposes, one machinery control room and one navigational and operational control room. Both control rooms should however be able to access the same systems in the event of an emergency. The vessel is also to be fitted with an escape vehicle in the event of a critical failure which leaves the vessel stranded on the sea floor or sinking to a larger depth than the hull crush depth.

This is to be connected to the vessel through a data link and have at least rudimentary access to the vessel control systems. In this way it is still possible to sail the vessel home even if both main control rooms are disabled.

Modern electronics and flat screen displays are very compact and powerful, so finding space for the hardware while maintaining an easy to use interface should not be an issue. All computer equipment, displays and equipment such as gyros and accelerometers are assumed to require 10 kW of electric power. The power required by sonars increases with the intended range, long range sonars can require 7 kW (24). This information is however close to 20 years old, so I expect that advances in signal processing technology have reduced the power required by sonar. The acoustic equipment is nonetheless assumed to consume 7 kW.

4.3.3 Battery Charging

According to (25) the maximum charge current in lithium ion battery cells is slightly larger than the discharge current. Consequently the batteries will take more or less the same time to recharge as to discharge if discharged at maximum capacity. The maximum power output of the batteries are however likely to be several orders of magnitude larger than the maximum battery load during normal operations. The limiting factor will therefore be the available surplus power. The batteries are assumed to be charged while using DP near the surface. Charging of the batteries is assumed to be done in parallel with HP air replenishment, resulting in a power surplus of 183 kW. This is larger than the expected battery load, so battery charging time is assumed to be smaller than the discharge time. There are five hours set aside for battery charging each day, hence the batteries cannot be drained more than 917 kWh per day if they are to be fully recharged during a single day.

4.4 Load Profiles

With the verification and specification of the power required by shipboard systems far more detailed and accurate load profiles can be specified. These can be used to calculate the total energy required to keep the vessel operating during a typical deployment. The minimum amount of fuel and battery capacity can then be established. The different load profiles are:

I. Transit
II. Inspection
III. Installation
IV. Airlock cycling
V. Maintenance
VI. Surfacing
VII. Submergence

Table 33 Electric load analysis summary

Category	$\dot{W}$ [kW]	$\dot{Q}$ [kW]	I	II	III	IV	V	VI	VII
Hotel									
Aircondition	4,9		X	X	X	X	X	X	X
Heating		19,1	X	X	X	X	X	X	X
Freshwater generation system	15		X	X	X	X	X	X	X
Acoustic systems	7		X	X	X	X	X	X	X
Ship control and general electric equipment	10		X	X	X	X	X	X	X
Hydraulic pump	22,2		X	X	X	X	X	X	X
Galley	10,0		X	X	X	X	X	X	X
Trim transfer pump	8,33		X	X	X	X	X	X	X
Coolant circulation pump	0,2		X	X	X	X	X	X	X
Seawater coolant pump	0,2		X	X	X	X	X	X	X
Fuel pumps	0,06		X	X	X	X	X	X	X
Payload and cargo handling									
Airlock drain pump	108,18					X			
ROV	126,82			X	X				
ROV tool	8			X	X				
Crane power	105,01				X				
Miscellaneous machinery									
Air compressor	21,95						X		
Trim deballasting pump	83,96					X		X	X
LP MBT blower	125,93							X	
Battery charging	183						X		
Propulsion and station keeping									
Transit propulsion power (8 knots)	163,98		X						
Transit at battery power (4 knots)	20,10							X	X
Stationkeeping	196,78			X	X	X	X		

Table 34 Load profile & available power summary

Load profiles	Load [kW]
Transit	241,83
Inspection	409,45
Installation	598,42
Airlock cycling	466,77
Maintenance	480,00
Surfacing	307,83
Submergence	181,91
Available fuel cell power	480,00 kW
Maximum peak load handled by batteries	118,42 kW

Table 33 displays which equipment pieces that are used in the different load profiles. The total loads and the maximum available power are summarized in Table 34. It is apparent that the only task that cannot be completed without battery power is installation. The expected "overload" duration is merely a few hours. The emergency conditions are therefore likely to pose the strictest requirements to the battery rack. More on this in section 4.5.4.

4.5 Endurance Analysis

4.5.1 Energy Consumption

Two mission scenarios were developed to establish the amount of fuel needed by the vessel. One is a typical inspection mission where the vessel only performs inspections, the other is a combined installation and inspection mission where a single large component is replaced. After replacing the module the rest of the mission is devoted to inspection. The energy consumption of the two mission types are listed in Table 35 and Table 36.

Table 35 Energy consumption for a 14 day inspection mission

Task	Time spent [hours]	Load [kW]	Energy [kWh]
Transit to and from the field	72	241,8	17412,0
Transit in-field	47,67	241,8	11527,4
Airlock cycling	14,67	466,8	6846,0
Inspection	154,0	409,4	63055,2
Maintenance	47,67	480,0	22880,0
Total time, peak load and total required energy	***336***	***480,0***	***121720,6***

Table 36 Energy consumption for a 14 day installation and inspection mission

Task	Time spent [hours]	Load [kW]	Energy [kWh]
Transit to and from the field	72	241,8	17412,0
Airlock cycling	14,67	466,8	6846,0
Removal of the old module	2,0	598,4	1196,8
Installation of the new module	2,0	598,4	1196,8
Deployment and recovery of tools	1,0	598,4	598,4
Inspection and cleaning of the module before replacement	8	409,4	3275,6
Transit in-field	49,24	241,8	11906,2
Maintenance	49,24	480,0	23633,3
Other structural inspection	137,9	409,4	56447,2
Total time, peak load and total required energy	***336***	***598,4***	***122513,1***

The schedules have been estimated based on past experience from an internship with Technip, earlier projects at NTNU and information gathered for the project thesis that established what intervention tasks for which a submarine is suitable. During inspection 14 hours per day are dedicated to inspection/ROV operations, 80 minutes to airlock cycling while the remaining time is split evenly between in-field transit and general maintenance. Please note that a pure inspection mission does not require battery power to handle expected peak loads. The installation operation is estimated to take 15 hours, only five of which are expected to need battery power. There should therefore be no problem to recharge the batteries.

The surfacing and submergence loads are neglected in these calculations as their durations are very short.

4.5.2 Fuel Tank Sizing

The fuel tanks are to be of the auto-refrigerating type of cryogenic tanks often used to store LNG. The pressure is kept constant by siphoning off the boil-off gas. This can then either be liquefied or used by the fuel cells. The tanks are in this study assumed to store the cryogenic liquids at atmospheric pressure. Increasing the storage pressure should however by gas-liquid-solid phase diagram theory increase the boiling point of the fluid, and it would be prudent to investigate if pressurization has a notable effect on the boil-off rate.

The tanks are to have an outer pressure shell. Within this shell there is to be insulation and an internal reactant containment tank. As a safety measure the tanks must have a safety release valve that allows boil off gas to be vented to the sea if, for some reason, the fuel cells are not operating for an extended amount of time. Based on the maximum strength of the insulation and/or internal tank a maximum allowable internal overpressure can be set.

The storage factor takes the added volume requirements due to insulation. Cryogenic LNG tanks are in widespread use today, the normal storage factor is between 2 and 3. The fuel tanks are not to be form fitted to the hull, but will be cylindrical tanks mounted both between the pressure hull and outer hull as well as in a main fuel module. This modular approach is to simplify construction and hopefully reduce costs. It also allows for optimum tank design. Even so the LH_2 requires very low temperatures, so the storage factor is set to 3.

If the unmodified energy consumption from the load analysis is used the tanks will by design be completely empty when returning. A margin factor is added in order to avoid this.

Originally the missions were intended to last three weeks, however these calculations revealed that the fuel tanks would require such a volume that the vessel would require very large fuel tanks (approx. 450 m^3). This is nearly as large as the pressure hull and would drastically increase the size of the vessel and the propulsion resistance. The mission length was therefore shortened to two weeks, reducing the tankage volume by roughly a third. The storage calculation results are listed in Table 37.

Table 37 Fuel storage volumes and weight according to load analysis max consumption

Energy margin factor	1,10 [-]
Stored fuel energy content	134764 [kWh]
Stored LH_2	6913 [kg]
Stored LO_2	55304 [kg]
LH_2 volume	98 [m^3]
LO_2 volume	48 [m^3]
Storage factor	3 [-]
Minimum LH_2 tank volume	296 [m^3]
Minimum LO_2 tank volume	145 [m^3]

4.5.3 Fuel Tank Design

The main challenge with the design of the tanks is the difference in density between LO_2 and LH_2. To simplify production it would be an advantage if all tanks were equal, however this is not possible with the density difference. This is solved by using two lateral tanks mounted outside the pressure hull to carry the “excess” LH_2. These are mounted high on the hull in order to raise the centre of buoyancy. They also provide the vessel with an increased area of inertia during surfacing and submergence. The LO_2 and most of the LH_2 will be stored in four identical tanks in the fuel section. This is of course on the premise that the tanks can be insulated enough to reduce hydrogen boil-off to a manageable level. The insulation material has not been selected, but the maximum insulation thicknesses and minimum overall thermal conductivities have been estimated. The two LH_2 tanks in the fuel module are located above the two LO_2 tanks to keep the centre of gravity as low as possible. The fuel tank characteristics are summarized in Table 38 and their location is illustrated in Figure 10.

Table 38 Fuel tank summary

Parameter	Module LO_2 tanks	Module LH_2 tanks	Lateral LH_2 tanks	Symbol
Number of tanks	2 [-]	2 [-]	2 [-]	n_{tank}
Pressure tank length	15 [m]	15 [m]	25 [m]	L_{PT}
Reactant tank length	13 [m]	13 [m]	24 [m]	L_{RT}
Pressure tank diameter	3,0 [m]	3,0 [m]	1,5 [m]	D_{PT}
Reactant tank diameter	2,3 [m]	2,3 [m]	0,9 [m]	D_{RT}
Insulation thickness	0,5 [m]	0,5 [m]	0,3 [m]	t_{ins}
Pressure tank volume	106 [m^3]	106 [m^3]	44,2 [m^3]	V_{PT}
Reactant tank volume	35,3 [m^3]	35,3 [m^3]	14,7 [m^3]	V_{RT}
Total Pressure tank volume	**212 [m^3]**		**300,4 [m^3]**	$\mathbf{V_{fuel}}$

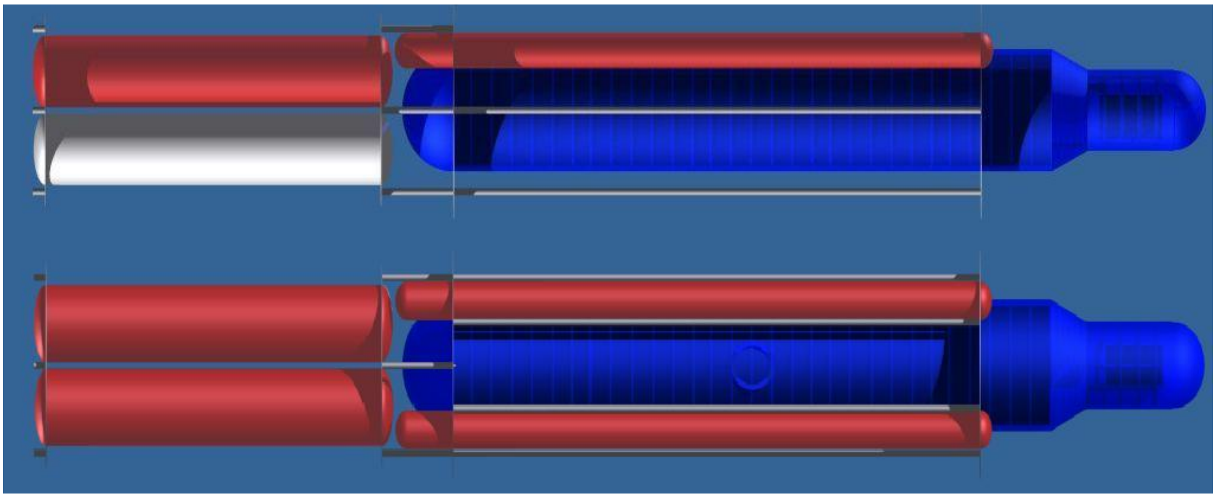

Figure 10 Side and top view of the fuel tanks as mounted in the fuel section and along the pressure hull. Hydrogen tanks are marked in red and oxygen tanks in white.

This configuration satisfies the volumetric requirements and leaves quite some space for insulation. The listed insulation thickness does not include the pressure tank wall thickness. Please see section 7.1.2.3 for the dimensioning calculations of the fuel tank pressure vessels. An estimate of the maximum overall thermal conductivities was established by setting the maximum hydrogen boil-off rate to the maximum hydrogen consumption of one fuel cell. As this is but a rough estimate intended to establish the order of magnitude of the thermal conductivities thin wall is assumed. The surface area is taken as the total surface area of the pressure tanks, though using the reactant tank length. Although the initial intention is to keep the fuel section tanks identical, the insulation required for LH_2 storage may be too expensive to use in the LO_2 tanks. The maximum thermal conductivity of the LO_2 tanks is therefore also estimated. Please note that the maximum LO_2 tank thermal conductivity is only half of that expected from insulated air-to-air bulkheads (16) and should therefore be relatively easy to achieve. The boil-off calculations are given in Table 39.

Table 39 Fuel boil-off rates

Parameter	LH_2	LO_2	Symbol/Formula
Lateral tank surface area [m^2]	121,3	-	$S_{TL} = \pi \left(\frac{D_{PT}{}^2}{2} + D_{PT} L_{RT} \right)$
Fuel module tank surface area [m^2]	155,4	155,4	$S_{TM} = \pi \left(\frac{D_{PT}{}^2}{2} + D_{PT} L_{RT} \right)$
Total surface area [m^2]	553,4	310,9	$S_{total} = 2 \cdot (S_{TL} + S_{TM})$
Required insulation thermal conductivity [W/m^2 K]	0,00545	0,0505	k_{ins}
Ambient temperature	276 ˚K	276 ˚K	T_{amb}
Reactant storage temperature	20˚K	90˚K	$T_{storage}$
Heat flow [kW]	0,77	2,92	$\dot{Q}_{lost} = k_{ins} \cdot S_{total} \cdot (T_{amb} - T_{storage})$
Heat of vaporization [kJ/kg]	452	213,2	Δh_{vap}
Amount boiled off [kg/s]	0,00171	0,0137	$\dot{m}_{evap} = \frac{\dot{Q}_{lost}}{\Delta h_{vap}}$
Single fuel cell consumption [kg/s]	0,00171	0,0137	-

4.5.4 Battery Rack

4.5.4.1 Battery Type

Even though the vessel has an all-AIP power plant it still needs batteries to provide emergency power and to handle peak loads. Lithium ion batteries are to be used as they have very high power and stored energy densities and several other advantages. They do not have any memory effect and have a low self discharge rate compared to other types of batteries (26). Buckingham (27) also states the following advantages and disadvantage:

Lithium ion batteries are:

- *Rated to a higher current than other battery types;*
- *Durable to experience a large number of full-charge cycles;*
- *Capable of sudden changes in demand;*
- *Shown to have an in-service reliability;*
- *Vulnerable to fire if over-charged.*

The retained charge level is highly dependent on the storage temperature; even a 100% charge will only drop to 94% over the course of one year if stored at 0˚C (27). The batteries are in this case to be stored outside the pressure hull in arctic waters, and are therefore expected to perform very well. Table 40 lists the main performance characteristics of the lithium ion battery.

Table 40 Lithium ion battery performance characteristics(27)

Power density	220 [kW/m^3]
Specific power	0,11 [kW/kg]
Energy density	270 [kWh/m^3]
Specific weight	0,12 [kWh/kg]
Density	2250 [kg/m^3]

4.5.4.2 Performance Requirements

There are several situations for which the batteries must be able to supply power. These will be used in conjunction with the information in Table 41 to establish the minimum size and weight of the battery rack. The minimum requirements are:

i. The batteries must be able to maintain life support functions in case of an emergency for at least 48 hours as per §34 of the PSA Activities Regulations(13). Minimum life support is interpreted as being able to sustain the hotel load. This assumes that all propulsion is lost and that the vessel is stranded at the bottom of the sea.
ii. In the event of a failure that leads to a complete loss of fuel cell power the vessel must still be able to transit back to base, albeit at a lower speed.
iii. The batteries must be able to handle the peak loads the fuel cells cannot handle during the most power demanding operations.

Table 41 Battery rack performance requirements

Parameter	i	ii	iii
Endurance period [hours]	48	74	5
Load [kW]	78	83	118
Expended energy [kWh]	3737	6182	592
Required volume for minimum power [m^3]	0,4	0,4	0,5
Required volume for minimum energy [m^3]	14	23	2
Required emergency battery weight [kg]	31140	51513	4934

The battery rack must at all times be able to provide power for both an emergency return and emergency life support. The minimum battery rack size must therefore be 39 m^3 and weigh in at 87,6 Te if the batteries are to handle the peak loads in addition to the emergency loads. Please note that the energy expended by peak loads is less than the energy available for battery charging (see section 3.3.3). Originally the battery rack was intended to weigh in at 100 Te as this would fulfil the minimum requirements with a certain safety stored energy margin. This safety margin is useful in several instances:

- The loss of a fuel cell can potentially happen any time, the worst case being during component installation or removal. By having a safety margin there is no need to panic or dump potentially expensive components as the vessel at least can retrieve the component still attached to the crane before returning to base.
- The cause of the failure can be very simple, and by having a safety margin one has the time to identify and possibly solve the problem so that the vessel can continue the mission.

It was however discovered that there was a large excess of buoyancy during the weight and buoyancy analysis. The vessel must be neutrally buoyant, and the only practical options available are significantly increasing the battery rack size or to increase the permanent ballast (see section 7). It was decided to double the size of the battery rack, as this would drastically improve the stored energy safety margin of the vessel.. This was not a problem, as there was more than enough volume in the free-flood space between the outer hull and pressure hull. The increased battery rack capacity also greatly increases the time available for problem solving in case of an emergency. The following example cases describe different situations that may arise. The time available before batteries are drained below the minimum emergency charge in the in the different cases are listed in Table 42.

A. A critical failure shuts down the fuel cells during an installation operation. How long can the batteries support the installation load profile on the available power reserve?
B. Assume that the same error occurs as in case A, but in this case the one is able to recover from the installation load profile in two hours. How long can the batteries support the hotel and stationkeeping loads while attempting a repair/troubleshooting on the available power reserve?

Table 42 Battery rack details

Parameter	Value
Battery weight	200 [Te]
Battery volume	89 [m^3]
Maximum power output	19556 [kW]
Stored energy	24000 [kWh]
Combined energy requirement for an emergency return and life support reserve	9919 [kWh]
Stored energy not intended for emergency situations	14081 [kWh]
Case A	24 [hours]
Case B	47 [hours]

5 Pressure Hull Design & Dimensioning

5.1 Dimensioning by ABS Rules

The main dimensions of the pressure hull were established in chapters 3.1 and 3.3. The pressure hull is to comply with the ABS rules for manned submersibles. It will be comprised of five main sections separated by bulkheads capable of withstanding the flooding of the neighbouring section. A conical section connects the main pressure hull section with the small diameter airlock. The conical transition is used to achieve a smooth diameter transition without large stresses. Ellipsoidal end caps close the pressure hull fore and aft, the fore cap being hinged in order to facilitate ROV launch and recovery. HY120 (yield stress is 120 kpsi) steel is chosen to keep plate thickness at a manageable level. HY120 is fairly common in submarine pressure hulls. A safety factor of 0,5 was applied to the material yield stress. The main cylindrical section, conical section, airlock section and ellipsoidal end caps were dimensioned for an operating depth of 500 metres according to ABS regulations using a spreadsheet (28) used in earlier projects at NTNU(29)(30)(31). This spreadsheet did however have to be expanded in order to include the conical section requirements. A full summary of the spreadsheet calculations can be found in appendix II, while the formulas can be found in the ABS *Rules for Building and Classing Underwater Vehicles, Systems and Hyperbaric Facilities, Section 6*. (21)The spreadsheet uses American units and was not converted to the metric system to save time and avoid possible conversion errors. Figure 11 displays the internal structure of the pressure hull.

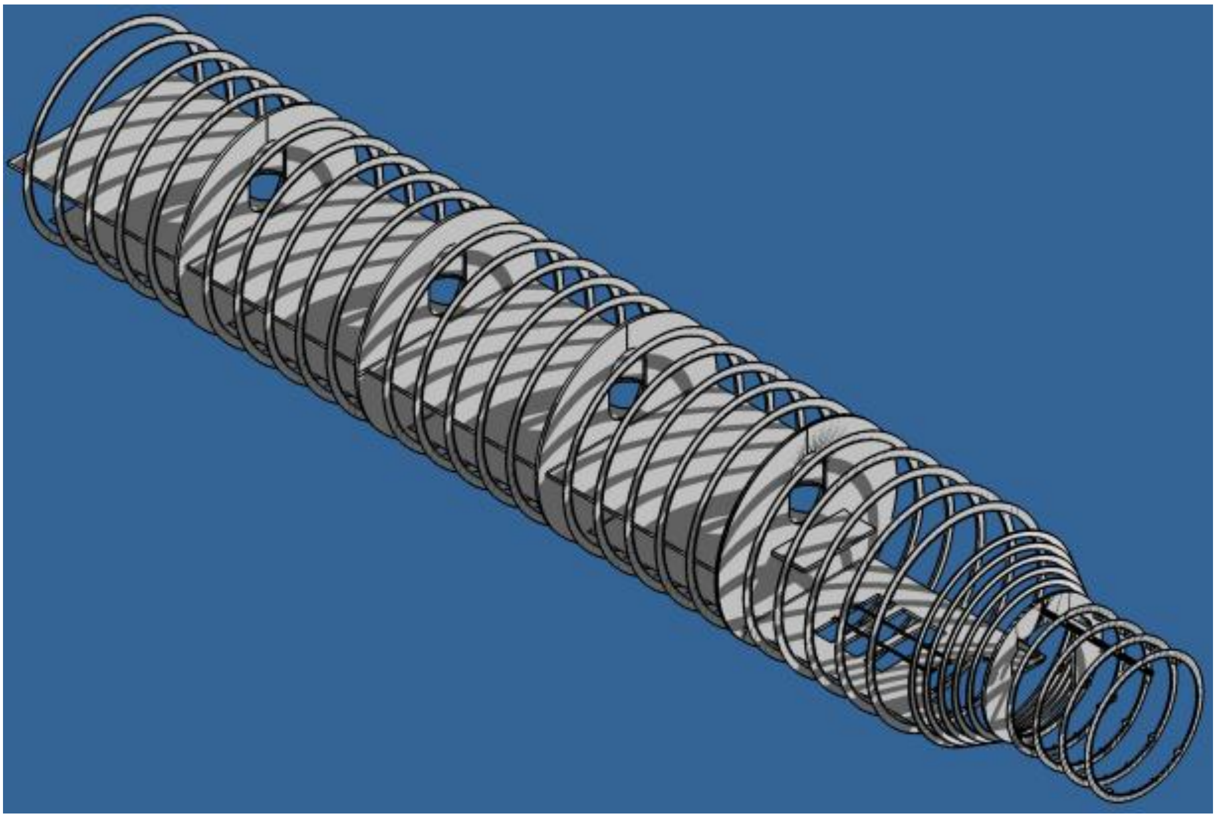

Figure 11 Stiffener, bulkhead & deck layout

The stiffeners are T-shaped steel rings. The steel density, Young's modulus and Poisson ratio are assumed to have typical values. The plate thickness should be equal where the sections are welded together. The thickness is for simplicity set to be constant, leading to much higher strength in the smaller diameter sections. One can apply a tapering thickness transition at a weld where there is a difference in required plate thickness if the increased steel cost is found to be too high. The taper is necessary in order to avoid high stress concentrations at the weld. The conical section is not listed with internal and external diameters as these will be equal to the main pressure hull section on one end and the airlock section on the other end. The calculation results are summarized in Table 43 and Table 44.

Table 43 Hull section dimensions

Parameter	Main Hull Section	Conical Section	Airlock Section
Young's modulus [GPa]	210	210	210
Poisson's ratio [-]	0,3	0,3	0,3
Yield stress after application of the safety factor [MPa]	413,7	413,7	413,7
Density [kg/m³]	8000	8000	8000
Length [m]	25	1,5	3,6
Bulkhead centre-to-centre spacing [m]	5	N/A	N/A
Internal diameter [m]	5	N/A	3,3
External diameter [m]	5,1	N/A	3,4
Shell plate thickness [mm]	50,8	50,8	50,8
Stiffener web height [mm]	177,8	152,4	152,4
Stiffener web thickness [mm]	50,8	50,8	25,4
Stiffener flange width [mm]	101,6	76,2	76,2
Stiffener flange thickness [mm]	25,4	25,4	25,4
Design depth [m]	500	500	500
Maximum depth allowed by ABS rules [m]	537,7	688,9	779,7
Section crush depth (all safety factors set to 1) [m]	1075,3	1684,3	2243,9

Table 44 Ellipsoidal head end caps

Parameter	Main hull section end cap	Airlock end cap
Young's modulus [GPa]	210	210
Poisson's ratio [-]	0,3	0,3
Yield stress after application of the safety factor [MPa]	413,7	413,7
Density [kg/m³]	8000	8000
Maximum inner diameter [m]	5	3,3
Maximum outer diameter [m]	5,1	3,4
Inside depth [m]	2,03	1,27
Shell plate thickness [mm]	50,8	50,8
Skirt length [mm]	152,4	101,6
Design depth[m]	500	500
Maximum depth allowed by ABS rules[m]	560,8	899,5
Head crush depth (all safety factors set to 1) [m]	1204,9	2177,2

Not unexpectedly the constant thickness gives the smaller diameter sections the ability to descend to greater depths than the main hull section. This surplus capacity may perhaps be unnecessary regarding the functional requirements to the vessel, but there are several points in favour of a constant thickness:

- The logistics of building the vessel is simplified by only having to order hull plates of a single thickness. This should help reduce overall building costs.
- Without a tapered thickness reduction the hull sections will likely be easier to handle and assemble due to a simpler geometry. This should also reduce building time and overall building costs.
- The hull sections with the added strength are located in the bow. While the vessel is to avoid collisions, the forward hull structure is the most vulnerable if one occurred. One example of this is the collision of the American fast-attack submarine USS San Francisco with an underwater mountain. The damages to the vessel can be seen on Figure 12. Several crewmembers were injured and one killed in the incident (32). It can therefore not be a disadvantage to have a reinforced bow section.
- The extra-strength sections comprise a very little part of the pressure hull, therefore the extra steel cost is likely to be small compared to the overall steel costs.

Figure 12 USS San Francisco after collision with an underwater mountain (33)

The vessel has a hull crush depth of more than double the required operational depth. This is largely due to the large factor of safety applied to the steel yield strength on top of the factors of safety in the ABS rules. This demonstrates that there is potential for operations at greater depths given some refinement of the pressure hull design. It is of course also advantageous that the crush depth is much larger than the design depth from a safety point of view.

5.2 Strength Control Calculations

Although the pressure hull fulfils the requirements from ABS it remains prudent to perform some simplified control calculations to verify the results. The basis of this check will be the von Mises yield criterion, while shell buckling and the existence of stiffeners are disregarded:

$$\sigma_y = \eta_{safety}\sigma_{vM}, \qquad \sigma_{vM} = \sqrt{{\sigma_1}^2 + {\sigma_2}^2 + {\sigma_3}^2 - \sigma_1\sigma_2 - \sigma_1\sigma_3 - \sigma_2\sigma_3}$$

Because the diameter of the hull is much larger than the plate thickness thin wall theory is assumed to be valid. The principal stresses for a cylinder with average radius *r* and wall thickness *t* exposed to an external pressure *P* then become:

$$\sigma_1 = \sigma_r = 0, \qquad \sigma_2 = \sigma_t = -\frac{r}{t}P, \qquad \sigma_3 = \sigma_z = -\frac{r}{2t}P, \qquad r = r_o - \frac{t}{2}$$

The thickness can now be expressed as a function of the outer radius and external pressure. This formula is regarded as valid as long as t/r is less than 0,1.

$$\sigma_{vM} = \sqrt{{\sigma_1}^2 + {\sigma_2}^2 + {\sigma_3}^2 - \sigma_1\sigma_2 - \sigma_1\sigma_3 - \sigma_2\sigma_3}$$

$$\Rightarrow \sigma_{vM} = \sqrt{\left(\frac{r}{t}P\right)^2 + \left(\frac{r}{2t}P\right)^2 - \frac{1}{2}\left(\frac{r}{t}P\right)^2}$$

$$\Leftrightarrow \sigma_{vM} = P\frac{r}{t}\sqrt{1 + \frac{1}{4} - \frac{1}{2}} = P\frac{r\sqrt{3}}{2t}$$

Introducing $r = r_o - \frac{t}{2}$

$$\sigma_{vM} = P\frac{\sqrt{3}}{2t}\left(r_o - \frac{t}{2}\right) = P\frac{r_o\sqrt{3}}{2t} - P\frac{\sqrt{3}}{4}$$

$$\Leftrightarrow \sigma_{vM} + P\frac{\sqrt{3}}{4} = P\frac{r_o\sqrt{3}}{2t}$$

$$\Leftrightarrow \frac{1}{t} = \frac{1}{r_o}\left(\frac{2\sigma_{vM}}{\sqrt{3}P} + \frac{1}{2}\right)$$

$$\Leftrightarrow t = r_o\left(\frac{2\sigma_{vM}}{\sqrt{3}P} + \frac{1}{2}\right)^{-1}$$

The use of this formula yields almost the exact same plate thickness as the ABS rules when the safety factor is applied to the yield stress. As this analysis disregards the fairly heavy stiffeners used the ABS hull calculations are deemed reasonable. The results are given in Table 45.

Table 45 Simplified pressure hull strength analysis

Parameter	Value	Symbol
Young's modulus	210 [GPa]	E
Yield strength	827,4 [MPa]	σ_y
Operating depth	500 [m]	$T_{operating}$
Safety margin	2 [-]	η_{safety}
Operating pressure	50,3 [bar]	$p_{operating}$
Estimated hull thickness without safety margin	26,2 [mm]	$t_{0\ hull}$
t/r	0,01 [-]	-
Estimated hull thickness with safety margin	52,1 [mm]	t_{hull}
t/r	0,021 [-]	-

6 Arrangement & Modelling

6.1 Hull Arrangement

The vessel will be comprised of five main sections. They are numbered with roman numerals starting at the forward section. These sections coincide more or less with different vessel functions, although section I by far is the most complex section. Figure 13 displays the sectional division.

I. The pressure hull section, main fuel section, cargo section and main propulsion section. The main components in the pressure hull section are the pressure hull, rescue vehicle, HP air storage cylinders and lateral LH_2 tanks.
II. The fuel section contains the main LO_2 and LH_2 tanks.
III. The cargo section is made up of the cargo hold and crane machinery.
IV. The main components in the propulsion section the propulsion pods and control surface machinery.

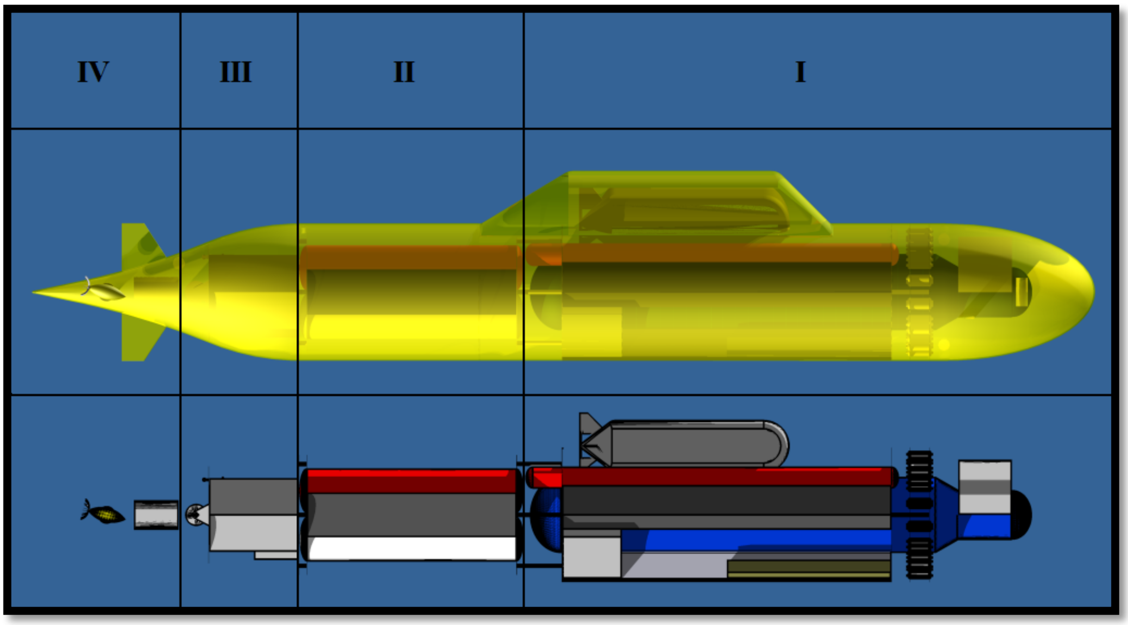

Figure 13 Hull sections with and without the outer hull

6.2 Pressure Hull Internal Arrangement

The pressure hull is designed with both safety and functionality in mind. The airlock is the most forward part of the pressure hull. Due to the size of the largest ROV it is to handle this was the only placement that did not require the vessel to have a large beam. The ROV workshop is necessarily placed directly aft of the airlock.

The machinery compartment contains pumps and other potentially noisy machinery and must therefore be placed in one of the extreme ends of the hull. As the front extremity is occupied by the airlock and ROV workshop the aftermost compartment is the only option. This is also close to the fuel tanks, minimizing the fuel piping length.

The sleeping quarters are placed in the middle of the vessel. This is a safety precaution as the compartments most likely to suffer a mishap are the machinery compartment and the ROV workshop/airlock. This placement puts two watertight bulkheads between these compartments and the sleeping quarters. Additionally, any sleeping crewmember is likely to be the last to be alerted and the most confused in the event of an emergency. These crewmembers should therefore have the shortest distance to travel to the escape vehicle. The escape vehicle entry hatch is located in sleeping quarters both because of this and because it is far from any probable point of damage.

The washroom is located just aft of the ROV workshop to avoid traffic through the sleeping quarters simply because a mechanic needs to wash his or her hands. Most of the crew is also expected to be in either the control room or ROV workshop, which is why the toilets also are forward. Similarly the sink in the galley can be used by mechanics working in the machinery compartment. Although some traffic cannot be avoided it is at least kept at a minimum. A graphical presentation is given in Figure 14 and Figure 15.

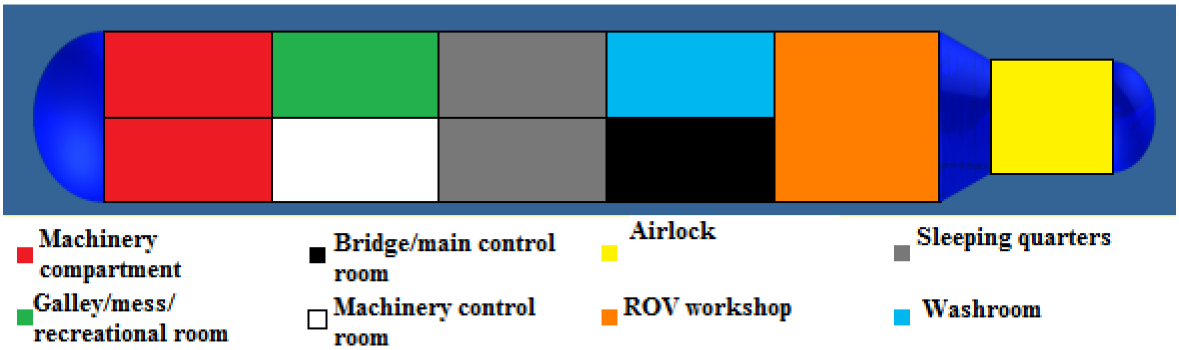

Figure 14 Pressure hull compartment classification

Figure 15 Upper and lower deck arrangement and side view

These drawings include all components in the parts list in Appendix III: Weight. This does not include items such as tables, chairs, lockers and so forth, although space has been set aside for these items. This is why the galley and sleeping compartments seem fairly empty. The exact location of each component can also be found in Appendix III: Weight. The approximate size of the components have in some cases been estimated in the previous chapters, but for the most part the required flow rates, delivery pressures etc. have been used to find existing components, the size of which have been used in the model.

The flounder diagram is another way to present the disposition of the pressure hull. The section area is plotted against the longitudinal position, and the area within the resulting curve represents the pressure hull volume. The approximate volume, longitudinal and vertical position of the vessel systems and components is plotted within this boundary. Figure 16 is the flounder diagram created for the pressure hull in the vessel designed in this thesis.

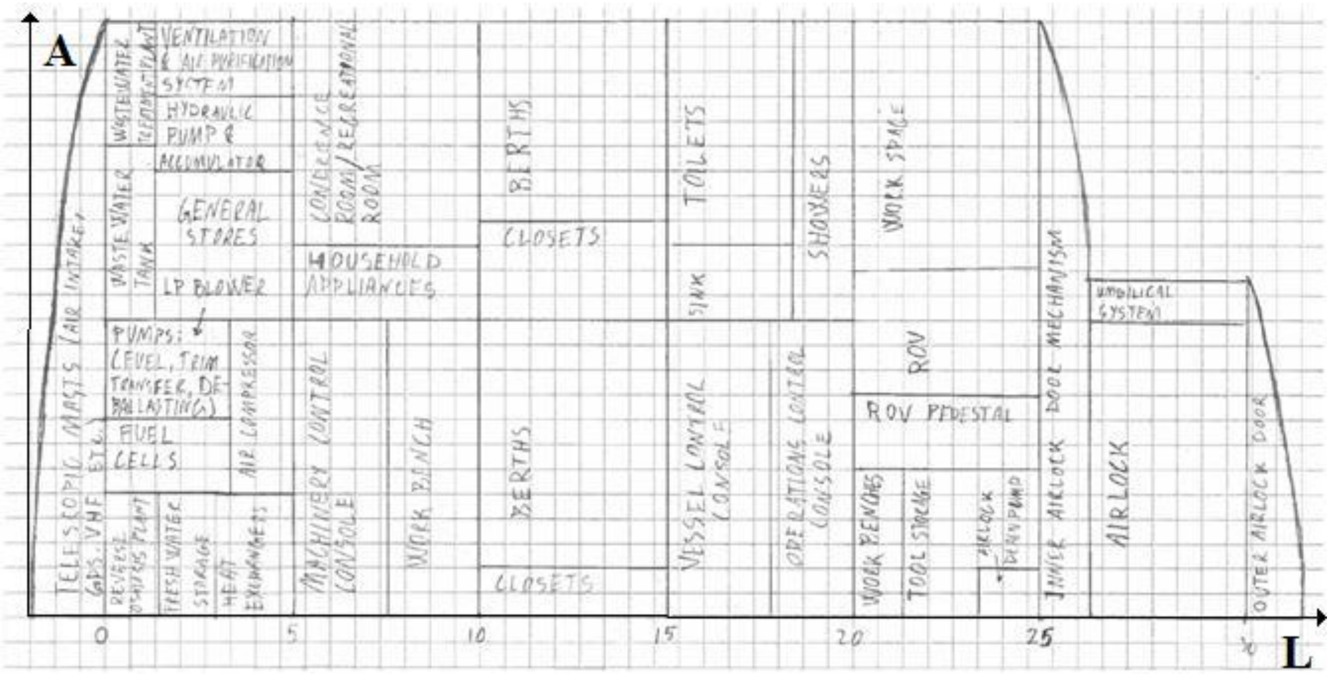

Figure 16 Flounder diagram

6.3 Custom System Design

The airlock is crucial for the vessel's ability to perform as intended. While designing an airlock may seem simple enough systems, certain difficulties arise when they are subject to the size- and functional limitations in this vessel. "Proof of concept" designs have therefore been developed to answer these difficulties.

It was also necessary to perform a more detailed design of the crane in order to determine the size and weight of the crane unit.

6.3.1 Airlock

The main challenges with designing the airlock were:

- Designing a simple solution for launching, recovering and maintaining the ROV.
- Designing a way to store the umbilical.
- Providing a way to avoid umbilical entanglement within the airlock.

The minimum workshop area and airlock dimensions were established in section 3.3.2. By bringing the deck of the workshop to the same level as the airlock "deck" two issues are solved:

- The minimum floor space requirement to the workshop is fulfilled. The raising of the deck compared also gives access to the underside of the ROV without the need to jack it up.
- By using a trolley the ROV can simply be wheeled in and out of the airlock.

The trolley is to be guided by rails mounted in the airlock and in the workshop. The inside airlock door is to be lowered down to form a "bridge" between the airlock and workshop deck, much like the loading platform common on most lorries. There is a guiding framework for the trolley within the airlock. This framework is designed so that the ROV will be correctly placed on the trolley during recovery. The airlock and workshop with the inner airlock door open are displayed in Figure 19 and Figure 20. The rail sections connecting the rails in the workshop, on the airlock door and inside the airlock are not included in the illustration.

The internal airlock door is designed as a plug type door like the doors on modern passenger aircraft. The edges of a plug door are slightly angled so that the seal becomes tighter with increasing pressure. To open and lower the door it must first be opened into the airlock and tilted sideways before it can be fully opened. The principle is demonstrated in Figure 17. The hydraulic arm and joint used to tilt and lower the door can be seen on Figure 20.

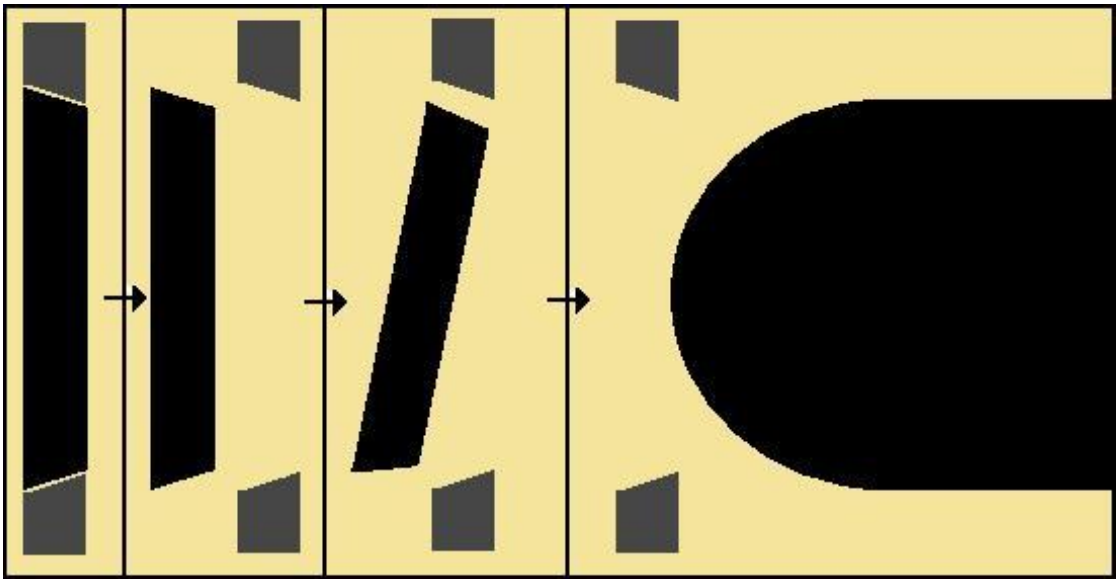

Figure 17 Airlock inner door opening sequence

There is 400 millimetre gap between the inner airlock bulkhead and the airlock set aside for this mechanism. Pythagoras' yields the distance required for the opening mechanism between the aft airlock bulkhead and the ROV was checked against the available space. These calculations are summarized in Table 46, and it is apparent that the gap is large enough.

Table 46 Airlock inner door space requirement

Parameter	Value	Symbol/formula
Door thickness	100 [mm]	t_d
Largest door width	1789,5 [mm]	w_{max}
Smallest hatch opening width	1769,5 [mm]	w_{min}
Minimum tilt depth	266,8 [mm]	$D_t = \sqrt{{w_{max}}^2 - {w_{min}}^2}$

The external door is simply the front ellipsoidal end cap. This will also be kept in place by the external pressure when closed, so the locking mechanism will not have to withstand large forces. This can consist of a few locking pins securing the head in both the open and closed positions. Figure 18 displays the opening principle.

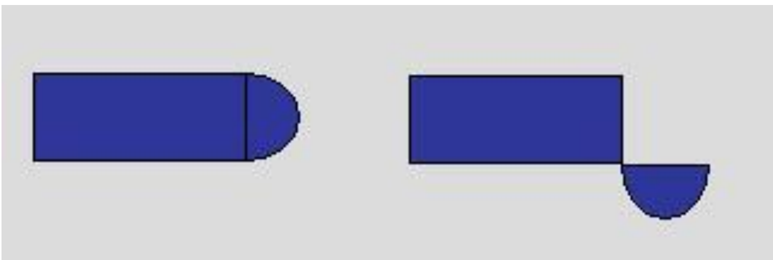

Figure 18 Top view principle sketch of the external airlock door

The umbilical drum is fitted above the hatch opening on the aft airlock bulkhead. As the vessel is submerged, the ROV will not require as long an umbilical as if the vessel was surfaced. A 1,4 metre long drum with a 0,5 metre diameter is therefore deemed sufficient. The umbilical will run through a sheave mounted on a hydraulic arm, keeping it from tangling itself within the airlock. The arm will extend outside the outer hull during ROV operations. This arm is not intended to handle any other loads than the tension in the umbilical. Both the drum and hydraulic arm are clearly visible on Figure 19.

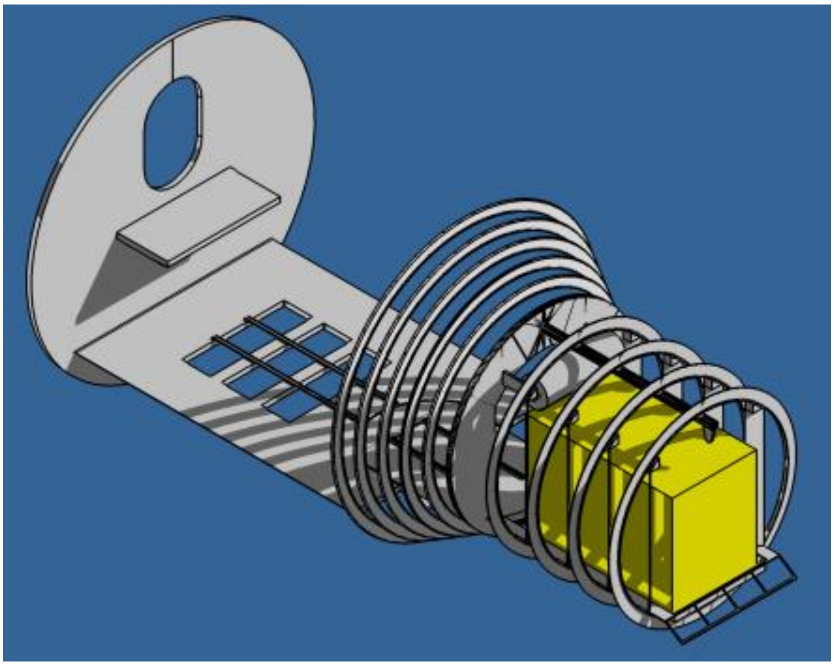

Figure 19 Airlock and workshop illustration 1

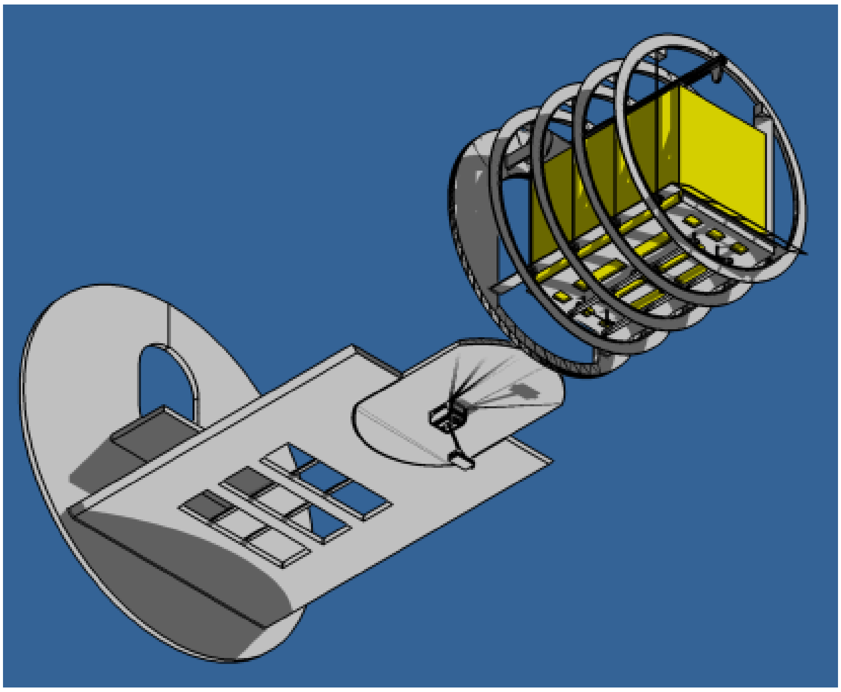

Figure 20 Airlock and workshop illustration 2

6.3.2 Gantry Crane

The crane beam consists of two steel I-beams with the hoist trolley traversing between them. In order to determine the approximate size of these I-beams the maximum allowable beam deflection was set to 50 millimetres. The problem was simplified to a 2D problem as displayed in Figure 21. Simple 2D beam deflection formulas were used to calculate the deflection (34). The steel is assumed to be HY120. The crane is to be able to handle a dynamic amplification factor (DAF) of 3. The gantry beam design with hoist trolley is displayed in Figure 22. The formulas and results are presented in Table 47.

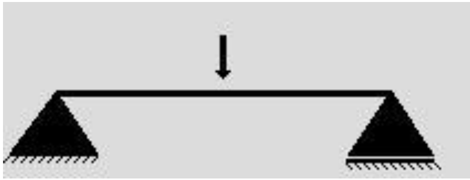

Figure 21 2D static beam model

Table 47 Crane boom strength

Parameter	Value	Symbol/ Formula
Design DAF	3 [-]	DAF
Number of beams	2 [-]	n
Hoist trolley weight	0,8 [Te]	m_t
Maximum static load per beam	42,8 [Te]	F_L
Maximum beam deflection	50 [mm]	δ_{max}
Beam length	3000 [mm]	L_B
Young's modulus	210 [GPa]	E
Required moment of section	34385214 [mm^4]	$I_{min} = \frac{DAF \cdot \left(\frac{F_L + m_t}{n}\right) \cdot L^3}{48 \cdot \delta_{max} \cdot E}$
HY120 Yield stress	826,8 [MPa]	σ_y
Steel density	8000 [kg/m^3]	ρ_s
Web height	270 [mm]	h_w
Web thickness	30 [mm]	t_w
Flange width	270 [mm]	w_f
Flange thickness	30 [mm]	t_f
Moment of section	52852500 [mm^4]	$I = \frac{{h_w}^3 \cdot t_w}{12} + 2\left(\frac{{t_f}^3 \cdot w_f}{12} + t_f \cdot w_f \cdot \frac{t_f + w_f}{2}\right)$
Maximum bending moment	162,1 [kNm]	$M = \frac{F_L \cdot L^2}{4}$
Maximum bending stress	460 [MPa]	$\sigma_m = \frac{M}{I} \cdot \left(\frac{h_w}{2} + t_f\right)$
Yield stress safety factor	1,8 [-]	$\frac{\sigma_y}{\sigma_m}$
Combined weight of both beams	1,2 [Te]	$m_b = 2 \cdot \rho_s \left(L \cdot h_w \cdot t_w \cdot 2\left(w_f \cdot t_f\right)\right)$

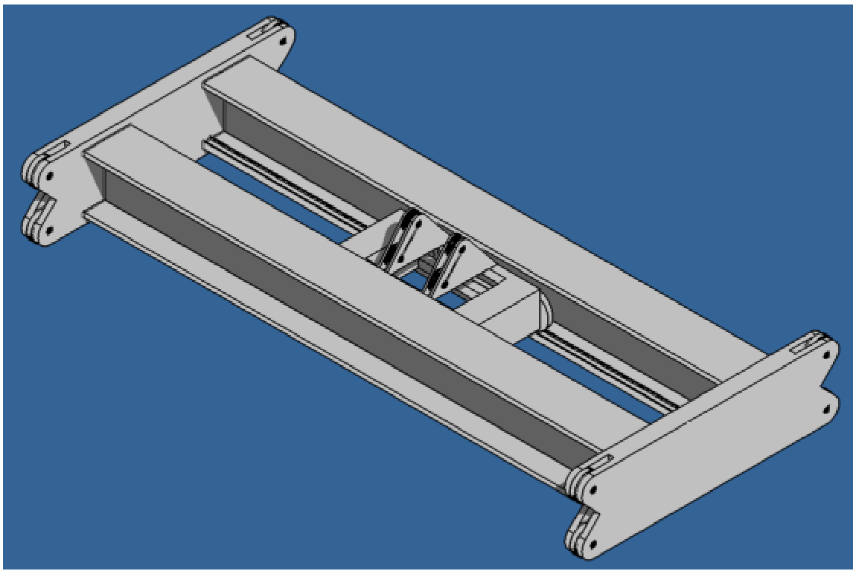

Figure 22 Crane beam & hoist trolley assembly

The hoist unit is to use two wires in a double fall configuration. This is to reduce the minimum wire diameter. This is in order to decrease the required wire drum diameter and therefore the overall space requirement. The material is again assumed to be HY120, however a safety factor of 3 is applied to the yield strength. The cable properties are listed in Table 48. The configuration of the wires and wire drums is displayed on Figure 23.

Table 48 Crane cable summary

Wire yield stress safety factor	3 [-]
Design yield stress	275,6 [MPa]
Number of wires	4 [-]
Max load per wire	642 [kN]
Minimum cross-sectional area per wire	582 [mm^2]
Minimum diameter	27 [mm]

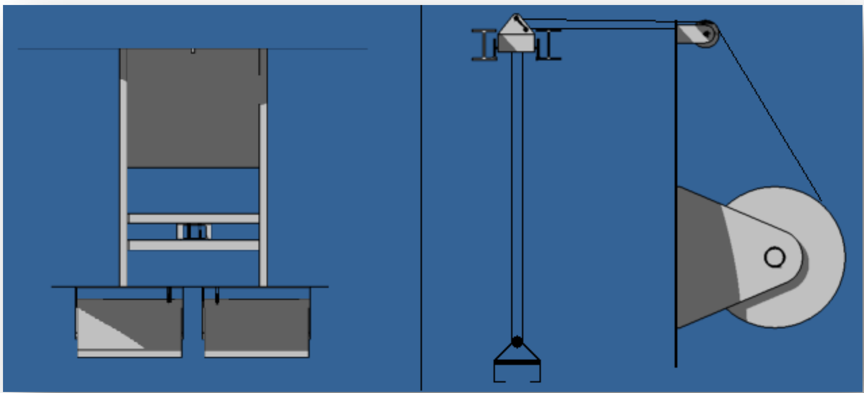

Figure 23 Wire & wire drum configuration

6.3.3 Outer Hull Hatches

The airlock, cargo hold and thrusters all require openings in the outer hull in order to function properly, however such openings would cause added resistance during transit. They are therefore to be covered by "sunroof"-style hatches. These will be mounted on rails inside the outer hull and are to be hydraulically powered. Figure 24 is a principle sketch of the mechanism. Such systems have been used in the car industry for several decades and should therefore be simple to adapt to this purpose. As these hatches are in the outer hull they do not need to form perfect seals or withstand external diving pressures, but they must be able to withstand ice loads when surfacing through ice.

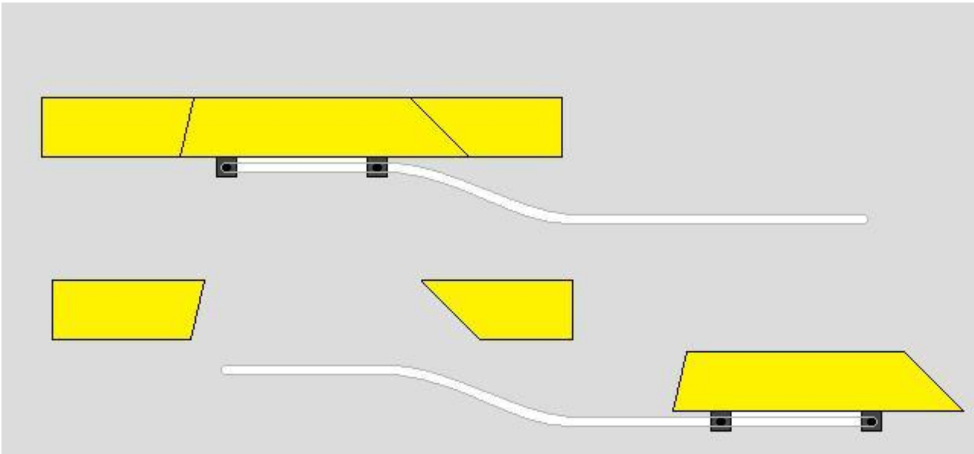

Figure 24 Outer hull hatch opening mechanism principle sketch

6.4 Emergency Systems

6.4.1 Fire Suppression System

Fire poses a very serious threat to a submarine and can quickly consume the limited oxygen in the submarine's atmosphere. Care must however be taken when combating fires, as submarines cannot take on an unlimited amount of water to douse the flames. Saltwater is not permitted as an extinguishing agent according to ABS rules (21), probably due to its conductive properties. Water mist fire-fighting systems such as HI-FOG (35) use high-pressure water to create a water mist which effectively and safely kills the fire. Tests have shown that water mist systems do not damage electrical equipment (36), however one should perform further studies due to the extreme damage potential of short-circuits on submarines. This is tragically demonstrated by the loss of the American fast attack submarine USS Thresher in 1963. It is theorized that a water mist caused by a small leak lead to short-circuits in the ship's electrical systems. This is assumed to have tripped the reactor fail-safes resulting in reactor shut-down, rendering the vessel unable to stop the descent to and beyond hull crush depth (37).

Another option is to use CO_2 as an extinguishing agent. The main issue with CO_2 is that the ventilation system is capable of quickly removing the CO_2 once the fire is put out and under control. The CO_2 is stored under pressure, greatly reducing the storage volume and eliminating the need for a dispersal pump. A mist system would require a high-pressure pump or accumulator and a water supply.

The centralized fire fighting system is to be a CO_2 deluge system as there is no danger at all for short-circuits and because it is very compact. The CO_2 system is only to be used if the crew is unable to control the fire with portable extinguishers. A water mist system can be used provided it is compact, can make do with the limited water supply and that it is proven beyond any doubt that no short circuits will occur.

6.4.2 Emergency Life Support

The ABS rules require that the life support system can maintain suitable concentrations of O_2 and CO_2 for at least 72 hours. The normal operating mode of the life support system uses oxygen stored in pressure vessels, while CO_2 is removed through filters. The system is designed so that no replenishment is required during the mission, and is so in full compliance with the ABS rules.

In the event of a main life support system failure emergency breathing masks are to be employed. These should be fairly standard and COTS units should therefore be available. These masks must comply with section 35.7 of the ABS rules, which aside from requiring that masks are available also state that:

> ***35.7.2 CO_2***
>
> *The system is to be designed such that CO_2 levels in the gas being breathed do not exceed 1.5 percent by volume referenced to standard temperature and pressure [a CO_2 mass of 0.0297 kg/m^3 at 1 atmosphere and 0˚C (0.00185 lbm/$ft^3$and 70˚F)].*
>
> ***34.7.3 Duration***
>
> *Untethered Submersibles. 150 percent of the time normally required to reach the surface from rated depth, but no less than two hours.*

By using the trim angle restriction of 25˚ from the stability requirement and a normal transit speed of 4,2 m/s the normal surfacing time is found to be:

$$t_{surface} = \frac{500\ m}{4{,}2\ {}^{m}/_{s} \cdot \sin 25} = 4{,}7\ minutes$$

It is apparent that the two hour minimum is the applicable requirement.

6.4.3 In Case of Emergency

Because the vessel will be operating far from the shore in an area with hostile surface conditions, several measures are taken to reduce the need to abandon ship and to limit the need for assistance in case of an emergency. The battery rack is sized so that a return on battery power is possible in the event of a critical failure which in, some way, disables the fuel cells. The battery rack stores enough energy to keep the vessel operational for quite some time before battery levels are too low for a return to base, leaving the crew time to perform repairs and make informed decisions. All critical systems are to be operable from all three main control stations: the operations control room, machinery control room and the deep sea rescue vehicle (DSRV) located within the sail. This is done in order to retain control of the vessel in case one or more of the control rooms suffer some sort of damage or equipment failure. The control rooms are also spaced from each other with at least one watertight bulkhead separating them. The vessel has two main propulsors, so even if one fails the vessel will not lose the ability to navigate. These built-in redundancies enable the vessel to perform the required repairs at sea or, if necessary, a safe transit back to base.

In the event of an accident which leaves the submarine stranded on the sea floor, the crew must have a safe way to abandon ship. Evacuation of crew from stranded submarines can either be done with individual escape pods or by using a DSRV. As the vessel is operating in the arctic and at great depths evacuation by survival suit only is not suitable. Individual escape pods would require a great deal of space in an already cramped pressure hull, not to mention that retrieving the pods on the surface may prove difficult. By using a DSRV the surface vessels would only have to look for one large vehicle, not to mention that the DSRV can be navigated would further simplify retrieving it, as opposed to an escape pod. The DSRV is therefore the preferred evacuation vehicle.

DSRVs are traditionally transported to the accident site by a parent vessel. As the submarine is operating in a remote region the only way to achieve a rescue within a few days is to have one permanently stationed at the field. Rather than having the DSRV on a stand-by vessel I propose to integrate it into the submarine. Given the size of DSRVs the best solution would be to integrate the DSRV into the conning tower/sail.

If the vessel is stranded the first order of action would be to release an emergency beacon buoy. By securing it with a cable it can be used both for communication and location. Ideally the crew will wait as long as possible/necessary within the submarine so that surface vessels can prepare to pick up the DSRV and receive the crew. Shtokman will have stand-by rescue vessels servicing the FPU(2), so this should be done within the 48 hour limit. Once all is ready for evacuation the crew will surface with the DSRV and rendezvous with a rescue vessel. If the damage to the vessel is critical and the crew has to evacuate immediately, rendezvousing with the rescue vessel will be more complicated, but the DSRV can stay submerged for some time and await the best possible retrieval opportunity, which still is preferable to surfacing 15 individual pods.

6.4.4 Deep Sea Rescue Vehicle

The DSRV used in the design is based on the Kockums R35 URF class submarine rescue vehicle, although the newer R351 S-SRV and R20 SRV designs are recommended for the final design. The R35 and DSRV model can be seen on Figure 25 and Figure 26. This is due to the ability to dive deeper (R351 can dive to 700 metres, R20 has no specific depth listed but is unlikely to have poorer diving abilities than its predecessors) and the simple fact that they implement improvements based on experiences gained with the R35 URF. The URF was chosen because of the amount of information available on the performance characteristics and main dimensions compared to the more recent designs (38). This information was used to design the DSRV bay on the submarine. Both the R35 and R351 DSRVs are able to rescue 35 persons, which is much more than required as the vessel crew size is maximum 15 persons. The spare room could be outfitted with electronics in order to facilitate the third control room function of the DSRV. Although very little is specified, the R351 and R20 are assumed to have equal or better performance than the R35. The technical data of the R35 are listed in Table 49.

Figure 25 Kockums R35 URF submarine rescue vehicle(38)

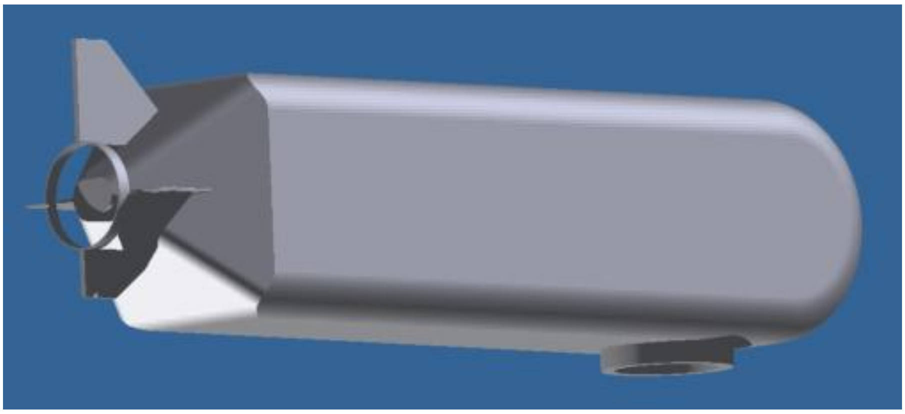

Figure 26 DSRV design used in the 3D model

Table 49 R35 URF Technical Details (38)

Hull	Double
Length over all	13.9 [m]
Beam	3.2 [m]
Displacement	52 [Te]
Propulsion	Single-shaft electric/hydraulic
Speed	3 [knots]
Diving depth	460 [m]
Submerged endurance	85 [hours]
Crew	3 [persons]
Rescue department	35 [persons]

The DSRV is to be kept neutrally buoyant in order to avoid unnecessary stresses on the link between the pressure hull and DSRV. This also enables a quick-release from the main hull if necessary as the DSRV will not need to adjust the trim much to avoid a rapid ascent immediately after detachment from the pressure hull. Some form of release mechanism must be designed to remove parts of the outer hull of the sail before the DSRV can be detached. The outer hull is only to withstand external loads; hence designing a quick release should not be too difficult. Due to time constraints a concrete design proposal has not been developed for this report.

7 Weight, Buoyancy & Stability

7.1 Weight & Buoyancy Distribution

Besides making sure that the vessel indeed can fit all of the equipment within its hulls, the 3D model is also the basis for determining the stability of the vessel. The estimated weight and placement of each component have been used to calculate the centre of gravity; the same approach was used on the hull sections to determine the centre of buoyancy. The displacement can also be accurately determined. Excluded from these calculations are systems that require detailed design such as electrical wiring, ventilation ducts and so on. These systems are expected to be fairly light (on the order of a few tonnes or less) and will therefore not have a significant impact on the centre of gravity, especially considering the effect of permanent ballast. Permanent ballast is required due to an excess of buoyancy and to correct the imbalance caused by the difference between the positions of the longitudinal centres of gravity and buoyancy. It is highly unlikely that the vessel equipment can be arranged to have a resultant centre of gravity in the same position as the centre of buoyancy. According to Burcher & Rydill (4) the permanent ballast is impossible to avoid in practice. At this stage the vessel requires 118,7 Te of permanent ballast. This will undoubtedly grow less as more systems and components are accounted for; nevertheless the permanent ballast size and placement is the key to attain a balanced and neutrally buoyant standard condition in the finished design as well. A complete summary of the weights and buoyancies can be found in Appendix III: Weight and Appendix IV: Buoyancy, however the most significant contributors will be accounted for in chapter 7.2. Table 53 lists the weight and location of the largest components, while Table 50 lists the buoyant sections and their centres of buoyancy.

Usually one would like to set the origin at the longitudinal centre of buoyancy when determining trim moments, but as there are several large components mounted outside the pressure hull the centre of buoyancy is yet to be determined. The longitudinal (x-axis) centre point is set at the end face of the cylindrical section of the pressure hull. This location was chosen as it was convenient to measure from in the model. The vertical (y-axis) and transverse (z-axis) centre points were set on the centre axis of the pressure hull cylinder for the same reason. This is illustrated on Figure 27.

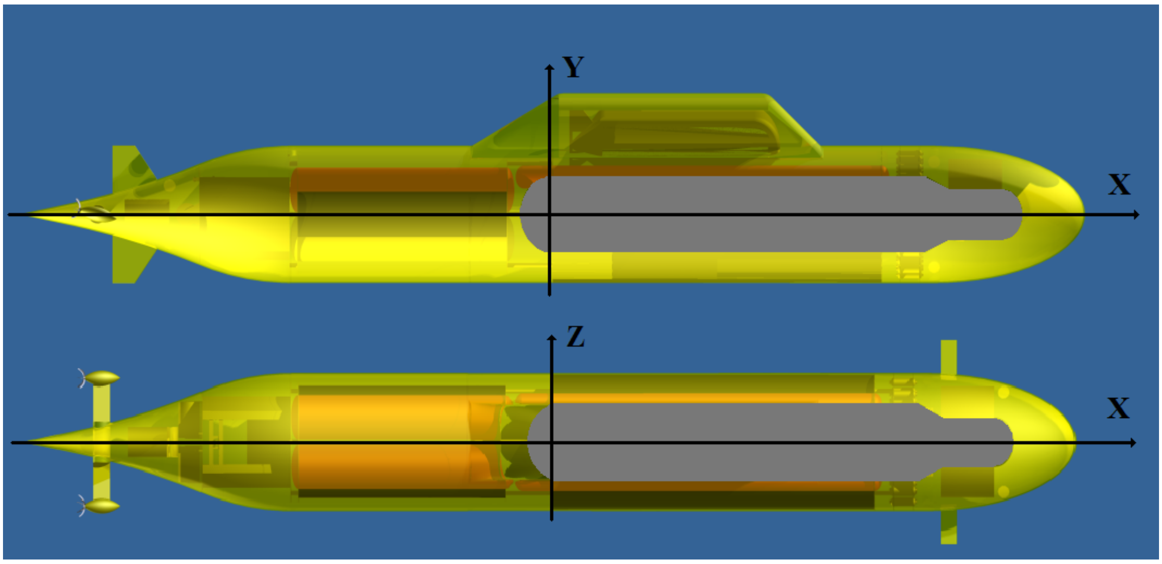

Figure 27 Origin location

7.1.1 Buoyancy

Only watertight modules and components were included in the buoyancy calculations. A seawater density of 1025 kg/m^3 was used. The volumes of the pressure hull and fuel tanks were determined by ordinary geometric formulas. The volume of the batteries is known from section 5.4.1. The wall thickness of the HP air cylinders was neglected, reducing their buoyant volume to the air storage volume of 258 litres. The conical pod simplification explained in section 7.1.2.3 was also used to determine the pod buoyancy. The frames and bracings were excluded as their exact shape and volume remains to be determined. The trim tanks are not included in these calculations; rather these calculations will be important when establishing the trim tank requirements. The DSRV is as previously mentioned neutrally buoyant and is therefore excluded. The external hull is also assumed to be neutrally buoyant. The reasoning behind this assumption is explained in section 7.1.2.

Table 50 Buoyancy summary

Module/equipment	Buoyancy [Te]	Vertical CB [m]	Longitudinal CB [m]	Transverse CB [m]
Main pressure hull	526,80	0,00	12,00	0,00
Main pressure hull ellipsoidal head	10,87	0,00	0,68	0,00
Conical section	22,14	0,00	25,20	0,00
Airlock	33,54	0,00	28,50	0,00
Airlock ellipsoidal head	5,77	0,00	30,52	0,00
Lateral LH_2 tanks	45,31	2,50	10,03	0,00
Main LH_2 tanks	217,20	1,63	-10,00	0,00
Main LO_2 tanks	217,20	-1,63	-10,00	0,00
Battery rack	91,10	-3,48	11,00	0,00
HP air bottles	11,11	0,00	23,90	0,00
Propulsion pods	1,44	0,00	-33,70	0,00
Standard vessel buoyancy/ resulting centres of buoyancy	**1182,46**	**-0,17**	**4,52**	**0,00**

7.1.2 Weights

7.1.2.1 Outer Hull

The outer hull is little more than a hydrodynamic fairing shell. It is not intended to resist diving pressures, and need therefore not be made of a high strength material like the pressure hull. It must however be strong and durable enough to survive potentially rough seas while the vessel is at periscope depth. Other unexpected events can also occur that forces the vessel to surface, therefore the outer hull must be able to survive the rough surface conditions. As it will operate in the arctic the vessel should also be able to surface through some ice sheets. Some military vessels are strengthened for ice and can surface through up to three metres of ice, while submarines without strengthening generally can penetrate one metre thick ice(39). As the Shtokhman field is not within the permanent arctic ice cap the ability to surface through one metre of ice seems sufficient. The outer hull must also be equipped with hard points and bollards for use during docking, although these can be recessed and covered during transit.

Taking all off this into account it is clear that the outer hull does not need to be as strong as the pressure hull, yet it must be durable enough to survive daily use in the arctic. I propose using composite materials in the outer hull. Composites are corrosion resistant and well suited for creating the complex geometries on parts of the pressure hull. Composites are also quite light, so the outer hull can be made virtually neutrally buoyant. The exact material is yet to be determined, but a thickness of 10 centimetres has been assumed to be enough to create a strong composite hull. It is also assumed to be neutrally buoyant.

7.1.2.2 Structural Components

The weight of all structural components has been estimated using the volume of the components and a standardized steel density of 8000 kg/m^3. The dimensions of most components can be found in section 5. The steel weigh of the bulkheads and main deck are assumed to be the equivalent of five centimetres thick steel plates. Simple geometric formulas could be used to determine the centres of gravity due to the high degree of symmetry and simple geometric shapes.

The weight of the structural components outside the pressure hull such as stiffeners and bracings was assumed to be equal to five percent of the combined weight of all other components except the permanent ballast.

7.1.2.3 Other Components

It is no surprise that the structural components provide the bulk of the displacement, while the individual machinery components are marginal in comparison. There are however a few other heavy components, namely the forward hydroplanes, fuel tanks, batteries, propulsion pods, HP air bottles and ballast water transfer piping.

The propulsion pods are in practice streamlined electric motors. The weight was estimated by simplifying the shape to two cones mounted end to end using the maximum diameter and half length of the pod. The density of this double cone is assumed to be 8000 kg/m^3. As an electric motor is not a single block of metal, thus this estimate will likely be too high. It is therefore assumed to include the weight of the aft hydroplane and hydroplane deflection actuator (assumed to be electric, but with a hydraulic emergency drive). The simplification is illustrated along with pod measurements on Figure 28.

$$m_{pod} = \rho \cdot V_{pod} = \rho \cdot 2 \cdot \frac{\pi {D_{max}}^2 L_{half\ max}}{12} = 8000\ {}^{kg}/_{m^3} \cdot \frac{\pi \cdot (1m)^2 \cdot 1{,}32m}{6} = 5529\ kg$$

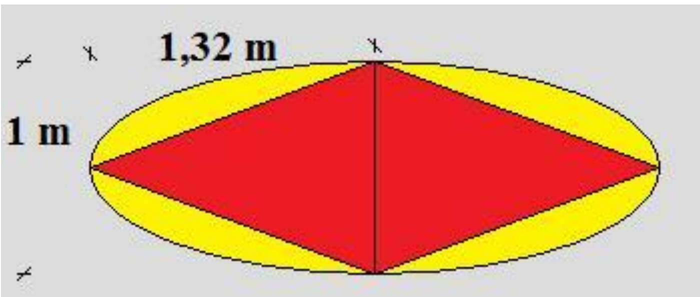

Figure 28 Conical pod simplification

The forward hydroplanes are slightly smaller than the aft hydroplanes and do not have propulsion pods attached. They are however retractable, and the retracting mechanism will require framework and actuators. The forward hydroplanes are therefore assumed to weigh as much as the aft hydroplanes and propulsion pods.

The fuel tanks are subjected to an overpressure and therefore require pressure bearing shells. The thicknesses of these were estimated by using the formula based on the von Mises thin wall yield criterion developed in section 5.2. The resulting thickness to average radius ratio is less than 0,1, hence thin wall is considered to be accurate. The thin wall approximation is also used to estimate the volume of the cylindrical shell. The ellipsoidal heads were simplified to flat disks. The material is assumed to be steel with a density of 8000 kg/m^3. Insulation materials are often very light; hence the insulation weigh is also neglected. The results of the tank wall thickness calculations are listed in Table 51.

Table 51 Summary of the wall thickness calculations for the fuel tanks

Parameter	
Young's modulus	210 [GPa]
Yield strength	250 [MPa]
Internal pressure	1 [bar]
Operating depth	500 [m]
Von Mises safety factor	2 [-]
Expernal pressure	50,3 [bar]
Estimated lateral pressure tank thickness	25,7 [mm]
t/r	0,034 [-]
Estimated main pressure tank thickness	51,4 [mm]
t/r	0,034 [-]

Each HP air bottle holds just under approximately 90 kg air (see Table 22). The type of bottle used is the same as the ones used by the Royal Navy, and so the dimensions and capacity is stated in a Defence Standard (40). Figure 29 displays one such bottle. As the defence standard was very specific, the bottles could be modelled accurately. By using volume measurements from the Inventor model and the density specified in the defence standard (7,89g/cm^3), the weight of one bottle is found to be approximately 5,5 Te.

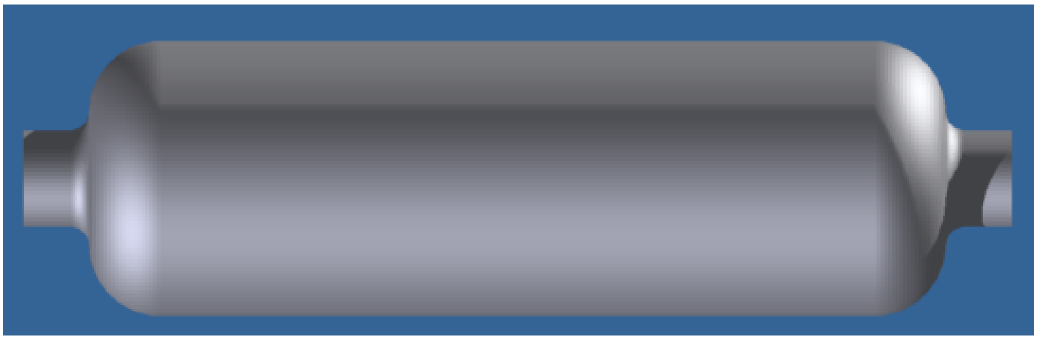

Figure 29 High pressure air storage cylinder

The weight of piping was estimated by using the same wall thickness formula as was used on the fuel tanks. The internal diameter of the pipes was estimated by imposing the 2 m/s flow speed limit recommended to avoid erosion (16). The fuel piping flow was so low that the weight was negligible, thus smaller pipes were included by imposing pipe weight margins. The ballast transfer pipes have to withstand the full diving pressure as they connect hard trim tanks mounted outside the pressure hull. They also have to accommodate a large flow. The distance between the forward and aft trim tanks is approximately 60 metres, while the piping length was estimated 30% longer than this to account for the inevitable twists and turns needed to fit the piping inside the hull. The centre of gravity is assumed to be located in the vessel origin, which probably is fairly accurate as the vessel origin is more or less right between the forward and aft trim tanks. The central trim tanks are located near the keel line, assuming that the vertical centre of gravity is at the vessel origin is therefore conservative. The material used is assumed to have the same properties as the steel used in the fuel tanks. The calculation results are found in Table 52.

Table 52 Trim ballast transfer piping weight calculation

Average diameter	564,3 [mm]
Pipe wall minimum thickness	9,6 [mm]
t/r	0,009 [-]
Wall thickness with a slight margin	10 [mm]
Piping length estimate	84 [m]
Piping weight	12 Te

A summary of the weight and location of major components are presented in Table 53 along with the resulting vessel centres of gravity. Please note that the permanent ballast is placed to fulfil its role and ensure that the longitudinal centre of gravity is co-located with the longitudinal centre of buoyancy, thus defining this as the natural condition of the vessel. The different loading conditions and trim capabilities of the vessel will be further explored in section 6.2. A complete list including the machinery components can be found in Appendix III: Weight.

Table 53 Weight distribution summary

Weight class	Weight [Te]	Vertical CG [m]	Longitudinal CG[m]	Transverse CG [m]
Structural Components				
Cargo hold bulkheads	17,3	0	-20,5	0
Main hull steel plates	161,21	0	12,5	0
Main hull stiffeners	30	0	12,5	0
Conical section steel plate	8	0	25,2	0
Conical stiffeners	3,71	0	25,2	0
Airlock steel plates	15,17	0	28,3	0
Airlock stiffeners	1,93	0	28,3	0
Main pressure hull ellipsoidal end cap	10,6	0	-0,7	0
Airlock ellipsoidal end cap	2,2	0	30,5	0
Main decks	45	0	12,5	0
Main bulkheads	31,4	0	12,5	0
External structural bracings	50,7	0	0	0
Permanent ballast	118,7	-4,0	12,5	0
Miscellaneous components				
Forward hydroplanes	11,1	-2,0	28,3	0
Fuel module tank steel weight	63,9	-	-	-
Fuel module LH_2 tanks	132,6	1,6	-10,0	0
Fuel module LO_2 tanks	208,3	-1,6	-10,0	0
Lateral tank steel weight	24,9	-	-	-
Lateral LH_2 tanks	51,9	2,5	10,0	0
Batteries	200	-3,5	11,0	0
Main propulsion pods	11,1	0	-33,7	0
HP air bottles	27,3	0	23,9	0
Ballast transfer piping	12	0	0	0
Standard vessel weight/ resulting vessel centres of gravity	**1182,5**	**-0,59**	**4,52**	**0**

7.2 Trim Capabilities

7.2.1 The Equilibrium Polygon

The equilibrium polygon is a graphical illustration of the trim capabilities of the vessel. The displacement is plotted on the y-axis and longitudinal moment on the x-axis. A positive moment would lower the bow relatively to the stern. All combinations of displacement and net moment that lie within the trim polygon can be compensated for by the trim and compensating (T & C) tanks. This can also be used to dimension the T & C tanks. The sign convention used in moment calculations and thus in the equilibrium polygon is as follows:

- The loading conditions present the weight of the vessel, and a positive net longitudinal moment in Table 54 would cause the stern to rise and bow to sink.
- The equilibrium polygon itself is an expression for the buoyancy capabilities of the vessel, hence a positive moment would cause the bow to rise and stern to sink.

The approach when dimensioning the T & C tanks is fairly simple:

1. Determine which weights that are constant and which that are variable. The same is done with the buoyancy (the airlock can for instance be viewed as a section with variable buoyancy).
2. Determine different loading conditions by varying the variable weights and buoyant sections. The density of the water can also be varied. The loading conditions are plotted in the diagram below and form a minimum requirement for the T & C tanks.
3. The trim polygon is established by first emptying the trim tanks successively, starting with the forward trim tank, plotting the resulting displacement and moment. Once all tanks are emptied the tanks are successively filled again, starting with the forward tank. The equilibrium polygon is formed by connecting these points. The size and location of the T & C tanks can then be established through an iterative process.

7.2.2 Loading Conditions

The vessel has four basic operational modes which were divided into three sub-conditions; standard, light and heavy. In the standard condition the vessel is fully stocked with consumables and floating in seawater with a density of 1025 kg/m3. In the extreme light condition all consumables are spent and the vessel is floating in dense water (1035 kg/m^3). Reversely the vessel retains all consumables while floating in low density water (1015 kg/m^3) in the heavy condition. The standard condition is only applied to the normal mode and payload mode as they closely resemble the loading condition at the start of any mission. Inspection mode and installation mode will however only be entered after transit to the field, hence the vessel will be somewhere between the extreme light and heavy conditions. The standard condition is therefore not applied to those operational modes. The respective masses, buoyancies and moments of the different loading conditions are listed in Table 54.

The different loading conditions are

- Normal mode: The majority of the missions are expected to be pure inspection missions; the normal mode is therefore recognized as a vessel without any payload in the cargo hold and an airlock that is free of water.
- Payload mode: The payload mode is equal to the normal mode with one exception; a payload of 20 Te is included. The added mass and drag of the crane design load bar is also included.
- Inspection mode: This is the expected state while performing structural inspection. The airlock is flooded and the ROV outside inspecting.
- Installation mode: This mode is equal to inspection mode with one exception; a payload of 20 Te is included. The added mass and drag of the crane design load bar is also included.

Table 54 Loading conditions summary

Loading condition	Vessel mass [Te]	Vessel buoyancy [Te]	Net longitudinal moment [Te·m]
Normal	1182,5	1182,5	-2,3
Light normal	1081,5	1194,0	761,5
Heavy normal	1182,5	1170,9	49,9
Standard payload	1234,6	1182,5	-1124,6
Light payload	1133,7	1194,0	-360,8
Heavy payload	1234,6	1170,9	-1072,4
Light inspection	1079,0	1154,3	1833,9
Heavy inspection	1180,0	1132,0	1100,2
Light installation	1031,2	1154,3	711,6
Heavy installation	1132,1	1132,0	-22,1

7.2.3 Trim & Compensation Tanks

The vessel is fitted with seven trim tanks; a forward trim tank, two central compensation tanks, three cargo compensation tanks and an aft trim tank. Only the forward trim tank and the central compensation tanks are directly connected to deballasting pumps, while the trim ballast transfer pump handles the flow in and out of the aft tank and cargo compensation tanks. The forward and aft trim tanks and the bottom cargo compensation tank are placed on the centre line, and can therefore only be used to change the trim of the vessel. Two of the cargo compensation tanks and the central compensation tanks are however placed to the sides of the vessel, and can therefore also compensate to a certain degree for heeling moments as well. Positioning components in the cargo hold so that their centres of gravity are centred perfectly on the centre line is unrealistic; hence some heeling moment must be expected. Other imbalances might occur due to damage. With the current T & C tanks such imbalances can be countered without the use of thrusters while stationary or hydroplanes while moving. The layout of the tanks is shown in Figure 30 and Figure 31 while their sizes are listed in Table 55.

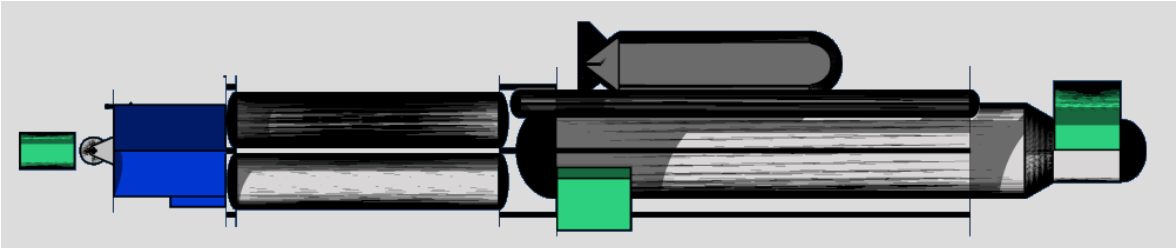

Figure 30 T & C tanks shape and placement seen from the side. Trim tanks are marked in green, cargo compensation tanks in blue.

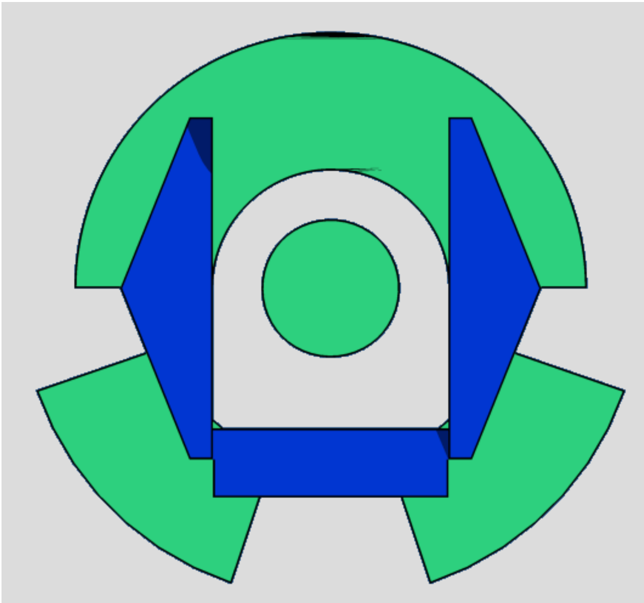

Figure 31 T & C tanks shape and placement as seen from aft. Trim tanks and cargo compensation tanks are again shown in green and blue.

Table 55 T & C tanks summary

Tank type	Longitudinal distance from centre point [m]	Volume [m^3]
Forward trim tank	28,5	60
Central compensation tanks	2	100
Bottom cargo compensation tank	-19,6	10,2
Lateral cargo compensation tanks	-20,5	49,64
Aft trim tank	-26,6	6,3

The loading conditions and equilibrium polygon are plotted in Figure 32. The chosen trim tank configuration is able to compensate for even the most extreme conditions envisioned. There is also a fairly large margin of safety.

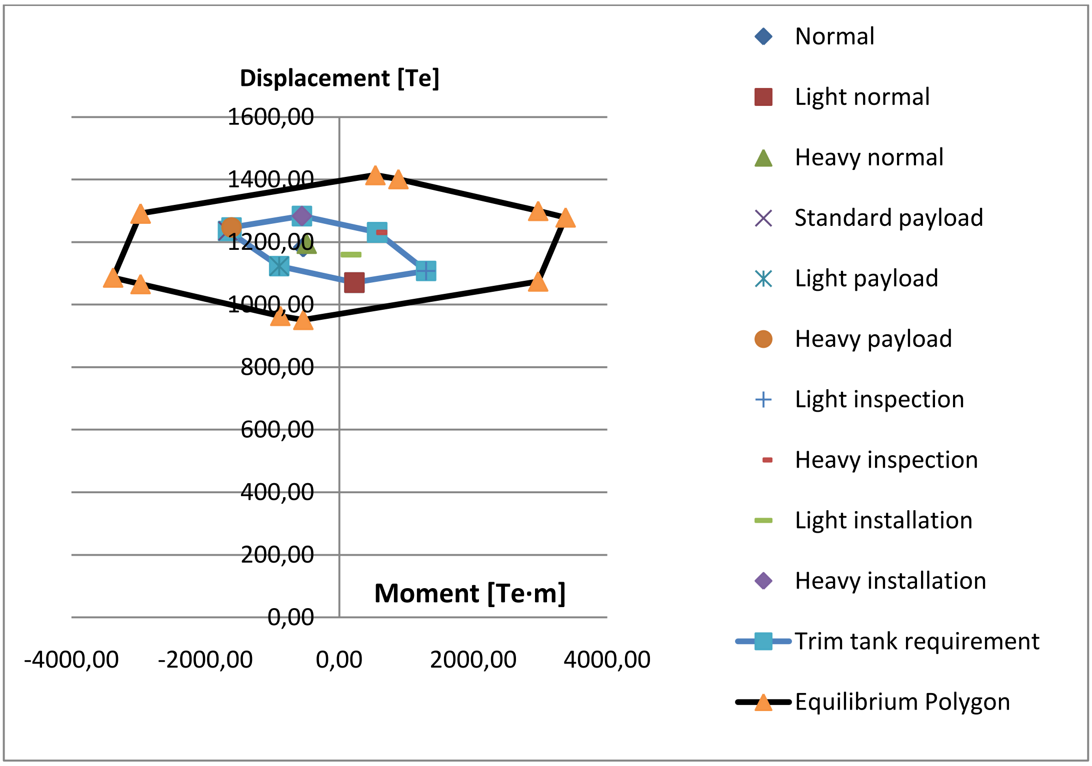

Figure 32 Equilibrium polygon

7.3 Main Ballast Tanks

The size of the main ballast tanks is very much up to the designer, and the relative size is often expressed through the "reserve of buoyancy"; the ratio between the volume of the main ballast tanks and the pressure hull volume. The buoyancy reserve is usually between 0,1 and 0,15 (4). However, as the vessel was expected to have a fair amount of equipment mounted outside the pressure hull, the buoyancy reserve was set to 0,25 during the initial estimates. This results in a total MBT volume of 135,6 m^3. The vessel as a whole might have grown considerably in weight and volume, but the pressure hull remains approximately the same size. The vessel will also be neutrally buoyant with full MBTs; there are therefore no reasons to drastically increase the MBT volume. The MBTs are however necessary to ensure adequate surface stability, but this is rather a question of how to shape the MBTs rather than how large they must be.

The MBTs are to be soft tanks and are thus not required to be very strong. They are placed along the sides of the pressure hull in such a way that they contribute to the surface stability (see section 7.4.2). The tank dimensions and placement can be seen in Figure 33 and Figure 34. The MBTs are coloured black for easy identification. The square grooves in the forward MBTs are there to accommodate the placeholder external longitudinal stiffeners.

Figure 33 Side view of the main ballast tanks

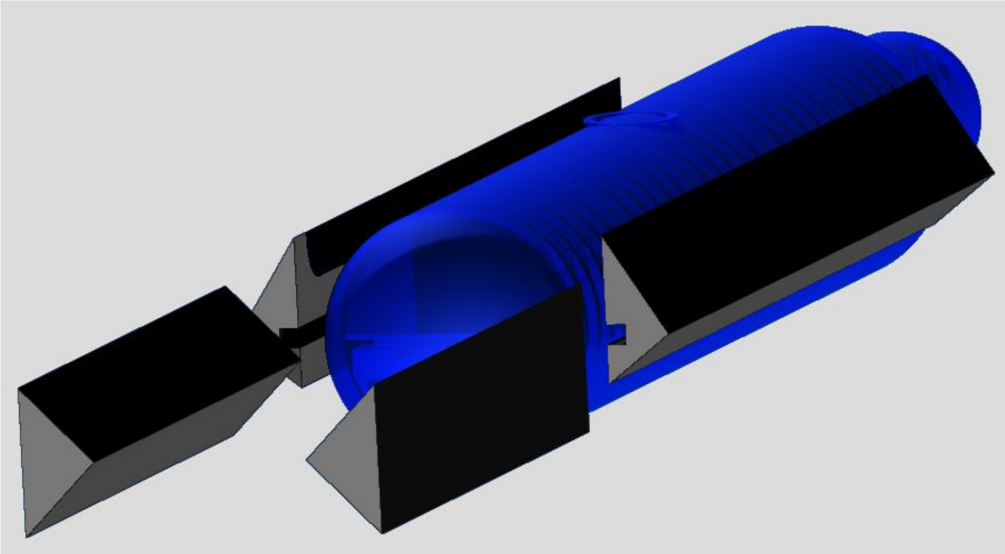

Figure 34 Stern quartering view of the main ballast tanks

7.4 Stability

7.4.1 Submerged Stability

The lack of waterline area makes the submerged stability of the vessel purely dependant on the distance between the centre of buoyancy and the centre of gravity. The ABS regulations state that:

*For all normal operational conditions of loading and ballast, the center of buoyancy is to be above the center of gravity by a distance **GB** which is the greater of either 51 mm (2 in.) or the height as determined below:*

$$GB_{min} = \frac{nwNd}{W \tan \alpha}$$

Where

n=0,1 (this represents10 percent of the people aboard moving simultaneously)

w=79,5 kg (175 pounds)per person (for passenger submersibles, w may be taken as 72,5 kg (160 pounds) per person)

d=interior length of the main cabin accessible to personnel, in mm(in.). This should not include machinery compartments if they are separated from the main cabin with a bulkhead.

N=total number of people onboard the submersible.

W=total weight (in units consistent with w) of the fully loaded submersible, not including soft ballast.

α=25 degrees (representing the maximum safe trim angle. A smaller angle may be required if battery spillage or malfunction of essential equipment would occur at 25 degrees. This assumes that each person has an individual seat that is contoured or upholstered so that a person can remain in it at this angle).

The free surface effect (FSE) from the fuel tanks and water tanks must also be taken into consideration. A free liquid surface causes an effective raise of the centre of gravity of the tank from g to g'. The influence on the overall centre of gravity of one free surface is described by the following formula:

$$\overline{gg'}_n = \frac{\rho_n \cdot i_n}{\Delta}$$

ρ is the density of the fluid in tank n, i is the sectional inertia of the free surface area in tank n and Δ is the total vessel displacement. The combined effect from all tanks can then be expressed as the following sum:

$$\overline{GG'} = \sum \frac{\rho_n \cdot i_n}{\Delta}$$

The FSE has been calculated for the worst case scenario for each tank, i.e. all fuel tanks are half empty as is the hotwater tank, while the waste water tank is almost empty. The sectional inertia is calculated in both the transverse and longitudinal directions. The stability calculation results are given in Table 56.

Table 56 Stability summary

Interior length of the main cabin accessible to personnel	25 [m]
Total number of people onboard the submarine	15 [persons]
Standard displacement	1182,5 [Te]
Minimum operational GB	0,76 [m]
LO_2 tank longitudinal FSE w/o internal subdivision	0,34 [m]
Transverse GG'	0,017[m]
Longitudinal GG'	0,24 [m]
GB corrected for worst case FSE	0,52 [m]
ABS absolute GB minimum requirement	0,051 [m]
ABS formula GB minimum requirement	0,005 [m]

The ABS rules are intended for passenger submersibles used for underwater sightseeing rather than offshore intervention submarines. The cabin length was therefore taken as the full length of the pressure hull, including the machinery spaces. Even so the absolute minimum GB requirement was larger than the formula requirement by an order of magnitude. The formula requirement was therefore only calculated for the standard loading condition, as the differences in displacement between the different loading conditions are marginal. The smallest GB is attained in the "Light ROV operation with cargo"-loading condition, the origin of which was described in section 6.2.2.

The FSE from a single LO_2 tank is almost as large as the smallest vessel GB which of course is unacceptable in the final design. The FSE can however be reduced to a manageable level by introducing two internal bulkheads in the LO_2 tanks.

7.4.2 Surfaced Stability

The vessel is expected to have the poorest stability during the early stages of surfacing and late stages of submergence. This is due to the loss of buoyant volume, thereby lowering the centre of buoyancy without a significant waterline area providing metacentric height. The surface stability and stability during surfacing/submergence are best investigated by using software to calculate the changes in the position of the centre of buoyancy. These calculations were not performed as such software was not available. Performing the calculations by hand would be very time consuming and at the cost of other aspects of the vessel design and were therefore not performed. The lateral LH_2 tanks were however positioned high on the pressure hull in order to provide the vessel with an increased waterline area and metacentric height in these critical stages.

In order to control that the surface stability was not totally unacceptable a very simplified analysis was performed. The waterline is assumed to be at the vessel xz-plane (see Figure 27). Only the main pressure hull cylinder and the MBTs are assumed to contribute to the waterline area. The worst case transverse FSE is also taken into account in order to ensure conservatism. The simplified water line area is displayed in Figure 35. The following formulas were used along with recalculations of the centres of buoyancy and gravity for the relevant water line:

$$G'M = BM + GB - GG'$$

$$BM = \frac{I_{WL}}{\nabla}$$

GM is the distance between the metacentre and the centre of gravity, GG' is the free surface effect correction to the centre of gravity, GB is still the distance between the centres of buoyancy and gravity. BM is the distance between the centre of buoyancy and metacentre. I_{WL} is the waterline area of section and ∇ is the standard volumetric displacement.

Table 57 Simplified surfaced stability

Waterline moment of section	1278 [m^4]
BM	1,11 [m]
GB	-0,55 [m]
GG'_T	0,017 [m]
$G'M_T$	0,54 [m]

The results of the simplified surface stability analysis are listed in Table 57. Although the case in question is completely unrealistic, the positive (and relatively large) transverse GM demonstrates that the vessel should be stable on the surface. If the draught is deeper the area from the pressure hull and main ballast tanks will decrease, but more area will be gained by the submergence of the main hydrogen tanks. The vessel is therefore assumed to have sufficient surface stability, though this must be verified in a later study.

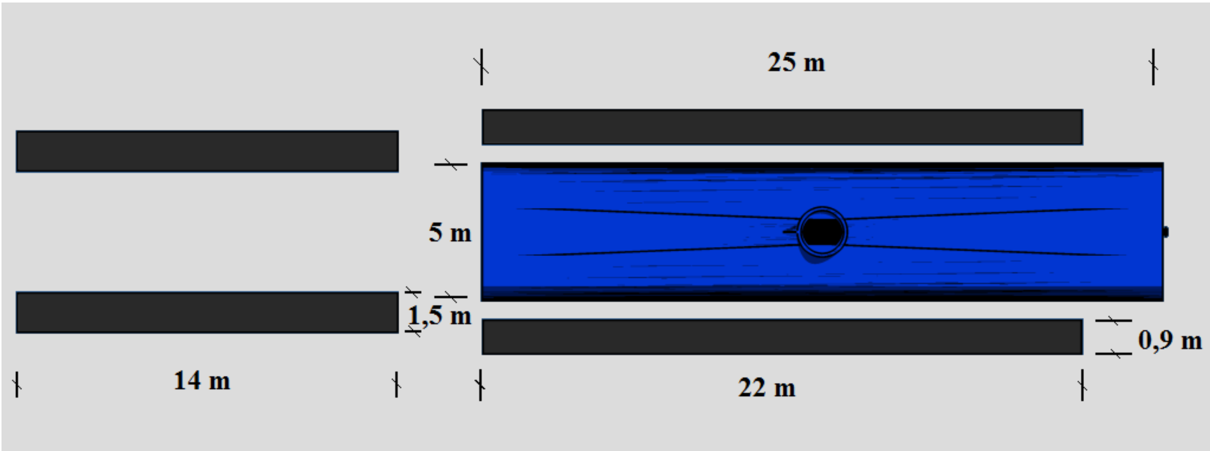

Figure 35 Simplified water line area

8 Resistance & Manoeuvring

The previous chapters can all be said to constitute the first lap in the design spiral (though logistic and economic aspects so far have been overlooked), resulting in a vessel with a certain shape, size and capability. This chapter represents the start of a new lap in the design spiral as the operational requirements have not changed. The results of the following calculations will be used to modify the capabilities of the vessels “as is”; performing a redesign is too time consuming and is therefore the subject for later work.

8.1 Control Surfaces

In order to maintain control of any vessel control surfaces are needed. While surface vessels only need to navigate a two-dimensional surface, a submarine must be able to maintain directional control in three dimensions. Once the vessel is submerged the rudder provides control in the horizontal plane while the hydroplanes provide control in the vertical plane. Because the propulsors are twin propulsion pods the only realistic rudder/hydroplane configuration option was the cruciform configuration.

It is not desirable to have rudders or hydroplanes that extend beyond the vessel keel plane or beam, as this will be problematic during mooring. This is the main reason why the forward hydroplanes are retractable, but not the only.

Submarines are vulnerable to a phenomenon known as “the Chinese effect”. The Chinese effect is caused by the movement of the so-called critical point, also known as the trim point (41). As speed is reduced the critical point will move aft, and when it passes the aft hydroplanes the controls are reversed, i.e. a hydroplane deflection which at high speeds cause an upward motion will now cause a downward motion. Additionally, any upward or downward force applied at the critical point will only cause a change in trim, not depth. This phenomenon usually occurs around two knots (4), and so the ability of the aft hydroplanes to control depth is reduced at low speeds. Forward hydroplanes are always ahead of the critical point and are therefore unaffected by this, which is why using them offers depth control even at low speeds.

There is also the neutral point to consider. This point is not dependent on speed and is located far forward. Any vertical force applied at the neutral point will not cause a change in trim, only depth. Consequently, the further away from the neutral point a vertical force is applied the larger the effect on the trim. At speed angling the submarine up or down and use the propulsive power of the vessel to change depth is much more efficient than changing depth by means of a lift force applied on the hydroplanes. The great distance between the aft hydroplanes and the neutral point renders the effect of forward hydroplanes negligible at speed. The forward hydroplanes thus become nothing more than a drag-inducing appendage, which is the other incentive to make them retractable. The critical point and neutral point placements are illustrated on Figure 36.

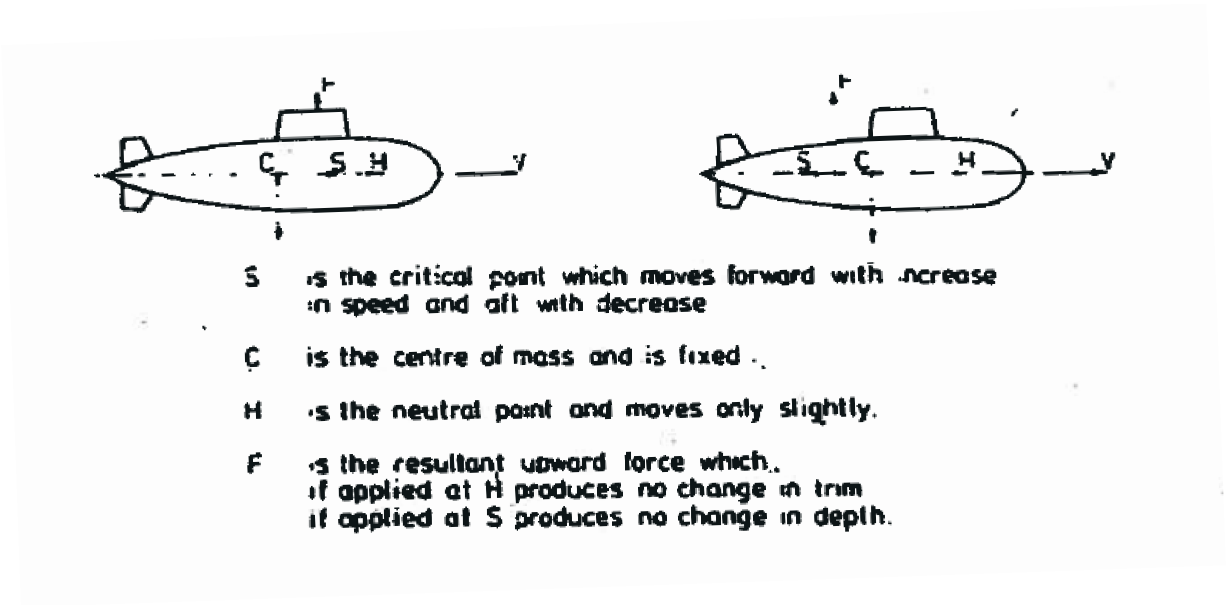

Figure 36 Critical and neutral point positions (41)

The required rudder and hydroplane surface area is of course dependant on the required performance, usually expressed as a minimum turning circle diameter and minimum vertical speed at given speeds and control surface deflection angles. Such analyses are extensive, and determining the optimum rudder and hydroplane size and shape is probably a comprehensive enough study to by itself justify a separate master's thesis. There are some rules of thumb ratios available for the initial sizing of the control surfaces. These are based on usual values on attack submarines (14):

- $\frac{S_{rudder}}{\nabla_{sub}^{\frac{2}{3}}} \approx 0{,}075$, where S_{rudder} is the rudder area and ∇ is the volumetric submerged displacement
- $\frac{S_{aft}}{\nabla_{sub}^{\frac{2}{3}}} \approx 0{,}05$, where S_{aft} is the aft hydroplane area
- $\frac{S_{fwd}}{\nabla_{sub}^{\frac{2}{3}}} \approx 0{,}04$, where S_{fwd} is the forward hydroplane area

The resulting control surface areas using the standard normal condition displacement are summarized in Table 58. The forward hydroplanes are placed just underneath the hull centre line. The placement will result in a slight drag moment which must be countered by a small plane deflection. They were placed there in order to ensure enough space for the retraction mechanism.

Table 58 Control surfaces summary

Appendage	Minimum area	Modelled area
Rudder fins [m^2]	8,25	12,8
Aft hydroplanes [m^2]	5,50	5,76
Forward hydroplanes [m^2]	4,40	5

8.2 Propulsion Resistance

8.2.1 Basic Submarine Resistance Theory

A major advantage submarines have over surface vessels is the loss of wave-induced resistance when adequately submerged. The wave resistance is the largest resistance component by far for displacement vessels (42). A submerged submarine will therefore be able to reach much higher speeds than a surface vessel of similar displacement and with the same propulsion power. The main remaining resistance is caused by viscous effects, namely friction and pressure differentials which are commonly known as drag effects. The 1957 ITTC friction line was used to determine the friction resistance as per the procedure outlined in Appendix 5 in Concepts in Submarine Design (4), while the form resistance (caused by the pressure differentials) was calculated by using Droblenkov's coefficient to relate the friction resistance to the form resistance(14). These calculations will be further described in section 7.2.2. Droblenkov's coefficient was determined from the curves in Figure 37.

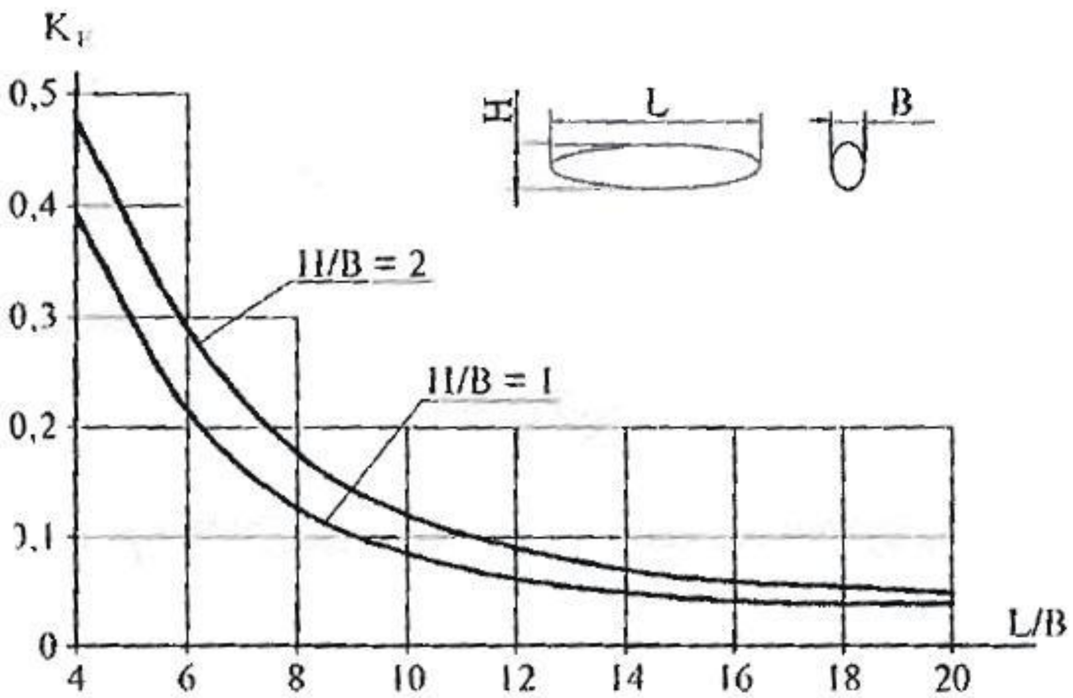

Figure 37 V.F. Droblenkov's curves of design values of the K_f coefficient for streamlined bodies depending on their relative elongation L/B and aspect ratio H/B (14)

The optimal hull shape for submerged vessel has long been recognized to be a teardrop-shaped axisymmetric body of revolution with a tri-axial ellipsoid bow section. The fullest section is one third of the overall length from the bow, with the stern tapering down to a sharp end with a half apex angle between 8-12 degrees. This is illustrated in Figure 38. This shape has favourable pressure gradients at the bow and stern which gives a low from resistance. This shape is of course difficult to achieve in practice due to the size and shape of internal components. The external hull would be very large in order to fit the internal structure within a "perfect" hull, resulting in a much greater surface area and frictional resistance.

This will probably negate the gain from having a “perfect” hull rather than a slightly imperfect hull, if not increasing the overall resistance. Appendixes such as the sail, rudders and hydroplanes also add resistance. Adding a cylindrical midbody increases the form resistance, but for a fixed hull volume the vessel length and total surface area is reduced. This can lead to a larger reduction in friction resistance than increase in form resistance (14) . The shape of the outer hull should therefore strive to attain the optimal shape, but not at any cost.

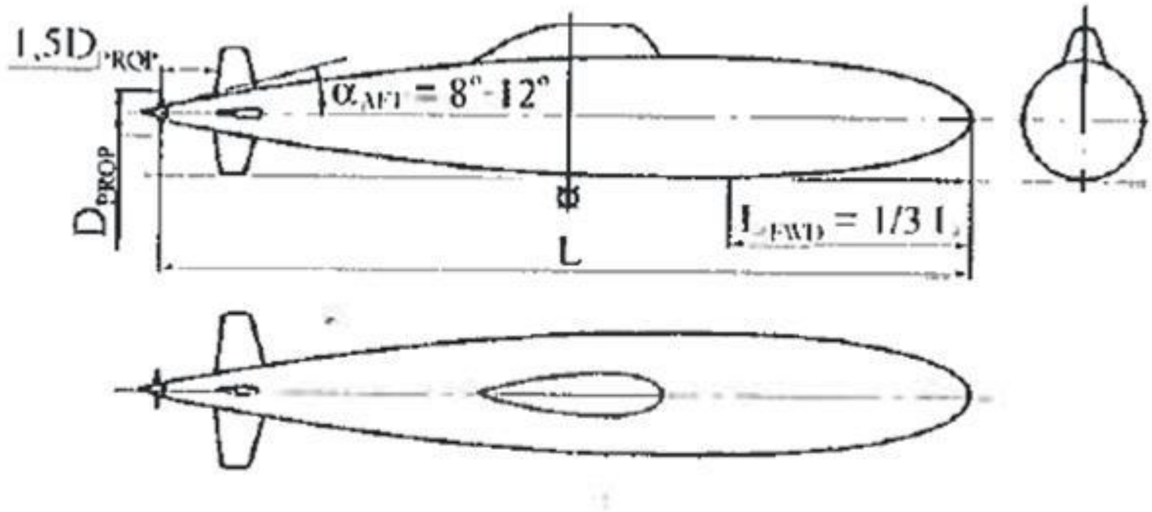

Figure 38 Submarine optimal hull shape (Kormilitsin & Khalizev, 2001)

8.2.2 Outer Hull Shape

The two largest sections, the pressure hull and main fuel module, are long constant diameter sections. A 43,1 metre long cylindrical midbody is used to accommodate these sections. The bow section is a 10,1 metre long tri-axially symmetrical ellipsoid. The 18,1 metre stern section is vaguely teardrop shaped and was designed with space allowances so that rudder and hydroplane actuators can be fitted at a later stage. The optimal shape is not used as the aft section would have to be much longer and have a much larger whetted area. The vessel has a rather large sail necessary to streamline the flow around the escape vehicle.

I would like to stress that this is but a preliminary design whose main functions are to identify potential problems with the design and to provide the grounds for a more accurate resistance calculation than that of the initial estimates.

8.2.3 Resistance Calculations

The total resistance of the vessel is, as mentioned earlier, comprised of two components, frictional resistance and form resistance. The latter is expressed as a function of the frictional resistance through the following relation(14):

$$R_f = K_f \cdot R_{friction}$$

The coefficient K_f is determined by using the curves in Figure 37. This relation simplifies the expression for the total resistance.

$$R_{total} = R_{friction} + R_f = R_{friction}\left(1 + K_f\right)$$

The friction resistance is proportional to the whetted area of the hull, water density, the square of the vessel speed and the friction coefficient. The 1957 ITTC friction line is used to determine the friction coefficient for each component.

$$C_{friction} = \frac{R_{friction}}{\frac{1}{2} \cdot \rho \cdot S_{whetted} \cdot V^2}$$

ITTC '57 :

$$C_{friction} = \frac{0{,}075}{(\log(R_N) - 2)^2}$$

R_N is the Reynold's number where V is the flow speed, υ is the kinematic viscosity of the fluid and L is the characteristic length of the object in question.

$$R_N = \frac{V \cdot L_c}{\upsilon}$$

The characteristic length of the vessel is of course not equal to that of the appendages, leading to different friction coefficients for the different components. This is taken into account by adding the different component coefficients after weighting them according to their contribution to the total whetted area (4). This produces the following expression for the total friction coefficient:

$$C_{friction_{total}} = C_{friction_{bare\ hull}} \cdot \frac{S_{bare\ hull}}{S_{whetted}} + C_{friction_{sail}} \cdot \frac{S_{sail}}{S_{whetted}}$$

$$+ C_{friction_{propulsion\ pod}} \cdot \frac{2 \cdot S_{single\ propulsion\ pod}}{S_{whetted}}$$

$$+ C_{friction_{aft\ plane}} \cdot \frac{S_{single\ aft\ plane}}{S_{whetted}} + C_{friction_{rudder}} \cdot \frac{2 \cdot S_{single\ rudder}}{S_{whetted}}$$

The bow planes are not included as they will be retracted at higher speeds. Once the total resistance is determined the required power from the power plant can also be determined:

$$P_{motor} = \frac{P_P}{\eta} = \frac{R_{total} \cdot V}{\eta_0 \cdot \eta_H \cdot \eta_s} = \frac{(1 + K_f) \cdot C_{friction_{total}} \cdot \frac{1}{2} \cdot \rho \cdot S_{whetted} \cdot V^3}{\eta_0 \cdot \eta_H \cdot \eta_s}$$

The surface areas and characteristic lengths are measured on the 3D model. The proposed value for standard seawater kinematic viscosity is from the resistance and propulsion reader used in the course TMR4247 (42). Because the placement of the propulsors is far from the hull, the combined hull and propulsor efficiency is increased to 0,75. The resistance calculation results are summarized in Table 59. The different lengths are explained in Figure 39.

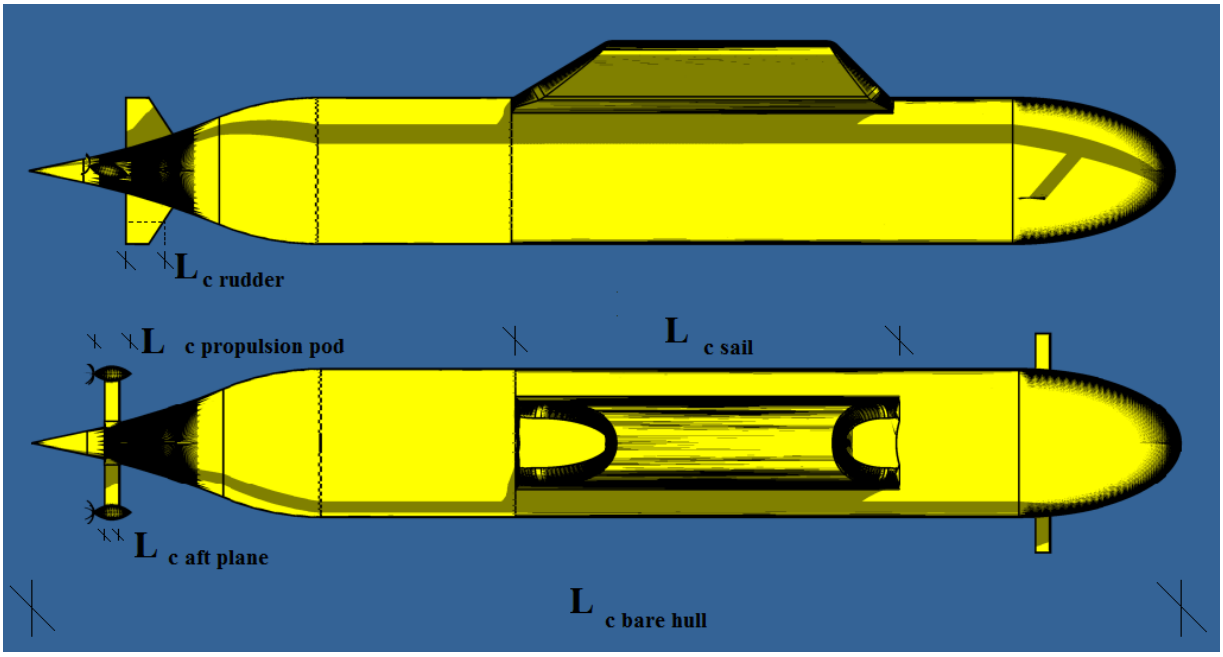

Figure 39 Aspect lengths

The required propulsion power increased with 41,8% (68,7 kW) compared to the early estimates. This is a significant increase; however it was not completely unexpected as the formula used in the early estimates is intended for military submarines without large volumes outside the pressure hull. This vessel does not only have a cargo hold, but more importantly uses a fuel whose volumetric energy density is very small compared to ordinary fuel oil. The net result is that the form volume is much larger than initially anticipated, thus increasing whetted area and resistance.

There are three main approaches to dealing with the resistance increase. The obvious and simple solution is to reduce the transit speed since propulsion power is proportional to the cube of the speed. If the power allotted to propulsion is kept at the initially estimated level the speed is reduced by two knots. The one way transit time increases with 11 hours and 42 minutes and requires 1918 kWh more energy. The vessel carries enough energy for a speed reduction when considering both the batteries and fuel, but there will hardly be any fuel storage safety margin. Increasing fuel storage volume should be considered if the speed is reduced. The battery load and capacity must also be revised due to the rather large increase in resistance. While there is no question that the battery rack can support the new propulsion load, the performance calculations performed in section 4.5.4.2 must be revised.

The second option is to tweak the hull shape and thus hope to reduce the form resistance. The by far largest resistance component is however the frictional resistance. This is best reduced by reducing the whetted area, i.e. reducing the size of the vessel. The major potential for size reduction is in using a fuel with higher volumetric energy density. This would also require a revision of the power plant.

A combination of the two is recommended when further developing the design, though a hull shape tweaking rather than a complete redesign probably will save a lot of time. The object is to reduce the resistance to allow a higher transit speed with the engine power allotted in the load analysis. The propulsors must therefore be able to deliver a combined thrust power of at least 120 kW. The pods are also to be used by the dynamic positioning system, subjecting them to a new set of requirements. These will be explained in section 8.3. The end result was that the chosen propulsors have a combined maximum thrust of 68,6 kN, giving the vessel a top speed of 8,4 knots.

Table 59 Resistance calculations summary

Length of vessel	71,3 [m]	L
Vessel beam	9,2 [m]	B
Vessel height	9,2 [m]	H
L/B	7,6 [-]	-
H/B	1 [-]	-
Target transit speed	8,2 [knots]	V
Water density	1025 [kg/ m^3]	ρ
Water kinematic viscosity	$1{,}2 \cdot 10^{-6}$ [m^2/s]	υ
Hull characteristic length	71,3 [m]	$L_{c\ bare\ hull}$
Sail characteristic length	23,3 [m]	$L_{c\ sail}$
Propulsion pod characteristic length	2,5 [m]	$L_{c\ propulsion\ pod}$
Aft hydroplane characteristic length	1,2 [m]	$L_{c\ aft\ plane}$
Rudder characteristic length	2,25 [m]	$L_{c\ rudder}$
Hull surface area not covered by appendages	1618,5 [m^2]	$S_{bare\ hull}$
Sail surface area	243,1 [m^2]	S_{sail}
Surface area of a single pod	5,8 [m^2]	$S_{single propulsion\ pod}$
Single aft hydroplane surface area	5,8 [m^2]	$S_{single\ aft\ plane}$
Single rudder surface area	12,8 [m^2]	$S_{single\ rudder}$
Total whetted area	1910,3 [m^2]	$S_{whetted}$
Bare hull friction coefficient	$1{,}8 \cdot 10^{-3}$ [-]	$C_{friction\ bare\ hull}$
Sail friction coefficient	$2{,}1 \cdot 10^{-3}$ [-]	$C_{friction\ sail}$
Propulsion pod friction coefficient	$3{,}1 \cdot 10^{-3}$ [-]	$C_{friction\ propulsion\ pod}$
Aft hydroplane friction coefficient	$3{,}5 \cdot 10^{-3}$ [-]	$C_{friction\ aft\ plane}$
Rudder friction coefficient	$3{,}1 \cdot 10^{-3}$ [-]	$C_{friction\ rudder}$
Total friction coefficient	$2{,}1 \cdot 10^{-3}$ [-]	$C_{friction\ total}$
Form resistance coefficient	0,15 [-]	K_f
Form resistance	5,3 kN	R_f
Frictional resistance	35,4 kN	$R_{friction}$
Total resistance	40,7 kN	R_{total}
Transmission efficiency	0,98 [-]	η_S
Hull & propulsor efficiency	0,75 [-]	$\eta_O \cdot \eta_H$
Required propulsor power	171 [kW]	P_P
Required engine propulsion power	232,7 [kW]	P_{motor}
Initial engine propulsion power estimate	164 [kW]	$P_{motor\ initial}$
Difference	68,7 [kW]	-
Required speed to reduce power to the estimate	6,2 [knots]	$V_{reduced}$
New transit time	47,7 [hours]	$T_{new\ transit}$
Increased transit time	11,7 [hours]	-
Flank speed	8,4 [knots]	V_{flank}

8.3 Dynamic Positioning & Thruster Power

The thrusters have two main tasks, maintaining position while using dynamic positioning (DP) and compensating for possible vertical forces during crane operations. These tasks require that the thruster configuration must be able to compensate for currents and/or transient loads in five of the six degrees of motion, as surge is compensated for by the propulsion pods. The thrusters are placed in the bow and stern sections, primarily to generate large moments with minimum thrust although these are also the locations where there is space available. I propose using rim-driven permanent magnet thrusters as these are very compact.

Sway- and yaw motion compensation is required to fine-tune the position and, once properly there, remain in position. The position of the thrusters enables them to generate a roll moment as well, although roll compensation beyond that offered by the trim tanks is not expected to be necessary. The vessel is always to keep the bow against the current in order to minimize the power required by the thrusters. This is not problematic during crane operations as the vessel uses a gantry crane to lower cargo. The vessel can rotate about the hoist wagon and lifting wire without causing much trouble, whereas any rotation would become translatory motions at the crane tip if the vessel was equipped with a boom crane. Yaw corrections are therefore expected to be most common. Performing a yaw correction in a current can be assisted by the rudders to some extent, but pure sway can only be corrected by the thrusters. The thruster capabilities are therefore be expressed as a maximum current flow speed in a beam current. In order to estimate the sway force the hull is simplified to a circular cylinder in a steady incident flow. This is of course far from the truth due to the large sail, rudders and other appendages, but it is a quick and simple method to get a rough idea of the sway force in a beam current. The following drag formula from Sea Loads ch. 6 (43) was used:

$$F = C_D \frac{1}{2} \rho D {U_\infty}^2$$

D is the outer diameter of the hull and $U_\infty$is the incident current flow speed. Falthinsen also presents plots of the drag coefficient variation with increasing Reynolds numbers for different surface roughnesses (Figure 40). The plots seem converge on a drag coefficient of approximately 1 at high Reynolds numbers. Because the hull cross-flow Reynolds number is larger than those plotted by Falthinsen a drag coefficient of 1 is assumed.

The propulsion calculations from section 8.2 were used to determine the maximum current speed when the vessel is in the current-waning DP-mode. The maximum current speeds are listed in Table 61.

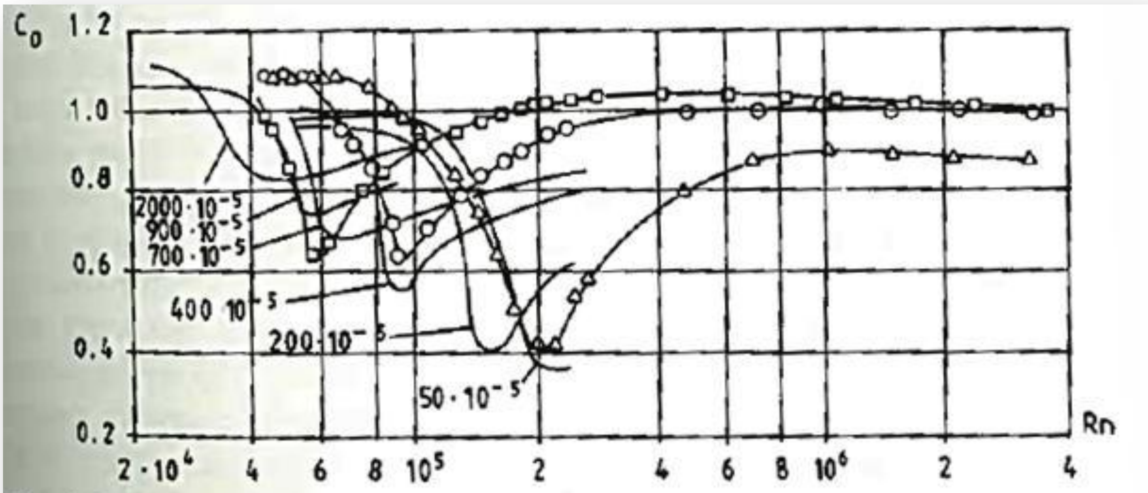

Fig. 6.3. Drag coefficient C_D of rough circular cylinders in steady incident flow for different surface roughness values k/D (k = average height of surface roughness, D = cylinder diameter, $Rn = U_\infty D/\nu$, U_∞ = incident flow velocity.) △, $k/D = 110 \cdot 10^{-5}$; ○, $k/D = 450 \cdot 10^{-5}$; □, $k/D = 900 \cdot 10^{-5}$; ——, Fage & Warsap (1929), (Achenbach, 1971).

Figure 40 Drag coefficient plots(43)

The main task of the vertical thrusters is to counteract hydrodynamic forces on an object being hoisted up by the crane and depth control when the speed is too slow for the hydroplanes to be effective. The flow resistance of the load during the hoist is transferred to the vessel through the crane wire which subjects the vessel to a trim moment. The force and moment from the crane load is counteracted by the trim tanks, but due to the relatively short duration of the hoist and on/off nature of the load resistance complementing the trim tanks with thrusters seems reasonable. This is due to the fast reaction times and simple tuning of the compensation force given by thrusters. Initially thrusters were intended to carry the entire hydrodynamic load, but the added mass of the design load bar would simply require too much power. Even at full power, which far exceeds the allotted thruster power, the selected thruster configuration cannot fully compensate for the hydrodynamic load. The vertical thrusters are therefore mainly used to counter lift force transients during hoists and for fine tuning the depth.

The thrust from each thruster was not supplied by Brunvoll. The thrust was therefore estimated by using the following formula (42):

$$T = \rho \cdot A_P \left[\frac{2 \cdot P}{\rho \cdot A_P}\right]^{1/3}$$

A_P is the propeller disk area, i.e. the cross sectional area of the thruster tunnel, ρ seawater density, T is the thrust and P is the propulsion power.

The vessel has six tunnel thrusters, four mounted in the bow and two in the stern. The stern thrusters are horizontal plane thrusters, the two azimuthing propulsion pods can be rotated in order to provide vertical thrust. Rotating the pods will reduce the forward thrust, thus 45˚ is set as the maximum tilt angle in order to maintain forward thrust. The usage of azimuthing pods rather than tunnel thrusters was chosen due to a lack of available space in the stern section. The tunnel thrusters positions are indicated in black on Figure 41.

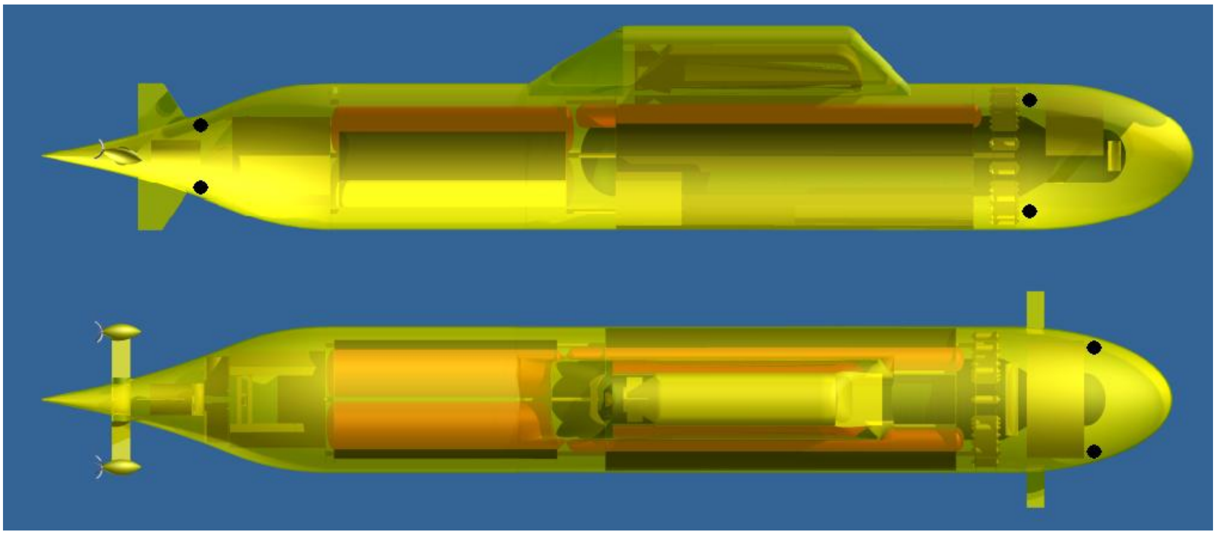

Figure 41 Tunnel thruster positions

The smallest Brunvoll rim driven thrusters (RDT) are assumed to be used in both the tunnel thrusters and the propulsion pods. The pods are modeled as ordinary propulsion pods and not RDT pods simply because the RDT concept was discovered late in the thesis work. RDTs seem well suited for both applications. It reduces the whetted surface of the propulsion pod, and is very easy to fit inside the hull as they eliminate the need of an external thruster motor. The same RDT size is used in both the tunnel thrusters and pods. An RDT pod as used on a Norwegian ferry is displayed on Figure 42.

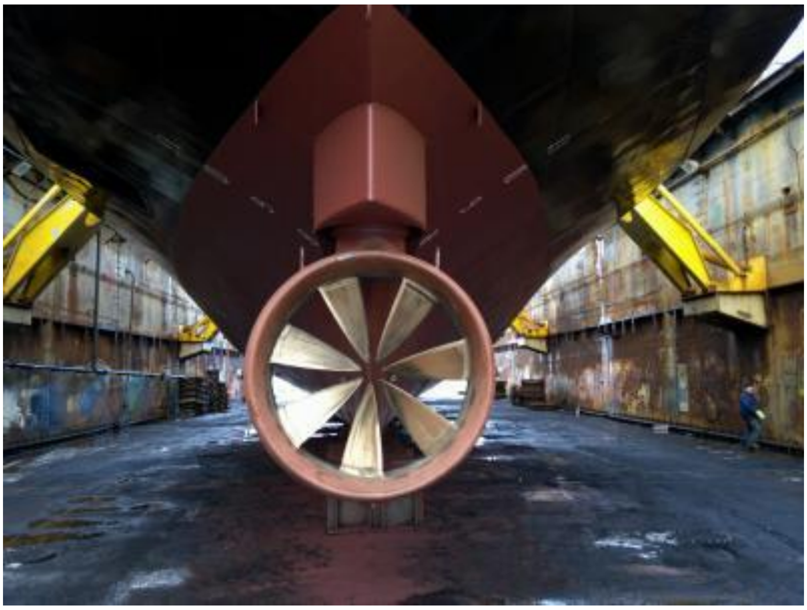

Figure 42 RDT propulsion pod (44)

Table 60 Thruster & propulsor summary

Parameter	Value
RDT propeller disk diameter	800 [mm]
Maximum RDT thrust power	200 [kW]
Assumed RDT total efficiency	0,7 [-]
RDT fluid power	140 [kW]
Maximum RDT thrust	34,3 [kN]
Number of vertical thrusters	2 [-]
Number of horizontal thrusters	4 [-]
Pod maximum vertical thrust fluid power	99,4 [kW]
Pod maximum vertical thrust	24,4 [kN]
Number of propulsion pods	2 [-]
Maximum vessel horizontal fluid thrust power	560 [kW]
Maximum vessel horizontal thrust	174 [kN]
Maximum vessel vertical fluid thrust power	478,8 [kW]
Maximum vessel vertical thrust	117,4 [kN]

Table 61 Thruster requirements summary

Horizontal plane calculations	
Maximum head current speed when current-waning	3,4 [m/s]
C_D	1[-]
Standard water density	1025 [kg/m^3]
Vessel maximum beam	9,2 [m]
Beam current flow Reynolds number	$2{,}4 \cdot 10^7$ [-]
Initially allocated thruster power	196 [kW]
Assumed thruster efficiency	0,7 [-]
Maximum beam current speed	3,1 [m/s]
Sway force	44,6 [kN]
Vertical plane calculations	
Crane hoist speed	0,25 [m/s]
Hydrodynamic force (drag and added mass from Table 10)	321,9 [kN]
Required vertical thrust power	80,5 [kW]

Table 60 and Table 61 present the capabilities of the thruster configuration and the expected loads from the incident current and crane operation. There is tremendous over-capacity in the horizontal plane however; the main reason for this is that the thrusters will generate a lot less noise when operating at low loads rather than full load. As the vessel is totally dependent on acoustic systems for accurate submerged navigation noise reduction is important. Another benefit is that the wear on the thrusters is lessened, hopefully increasing service life and reducing maintenance costs. It should also be noted that the maximum current speeds in the horizontal plane assumes that all no corrections are made in the vertical plane, i.e. all power is used by the horizontal thrusters.

9 Conclusions

9.1 Vessel Capabilities

The design process has been performance driven, thus the vessel has most of the capabilities listed in the design requirements. The vessel is intended to perform structural inspection missions with an ROV and replace components weighing up to 10 Te, though the only real limits for which operations that can be performed are your imagination, ROV size, cargo hold size and crane capacity. The vessel can for instance be transformed to a trenching vessel by lowering a trencher from the cargo hold. The only operational requirement that has not been fulfilled is the transit speed. The underestimation of the required propulsion power is a direct consequence of the underestimation of the form volume. This has two main causes:

1. The formula used to estimate propulsion power is based on experience with military vessels where the pressure hull is by far the largest part of the vessel.
2. The use of fuel cells that use pure hydrogen and oxygen as fuel. This requires much larger fuel tanks than conventional fuels.

It was not unexpected that the propulsion power would be higher than indicated by the estimation formula, but due to a lack of options it was used with along conservative efficiency estimates. Once the vessel was modelled far more accurate resistance calculations could be performed. While the vessel has the thrust required for the 8,2 knot transit speed it does not carry enough fuel to sustain it. The vessel can however sustain a speed of 6,2 knots with the power initially allotted to propulsion. This increases the transit time, decreasing the time on the field.

The vessel is spacious for a submarine with all modern facilities a crew can ask for. Even though submarining is inherently dangerous precautions have been taken so that the vessel is as safe to operate as possible. The vessel capabilities and main dimensions are summarized in Table 62.

Table 62 Vessel summary

Mission length	14 [days]
Crew size	15 [persons]
Transit speed	6,2 [knots]
Flank speed	8,4 [knots]
Emergency return speed	4 [knots]
Length overall	71,3 [m]
Maximum height	12,7 [m]
Outer hull diameter	9,2 [m]
Pressure hull diameter	5 [m]
Pressure hull length overall	33,6 [m]
Maximum payload weight	20 [Te]
Cargo hold storage area size (LxW)	3x3 [m]
Crane maximum rated load	43 [Te]
Maximum head current speed on DP	3,4 [m/s]
Maximum beam current speed on DP	3,1 [m/s]
Maximum operational depth	537 [m]
Hull crush depth	1075 [m]

9.2 Design Improvements

It was discovered in chapter 8 that the vessel has a significantly higher resistance than projected, increasing fuel consumption to such a level that the projected transit speed cannot be sustained. I propose a lengthening of the main fuel tank section in addition to a general streamlining of the vessel in order to compensate for the increased consumption.

Another option is to store the hydrogen in another compound, for instance methanol. This would reduce the fuel storage volume, though a reformer would be needed to separate the hydrogen from the other constituents of the compound. The vessel would also need a way to dispose of this residual material. As the machinery compartment already is quite cramped the inclusion of these components would demand a larger machinery compartment.

The fuel supply is thus far dependant on two fuel pumps, one for the hydrogen and one for the oxygen. These pumps are a possible weak spot in the design, as the failure of one would limit the fuel flow to the boil off rate. One can however argue that such pumps must be expected to be operable for long periods of time without breaking down. Further, the vessel can still return to base on battery power if the pumps break down in a way that cannot be repaired in situ. Combine this with a proper maintenance scheme and having redundant fuel pumps appear unnecessary. The pumps are quite small, so if it is decided that a duplicate fuel pump set is necessary this will not be difficult do place.

The vessel is only fitted with one air compressor, one hydraulic pump and one hydraulic accumulator. Their common denominator is that is that they are fairly large and that their failure is not a direct threat against the crew's safety. The hydraulic system is intended to power mission-critical systems like cargo hold hatches, airlock door opening mechanisms and so forth, but it is only used in back-up systems for critical systems like rudder actuators. Similarly only critical use of the HP air is for surfacing, and the vessel intentionally carries much more HP air than needed for one surfacing. In these cases one would *like* to have more than one unit for redundancy, but it seems than only one is *needed*. It was therefore decided to use only one unit due to the limited space available. It should be noted that a proper risk analysis has not been performed yet, and this may conclude that redundancy is needed in all or some of these cases.

Thus far the vessel has been somewhat luxuriously furnished compared to the military vessels visited by the author. The vessel has a rather large washroom with several showers and toilets and a rather large combined galley and recreational room. Hot-bunking is also not practiced. Though it may not be a direct improvement per se, these "luxuries" can be reduced or removed in order to reduce vessel size and cost.

9.3 Other Design Possibilities

During the initial phase of the design several different concepts were evaluated and found unsuitable for this application. This section will present these and the reasons for their rejection.

The use of a docking station was proposed. The intention was that this docking station could provide a stable foundation for crane operations (using a boom crane). The docking station would also connect the vessel to the subsea network enabling information sharing, communication and power transfer. The docking station concept was rejected as it would require the installation of a docking station next to each structure to be serviced, drastically increasing the investment costs. Reducing the autonomy of the vessel by making it dependant on power from the subsea system was also undesirable.

A similar concept involved fitting the vessel with retractable "legs" just like mobile cranes on land. This would require the vessel to land next to the subsea structure and deploy the legs before using a boom crane to install equipment. This would require pin-point accuracy in order to manoeuvre the vessel close enough to use the crane without crashing into the structure. The soil conditions must also be favourable as supporting legs are of little use if the ground yields. Soil suction could also be a problem when the vessel tried to depart. Support legs were rejected because the alternative of simply hovering above the structure seemed much simpler and safer than landing the vessel close to a subsea structure.

An altogether different topic is whether or not a commercial submarine has applications in other sectors than the offshore oil and gas industry. Marine biologists may be interested in the vessel as a mobile habitat used for ocean exploration. Similarly marine archaeologists may have use of the long submerged endurance of the vessel, not to mention the large cargo hold and ROV. As of today they usually have to rent the same equipment as the oil companies and at the same rates. The independence from the surface conditions may be just as enticing for these possible users as the offshore industry, perhaps even more so as scientific organizations usually do not possess the economic muscles of the offshore business. The offshore business uses such vessels and equipment to maintain what in practice is a dollar bill printing press and will only see a drop in profit due to a few days waiting for perfect weather. Scientific expeditions can however usually only afford to rent the vessels a certain amount of time, and if most of that time is spent waiting on the weather they would only have wasted their money compared to what could have been. A submarine would, with the condition that the operation is to take place at sufficiently large depths, allow for an efficient use of all the expedition days at any time of the year.

9.4 Further Work

The vessel has been designed as if storing and handling large quantities of cryogenic liquid hydrogen and oxygen not much different from storing diesel fuel, albeit using rather thick insulation in the storage tanks. This is by no means a given fact, so performing research on the fuel storage system and power generation system as a whole is strongly recommended.

The main topics not covered by or touched upon in this report are as follows:

- Detailed structural design
- Outer hull material selection
- Risk analysis
- Cost analysis

So far not cost estimates have been made. The impact of overall cost on the feasibility of the design has not been forgotten though. Throughout the design process unnecessarily expensive solutions have been rejected, and general tactics taught at NTNU for cost reduction have been employed. A proper cost analysis is however yet to be performed. This would not only have to take the building and maintenance costs of the submarine into account, but also the potential savings in time as well as money by utilizing the winter months for inspection. This analysis is critical in order to determine whether or not the vessel is commercially feasible.

Similarly important are the risk analysis and the hazard and operability analysis. Thought have been given during the design process to lower the potential risks, but this has to be quantified in order to properly evaluate the safety of the vessel.

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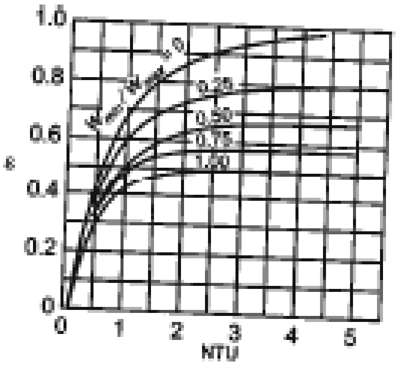
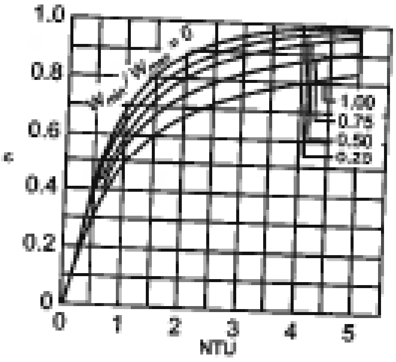
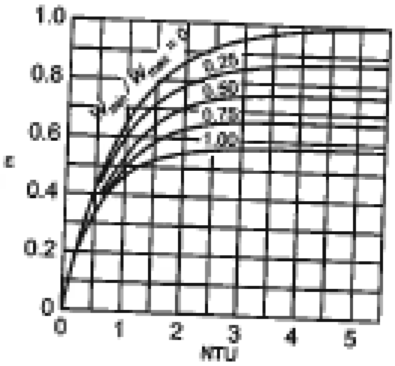
Appendix I: Heat Exchanger Efficiency Plots



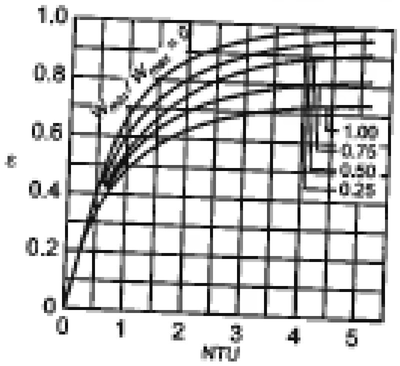
Effectiveness of a parallel-flow heat exchanger



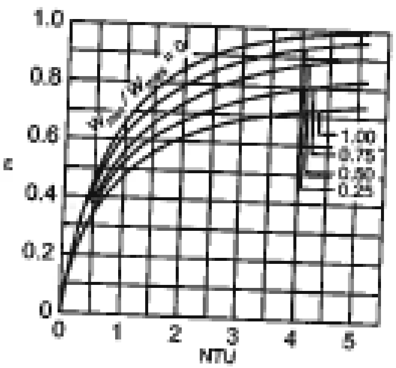
Effectiveness of a counterflow heat exchanger



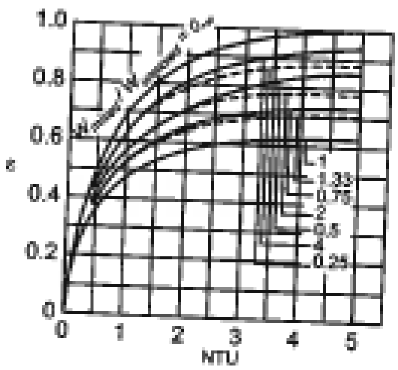
Effectiveness of a shell-and-tube heat exchanger with one shell and any multiple of two tube passes



Effectiveness of a shell-and-tube heat exchanger with two shell passes and any multiple of four tube passes



Effectiveness of a single-pass, cross-flow heat exchanger with both fluids unmixed.



Effectiveness of a single-pass, crossflow heat exchanger with one fluid mixed and the other unmixed.

Figure 43 Heat exchanger efficiency plots from machinery BC

Appendix II: ABS Pressure Hull Spreadsheet

Table 63 ABS Cylindrical shell spreadsheet

ABS ANALYSIS - METALLIC PRESSURE BOUNDARY		
Cylinderical Shells Under External Pressure		Rev: 5/18/2009
Section ==>	5m Section	ROV airlock
Material	HY120	HY120
E = Young's Modulus of Elasticity (psi)	30 461 270	30 461 270
ν = Poisson's Ratio	0,30	0,30
σ_y = Yield Stress	60 000	60 000
ρ = Density (lb_m/in^3)	0,289	0,289
Shell Geometry		
D_o = Outside Diameter of Shell (in)	200,90	133,90
L_s = Center to center spacing of stiffeners (in)	32,83	28,34
L_b = Unsupported spacing between stiffeners(in)	30,8300	27,3400
L = Greater of L_b or L_s	32,8300	28,3400
L_c = Length Between Bulkheads (in)	197,00	141,70
R = Radius to Midplane of Shell (in)	99,5	66,0
D = Diameter of Midplane of Shell (in)	198,9	131,9
t_w = Stiffener web Thickness (in)	2,00	1,00
L_w = Stiffener web width (in)	7,00	6,00
t_f = Flange Thickness (in)	1,00	1,00
L_f = Flange Width (in)	4,00	3,00
R_f = Radius to tip of the stiffener away from the shell (in)	90,450	57,950
R_o = Outside Radius of Shell (in)	100,450	66,950
$\bar{z}$ = Dist. of centroid of stiffener alone to the closer shell surface	4,389	4,167
R_s = Radius to centroid of stiffener cross section only	94,061	60,783
t = Shell Plating Thickness (in)	2	2
n = Num. of Circumferential Lobes for Failure Calculation	2	2
Stiffener Properties		
A_s = Area of Frame section only (in^2) (Zero for no stiffner)	18,0000	9,0000
A_t = Area of Frame and Shell section L_e (in^2)	60,3096	43,4543
$L_e = 1.5*(Rt)^{0.5}$ (First Equation)	21,1548	17,2272
$L_e = 0.75L_s$ (Second Equation)	24,6225	21,2550
L_e = Effective lengh of cylinder shell acting with stiffener (in)	21,1548	17,2272
I = Moment of inertia for combined section (in^4)	488,0915	244,7256
I_z = Moment of stiffener alone (in^4)	74,0000	39,5000

Shell Parameters

M	2,3278	2,4676
θ	2,9922	3,1719
Q	1,4961	1,5859
N	1,0882	1,0903
G	0,4948	0,4225
H	0,9705	1,0051
Inter-Stiffener Strength (assuming internal stiffeners)		
a) Inter-stiffener strength is to be able to be obtained from the following equations		
A	20,1216	10,5951
F	0,1177	0,0738
P_y = Yield Pres. at midbay and midplane of a cylinder (psig)	1367,6	1964,5
P_m = Von Mises shell buckling pressure for a cylinder (psig)	6688,5	14048,4
P_c = Cylinder inter-stiffener limit pressure (psig)	1139,7	1637,1
η = for maximum allowable working pressure	0,80	0,80
P_a = Maximum Allowable working pressure based inter-stiffener strength (psig)	**911,8**	**1309,7**
b) The limit pressure corresponding to the longitudinal stress at stiffeners reaching yield, is given by the following:		
γ	0,2379	0,1747
P_l = Cylinder stiffener long. yield stress pressure (psig)	1312,6	2222,3
η = for maximum allowable working pressure	0,67	0,67
P_a = Maximum Allowable working pressure based on longitudinal stress at the frame (psig)	**879,4**	**1489,0**
Overall Buckling Strength (General Instability)		
A_2	3,0000	3,0000
λ	1,5859	1,4622
A_1	0,035004892	0,029816396
P_n = Cylinder overall instability pressure (psig)	22825	30295
η = for maximum allowable working pressure	0,50	0,50
P_a = Maximum allowable working pressure based on overall buckling (psig)	**11412,6**	**15147,3**

Non-Heavy Stiffeners

(a) Stress Limits		
c = distance from the outer surface of the stiffener flange to the neutral axis of the combined stiffener and effective shell section (in)	7,3916	6,9299
Out-of-roundness as percent of R	0,50	0,50
δ = Allowable out of roundness, in	0,4973	0,3298
P_t = Yield pressure (psig)	1712,2	2217,0
η = for maximum allowable working pressure	0,50	0,50
P_a = Maximum Allowable working pressure based on stiffener stress (psig)	**856,1**	**1108,5**
(b) Stiffener Tripping		
σ_T = Circumferential tripping stress (psia), OK if > σ_y	286 911	486 518
$\sigma_T > \sigma_y$ (not applicable if no stiffner)	OK	OK
(c) Local Buckling		
Web Depth/Thickness	3,5	6,0
$0.9*(E/\sigma_y)^{0.5}$	20,3	20,3
Web Depth/Thickness < $0.9*(E/\sigma_y)^{0.5}$	OK	OK
Flange Width / Flange Thickness	4,0	3,0
$0.3*(E/\sigma_y)^{0.5}$	6,8	6,8
Flange Width/Thickness < $0.3*(E/\sigma_y)^{0.5}$	OK	OK
(d) Inertia Requirment		
Assumed Max operating depth (ft) <== Change this to see how I_{min} changes	1640,0	1640,0
ρ_w = Water Density (lbm/ft^3)	62,4	62,4
P at assumed maximum depth (psig)	710,7	710,7
η = usage factor	0,5	0,5
I_{min} = Minimum required combined Moment of intertia (in^4)	453,8034	109,0300
I = Moment of inertia for combined section (in^4)	488,0915	244,7256
I >= I_{min} ?	Yes	Yes
P_a = Maximum External Pressure based on Moment of Inertia calculation (psig)	**764,4**	**1595,1**
Minimum of all Pa's (psig)	**764,4**	**1108,5**
Maximum depth (ft)	**1764**	**2558**

Table 64 ABS Conical shell spreadsheet

ABS ANALYSIS - METALLIC PRESSURE BOUNDARY	
Conical Shells Under External Pressure	Rev: 5/18/2009
Section ==>	Conical transition to ROV airlock
Material Definition	
Material	HY120
E = Young's Modulus of Elasticity (psi)	30 461 270
ν = Poisson's Ratio	0,30
σ_y = Yield Stress	60 000
ρ = Density (lb_m/in^3)	0,289
Shell Geometry	
D_o = Outside Diameter of Shell (in)	200,90
R1= mean radius at the small end	64,50
R2= mean radius at the large end	99,50
Lstiff= length between stiffeners	10,0000
Rb	82,0
L_s = Center to center spacing of stiffeners (in)	10,0000
L_b = Unsupported spacing between stiffeners(in)	8,0000
L = Greater of L_b or L_s	10,0000
L_c = Length of section	60,00
R = Radius to Midplane of Shell (in)	99,5
D = Diameter of Midplane of Shell (in)	198,9
t_w = Stiffener web Thickness (in)	2,00
L_w = Stiffener web width (in)	6,00
t_f = Flange Thickness (in)	1,00
L_f = Flange Width (in)	3,00
alfa= half apex angle	0,528
R_f = Radius to tip of the stiffener away from the shell (in)	91,450
R_o = Outside Radius of Shell (in)	100,450
z = Dist. of centroid of stiffener alone to the closer shell surface	3,700
R_s = Radius to centroid of stiffener cross section only	94,750
t = Shell Plating Thickness (in)	2
n = Num. of Circumferential Lobes for Failure Calculation	2
Stiffener Properties	
A_s = Area of Frame section only (in^2) (Zero for no stiffner)	15,0000
A_t = Area of Frame and Shell section L_e (in^2)	30,0000
$L_e = 1.5*(Rt)^{0.5}$ (First Equation)	17,0367
$L_e = 0.75L_s$ (Second Equation)	7,5000
L_e = Effective lengh of cylinder shell acting with stiffener (in)	7,5000
I = Moment of inertia for combined section (in^4)	236,3250
I_z = Moment of stiffener alone (in^4)	43,0000

Shell Parameters

M	0,7091
θ	0,9114
Q	0,4557
N	0,4540
G	0,9929
H	0,1378

Inter-Stiffener Strength (assuming internal stiffeners)

a) Inter-stiffener strength is to be able to be obtained from the following equations

A	15,9120
F	0,3617
P_y = Yield Pres. at midbay and midplane of a cylinder (psig)	1890,5
P_m = Von Mises shell buckling pressure for a cylinder (psig)	155694,3

Inter-Stiffener Strength

Pmo	60923,7506
Pyo	1632,1701
Pmo/Pyo=Pratio	37,3268
If Pratio=<1, Pco=	N/A
If 1<Pratio=<3, Pco	1610,3069
If Pratio>3 Pco=	1360,1418
safety factor	0,7200
Pa	979,3021

b) The limit pressure corresponding to the longitudinal stress at stiffeners reaching yield, is given by the following:

γ	0,3698
P_l = Cylinder stiffener long. yield stress pressure (psig)	2133,4
η = for maximum allowable working pressure	0,67
P_a = Maximum Allowable working pressure based on longitudinal stress at the frame (psig)	**1429,4**

Overall Buckling Strength (General Instability)

A_2	3,0000
λ	3,7086
A_1	0,06076315
P_n = Cylinder overall instability pressure (psig)	41187
η = for maximum allowable working pressure	0,50
P_a = Maximum allowable working pressure based on overall buckling (psig)	**20593,6**

Non-Heavy Stiffeners

(a) Stress Limits

c = distance from the outer surface of the stiffener flange to the neutral axis of the combined stiffener and effective shell section (in)	5,6500
Out-of-roundness as percent of R	0,50
δ = Allowable out of roundness, in	0,4973
P_t = Yield pressure (psig)	2286,4
η = for maximum allowable working pressure	0,50
P_a = Maximum Allowable working pressure based on stiffener stress (psig)	**1143,2**

(b) Stiffener Tripping

σ_T = Circumferential tripping stress (psia), OK if > σ_y	287 812
σ_T > σ_y (not applicable if no stiffner)	OK

(c) Local Buckling

Web Depth/Thickness	3,0
$0.9*(E/\sigma_y)^{0.5}$	20,3
Web Depth/Thickness < $0.9*(E/\sigma_y)^{0.5}$	OK
Flange Width / Flange Thickness	3,0
$0.3*(E/\sigma_y)^{0.5}$	6,8
Flange Width/Thickness < $0.3*(E/\sigma_y)^{0.5}$	OK

(d) Inertia Requirment

Assumed Max operating depth (ft) <== Change this to see how I_{min} changes	1640,0
ρ_w = Water Density (lbm/ft^3)	62,4
P at assumed maximum depth (psig)	710,7
η = usage factor	0,5
I_{min} = Minimum required combined Moment of intertia (in^4)	134,1716
I = Moment of inertia for combined section (in^4)	236,3250
I >= I_{min} ?	Yes
P_a = Maximum External Pressure based on Moment of Inertia calculation (psig)	**1197,4**
Minimum of all Pa's (psig)	**979,3**
Maximum depth (ft)	**2260**

Table 65 ABS Spherical/Hemispherical/Ellipsoid head spreadsheet

ABS ANALYSIS - METALLIC PRESSURE BOUNDARY Spherical Shells Under External Pressure		
Section ==>	Main PH end cap	ROV lock outer hatch
Material Definition		
Material	HY120	A516 gr 70
E = Young's Modulus of Elasticity	30 461 270	30 461 270
ν = Poisson's Ratio	0,30	0,30
σ_y = Yield Stress	60 000	60 000
ρ = Density (lbm/in^3)	0,289	0,2835
Shell Geometry for Sphere or Hemisphere Head		
D = Diameter of Midplane of Sphere (in)	198,00	128,00
D_i = Inside Diameter of Sphere (in)	195,000	126,000
D_o = Outside Diameter of Sphere (in)	201,000	130,000
R_o = Outside Radius of Sphere (in)	100,500	65,000
t = Sphere Plating Thickness (in)	3	2
a) Limit pressure for spherical shells and hemi. heads		
P_{es} (psig)	32855	34908
P_{ys} (psig)	3582	3692
P_{es}/P_{ys}	9,172	9,454
P_{cs}/P_{ys}	0,695	0,697
η = for maximum allowable working pressure	0,67	0,67
P_a = Maximum Allowable working pressure (psig)	**1667,2**	**1724,4**
b) Shape Limination Test		
Maximum Wall thickness, in	31,680	20,480
Minimum Wall thickness, in	0,040	0,026
Maximum wall thickness test	Pass	Pass
Minimum wall thickness test	Pass	Pass
Ellipsoidal Head		
D_i = Inner diameter of head (in)	197,00	129,00
D_o = Outer diameter of head (in)	201,00	131,00
D = Mean diameter of head (in)	199,00	130,00
h = head inside depth, measured along the tangent line (in)	80,00	50,00
R_e = Equvalent spherical radius, in	123,74	84,50
t = Shell Plating Thickness (in)	2	2
l_h = Skirt dimension (in)	6	4
a) Limit pressure for eliptical heads		
P_{es} (psig)	9632	20658
P_{ys} (psig)	1940	2840
P_{es}/P_{ys}	4,966	7,273
P_{cs}/P_{ys}	0,614	0,672
η = for maximum allowable working pressure for head	0,67	0,67
P_a = Maximum Allowable working pressure oor head (psig)	**797,5**	**1278,7**

b) Shape Limination Test

Maximum Wall thickness, in	159,200	104,000
Minimum Wall thickness, in	0,040	0,026
Maximum wall thickness test	Pass	Pass
Minimum wall thickness test	Pass	Pass
Minimum h, in	35,820	23,400
Minimum h test	Pass	Pass
Minimum l_h, in	4,000	4,000
Minimum l_h test	Pass	Pass

c) Calculated maximum depth

Minimum Pa , psig	**797,5**	**1278,7**
Maximum depth (assuming density = 62.4 lbm/ft$^{3)}$	**1840**	**2951**

Appendix III: Weight

A numbering system has been developed which identifies in which equipment category a component belongs and in which vessel section it is placed. The first number is a Roman numeral which defines the vessel section, while the letter defines the equipment category. If the component is located in the pressure hull the Roman numeral is augmented by a number between 1 and 6 and the letter A or B. This identifies the pressure hull section and deck the component is located on. The placement according to the origin defined in Figure 27 and the resulting moments used in the stability and trim calculations are also listed. The division is displayed on Figure 44.

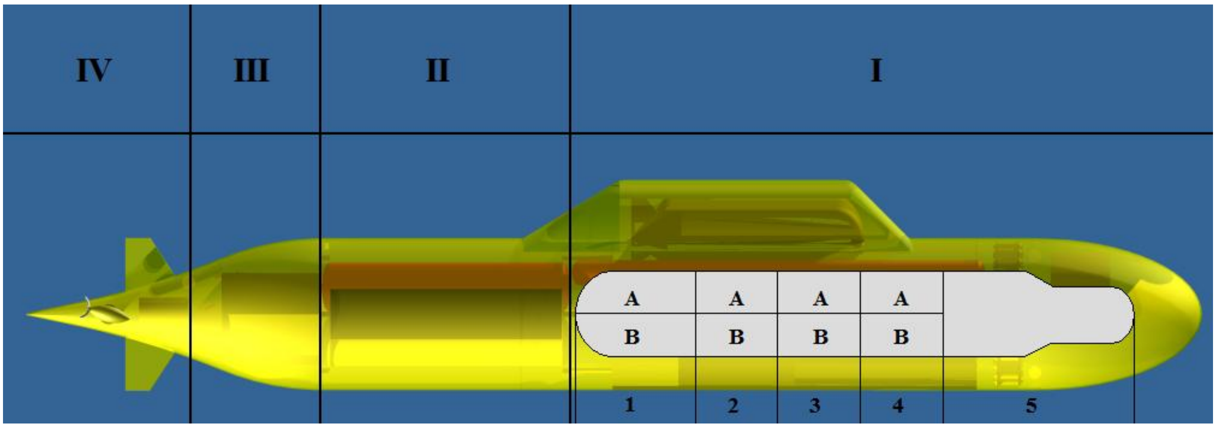


Figure 44 Numbering system

No.	Equipment	Weight [Te]	VCG [m]	Vertical moment [Te m]	LCG [m]	Longditudinal moment [Te m]	TCG [m]	Transverse moment [Te m]
Life Support								
I-1B-L	Reverse osmosis sys. with pump	0,215	-0,87	-0,19	0,66	0,14	-1,51	-0,33
I-1B-L	Main freshwater tank	0,7	-0,71	-0,49	2,07	1,45	-1,47	-1,03
I-1B-L	Hot freshwater tank	0,3	-1,41	-0,42	2,07	0,62	-1,42	-0,43
I-1A-L	Ventilation & air purification sys	0,5	1,96	0,98	1,50	0,75	0,00	0
I-1A-L	Waste treatment sys. & waste tank	6	0,96	5,76	0,50	3	0	0
I-1A-L	Stores	6	0,90	5,40	2,25	13,50	0,50	3,00
I-2A-L	Kitchen appliances	1	0,60	0,60	7,50	7,50	-1,63	-1,63
I-3B-L	Bed rack 1	0,08	-1,23	-0,10	11,45	0,92	0,72	0,06
I-3B-L	Bed rack 2	0,08	-1,23	-0,10	12,45	1,00	-0,72	-0,06
I-3A-L	Bed rack 3	0,08	1,23	0,10	12,45	1,00	-0,72	-0,06
I-4A-L	Shower unit	0,01	1,00	0,01	18,47	0,18	1,12	0,01
I-4A-L	Starboard Toilet unit	0,03	0,45	0,01	16,02	0,48	-1,52	-0,05
I-4A-L	Port toilet unit	0,015	0,45	0,01	15,51	0,23	1,52	0,02
I-4A-L	Washroom sink	0,01	0,55	0,01	17,02	0,17	1,67	0,02
I-L	DSRV (neutrally buoyant)	0		0		0	0	0

No.	Equipment	Weight [Te]	*VCG [m]*	*Vertical moment [Te m]*	*LCG [m]*	*Longditudinal moment [Te m]*	*TCG [m]*	*Transverse moment [Te m]*
Propulsion, Power & Steering								
I-P	Bow planes	11,1	-2,00	-22,20	28,25	313,58	0	0
I-1B-P	Top port fuel cell	1	-1,15	-1,15	1,38	1,38	0,86	0,86
I-1B-P	bottom port fuel cell	1	1,70	1,70	1,38	1,38	0,86	0,86
I-1B-P	top starboard fuel cell	1	-1,15	-1,15	1,38	1,38	-0,86	-0,86
I-1B-P	bottom starboard fuel cell	1	-1,70	-1,70	1,38	1,38	-0,86	-0,86
II-P	Fuel module LH2 tanks	132,6	1,63	215,48	-10,00	-1326,00	0	0
II-P	Fuel module LO2 tanks	208,3	-1,63	-338,52	-10,00	-2083,20	0	0
I-P	External LH2 tanks	51,9	2,50	129,70	10,03	520,36	0	0
I-P	Batteries	200	-3,48	-696,00	11,00	2200,00	0	0
IV-P	Main propulsion pods	11,1	0	0	-33,70	-374,07	0	0
ROV & Payload								
III-R	Crane boom	1,2	2,40	2,88	-23,50	-28,20	0	0
III-R	Crane lifting gear	0,8	2,40	1,92	-23,50	-18,80	0	0
I-R	ROV and trolley	3,2	0	0	28,25	90,40	0	0
I-5-R	ROV workshop workbenches	1	-0,50	-0,50	22,50	22,50	0	0
III-R	Payload	0,00	0					
Vessel Control & Information Systems								
I-2B-C	Machinery control console	0,5	-0,75	-0,38	7,50	3,75	-1,55	-0,78
I-2B-C	Machinery control work bench	0,5	-0,76	-0,38	7,50	3,75	1,55	0,78
I-4B-C	Command console 1	0,5	-0,75	-0,38	17,50	8,75	-1,55	-0,78
I-4B-C	command console 2	0,5	-0,75	-0,38	17,00	8,50	1,55	0,78

No.	Equipment	Weight [Te]	*VCG [m]*	*Vertical moment [Te m]*	*LCG [m]*	*Longditudinal moment [Te m]*	*TCG [m]*	*Transverse moment [Te m]*
Auxilliary Machinery								
I-1B-A	HP air compressor	1	-0,54	-0,54	2,07	2,07	- 1,56	-1,56
I-1B-A	Seawater cooling system pump	0,015	-1,01	-0,02	0,73	0,01	1,62	0,02
I-1B-A	FC coolant circulation pump	0,015	-1,01	-0,02	1,02	0,02	1,62	0,02
I-5-A	Airlock drain pump	0,025	-1,05	-0,03	24,35	0,61	0,99	0,02
I-1B-A	Trim tanks deballasting pump	0,025	0,56	0,01	0,97	0,02	1,61	0,04
I-1B-A	Trim tanks transfer pump	0,18	-0,50	-0,09	0,18	0,03	1,61	0,29
I-1B-A	MBT LP blower	0,18	-0,50	-0,09	0,59	0,11	1,61	0,29
I-1B-A	O2 fuel pump	0,015	-1,01	-0,02	0,09	0,00	1,62	0,02
I-1B-A	H2 fuel pump	0,015	-1,01	-0,02	0,44	0,01	1,62	0,02
I-1A-A	Hydraulic booster pump	0,05	0,55	0,03	1,24	0,06	- 0,88	-0,04
I-1A-A	Hydraulic accumulator	2	0,67	1,34	2,76	5,51	- 0,87	-1,75
I-1B-A	HVAC heat exchanger	0,02	-1,69	-0,03	3,28	0,07	- 0,97	-0,02
I-1B-A	seawater heat exchanger	0,8	-1,20	-0,96	4,15	3,32	- 0,87	-0,70
I-A	HP air bottles	27,3	0	0	23,90	652,47	0	0
I-1-A	Engine room piping	0,75	0	0	2,50	1,88	0	0
A	Ballast transfer piping	12	0	0	0	0	0	0
Structural Components								
III-S	cargo hold bulkheads	17,3	0	1,92	- 20,50	-354,65	0	0
I-S	Main hull steel plates	161,21	0	0	12,45	2007,06	0	0
I-S	Main hull stiffeners	30	0	0	12,45	373,50	0	0
I-5-S	Conical section steel plate	8	0	0	25,20	201,58	0	0
I-5-S	Conical stiffeners	3,71	0	0	25,20	93,48	0	0
I-5-S	ROV lock steel plates	15,17	0	0	28,25	428,55	0	0
I-5-S	ROV lock stiffeners	1,93	0	0	28,25	54,52	0	0
I-1-S	Main PH end cap	10,6	0	0	-0,68	-7,18	0	0
I-5-S	ROV lock end cap	2,2	0	0	30,52	67,14	0	0
I-S	Main deck	45	0	0	12,50	562,50	0	0
I-S	Main bulkheads	31,4	0	0	12,50	392,50	0	0
S	External structural bracings	50,66	0	0	0,00	0,00	0	0
I-S	Permanent ballast	118,67	-4	-474,70	12,50	1483,43	0	0
	Total weight and resulting CGs	***1182,46***	***-0,99***	***-1172,67***	***4,52***	***5346,41***	***0,00***	***-3,80***

Appendix IV: Buoyancy

Table 66 Standard condition buoyancy summary

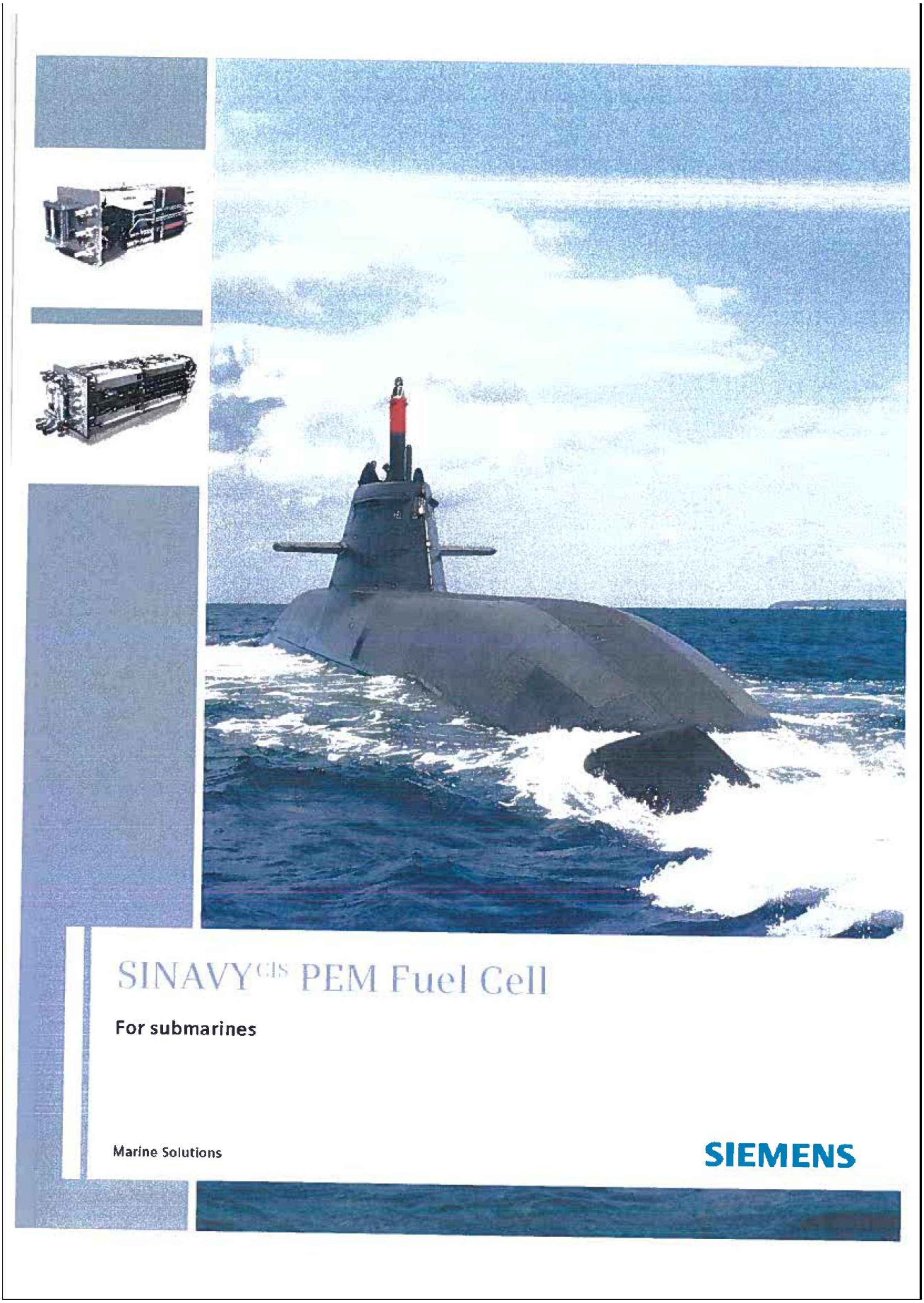
Module/equipment	Buoyancy [Te]	VCB [m]	Vertical moment [Te m]	LCB [m]	Longditudinal moment [Te m]	TCB [m]	Transverse moment [Te m]
Main pressure hull	526,80	0	0	12,00	6321,60	0	0
Conical section	22,14	0	0	25,20	557,93	0	0
Airlock	33,54	0	0	28,50	955,83	0	0
Lateral LH_2 tanks	45,31	2,50	113,26	10,03	454,41	0	0
Fuel module LH_2 tanks	217,20	1,63	352,95	-10,00	-2172,00	0	0
Fuel module LO_2 tanks	217,20	-1,63	-352,95	-10,00	-2172,00	0	0
Battery rack	91,10	-3,48	-317,03	11,00	1002,12	0	0
HP air bottles	11,11	0	0	23,90	265,45	0	0
Propulsion pods	1,44	0	0	-33,70	-48,36	0	0
Airlock ellipsoidal head	5,77	0	0	30,52	176,12	0	0
Main pressure hull ellipsoidal head	10,87	0	0	0,68	7,36	0	0
Sum/ resultant CB	**1182,46**	**-0,17**	**-203,77**	**4,52**	**5348,47**	**0**	**0**

Appendix V: Variable Weights and Buoyancies

Table 67 Variable weights and buoyancies

Variable weights
Main freshwater tank
Hot freshwater tank
Waste tank
Stores
Fuel module LH_2
Fuel module LO_2
Lateral LH_2
ROV
Payload
Variable buoyancies
Airlock
ROV lock ellipsoidal head

Appendix VI: Siemens PEM Fuel Cell Information Leaflet



Introduction

Fuel cells allow the direct generation of electric power from hydrogen and oxygen with a considerably better efficiency and no pollutant emission compared to conventional combustion engines. Their operation is noiseless.

The house's central building block: Completely Integrated Solutions.

The portfolio offered by the Integrated Solution House for Industry is bundled in our Completely Integrated Solutions. These solutions are based on tested standards that we "refine" to meet our customers' specific requirements. They combine best-in-class products, systems and services in a smart overall solution – resulting in a high degree of availability and readiness for new as well as revamped and modernized frigates and submarines.

In particular, Completely Integrated Solutions help you by:

- **Optimizing overall ship operation –** in all electrotechnical equipment and systems for propulsion, power generation and distribution in all parts of a ship. Our comprehensive approach ensures maximum operational safety, dependability under all conditions, low energy consumption and full compliance with environmental regulations.
- **Connecting all IT and automation levels –** from sensors to the automation and control level, with maritime-specific IT solutions that gather and process all the data from the entire ship, thus providing a solid basis for decision making.
- **Supporting you throughout the life cycle –** for everything from consulting and system integration to commissioning, documentation, training, and spare-parts procurement, as well as modernization.

Application Potential

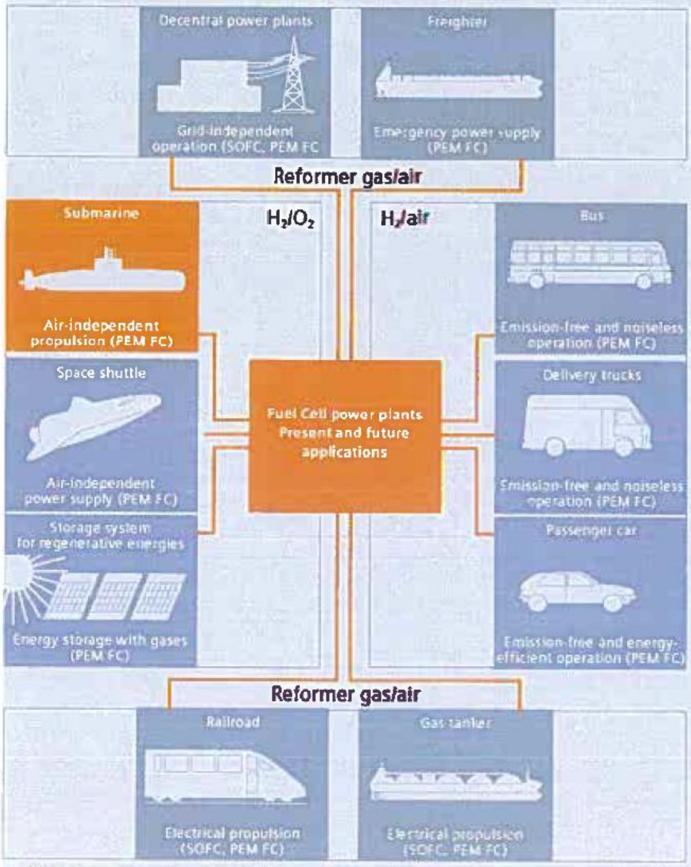


Fig. 1: Possible applications for fuel cell power plants

In addition to these basic advantages, the fuel cell with a solid, ion-conducting, polymeric membrane (Polymer Electrolyte Membrane – PEM) has further positive properties:

- Quick switch-on, switch-off behavior
- Low voltage degradation and long service life
- Favorable load and temperature cycle behavior
- Overload possibility
- Low operating temperature (80 °C)
- Absence of a liquidcorrosive electrolyte.

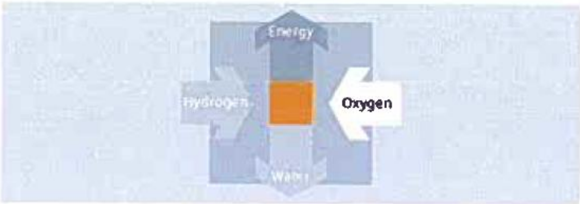


Fig. 2

All these characteristics make the SINAVY^{cs} PEM Fuel Cell an ideal power unit.
Aboard submarines they show their outstanding advantages against conventional AIP systems (Air-Independent Propulsion) using oxygen and hydrogen, carried on board.
Siemens has two types of SINAVY^{cs} PEM Fuel Cell modules for you to choose from. The BZM 34, with a rated power of 34 kW, as well as the BZM 120, with a rated power of 120 kW.
The new submarines of class U 212 A (six for the German Navy, and four for the Italian Navy) are equipped with BZM 34 modules, which have been developed since 1985 on behalf of the German Ministry of Defense. The new 209PN-class and 214-class submarines – up to now for the Hellenic Navy, Republic of Korea Navy and Portuguese Navy – are fitted with BZM 120 modules which have been developed by Siemens in a next step.

Existing submarines can be upgraded with an additional fuel cell power plant during refit, thus getting the benefits of the Air-Independent Propulsion (AIP) at a much lower price than for new submarines.
The Hellenic Navy has placed an order for modernizing three submarines of class 209 by installing fuel cell AIPs – among other measures of refit.
The suitability of fuel cell technology on board submarines has been demonstrated by earlier tests and now on board of submarines of classes U 212 A and 214.
Further possible applications of SINAVY^{cs} PEM Fuel Cell for power generation are listed below (see also fig. 2):

Using hydrogen and oxygen

- Operation in spacecrafts
- Component in a long-term energy storage system (consisting of solar cells, an electrolyser system and a hydrogen/oxygen storage system)

Using hydrogen and air

- Zero-emission operation of electrically driven vehicles

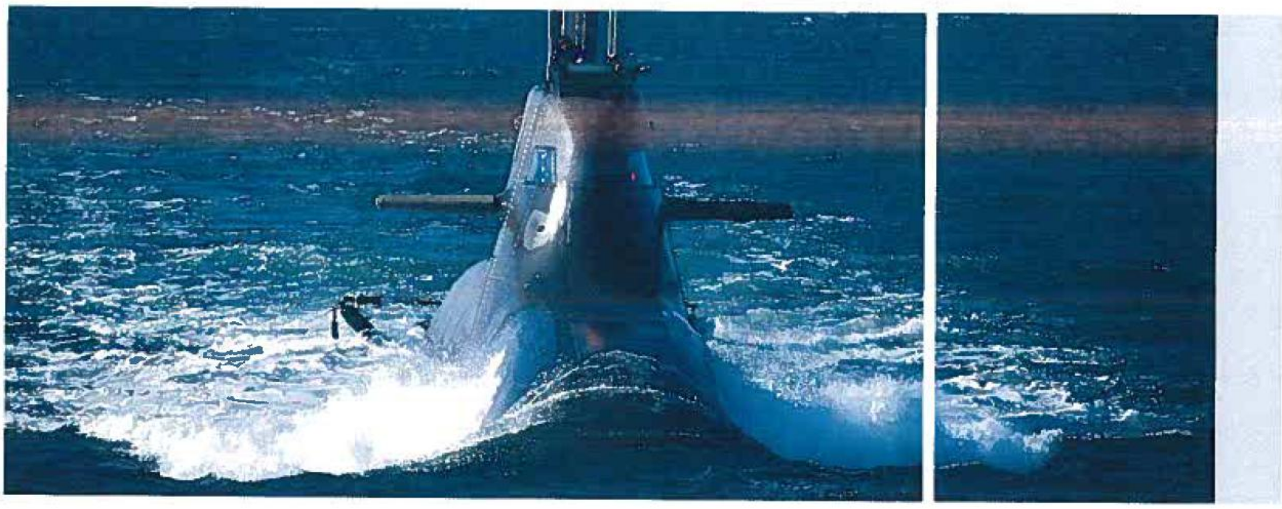
Using reformer gas and air

- Power supply far distant from a public power supply system
- Safe, low-emission power supply on cargo vessels especially in harbor
- Utilization of boil-off gases aboard gas tankers
- Power supply e.g. for drives on rail vehicles

Concentrating on manufacture and development of fuel cells for AIP applications, Siemens demonstrated its technological competence in projects for air-breathing fuel cells, e.g.

- Fork lift truck
- Micro co-generation
- Propulsion systems for busses.

The Siemens R&D activities in regard of other types of fuel cells like Solid Oxide Fuel Cells (SOFC) are not presented in this brochure.



PEM Fuel Cell: function and design

Both the basic function and the design of the SINAVY CIS PEM Fuel Cell are very simple (fig. 3): the electrochemical element at which the chemical energy is converted into electrical energy is the membrane electrode unit. It consists of the polymer electrolyte, the gas diffusion electrodes with a platinum catalyst and carbon sheets on each side.

After the abstraction of the electrons from hydrogen – they flow from the anode via the electrical load to the cathode – the resulting protons migrate from the anode to the cathode where they combine with oxygen (and the electrons) to water.

The theoretical voltage of an H_2/O_2 fuel cell is 1.48 V (referred to the upper heat value of hydrogen). At zero load conditions, slightly more than 1 V per cell is available.

The cooling units or bipolar plates in combination with carbon diffusion layers distribute the reactants uniformly across the area of the cell, conduct the electrons across the stack, remove the heat from the electrodes and separate the media from each other.

Fig. 4 shows the two core components of a cell with outside dimensions of 400 x 400 mm. As used in BZM 34 modules.

Fig. 5 compares the bipolar plate of the BZM 34 modules to the BZM 120. Two cells of the BZM 120 produce about twice the power of one cell of the BZM 34 type with nearly the same active area.

The in principle high development potential in regard to the membrane material is shown in fig. 6. With improved materials the power density can nearly be doubled. The voltage of a SINAVY CIS PEM Fuel Cell referred to the operating time is stable, degradation rates are less than 2 V/h for the BZM 34 module (fig. 7).

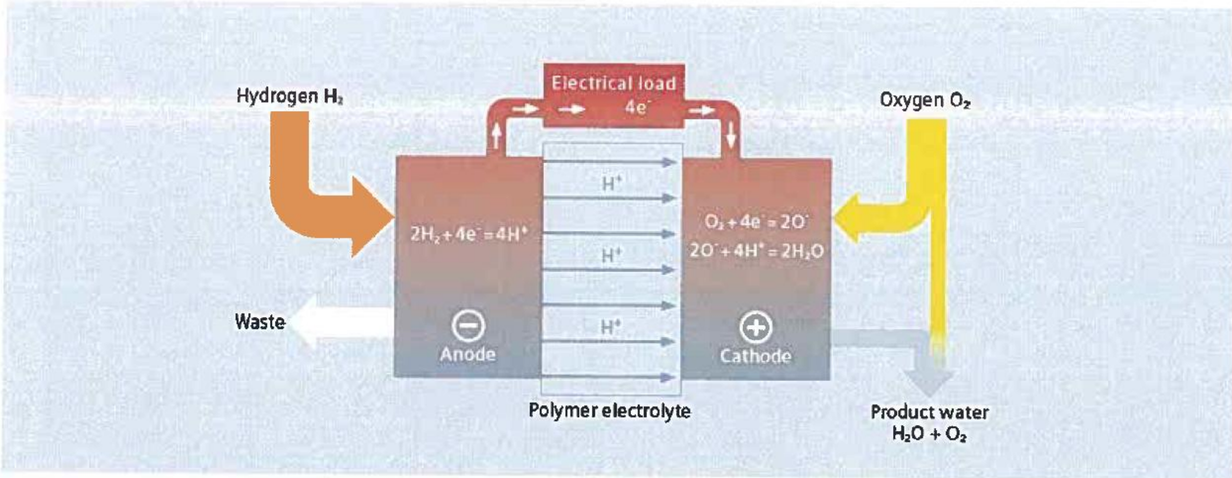


Fig. 3: Functional principle

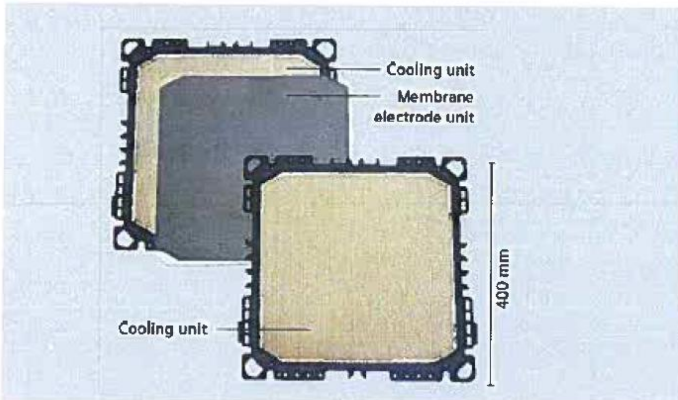


Fig. 4: Components of cell

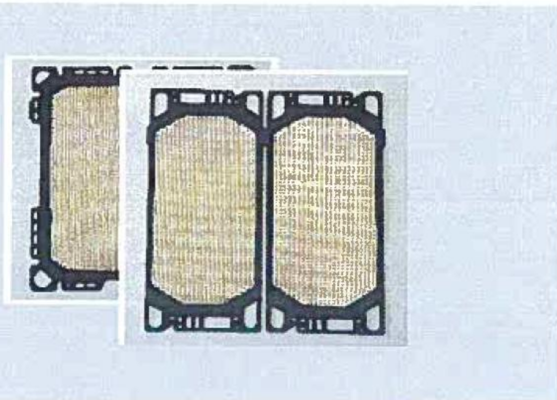


Fig. 5: Comparison of cells:
BZM 34 type (back), BZM 120 type (front)

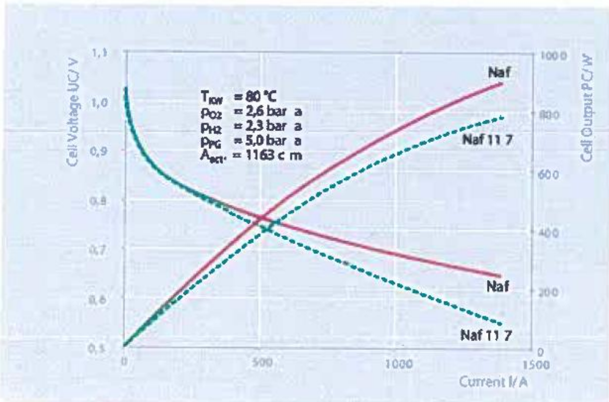


Fig. 6: Potential output increases by using various electrolytes

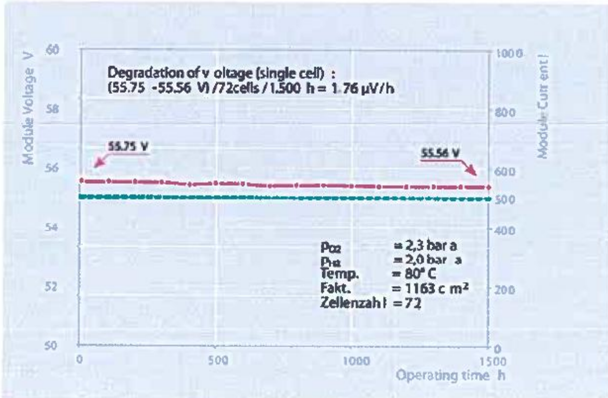


Fig. 7: Voltage degradation

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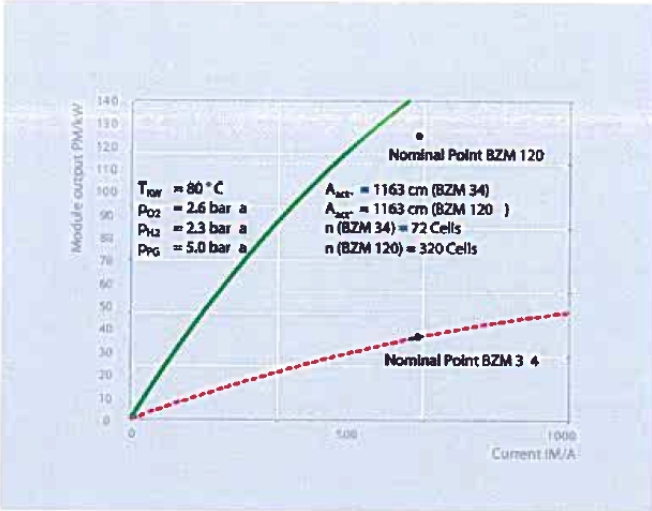

Fig. 8: Module output refers to load current

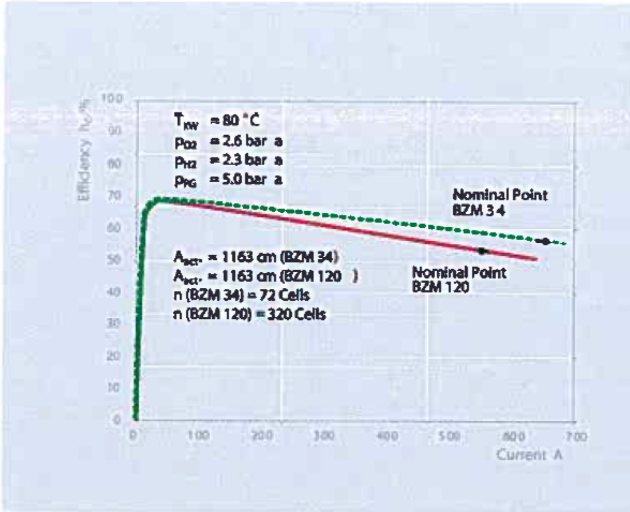

Fig. 9: Efficiency

PEM Fuel Cell modules and power plant

PEM Fuel Cell modules
The fuel cells need additional auxiliaries for their operation. The PEM Fuel Cell stack, valves, piping and sensors form the PEM Fuel Cell module, the corresponding module electronics controls the proper operation of the PEM Fuel Cell process. The ancillaries comprise the equipment for supplying H_2, O_2 and N_2, for reactant humidification, for product water, waste heat and residual gas removal. The PEM Fuel Cell stack and the ancillaries are installed in a container which is filled with inert gas (N_2) at 3.0 bar abs. to prevent a release of H_2 and/or O_2 in the case of leakage.

The PEM Fuel Cell module can be operated at various static load currents. Currents below 650A for BZM 34 modules or below 560A for BZM 120 modules respectively can be applied in continuous operation. The output power/current characteristics for BZM 34 modules are shown in fig. 8.

For currents above the rated current the loading time is limited due to the insufficient heat removal at such working points. Even loads up to the double of the rated current can be applied for a short time.

At the rated operating point, the overall efficiency is approximately 59 % referred to the lower heat value of H_2 (LHV). It increases in the part load range, reaching a maximum of approximately 69 % at a load factor of some 20 % of the rated current (approx. 100 A) (fig. 9).

The properties of the BZM 34 and BZM 120 modules are listed in the table.

PEM Fuel Cell power plant
Suitable operating conditions for fuel cell modules are provided for submarine application by a fuel cell system in which fuel cell modules are connected

- to the hydrogen and oxygen supply
- to disposal units such as for
  - cooling
  - residual gas
  - reaction water
- to auxiliary systems such as for
  - inert gas drying
  - nitrogen supply
  - evacuating system
- to the propulsion/ship's system as the purpose of the whole PEM Fuel Cell system.

Operator control and visualization of the fuel cell system are effected by the integrated platform management system, or directly by the control panel of the fuel cell system. Fig. 10 gives a simplified impression of the AIP system.

The fuel cell system in its entirety – the complete fuel cell power plant, especially the supply and disposal systems described above for AIP operation including spatial and functional integration on board – has been developed by HDW (Howaldtswerke Deutsche Werft AG).

The new submarine classes U 212 A and 214 are equipped with the new fuel cell power plant by HDW with the SINAVY^{cs} PEM Fuel Cell modules by Siemens. An AIP section with SINAVY^{cs} PEM Fuel Cell modules can be added into existing submarines.

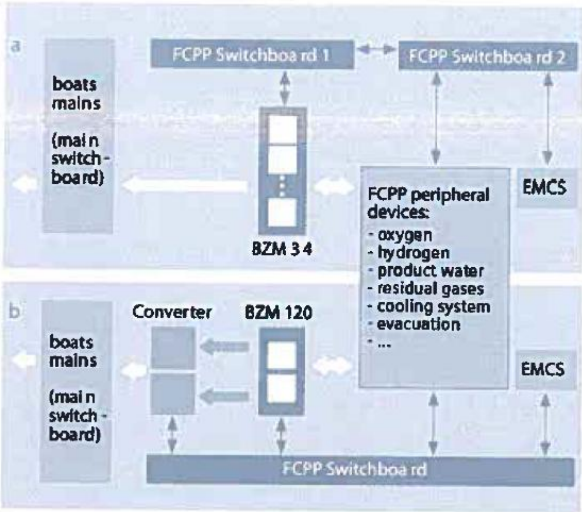

Fig. 10: Two types of fuel cell power plants (FCPP)
a: fuel cell battery with BZM 34; direct coupling of FC voltage to boats mains
b: fuel cell battery with BZM 120; coupling via converter

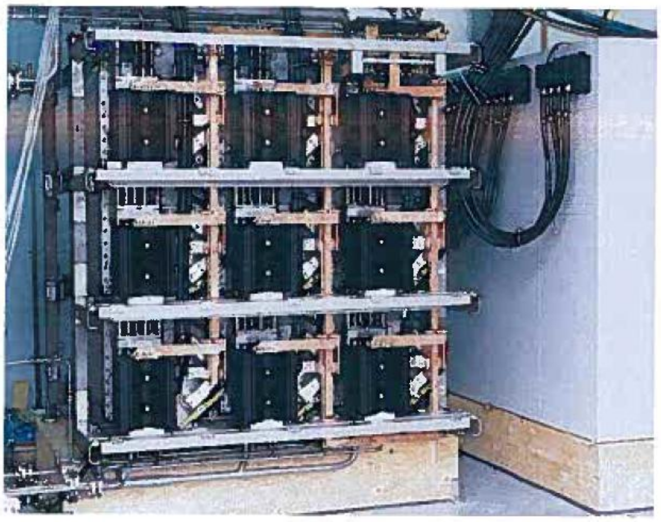

Fig. 11: PEM Fuel Cell modules assembled in a test rack

Technical data	BZM 34	BZM 120
Rated power	34 kW	120 kW
Voltage range	50–55 V	208–243 V
Efficiency at rated load, approx.	59%	58%
Efficiency at 20% load, approx.	69%	68%
Operating temperature	80 °C	
H_2 pressure	2.3 bar abs.	
O_2 pressure	2.6 bar abs.	
Dimensions	H = 48 cm	H = 50 cm
	W = 48 cm	W = 53 cm
	L = 145 cm	L = 176 cm
Weight (without module electronics)	650 kg	900 kg

Outlook

After the successful development the SINAVY^cis PEM Fuel Cell modules are now ready for application. They have proven their performance and reliability in extensive tests including long-term tests and on board of submarine U31 of the Federal German Navy. They are an integral part of an AIP system for modern submarines like that of class U 212 A and 214. An AIP section with SINAVY^cis PEM Fuel Cell modules can be added into existing submarines.

The field for use of SINAVY^cis PEM Fuel Cell will be widened when suitable reformers produce hydrogen from liquid fuels, e.g. methanol. Then it may be possible that fuel cells can become the sole power source of submarines of the future.

Using SINAVY^cis PEM Fuel Cell and replacing oxygen with air, they are an interesting alternative for environmental-friendly power generation, e.g. for vehicles in cities.

A transportable 160-kilowatt fuel cell system for emission-free power generation on board of ships has been designed. The system is housed in a container, which allows it to be brought for demonstrations and tests and to be easily connected to the ship's power supply. It is to be delivered in spring, 2005.

In general: the excellent operating performance of SINAVY^cis PEM Fuel Cell like high efficiency and noiseless operation can lead to a promising future upon further reduction in manufacturing and operating costs.

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Order No.: E10001-P19-A14-V2-7600
Printed in Germany
Dispo No.: 16600 K No.: 37306
GB C-IPMIN520M07 PA 11061.5
Subject to change without prior notice

Appendix VII: Assignment Text

NTNU
Norwegian University of Science and Technology

Faculty of Engineering Science and Technology

Department of Marine Technology

MASTERS ASSIGNMENT – SPRING 2011

for

Henrik Carlberg

Concept Design of a Commercial Submarine

Objective

The intent of this assignment is to prepare a concept design report for an autonomous manned intervention submarine for use in the offshore subsea oil and gas industry. This study will provide an opportunity to explore new concepts and innovations for use in deep water or under the arctic ice. An intervention submarine can be used to assist with tasks such as installation of new subsea facilities, intervention on existing fields for inspection, repair and maintenance, or well intervention. The submarine will consist of a one atmosphere habitat where the crew can live and work subsea for extended periods of time.

Some factors to consider:-

- Mission operations, mission profiles, and factors influencing design and requirements. Overview of current design, advanced concepts, production, and market factors
- Development of requirements into a concept meeting the constraints of underwater operations
- Concept selection/initial definition and sizing. Table of principle characteristics
- Structural design that will resist the hydrostatic and hydrodynamic forces to be encountered
- Relationships of weight, buoyancy, volumes, hydrostatics, trim, stability, tank capacity, anchoring, foundations, material selection etc.
- Health, safety and environmental issues. Subsea safety and its influence on design
- Manning, habitation, work tasks, space, area/volume summary, arrangements.
- Major systems and equipment characteristics and description. Mechanical and electrical systems. Life support systems. Heating, ventilation and air conditioning
- Determination of electric and thermal power requirements. Electrical load analysis, electrical power generation
- Illustrations and drawings, including hull form development, lines drawings, general arrangements, section drawings, arrangement sketches for work areas, crew accommodation, berthing messing and sanitary spaces. Machinery arrangement

- Rules, regulations and classification
- Weight estimates (light ship and load conditions)
- Cost analysis

Report Requirements

The report is to address the following eight areas of competency:

Fundamental Competencies:

- General Arrangement and Overall Hull and System Design
- Speed, Power and Endurance Analysis (including propulsion system selection)
- Weight, Buoyancy and Stability
- General Structural Design and Strength

Specialized Competencies:

- Mooring/station keeping. Design of dynamic positioning and anchoring systems..

Mechanical and Mission Systems:

- Definition, development, and mechanical design of specialized mission work systems. (Such as intervention tooling, pipeline monitoring and repair, handling of ROV & AUV’s, remote operated tooling etc.)

Miscellaneous:

- Life support, health, safety and environmental issues.
- Transport to/from surface, emergency evacuation, communications.

The following competencies may be touched upon while through the eight main areas of competency:

- Mechanical Engineering: Choice, arrangements and sizing of mechanical engineering systems and equipment.
- Power generation, distribution, and electric load analysis.
- Interior design and arrangement for work and habitation.
- Hull Form Development
- Cost Analysis
- Construction, fabrication and installation. Material selection.

Within 14 days of starting the assignment the candidate shall send the department a detailed plan for carrying out the work, for evaluation and discussion with the supervisor/contact persons. The thesis should be formulated as much as possible as a research report, with abstract, conclusions, reference list, contents. etc.

When preparing the thesis the candidate should place emphasis on making the text easy to read and well written. To help when reading the thesis it is important that the necessary references are made from corresponding points in the text to tables and figures. When grading the thesis a large weight is put on thorough processing of the results, and that they are presented graphically or in tables in a well arranged way and are fully discussed.

The work often forms part of a larger investigation at the department, which reserves itself the right to use all results in the masters assignment in connection with teaching, publications or other activities.

The thesis is to be submitted in 2 examples. Additional copies to co-supervisors/contact persons from cooperating companies shall be agreed with and delivered directly to them. A complete copy of the thesis shall be delivered to the department on a CD-ROM in Word-format.

Assignment handed out: 17/01/2011

Assignment to be handed in:14/06/2011

Assignment handed in:

Maurice F. White
Academic Supervisor