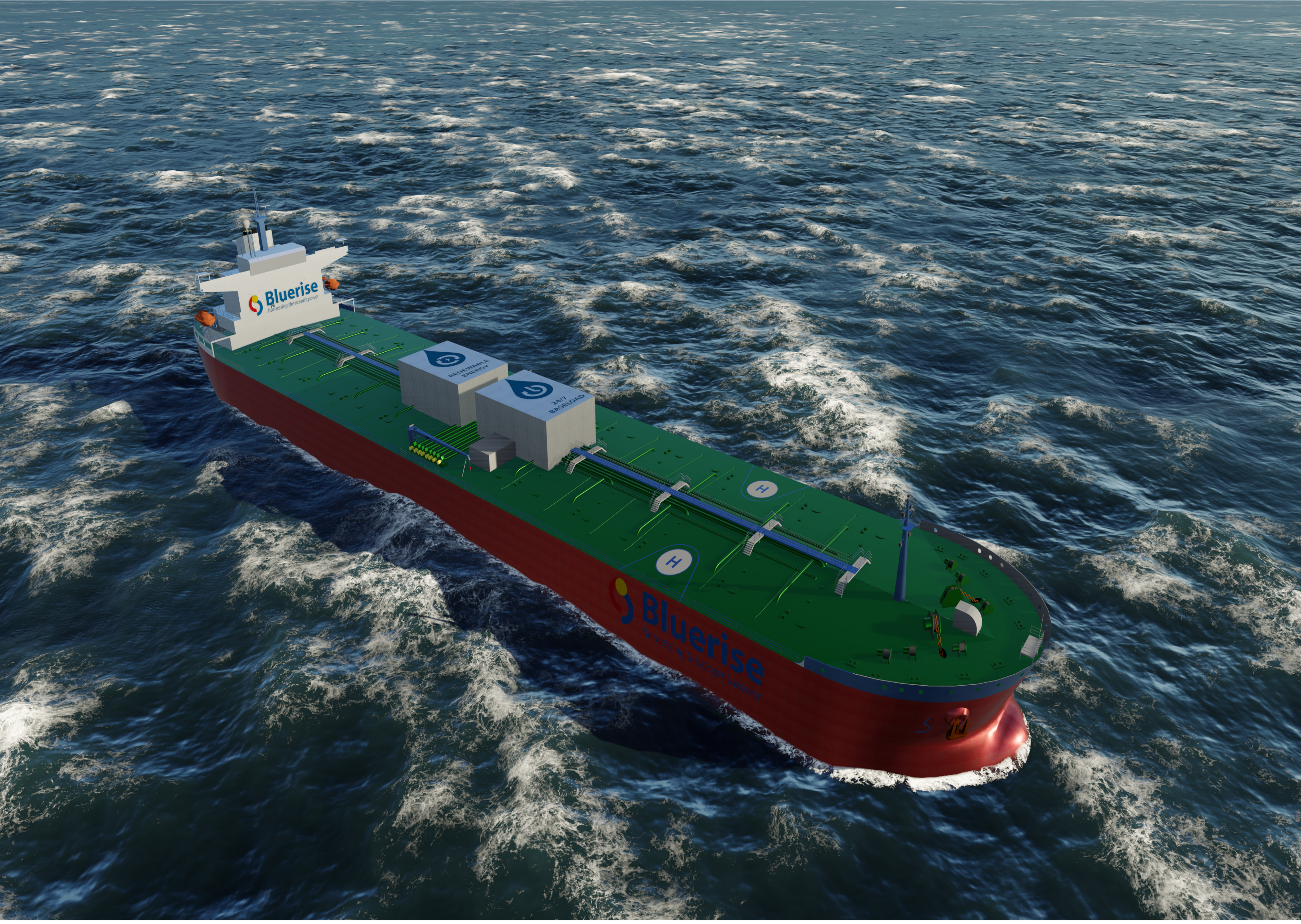

Concept Design of a 10 MW Ship-Shaped OTEC Plant

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Master's Thesis No. SDPO.17.039.m

TU Delft

Bluerise
harnessing the ocean's power

Concept Design of a 10 MW Ship-Shaped OTEC Plant

by

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This thesis is confidential and cannot be made public until December 31, 2022.

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Abstract

The purpose of the research is to provide insight into the conversion of a second-hand vessel into an Ocean Thermal Energy Conversion (OTEC) plant. To this end, a 10 MW ship-shaped OTEC platform, located in the waters around Curaçao, has been designed.

At present, only a few of OTEC plants are in operation, primarily with the purpose of research and development. These projects consist of small scale onshore platforms, the biggest of which produces a maximum of 100 kW of continuous electricity. The lack of existing large offshore OTEC installations, together with the time frame of the project, limit the project to a concept design. Ultimately, the report will serve as a guideline for the implementation of ship-shaped OTEC platforms.

The use of Systems Engineering (SE) has been motivated by the need to establish a clear structure to address effectively the challenges posed by the complexity of the installation. This methodology facilitates the revision of the performance of the ship-shaped OTEC throughout the design process.

From the initially defined Top Level Requirements (TLRs) a plant architecture has been developed, covering the engineering of the power plant, selection of mooring system and dimensioning of the second-hand Panamax bulker to be converted. The research has brought forward the main drivers of the design: process ducting, water pipes and, particularly, heat exchangers. As a result, the integration of process equipment, mooring system and hull has rendered an optimized platform layout.

The proposed solution was, then, assessed with regards to the utilization of space on board and the stability of the hull. While the ratio of occupied versus empty space is deemed adequate, it is necessary to add ballast water to increase the draft and the use of alternative solutions, such as bilge keels or fixed ballast in the top side tanks, to dimish the excess of stability of the hull.

Overall, the findings of the research suggest that the construction and operation of a ship-shaped OTEC plant is feasible. Nevertheless, a holistic validation demands for the incorporation of all new components, followed by structural and hydromechanic analysis of the platform. Lastly, it is recommended to study the use of different heat exchanger types and power ouputs that could derive in more space-efficient design.

Nomenclature

Acronyms

CALM Catenary Anchor Leg Mooring

DWP Discharge Water Pipe

ET External Turret Mooring

FS Free Surface

HX Heat Exchanger

IT Internal Turret Mooring

JSY Jacket Soft Yoke

LCG Longitudinal Center of Gravity

LMTD Logarithmic Mean Temperature Difference

LWL Length of ship at waterline

P&ID Piping and Instrumentation Diagram

RTM Riser Turret Mooring

SAL Single Anchor Loading Yoke

SALM Single Anchor Leg Mooring

SM Spread Mooring

SMP Submerged Mooring Pontoon

SPM Single Point Mooring

TEU Twenty-foot Equivalent Unit

TY Tower Yoke

VCG Vertical Center of Gravity

WWP Warm Water Pipe

AHTV Anchor Handling Tug Vessel

BIMCO Baltic and International Maritime Council

BV Bureau Veritas

CONOPS Concepts of Operations

CWP Cold Water Pipe

FPSO Floating Production Storage Offloading

IMO International Maritime Organization

MARPOL International Convention for the Prevention of Pollution from Ships

MCA Multi-Criteria Analysis

MPV Multi-Purpose Vessel

MS Maintenance Space

OPEC Organization of Petroleum Exporting Countries

OSV Offshore Supply Vessel

OTEC Ocean Thermal Energy Conversion

PCTC Pure Car and Truck Carriers

SE Systems Engineering

SOLAS International Convention for the Safety of Life at Sea

SWAC Seawater Air Conditioning

TLR Top Level Requirement

UKOOA United Kingdom Offshore Operators Association

Symbols

$\dot{m}$ Mass flow

$\dot{Q}$ Heat input

$\dot{W}$ Work output

ϵ/D Relative roughness of the pipe

η Thermal efficiency

μ Dynamic viscosity

ρ Fluid density

A Area

B Beam

C_b Block coefficient

C_p Specific heat

D (1) Diameter

D (2) Depth

E Lloyd's Equipment number

f Darcy factor

g Gravitational acceleration

GM Metacentric height

GZ Righting lever arm

H Height

h Head loss

K (2) Weight calculation coefficient

K (2) Head loss coefficient

L Length

l_1, h_1 Length and height of full width erections

l_2, h_2 Length and height of less than full width erections

Q Volumetric flow rate

Re Reynolds number

S_{min} Minimum submergence

T	(1) Temperature
T	(2) Draft
U	Heat transfer coefficient
U_f	Velocity of internal seawater
v	Flow velocity
W	(1) Width
W	(2) Weight

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1 Introduction

1.1 Background

According to UN's report on World Population Prospects [66], the world's population is projected to reach 8.5 billion by 2030 and exceed 9.5 billion by 2050, which will result in a significant increase in global energy consumption. The consequent acceleration of climate change, together with the foreseeable depletion of oil fields, calls for new energy sources to gradually take the place of hydrocarbons in the energy mix.

Small islands remain a special case for sustainable development. Tropical island countries are particularly vulnerable due to their relative geographical isolation, high population density which contrasts with the limited natural resources, and susceptibility to potentially more frequent and intense natural disasters. Additionally, their dependence on imported fuels can only aggravate the situation of these nations, already vulnerable to external economic shocks [81]. Against this background, governments in these regions are increasingly pushing towards a more resilient energy model, involving the integration of alternative renewable technologies.

Ocean energies are especially suitable for these islands, whose surrounding waters contain great amounts of untapped energy resources. In the tropics, the thermal energy from solar radiation stored in the ocean waters can be harnessed by means of Ocean Thermal Energy Conversion (OTEC), recognized as having the largest resource and economic potential of all ocean energy technologies [44].

1.1.1 Ocean Thermal Energy Conversion (OTEC)

OTEC is a clean and sustainable source of renewable energy utilizing the natural thermal gradient of the ocean to provide constant (24/7) and year-round power without energy storage.

The working principle of OTEC is based on a thermodynamic cycle that converts thermal energy into electricity. Warm surface water from the ocean is used to evaporate a working fluid with a low boiling point (e.g. ammonia). This vapor drives a turbine that is connected to an electricity generator. The use of a closed cycle, i.e. a Rankine cycle, as in Figure 1.1, enables the reutilization of the vapor, which will be condensed again using the cold deep seawater in a second heat exchanger [51].

Several studies have been carried out to prove the feasibility of OTEC technology. Currently, a number of OTEC plants are in operation, primarily with the purpose of research and development. These projects consist of small scale onshore platforms, the biggest of which produces a maximum of 100 kW of continuous electricity. Due to the required difference in seawater temperature, and, consequently, large ocean depth, the most applicable configuration for OTEC is an offshore, floating plant, which is the subject of this thesis.

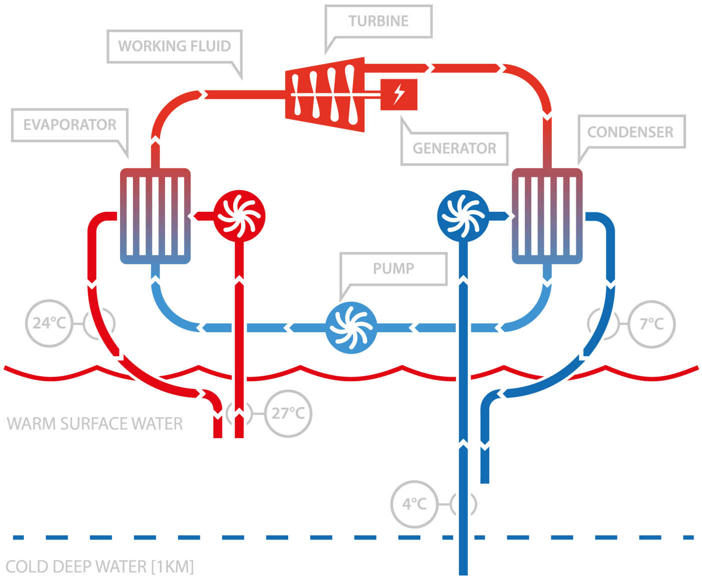


Figure 1.1: OTEC Basic Working Principle

1.1.2 Bluerise BV

Bluerise is a spin-off company from TUDelft located in YES!Delft. Bluerise is specialized in OTEC, Seawater Air Conditioning (SWAC) technologies and related Deep Sea Water applications. Bluerise is currently involved in the development and worldwide implementation of OTEC technology and systems, amongst others, in Curaçao, Sri Lanka and Colombia. This research will focus on the design of a ship-shaped OTEC plant in Curaçao, an island in the Caribbean Sea [2].

1.1.3 Motivation and State-of-the-Art

When compared to other floating strucures, the low fabrication and installation costs, the disconnectability or the off-the-shelf availability are the main upsides of ship-shaped plants [15].

The benefits of this concept have encouraged companies to invest time and effort in developing designs in the past, one of which is shown in Figure 1.2. As a result, a wide variety of solutions have been explored including different mooring and seawater pipes configurations. The lack of convergence of the designs constitutes a major motivation for this project.

Figure 1.2: Ship-shaped OTEC plant concept [19]

A second hand vessel would help keep the capital cost of an OTEC system to a minimum. Therefore, this work will focus on the conversion of an existing vessel, rather than on the design of a new build. Since the first built Floating Production Storage Offloading (FPSO) in 1977, the offshore industry has come a long way in the conversion of tankers. Now that FPSOs are proven technology, ship conversions have become an even more attractive option for OTEC systems. While the existing know-how from tanker conversions can be partially applied to ship-shaped OTEC plants, the divergence in requirements of both installations suggests the inclusion of other ship types for consideration.

1.2 Thesis Statement

The aim of the thesis is to evaluate the feasibility and actual benefits of converting a second-hand vessel into an OTEC plant. For this purpose, a concept design of a 10 MW ship-shaped OTEC plant will be developed. This thesis covers the engineering of the power plant, dimensioning and configuration of the floating structure, hull layout, mooring system and the seawater pipes.

The major challenge is to define a general arrangement in which the complex interrelations among systems are organized in a feasible design solution. The present work will explore different possible solutions for the aforementioned design drivers, taking into account their sensitivity and influence on the motions and stabilit of the plant.

The necessary capabilities of the mooring system arise from the weather conditions of the specific location of the ship in a tropical region. Therefore, several possible mooring systems must be analyzed to find the optimal solution for the ship-shaped OTEC plant. Furthermore, the dimensions of the hull and the location of the pipes have an impact on the arrangement of the equipment needed in the Rankine cycle. The final layout of the equipment and the location of the pipes must be defined following and design process that considers the limited space inside the hull and the interdependencies among the systems.

Lastly, the proposed concept must be assessed from a technical perspective in order to evaluate the feasibility of the design.

1.3 Research Questions

The prime research question of this thesis is:

What is the feasibility of converting a second-hand vessel into a 10 MW ship-shaped OTEC plant?

This can be broken down into several subquestions, which will be addressed throughout the report:

1. Which requirements, specific to a ship-shaped OTEC, will define its operability?
2. What systems are necessary for the functioning of the installation?
3. Which are the main drivers of the design?
4. What is the most efficient manner to integrate all systems in a single platform?

1.4 Report Structure

The aim of this work is to give an answer to the preceding subquestions (S.Q.) and, ultimately, the main research question. The structure of the report is, hence, broken down as in Table 1.1.

Table 1.1: Report Structure

Question	**Chapter**	**Content**
S.Q 1	Project Specifications	Preliminary analysis of design inputs Methodology, requirements and plant architecture
S.Q 2 & 3	Power Plant Engineering	Size and type of components and structure of power plant
S.Q 2 & 3	Mooring System	Selection of mooring system based on mission and location
S.Q 4	Ship Type Selection	Multi-criteria analysis of various ship types
S.Q 4	Platform Design	Dimensioning and layout of the installation

2

Project Specifications

The idea of a vessel as a complex structure is well consolidated in the industry. In particular, the innovative nature of a ship-shaped OTEC plant will bring forth unexpected challenges to the project. The large number of interfaces among systems, the company specifications, future uncertain scenarios or the environmental conditions in the region add complexity to the design. A comprehensive methodology, capable of handling the vast amount of information required to attain a robust design solution is, therefore, required. Additionally, a systematic strategy to keep track of the design process of the platform and monitor its operation after is installation must be developed.

2.1 Systems Engineering (SE)

Systems engineering can be described as "a logical sequence of activities and decisions that transforms an operational need into a description of system performance parameters and a preferred system configuration" [65].

The increasing complexity of engineering processes has spurred the establishment of systems engineering as a standalone discipline over the last few decades. Systems engineering meets the associated need for structure and clarity by providing a basis to solve problems and track requirements through the design process. As a consequence, there is a substantial reduction of unnecessary rework, while the viability of the design is ensured.

The so-called "V-model" depicts the key steps of the systems engineering process. The scope of the project is limited to left side of the diagram, specifically to the steps marked in red in Figure 2.1. The output is a design configuration, whose level of detail is dependent both on the available data and the phase of development. To ensure an optimal design and a safe and efficient operation of the plant the remaining steps of the "V-model" must be completed hereinafter.

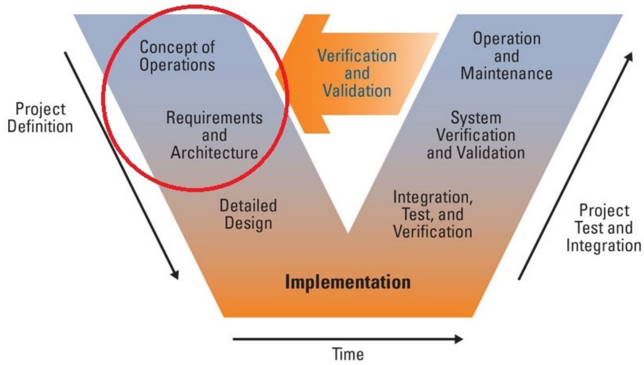


Figure 2.1: "V-Model" of the Systems Engineering Process [71]

2.1.1 Application of SE to the Ship-Shaped OTEC

Today, “ship designers are filling the role as Systems Engineers and merging the two processes together” [78]. SE helps ship designers by providing a solid basis to manage complexity and reduce risk during the design process. In particular, SE is able to approach the design of a new build vessel covering these aspects: structural, the arrangement and interrelations of systems; behavioral, the form-function mapping; contextual, the external circumstances; temporal, the uncertainties and changes over time; and, perceptual, the interpretation of the system by the stakeholders [30].

Due to the time frame and available resources, the scope of this thesis is limited to a Concept Design. The V-model in Figure 2.1 will be adjusted accordingly. As a result, the engineering process of the ship-shaped OTEC will encompass four phases:

1. Concept of Operations - analysis of the state-of-the-art technology and additional requirement inputs.
2. Requirements and Architecture - definition of requirements and functional and physical architecture of the installation.
3. Detailed Design - Concept Design, i.e. transform the architecture of the ship-shaped plant into a "preferred system configuration".
4. Verification and Validation - Evaluation of the final design.

The focus of this work will be on Point 3, but the previous two are necessary to attain a clear design cycle through which requirements can be periodically revisited.

Additionally, several inbuilt features, which differ from a conventional ship design project, will complicate the design and need to be taken into consideration:

- Limited information about former similar design projects and nonexistence of operational ship-shaped OTEC plants.
- The necessary electrical and process equipment depend solely on the required power output of the plant and cannot be adapted to the structure of existing hulls.
- Fixed structure shape not purposely designed for its newly requested function and permanent location.

2.2 Preliminary Analysis of Design Inputs

There are multiple sources of information on the systems requirements, the primary ones being customer objectives, as given by Bluerise, environment in which the plant operates, and existing technology base. Altogether, company specifications, environmental conditions in the operation site and the available technology limit the amount of feasible designs that may result from the research. This analysis will provide basis for the posterior definition of requirements.

2.2.1 Company specifications

The project terms given by Bluerise start from the premise that the present concept design is the first step towards the implementation of a ship-shaped OTEC plan in Bluerise, and that very limited information about similar works is accessible.

- Capacity: 10 MW
- Location: Curaçao
- Design life: at least 20 years
- Ship-shaped
- Motions within acceptable values for safe operations

In the present research , these specifications, which describe the characteristics of the proposed system from the company's perspective, will substitute the Concept of Operations (CONOPS) in Figure 2.1. In a real tender for a contract these CONOPS will outline the project specifications as stated by the future user.

2.2.2 Technology base

At present, very few OTEC plants in the world are operational, the biggest of which generates only 105 kW and is located on Big Island, in the Hawaiian archipelago. The low electrical capacity of this largest installation indicates that OTEC technology is still in a pilot phase. What is more, all existing plants are land-based, i.e. there are no offshore operating OTEC platforms.

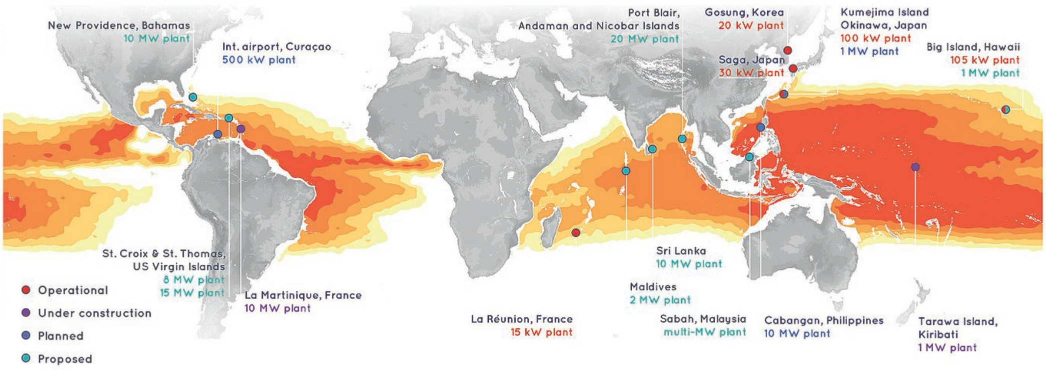


Figure 2.2: Current State of OTEC Plants Worldwide [34]

Nevertheless, several designs of offshore OTEC platforms have been carried out by engineering companies like Lockheed Martin [58]. These projects demonstrate to some degree the feasibility of the concept, which needs, however, to be constructed and operate for a certain period of time to prove definitively the practicality of the technology.

The OTEC-1 platform, *Chepachet*, became, in 1979, the greatest step towards the development of a practical system to extract the thermal energy of the ocean [72]. Even though the prime objective was to test heat exchangers, and not the viability of a ship-shaped platform, the construction involved the conversion of a tanker into an OTEC plant. The operation of *Chepachet* was suspended due to lack of funding but the engineering activities for its implementation can be, in a way, extrapolated to the present work.

Other companies, among them, SBM, have attempted the design of a ship-shaped OTEC plant ([49], [59]). The outcome of these studies has been the development of a variety of possible solutions for the arrangement of equipment on board. In spite of not containing much details, these projects do address the major obstacles for the concept examined, such as the Cold Water Pipe (CWP) deployment or the selection of the vessel to be converted. The interest and effort of large corporations as SBM on the concept is a good sign of its relevance.

Lastly, as formerly stated, FPSO conversion share certain features that make them, conceptually, an interesting starting point of the study. For instance, despite their smaller diameter, the long risers used for oil extraction are nowadays the closest proven technology to the CWP. Additionally, the mooring system, or the refit of the equipment on board, while devised following different requirements and constraints constitute a useful basis for the design of the present concept.

2.2.3 Environmental Data

To ensure an efficient cycle performance, the temperature difference between surface and bottom water has to be no less than 20 °C. Accordingly, OTEC is only suitable for regions where the thermal gradient of seawater is high and constant. The tropics are, thus, the areas with the largest resource potential 2.2. In particular, tropical islands, which currently import fossil fuels at a very high price, can benefit significantly from this technology. This, together with the previous experience of Bluerise in Curaçao, where they are currently building a 500 kW facility, were the main reasons for the selection of Curaçao as geographical location of the ship-shaped OTEC.

The temperature of the surface water around Curaçao, as depicted in Table 2.1, is indeed constant throughout the year. On the other hand, seawater at depths of 1000 m maintains a steady temperature of 5 to 7 °C.

Table 2.1: Seawater Temperature in Curaçao [3]

Month	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**
T (°C)	26	26	26	26	27	27	27	28	28	28	29	27

The required temperature profile implies, in turn, the need to install the platform in very deep waters. As it can be observed in Figure 2.3, the slope of the seabed around Curaçao is sufficiently steep to locate the platform at a close distance from shore. As a result, the costs and effort of the cable laying operations will be reduced.

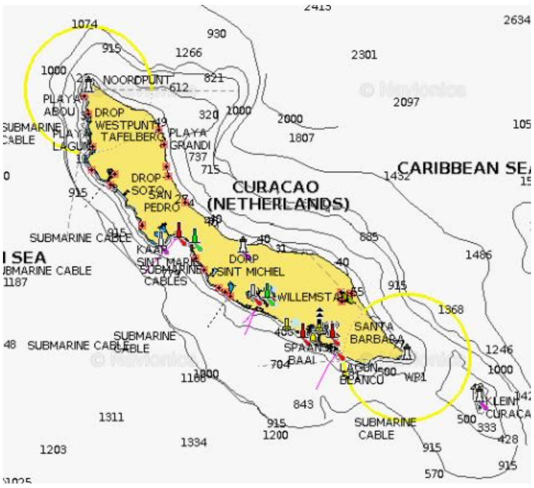


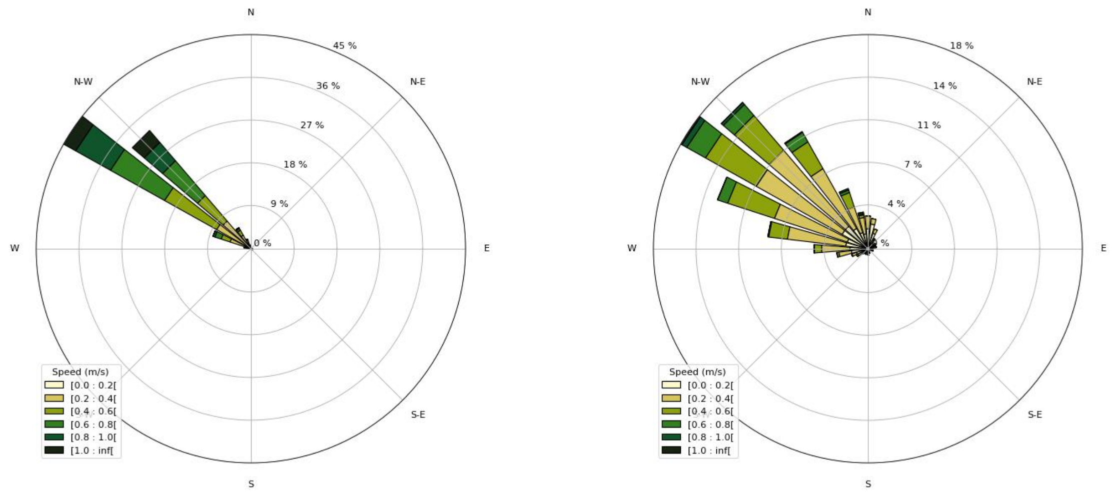
Figure 2.3: Water Depths around Curaçao [69]

During operation the projected installation has to withstand the environmental loads of the specific site. These loads are mainly caused by currents, wind and waves of the location. In this study, no specific location will be chosen. Nevertheless, the environmental conditions in the waters surrounding the island of Curaçao are examined in the following lines, to provide insight into the weather phenomena that the installation will have to endure.

- Current

An inflow of water, the Caribbean Low Level Jet (CLLJ) enters the Southern Caribbean from the East and is later divided into four streams that travel in a north-westerly direction to the Yucatan Channel [47]. The velocity of the jet in the winter, i.e. in the months between December and March. A maximum current of 2 $\mathrm{m\,s^{-1}}$ is found in the south of the Venezuela Basin, between the mainland of Venezuela and Curaçao.

An exhaustive exploration of the currents in an area of 100 km of radius around Curaçao was carried out by Lems-de Jong in [54]. The currents travelled north-west with average velocities of 0.5 $\mathrm{m\,s^{-1}}$ in the sheltered region, i.e. the southern part, of Curaçao, and of 0.3 $\mathrm{m\,s^{-1}}$ in the unobstructed area, i.e. the northern part of the island. Additionally, the study points out the presence of subsurface variable currents, both in the north and south of the island, featuring a velocity of 0.07 $\mathrm{m\,s^{-1}}$ and 0.13 $\mathrm{m\,s^{-1}}$, respectively.



(a) Channel Region, defined as [69.43° W, 11.6° N]to [69.03° W, 12.2° N]

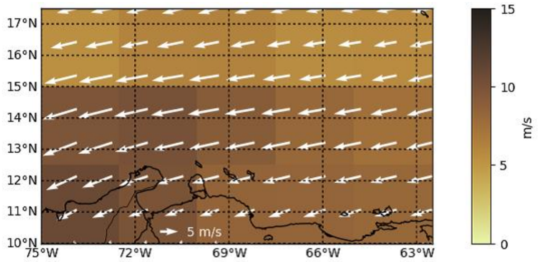
(b) Unobstructed Region, defined as [69.03° W, 12.2° N]to [68.5° W, 13.0° N]

Figure 2.4: Rose plots of surface current velocity [54]

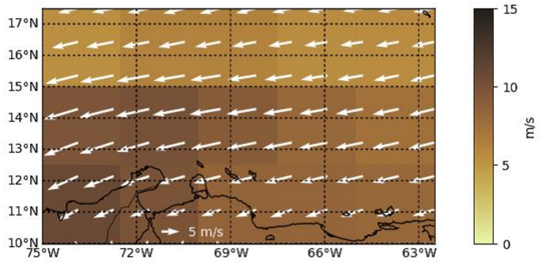
The characterization of currents surrouding the island was then used to predict the effect of currents on the CWP. This analysis showed that, for a selected location, a maximum deflection of 370 m at the tip could be expected, while the pipe would experience a stress of 6.7 MPa at the hull-pipe interface, well below the limit of the material (HDPE). Consequently, it can be assumed that wind and, most importantly, waves will be more compromising for the operability and safety of the plant.

- Wind

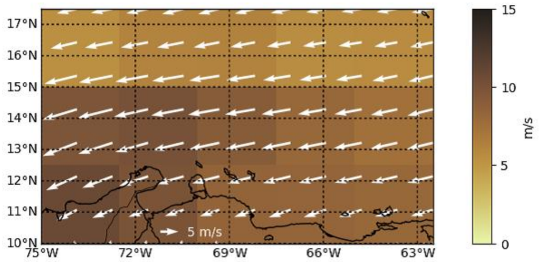
The predominant wind direction over the year is from the East. The mean wind speed is around 7 $\mathrm{m\,s^{-1}}$ (13.6 kn), which corresponds to a value of 4, or "Moderate breeze" in the Beaufort scale (Figure 2.5). However, the maximum wind speed registered over the period between 1987 and 2011 was above 20 $\mathrm{m\,s^{-1}}$ (38.9 kn), a value of 8, or "Fresh gale" in the Beaufort scale, close to the North of the island as illustrated by Figure A.1 in Appendix A.



(a) Mean Wind Speed from December to March



(b) Mean Wind Speed from April to August



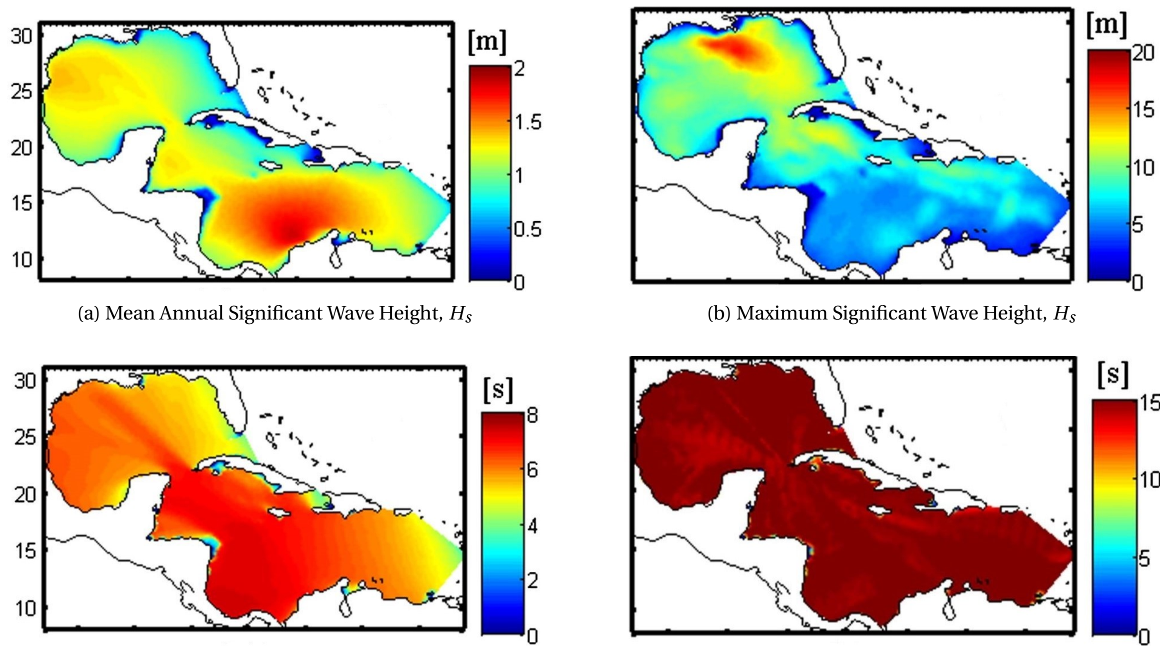
(c) Mean Wind Speed from September to November

Figure 2.5: Long Term Wind Speed in Caribbean Sea [54]

- Waves

Figure 2.6 shows the mean significant wave heights and maximum wave heights in the Gulf of Mexico and the Caribbean Sea. The figure indicates the moderate wave conditions around Curaçao, with a mean H_s around 1.25 m, and a maximum wave height in storm around 4 m, corresponding to a degree 3 (slight) and 5 (rough) of the Douglas Sea Scale, respectively. The extreme wave climate in Curaçao is shown seasonally in Figure A.2 in Appendix A. The wave periods are of particular relevance in relation to the natural frequency of the installation. The direction of the waves varies largely for each specific position. Nonetheless, a certain level of correlation with the direction of the wind can be expected.

Despite the mild environment at the edge Southern Caribbean Sea, Curaçao is positioned on the southern fringe of the hurricane belt. Historically, tropical cyclones on their path through the area have affected the islands roughly every 100 years. Once every four years a hurricane passes within a radius of 150 km, having only minor consequences on the islands [67] .



(a) Mean Annual Significant Wave Height, H_s

(b) Maximum Significant Wave Height, H_s

(c) Mean Peak Wave Period

(d) Maximum Peak Wave Period

Figure 2.6: Wave Climate in Gulf of Mexico and Caribbean Sea 1979 - 2008 [10]

Altogether, the platform and mooring system will need to be designed to withstand the loads induced on the structure by the weather conditions in the location and for a period of time of 20 years. The lack of available data for concrete locations will limit to a certain extent the feasibility of the subsequent design, which will have to be tested for the final operational site in the Verification phase.

Table 2.2 summarizes the core design considerations based on the aforementioned aspects.

Table 2.2: Design Inputs for the Definition of TLRs

Specification	**Detail**
Capacity	Annual production of 10 MW. The possibility of future upgrades of the plant is not considered within the scope of the project
Location	Curaçao
Design life	Minimum of 20 years
Operability	Platform motions in waves up to 5 m must be kept within allowable values for the performance of the components on board
Availability	90 % uptime (10 % downtime for environmental factors, maintenance and repair)
Accelerations	The lateral accelerations may not exceed 0.19 g in 100 year events. The vertical acceleration of the process equipment may not exceed 0.08 g [Lockheed]
Accessibility	The plant is originally designed to be unmanned, however, accessibility is required for maintenance and monitoring
Survivability	100 year storm. In case of extreme weather occurrence the platform must remain in place without CWP detachment

2.3 Top Level Requirements (TLR)

The definition of requirements as “statements of the problem to be solved” [55], illustrates the importance of a clear and unambiguous formulation. They represent the customer needs and the expected performance throughout the system’s life-cycle. The interfaces of the system, both internally and externally, give an indication of the constraints and boundaries of the system. Simply put, requirements must answer three questions: “what”, “how” and “how well”.

Requirements are meant to be achievable, objectively verifiable, consistent with other requirements and appropriate for the ongoing design phase. Henceforth, for the purpose of this work, only TLRs are defined, in accordance to Section 2.2.

1. To produce a yearly average of 10 MW net of electricity for the island of Curaçao.
2. To integrate all components necessary to create the specified energy output in the most efficient arrangement possible.
3. To transfer the electricity generated to a substation onshore.
4. To be stable and to maintain platform motions within allowable margins for the equipment to operate safely.
5. To remain in place for at least 20 years. The platform must be able to withstand extreme environmental conditions (100-year storm) without detachment.
6. To be accessible, by air or sea, for maintenance and monitoring.

2.4 Plant Architecture

The aim of functional allocation is to translate the stated requirements into system functions, and these, in turn, into physical solutions. Because the equipment of the platform is, to a large extent, known, this section will cover the association of functional characteristics to components.

2.4.1 Functions Definition

By clustering subsystems in functional groups, the functional interfaces that were initially undefined become more clear. In Figure 2.7 four first-level functional groups are identified. The subfunctions arising from the decomposition of each of these into a second level will be later on linked to the plant systems. Figure 2.7 provides a coherent basis of system functionality that will guide the future Detailed Design.

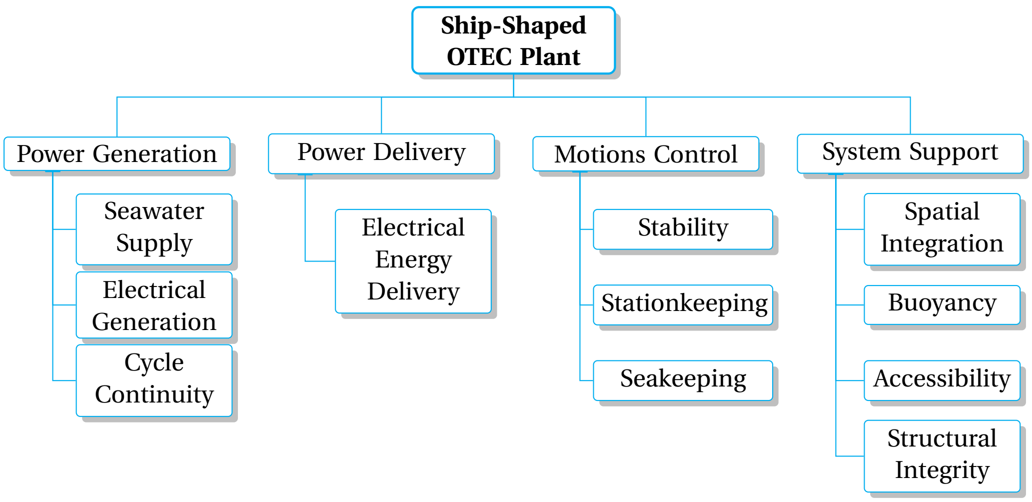


Figure 2.7: Functional Architecture of the Ship-Shaped OTEC Plant

Each of the prime functional segments in Figure 2.7 are defined below:

- Power Generation: to supply the necessary warm and cold seawater to create the required electricity production per year. To utilize this water to extract the thermal energy from the temperature difference and transform it into electricity.

- Power Delivery: to transmit the generated electrical energy in the floating offshore structure to a substation onshore, while minimizing energy losses.

- Motions Control: to maintain platform motions within allowable limits to ensure the operability and survivability of the plant.

- System Support: to physically support all of the other OTEC systems of the plant and to act as interface between them.

In Table 2.3, the TLRs are linked to the functions of the ship-shaped OTEC plant identified above.

Table 2.3: Correlation of TLRs and Functions

TLRs	Functions
1	Power Generation
2	Systems Support
3	Power Delivery
4	Motions Control
5	Motions Control
6	Systems Support

2.4.2 Systems Architecture

The next stage is to define a System Architecture hierarchy. Subsection 2.1.1 put forward that both the structure and the equipment of the plant are fixed. Consequently, the System Architecture of the concept is the same as that of other offshore installation designs, with the sole difference of the shape of the hull.

This structure differs from the preceding diagram in that now the system is separated from a physical perspective into smaller more manageable units. It is shown in Figure 2.8 that the design is confined to a system level, and thus the focus is on system integration rather than component engineering.

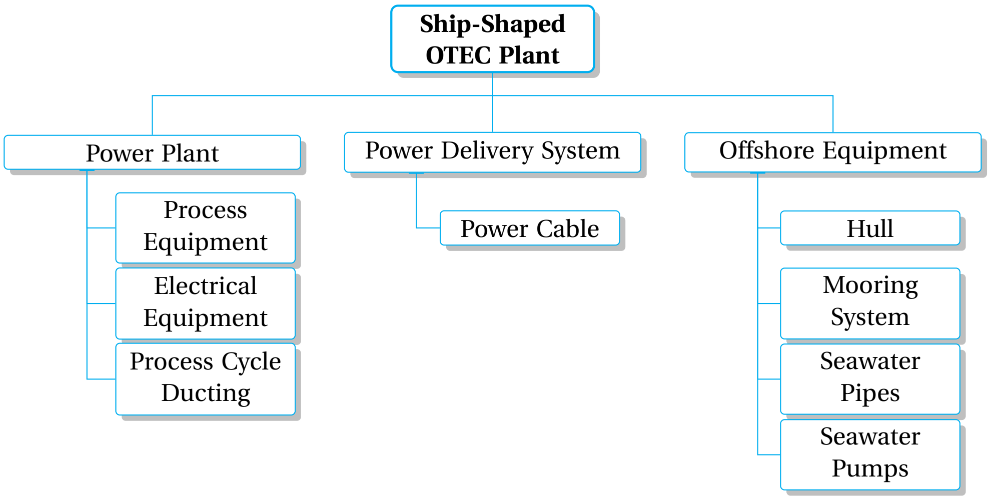


Figure 2.8: Equipment Scheme of the Ship-Shaped OTEC Plant

2.4.3 Functional Allocation

The foregoing functions are allocated to systems in Table 2.4 and can be readily tracked and verified during the design process.

Even though most functions are performed by individual subsystems, it can be seen that some of them are carried out by the combination of subsystems, especially in the case of the hull and the mooring system. Both components are responsible for the operability, survivability and accessibility, due to their intrinsic relation to platform motions. Even more so, while the stability and spatial integration are subfunctions carried out by the hull, the selection of the mooring system will have a direct relation with them.

While keeping in mind the necessary integration of all functions, the focus of the research is on the cells colored in yellow in Table 2.4, deemed more relevant from a ship design point of view.

The underlying rationale behind this is based on the following considerations:

- Structural and hydromechanic studies, e.g. fatigue or seakeeping analysis, will not be included in this design approach, and are considered an area for future work.

- The layout of the complete power delivery system, specifically the power cable and electrical components must be addressed from an electrical engineering perspective. Hence, only solutions for the attachment point of the power cable to the hull will be explored.

Table 2.4: Correlation of Plant Functions and Systems

		Power Plant			**Power Delivery**	**Offshore System**		
		Process Equipment	Electrical Equipment	Process Ducting	Power Cable	Hull	Mooring System	SW Pumps & Pipes
Power Gen.	Seawater Supply							X
	Electrical Generation	X	X					
	Cycle Continuity			X				
Power Delivery	Electrical Power Delivery				X			
Motions Control	Stability					X		
	Seakeeping					X	X	
	Stationkeeping						X	
Systems Support	Spatial Integration					X		
	Buoyancy					X		
	Accessibility					X	X	
	Structural Integrity					X		

The output the Functional Allocation becomes hereafter the input of the subsequent Concept Design. Bearing in mind the defined relations between systems and functions, the next chapters will examine possible engineering solutions to achieve the most optimal design for the ship-shaped OTEC plant.

2.5 Design Process

The structure of the design process is conditioned by two main aspects:

- The primary mission of the platform is to supply a yearly average of 10 MW to the island of Curaçao. The use of a second-hand vessel is a specific prerequisite of this work, but in reality is one of the many possible solutions for the installation of the power plant.

- The engineering process has to be approached from a ship design perspective. That is to say that the aforementioned challenges have to be adressed by finding the most suitable ship for conversion and adjusting the layout to the requirements of process equipment and mooring system components, and not otherwise.

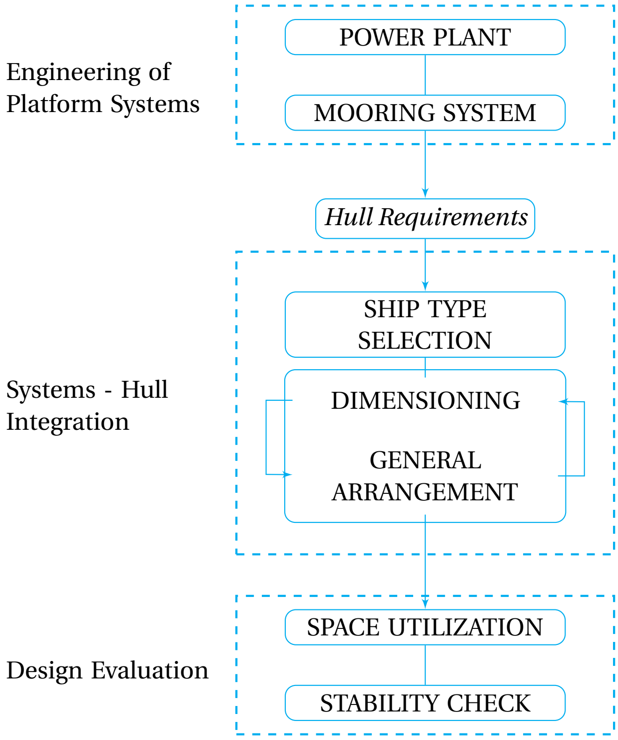


Figure 2.9: Design Process Structure

3

Power Plant Engineering

The present chapter describes the calculations and equipment selection of the power plant, and, in particular, the process cycle components. This engineering process will bring forward the characteristics and spatial requirements of the later hull selection and general arrangement.

It must be remembered that the purpose of this research is not to develop a detailed power plant layout, but to examine the benefits of converting an existing vessel into a 10 MW OTEC plant. Moreover, the introduction of the state-of-the-art OTEC technology in Subsection 2.2 anticipated the paucity of reports discussing OTEC plants of this size, especially in the case of ship-shaped platforms. As a consequence, the results obtained are constrained by the limited availability of data sources.

3.1 Basic OTEC Calculations

As stated in Subsection 2.3, the first TLR of the platform is the generation of 10 MW of electricity per year. Some steps further, the function defined as "Power Generation", divided in "Seawater supply" and "Electrical Generation", is directly correlated to the seawater pumps and the process and electrical equipment, respectively. The following lines will therefore be a transition between the functional and physical architecture of the power plant.

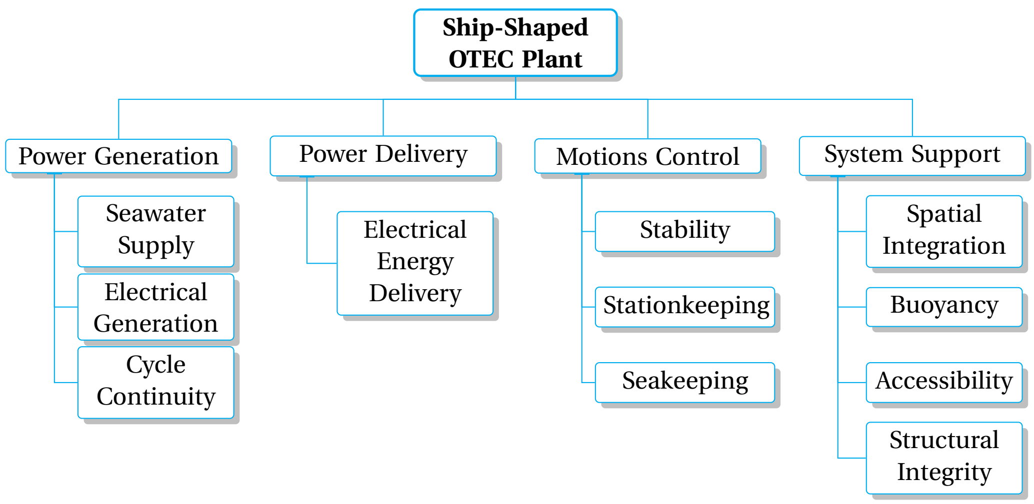


Figure 3.1: Functional Architecture (repeated from Figure 2.7)

The working principle of OTEC has been outlined in Subsection 1.1.1. The Rankine closed cycle makes use of the temperature difference between the hot source, i.e. warm water from the surface, and the cold source, i.e. deep ocean water. Paralleling the process to a conventional heat engine, the maximum thermal efficiency according to Carnot is:

$$\eta_{max} = \frac{W}{\dot{Q}_H} = 1 - \frac{T_C}{T_H} = 1 - \frac{278}{300} = 7.33\% \tag{3.1}$$

Where:

η_{max} = maximum thermal efficiency
W = work output (W)
$\dot{Q}_{\mathrm{H}}$ = heat input (MW)
T_{H} = temperature of the warm water reservoir (K)
T_{C} = temperature of the cold water reservoir (K)

As illustrated in the Carnot heat engine diagram in Figure 3.2.

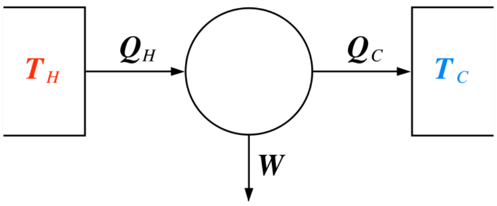


Figure 3.2: Diagram of Carnot Heat Engine

This value is further decreased when the irreversibility and inherent inefficiencies (e.g. turbine, heat exchangers, etc.) of the cycle are accounted for. The resulting final gross efficiency is around 3 – 3.5 %.

At the same time, analysis show that around 20 to 25 % of the electricity production of the plant is needed to operate the water and working fluid pumps and to supply power to the auxiliary systems and accommodation block of the platform.

$$Net\ Power = Gross\ Power - Pump\ Consumption = 0.80 \times Gross\ Power \tag{3.2}$$

Since the originally stipulated power capacity is 10 MW, the gross work output of the plant has to be:

$$Gross\ Power = \frac{10}{0.8} = 12.5\ \text{MW} \tag{3.3}$$

Which according to Equation 3.1 results in a required heat input from the hot reservoir of:

$$\dot{Q}_H = \frac{W}{\eta_{max}} = \frac{12.5}{0.035} = 357.1\ \text{MW} \tag{3.4}$$

And a heat output to the cold reservoir of:

$$W = \dot{Q}_H - \dot{Q}_C \rightarrow \dot{Q}_C = 357.1 - 12.5 = 344.6\ \text{MW} \tag{3.5}$$

The variation of the temperature of both warm and cold water will subsequently determine the mass flow of warm and cold water needed to attain the heat input and output from Equation 3.6. Temperature data is as taken from simulations performed using an ammonia-water mixture as working fluid [50].

$$\dot{Q} = \dot{m}_w C_p \Delta T \tag{3.6}$$

Where:

$\dot{Q}$ = heat transfer (MW)
$\dot{m}_\mathrm{w}$ = mass flow of water ($\mathrm{kg\,s^{-1}}$)
C_p = specific heat (J/kgK)
ΔT = temperature variation of water through the system (K)

The temperature variations and resulting water mass flows are included in Table 3.1.

Table 3.1: Mass Flows of Warm and Cold Water

	Warm Water	**Cold Water**
Δ T (K)	3.08	7.42
C_p (J/kgK)	4179	4205
$\dot{Q}$ (MW)	357.1	344.6
$\dot{m}_w$ ($\mathrm{kg\,s^{-1}}$)	27747.2	11045.9

It must be here noted that the higher temperature variation of the cold water derives in a much lower mass flow, over twice that of the warm water.

3.2 Redundancy and Modularity

Redundancy is customarily implemented to increase system reliability and availability. Parallel structures or k-of-n structures, as in Figure 3.3, are typical examples of equipment redundancy. These configurations are introduced for components deemed vital for the operational efficiency or the safety of the plant. For all that, improved reliability incurs higher costs, which means that some trade-offs have to be considered [18].

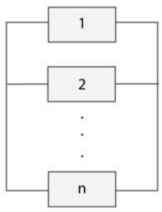


Figure 3.3: Simple Parallel System Configuration

It is evident that making the power plant fully redundant would not be plausible, since it would imply doubling the total capacity of the installation. However, it is equally clear that interrupting the supply of steady baseload electricity to the island due to the failure or maintenance of components would not be acceptable.

As a solution, in the present research it is proposed to project two independent modules, each of them generating 50 % of the required rated capacity of the plant, i.e. 5 MW. Moreover, redundancy will be introduced, in a smaller scale, for specific components of the system, e.g. the water pumps.

Other side benefits arising from a modular concept are:

- The engineering procedure of a single module is simpler and can be replicated for the second one. Indirectly, this may also enhance the scalability of the power plant.
- The failure of one of the modules will not affect the operation of the other module, thus considerably reducing the risk of a complete blackout. Conversely, the possibility of shutting down half of the plant makes maintenance scheduling easier.

- The installation time and cost may be decreased, thanks to the lower system complexity. Even more, the platform could start delivering electricity prior to the installation of the second module.
- The piping system is expected to be less complicated for a modular system than for a fully redundant installation.

Section 3.3 and 3.4 set out more detailed information regarding specific components of the process cycle. The implications of the reliability scheme suggested above are reflected in the next two sections, as well as in Section 3.7.

3.3 Seawater Supply

The large warm and cold mass flows of water in Table 3.1 are demanded continuously for the process to generate electricity without disruptions. The supply and discharge of seawater is, hence, key to the efficient functioning of the power plant (see Figure 2.7).

3.3.1 Pipes

For simplicity of the pilot plant only one pipe for warm, cold and discharge water, i.e. three pipes in total, are incorporated in the design. This choice also arises from the foreseeable complexity of the motion coupling of platform and pipes and the importance of maintaining the structural integrity of the hull. Multi-pipe solutions could be analyzed once the practicality of building a ship-shaped plant has been proven or for other floating structures that allow for a more flexible system arrangement.

The location of the warm and cold water intakes and the discharge outlet have a strong influence in the layout of components due to their direct link to the evaporators and condensers. Additionally, the diameter of the pipes will contribute to the spatial requirements of the platform. This will be studied in depth in Chapter 6.

These diameters can be found from the definition of volumetric flow rate in Equation 3.7. The relation between the volumetric and mass flow rate is presented in Equation 3.8.

$$Q = Av \tag{3.7}$$

Where:

$$\begin{aligned} &Q &&= \text{volumetric flow rate } (\mathrm{m^3\,s^{-1}}) \\ &A_{\text{pipe}} &&= \text{cross sectional area of the pipe } (\mathrm{m^2}) \\ &v &&= \text{flow velocity of water inside the pipe } (\mathrm{m\,s^{-1}}) \end{aligned}$$

$$Q = \frac{\dot{m}}{\rho} \tag{3.8}$$

Where:

$$\rho = \text{fluid density } (\mathrm{kg\,m^{-3}})$$

Table 3.2: Diameter of Water Pipes

	CWP	**WWP**	**DWP**
v ($\mathrm{m\,s^{-1}}$)	2	1.5	1.5
Q ($\mathrm{m^3\,s^{-1}}$)	10.72	27.07	37.79
A ($\mathrm{m^2}$)	5.36	18.05	25.20
D (m)	2.61	4.79	5.66

A normal flow velocity of 1.5 $m s^{-1}$ of water inside the pipe has been applied as in previous studies ([11], [15]). However, in the case of the cold water pipe, a higher value is assumed to make a reduction of the diameter possible. In light of the state-of-the-art technology, a smaller diameter may play a significant role in the feasibility of the installation of the long cold water pipe.

The length of the water pipes are, on the other hand, provided by Bluerise, as follows:

- CWP: 1000 m
- WWP: 20 m
- DWP: 90 m

The CWP, being the longest one, is considered a critical element of the installation. The CWP is made of high density poly-ethylene (HDPE), which is easy to manufacture and deploy and readily available for diameters up to 2.5 m, slightly smaller than the previously calculated diameter, 2.6 m. The density of HDPE is 950 $kg m^{-3}$, which means that it is necessary to add ballast to counteract the buoyancy force. Additionally, this ballast weight, attached to the tip of the pipe, as illustrated in Figure 3.4, will limit the lateral deflection of the pipe.

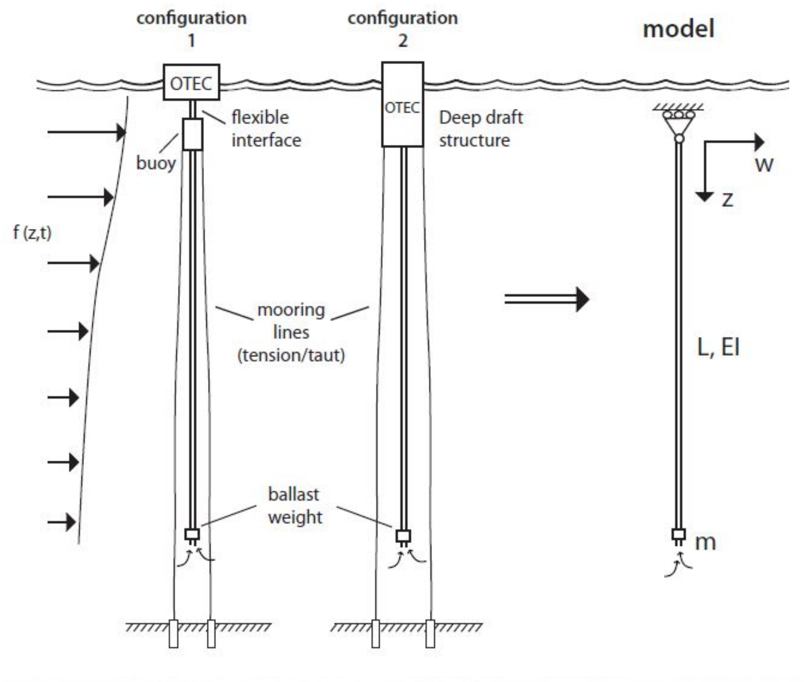


Figure 3.4: Platform - pipe configurations and model for a 10MW offshore OTEC plant [51]

Taking a wall thickness of 0.3 m, the volume and, subsequently, the weight of the CWP can now be calculated.

$$V = \frac{\pi(D_{out}^2 - D_{in}^2)}{4} L \tag{3.9}$$

$$V = 1154.5 \text{ m}^3$$

Where:

D_{out} = outer pipe diameter (m)
D_{in} = inner pipe diameter (m)
L = pipe length (m)

From this, the weight of the pipe, i.e. 1096.8 t, and the buoyancy, i.e. 1183.4 t, are obtained by multiplying the volume by the correspondent densities. Moreover, the ballast attached to the tip of the pipe adds a weight of 742 t [51]. Finally, if all forces are summed, the total load that the CWP exerts on the hull will be 656 t. This value will be taken into consideration when the weight of power plant systems is studied.

In addition to the foregoing, the large dimensions of the water pipes derive in a non-negligible flow around them which, together with the foreseeable coupling between platform and pipes, confirms the need to minimize their motions. In Figure 2.4 the function "Motions Control" was correlated with the hull and the mooring system, inferring that an optimal solution should be approached from both sides.

In conclusion, given the decisive influence of the pipes on most of the platform functions they will be considered design drivers hereafter.

3.3.2 Pumps

As discussed above, "Cycle Continuity" is an indispensable prerequisite for the functionality of the plant. The water pumps have to overcome the head losses in the system and ensure the water flow inside the pipes. The discharge of water is assumed to be done by gravity.

Table 3.3 breaks down the head losses in the system, caused mainly by the inlet and oulet losses, friction inside the pipes, hydrostatic head and pressure drop inside the heat exchangers. More detailed calculations can be found in Appendix B.1.

Table 3.3: Head Loss in Seawater Pipes

	CWP (27 °C)	**WWP (5 °C)**	**DWP (20 °C)**
Inlet loss (m)	0.0408	0.0229	-
Friction loss (m)	1.0443	0.0057	0.0210
Hydrostatic head (m)	2.3	-	-
Pressure drop (HX) (m)	3.5	3.5	-
Outlet loss (m)	-	-	0.1147
Total head (m)	6.89	3.52	0.14

In line with Section 3.2, a pump station, shared by the power modules, will be installed to provide 150 % of the total water flow required, both for cold and warm water. To achieve system redundancy the pumps will be arranged in parallel (Figure 3.5). The capacity of each of the pumps will cover the already mentioned 50 %.

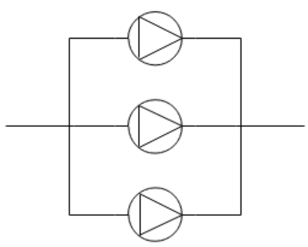


Figure 3.5: 3-Pump Configuration

With these requirements, i.e. flow rate, head and configuration, Flowserve was contacted during the design to attain reliable data of a suitable type of pump and its corresponding dimensions. The data of the axial flow pump as given by the manufacturer is attached in Appendix C.1.

Table 3.4: Dimensions of Water Pumps, Flowserve (Appendix C.1)

	CW	**WW**
Discharge Diameter, D (m)	1.37	1.98
Inlet Diameter, D_{inlet} (m)	1.87	2.79
Height, H (m) (excluding motor)	8.74	9.59

Figure 3.6: Axial Flow Pump

The total height and minimum submergence of the pumps will determine the height of the power plant deck. Despite this, once the deck height is determined, the location of the pumps is dependent on the water pipes and not vice-versa. Thus, from a layout perspective, they are not considered a design driver on their own.

3.3.3 Sumps

A proper intake should be achieved to maintain an optimal performance of the components in charge of the "Seawater Supply" in the system. In this case, sumps will be constructed to connect the pipes and pumps and integrate them in one structure, which will contribute to the spatial requirements of the platform.

According to the American National Standard for Pump Intake Design [38], there are several hydraulic phenomena that can adversely affect the operation of the pumps:

- Submerged vortices
- Free surface vortices
- Excessive pre-swirl of flow entering the pumps
- Non-uniform spatial distribution of velocity at the impeller eye
- Excessive variations in swirl and velocity with time
- Entrained air or gas bubbles

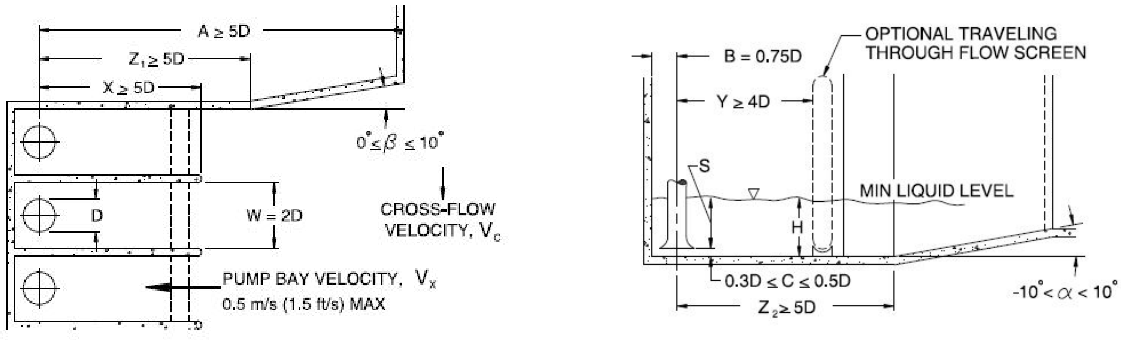
These conditions may cause reduced flow rate and head, and increased vibration and noise. Thus, these harmful effects have to be kept within allowable values. The aim of this subsection is not to perform a detailed analysis of all hydraulic phenomena, but rather to propose an initial sump arrangement that minimizes them.

The shape of the sump will affect the formation of swirls and vortices inside the sump and inlet pipes of the water pumps. Besides these complex flow patterns, the movement of the floating platform poses an additional challenge for the design: free surface (FS) effect. In this regard, two different sump configurations are examined: circular and rectangular sumps.

A circular sump, built as a caisson, offers a more compact solution and, accordingly, a smaller water surface, which will ultimately result in a smaller FS effect. On the other hand, circular shapes are expected to generate more swirl flow and vortex than a rectangular shape. For that reason, the latter is used in most pumping stations, although this solution is not always fully effective and needs to be tested for each design [77]. Both sump types are constrained by the maximum individual capacity of each pump [76]. This limitation is, however, smaller for circular sumps. Altogether, a rectangular sump intake appears to be the best solution for this project.

Following the guideline from the American National Standard [38], dividing walls between the pumps have to be placed for pumps with flows greater than 315 Ls^{-1} (1134 m^3h^{-1}), a value largely exceeded here. If this practice proves to be excessively complex, other options could be to install separate modules for each pump or increase the spacing between them.

Most guidelines found for the design of sump intakes deal with pump stations with a perpendicular water inflow (Figure 1.6) in contrast with the vertical inflow from the water pipes. While this might have an impact on the formation of vortices inside the sump, the dimensions indicated therein can be used as basis to prevent flow interaction close to the inlets of the pumps.



(a) Top View (b) Side View

Figure 3.7: Example of Rectangular Sump with Perpendicular Incoming Flow [38]

For symmetry of the closed sump, the clearance from the back and front walls to the centerline of the pump inlet will be the same. More conservative distances have been taken, as can be observed in Figure 3.8. Consequently, there are several differences between the open structure in Figure 3.7 and the final closed caisson.

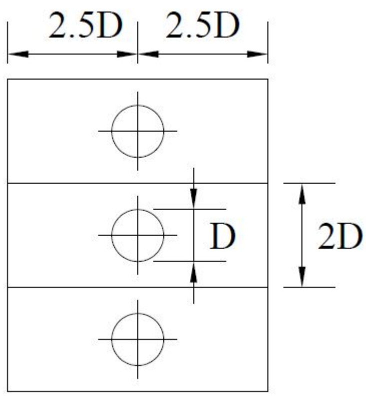


Figure 3.8: Rectangular Sump Design for the Ship-Shaped OTEC

This design will be used for the cold and warm water pump stations. The dimensions are listed in Table 3.5. By contrast, the discharge of water is assumed to be done by gravity, which implies that a discharge sump may not be needed. Nevertheless, to allow for a proper mix of the water flows at different temperatures, a similar structure, without dividing walls will be installed. The dimensions will be taken from the biggest of the cold and warm water flows, and could be increased later on.

Table 3.5: Dimensions of Cold and Warm Water Sumps

	CW	**WW**
Diameter, D (m)	1.87	2.79
Bay Width, w (m)	3.74	5.58
Total Width, W (m)	11.22	16.74
Length, L (m)	9.35	13.95
Minimum Submergence, S_{min} (m)	2.91	3.99
Pump Height, H (m) (anchor bolts to inlet)	4.80	4.59
Floor Clearance, C (m)	0.62	0.93
Sump Height, H_{sump} (m)	5.42	5.52
Sump Volume, V_{sump} (m)	579.1	1289.1

∗ Given by Flowserve (Appendix C.1)

Some aspects from Figure 3.8 and Table 3.5 must be pointed out:

- The volume of the WW sump is more than twice that of the CW sump, in line with the ratio between WW and CW flow rates. This volume can be enlarged in length or width if it is not sufficient.
- The height of the equipment deck coincides with the height of the sumps, which, in turn, depends on the requirements of the water pumps, as stated in Subsection 3.3.2. Therefore, to avoid placing equipment on two levels, the most restrictive value, i.e. 5.52 m will be chosen. This height is initially measured from the tank top to maintain the structural integrity while minimizing conversion effort. Even so, in reality this would depend on the connection of the water pipes.
- To maintain a symmetrical flow, the water pipes will be located in the center for all three sumps.

With this, the preliminary engineering of the "Seawater Supply" is concluded. The next section examines another sub-function comprising “Power Generation”, “Electrical Generation”, which refers specifically to the production of usable energy in the plant.

3.4 Electrical Generation

The sub-function “Electrical Generation” is to be performed by the process equipment (Table 2.4), i.e. heat exchangers (condensers and evaporators), turbine and generator. Other cycle components, e.g. working fluid pumps and separators will be included in the design, but are not looked at in depth, because their impact on the design is smaller compared to the larger process modules.

3.4.1 Heat Exchangers

The total heat input, $\dot{Q}_H$, and output, $\dot{Q}_C$, in Figure 3.2 take place in the evaporators and condensers, respectively. They are, therefore, a key element of the power plant and have to be carefully examined.

In [15] a number of heat exchanger types were analyzed. Plate-frame heat exchangers (Figure 3.9) are selected for the present design, despite the advantages of plate-fin and plate-channel over the chosen type. This is mainly due to the importance that the durability and the ease of cleaning and maintenance have for the operational life of an offshore OTEC plant, engineered to remain in place for an extended period of time. Apart from these gains, the main benefits of installing plate-frame heat exchangers are the high thermal efficiency and the up-to-date operational experience.

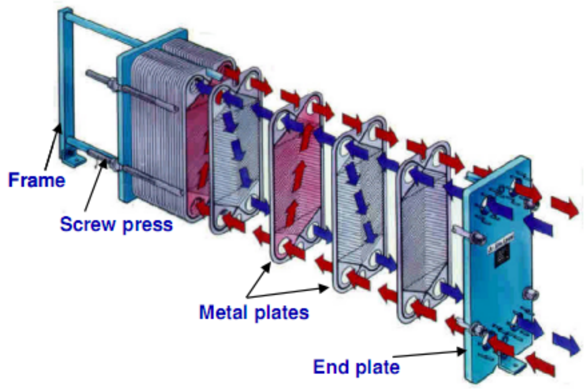


Figure 3.9: Plate-Frame Heat Exchanger

On the other hand, the high pressure drop and limited flow capacity of this type of heat exchangers makes them very difficult to scale. Accordingly, a great amount of components operating in parallel, will be required, thus enhancing the intricacy of the piping, the area for equipment placement and the complexity of the arrangement overall. To minimize the number of heat exchangers in the plant, a very large plate heat exchanger, the Alfa Laval T50 (Appendix C.2) will be used.

Equation 3.10 renders the maximum heat exchanged in a single unit.

$$\dot{Q}_{hx} = U A_e \Delta T_{LMTD} \tag{3.10}$$

Where:

Q_{hx} = heat transfer per heat exchanger (W)
U = heat transfer coefficient (W/m^{2}K)
A_{e} = heat transfer surface (m^2)
ΔT_{LMTD} = Logarithmic Mean Temperature Difference (LMTD) (K), from Appendix B.2

The heat transfer coefficient U, is assumed to be 2500 W/m^{2}K for both condensers and evaporators. Furthermore, the temperature data is again taken from the cycle simulations carried out in [50]. The calculation of the ΔT_{LMTD} is included in Appendix B.2.

The total number of evaporators and condensers can then be obtained by dividing the total heat exchanged from Section 3.1 by these values.

Table 3.6: Number of Condensers and Evaporators

	Condensers	**Evaporators**
U (W/m^{2}K)	2500	
A_e (m^2)	2880	
ΔT_{LMTD} (K)	3.27	3.69
$\dot{Q}$ (MW)	23.6	26.6
Nr. of Units	**15**	**14**

Notwithstanding this result, the maximum flow rate allowed through the Alfa Laval T50 is 5000 m^{3}h^{-1}. In view of this, the number of evaporators needs to be increased to 20 units for the system to function adequately. That is, a total of 35 plate heat exchangers have to be accommodated inside the hull. However, to maintain symmetry between modules, an extra condenser has been added to the design, making a total of 36 heat exchangers. This proves that the heat exchangers and the piping connected to them take up most of the space available on board.

If we now take into account the recommended space for maintenance, illustrated in Figure 3.10, the great number of heat exchangers makes clear their role as design drivers. This will be explained more in detail in Chapter 6.

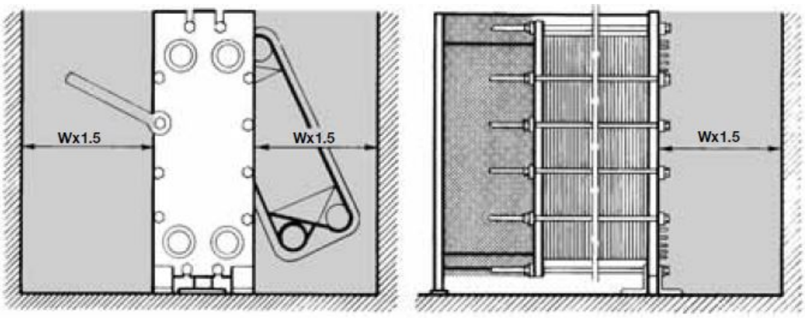


Figure 3.10: Recommended Space for Maintenance of the Plate Heat Exchangers [8]

3.4.2 Turbine and Generator Package

The working fluid leaves the evaporators and drives the turbine, which in turn actuates a generator. An integral turbine + generator configuration is preferred in order to simplify installation and reduce costs.

As it has been done in earlier OTEC studies [58], a turboexpander, i.e. radial inflow turbine, is selected for reference. The prime reason for their selection over axial inflow turbines is their better performance at smaller output capacity [60]. In accordance with Section 3.2, two modules of 7 MW each will be incorporated to the design. The specific model chosen for the present application is the GE Frame 40 turbo-expander (Appendix C.3).

The turboexpander-generator packages are not viewed as major design drivers, owing to their small size and consequent minor impact on the functions "Motions Control" and "Systems Support". These modules are, nonetheless, relevant for the "Power Delivery" and need to be located as close as possible to the power cable attachment and intermediate electrical equipment (switchboards, transformers, etc.).

Table 3.7 summarizes the dimensions of the equipment responsible for the generation of electricity.

Table 3.7: Electrical Generation Components

<table>
<tr><th></th><th>Condensers</th><th>Evaporators</th><th>Turbine+Gen</th></tr>
<tr><td>Nr. of Units</td><td>20</td><td>16</td><td>2</td></tr>
<tr><td>L (m)</td><td colspan="2">7.080</td><td>6.350</td></tr>
<tr><td>W (m)</td><td colspan="2">1.055</td><td>3.658</td></tr>
<tr><td>H (m)</td><td colspan="2">4.095</td><td>3.048</td></tr>
</table>

3.5 Cycle Continuity

Cycle continuity is ensured through the pipes that transport water and ammonia, liquid or vaporized, inside the Rankine the cycle.

The calculation of the exact diameter of these pipes is not within the scope of the thesis. In most reports, due to the use of tube shell heat exchangers, the choice was made to install a single, or sometimes two pipes to carry the fluids. Plate heat exchangers, might, on the other hand, required the fitting of several parallel lines.

This and the lack of reliable values found in similar project, leads to the assumption of a margin to make space for the ductings. The general arrangement of a 10 MW OTEC plant was investigated in [6]. This report proposes a diameter of the working fluid pipes of 1.5 m between the evaporators and the condensers, and 0.65 m from the separator to the condensed flow. Another study of a 10 MW semi-submersible OTEC by LM [58] estimates the diameter of the cold water ductings to be 4.35 m. These numbers are calculated for 10 MW installations. This means that halving these values could be a valid approximation of the margin taken for each of the power modules.

Based on the above, a total distance of 2.5 m next to the heat exchangers is considered acceptable. A clearer view of the implementation of this margin in the arrangement is illustrated later on.

It is evident that assuming the diameter of the piping decreases the accuracy of the dimensioning of the platform, and thus, of the vessel to be converted. This notwithstanding, for the purpose of the present work, a basic approximation of the size is deemed sufficient.

3.6 Weight Analysis

When dealing with the process equipment, the focus has been put on the space requirements of the plant. However, the effect of the distribution of weights in a floating structure cannot be disregarded. The hull is expected to provide sufficient buoyancy to accommodate the equipment and to be stable to ensure the safe operation of the plant (Table 2.4).

In Table 3.8 a summary of the estimated weights of the main equipment is given. Even though only the weights of the prime components, examined previously, are shown here, it must be borne in mind that in the Detailed Design stage the weights of associated piping and foundations and additional equipment (separators, transformers, etc.) will have to be considered.

Table 3.8: Weight Breakdown of Main Plant Equipment

	CW Pump	**WW Pump**	**Heat Exchanger**	**Turbine-Generator**
Nr. of units	3	3	36	2
Unit weight (t)	2.8	9.3	22.7	20.9
Component group weight (t)	9.3	37.4	816.5	41.8
Total weight (t)			900.1	

It must be here noted that, since the weight of the water pumps was not provided by the manufacturer, the weight of similar pumps, made available by Fairbanks Nijhuis (Appendix C.1), has been taken with a margin of 20 %. In addition, the weight of the non-buoyant volumes of the water sumps, i.e. 3236.1t has to be included.

Several conclusions can be drawn from the weight analysis:

- The largest contributors in weight and space are the heat exchangers, which amount to more than 80 % of the equipment weight. This reinforces the relevance of this component group as a design driver.
- The weight resulting from the analysis, 4136.2 t (900.1 + 3236.1), will be the required minimum deadweight of the vessel.

- The light weight of the equipment anticipates stability issues of the platform, and, consequently, the need to install fixed ballast.

- The contribution of the CWP to the overall weight, 656 t, is substantial, and thus, has to be included in the stability study.

All these will have an impact on the selection of the ship for conversion and the layout of systems on board, as it will be seen further on.

3.7 Power Plant Configuration

In the preceding sections, the calculations and technical specifications of the equipment comprising the process cycle have been outlined. The result of bringing all together is the simplified Piping and Instrumentation Diagram (P & ID) of a 5 MW module in Figure 3.11. Two identical modules of 5 MW each will then be accommodated on the platform.

It should be noted that, despite the differentiated outlets of warm and cold water, the discharge pipe is shared by both water flows.

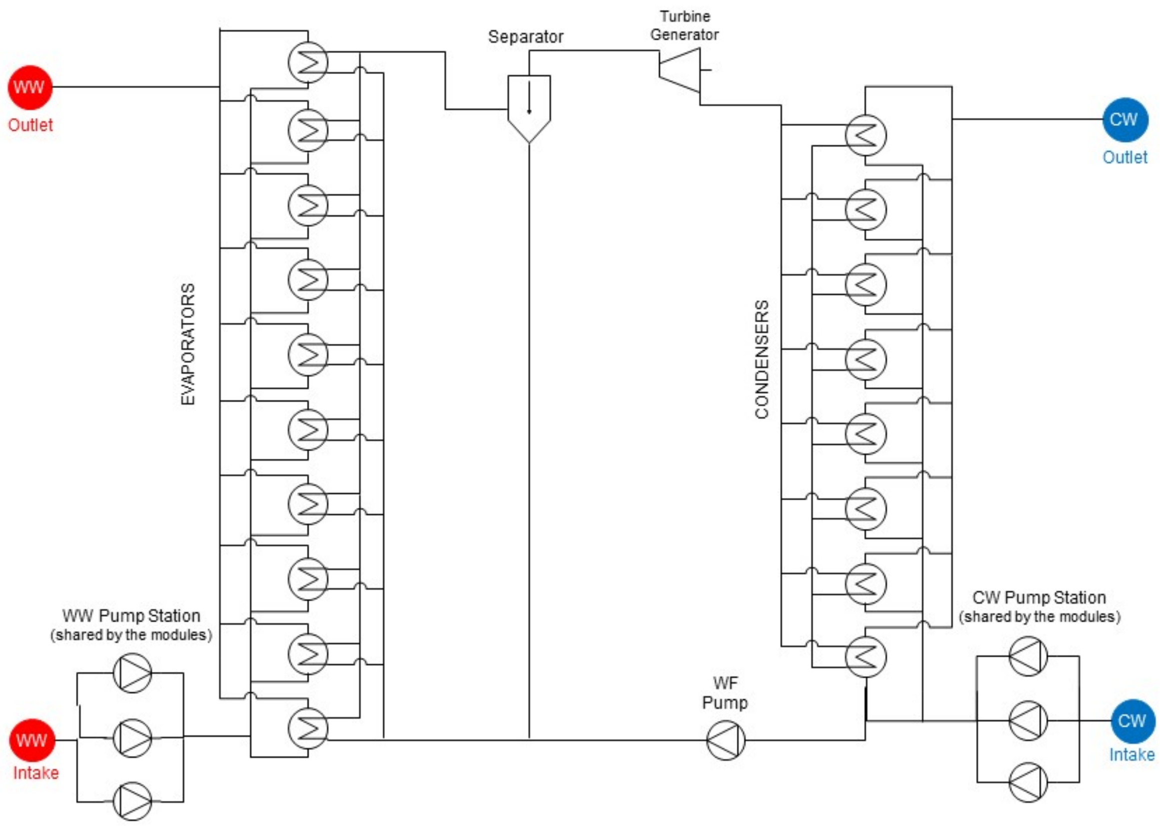


Figure 3.11: P & ID of a 5 MW Module

3.8 Hull Characteristics

The power plant systems and the piping connections shown in Figure 3.11 determine to a great extent the necessary functional features of the hull:

- Good motion performance to guarantee that the process equipment functions within acceptable operability limits, and to prevent high loads and inertia forces originating from the coupling of water pipes and platform. Restriction of platform motions is also largely related to the mooring system, which will be chosen later on.
- Simple hull configuration with big open spaces allowing for an easy power plant layout and system installation on board.
- Structure capable of withstanding point loads from the installation of heavy components.

The slender structure of a ship's hull complicates the arrangement of the systems on board. Henceforth, the layout of components and the selection of the vessel will depend on each other. The result is an iterative design loop that will have as output the selection of ship type, the dimensioning of the platform and, ultimately, the general arrangement of the plant.

4

Mooring System

The ship-shaped OTEC plant has to remain on site during its operational life to generate 10 MW of electricity per year. The design life of the platform is scheduled to be at least 20 years, as stipulated in the TLR in Subsection 2.3. The selection of the optimal mooring system is, thus, of great importance for the functioning of the installation.

4.1 Application to the Ship-Shaped OTEC Plant

Being an offshore structure, the platform will experience the environmental conditions of the waters surrounding Curaçao throughout its lifecycle. These environmental conditions, waves, wind and current were summarized in Subsection 2.2. Waves, wind and current induce platform motions, affecting the operability of the process and hampering the landing of helicopters or the access by lifeboat. In other words, the mooring system and its coupling with the hull are primarily responsible for the seakeeping, stationkeeping and accessibility of the plant. Even more so, bringing back again the TLRs in Subsection 2.3, the mooring system can be directly linked to the last four requirements, and indirectly to the first one.

While the hydromechanic calculations and modelling of the mooring system are not within the scope of this thesis, these aspects will be taken into account over the selection process.

Since the first installation of an FPSO in water depths of 117 m in the Mediterranean Sea, the offshore industry has made remarkable advances in the engineering of mooring systems. An example of this is the FPSO Turritella, anchored in water depths of 2896 m, which pushed even deeper the limits of mooring technology [5].

Conceptually, FPSOs as well as the studied concept are based on the idea of a stationary ship-shaped offshore structure, moored to the seabed for an extended period of time. With this in mind, the off-the-shelf mooring technology developed by the offshore industry can be applied to the floating OTEC plant.

There are, however, several fundamental differences that need to be considered in order to choose the most suitable mooring system for the present design:

- Unlike conventional FPSOs, offloading of liquid products is not a major decision driver for the ship-shaped OTEC. Because oil transfer will not take place for the latter, weathervaning to avoid collisions with shuttle tankers is not a key requirement anymore.

- The forces imposed by the coupling of the CWP to the hull can induce large accelerations on the floating structure. Despite the large numbers of risers that an FPSO can accommodate, in some cases over a hundred [4], the inertia forces caused by these are negligible when compared to the 1000 m long and 4 m diameter CWP.

- OTEC equipment, i.e. electrical and process cycle components, features different operability limits than those of specific FPSO systems.

These differences call for a clear definition of the main priority requirements for the mooring system of a ship-shaped OTEC plant:

- Low design complexity, particularly relevant owing to the lack of previous experience in the conversion of a bulk carrier into a stationary offshore structure. Not only this, but also the design of the ship-shaped plant as a prototype demands the design to be as simplified as possible.
- Wide flexibility to arrange plant equipment on board in an optimal manner, avoiding, when possible, complicated interfaces with other systems.
- Specific reduction of CWP motions. The influence of a pipe of such dimensions on the motions of the platform is yet unknown and will depend on the size of the selected hull, making it advisable to decrease them to minimal values.
- Cost effectiveness, since the ultimate goal of the project is not the quest for economic profit, but to demonstrate the feasibility of the design.

In conclusion, both applications pose distinctive engineering challenges and these dissimilarities have to be analyzed prior to taking the final decision.

This been said, the next section spells out the characteristics and advantages and disadvantages of existing mooring systems in relation with the design of a ship-shaped OTEC plant.

4.2 Single Point Mooring (SPM)

Single Point Mooring (SPM) systems are designed to allow the vessel to freely weathervane around a fixed point to adjust to the prevailing weather conditions at the location. As a result, motions and accelerations, and the associated loads on the structure are minimized. Additionally, the offsets can be adjusted to desired values to decrease the excursion rate of the platform.

The capability to adapt to the momentary wind, waves or current, thus reducing platform motions, involves other additional advantages:

- Low roll angles facilitate the landing of helicopters or the approximation to the platform by boat, subsequently enhancing platform accessibility. Furthermore, the operability of process equipment is improved, deriving in lower downtime.
- The structure can withstand extreme environmental conditions.
- The ability of the vessel to head incoming waves lowers the probability of experiencing green water on deck. The equipment is, therefore, more protected against potential damage from waves.
- Small wave encountering angles resulting from the weathervaning capacity of the platform also imply smaller sizes of mooring components, since they are projected to stand lower loads.

In addition to these benefits, SPM systems can be equipped with proven disconnectable capabilities in the case of a tropical storm or a hurricane.

SPM systems are, on the other hand, more complex to construct and install than passive mooring systems, which generally brings along a higher associated cost. The integration of OTEC equipment with mooring components becomes more intricate, involving an extensive engineering process. Moreover, these systems require a labor-intensive conversion that can delay the delivery time of the plant. Above all, if the vessel encounters non-collinear environmental conditions, the platform can experience large accelerations, deriving in greater loads in the mooring lines. Likewise, the beam sea condition can lead to wave slamming issues and green water on deck.

The majority of SPM systems fall into two categories: those presenting a mooring system integrated with the hull and those with the attachment point located at the bow. There is a wide variety of systems within these two types, but some of them can be disregarded a priori for the current application ([85]):

- Catenary Anchor Leg Mooring (CALM) (Figure 4.1) and Single Anchor Leg Mooring (SALM) (Figure 4.2), deployed only in shallow waters near shore devised mainly for loading and offloading of oil products.

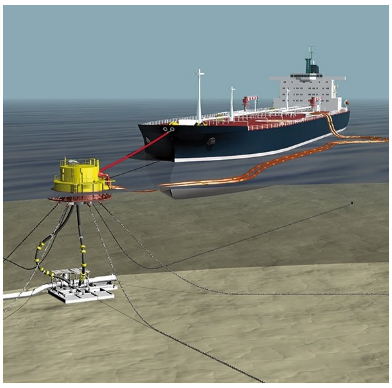


Figure 4.1: Catenary Anchor Leg Mooring (CALM) System

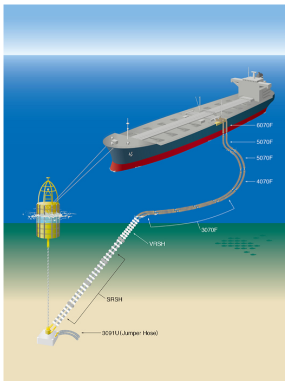


Figure 4.2: Single Anchor Leg Mooring (SALM) System

- Yoke systems (Figure 4.3), comprising Jacket Soft Yoke (JSY), and Tower Yoke (TY) and Single Anchor Loading (SAL) Yoke systems, all of them variations of the same concept to be installed for projects of limited depth.

Figure 4.3: Yoke Mooring System

- Riser Turret Mooring (RTM) (Figure 4.4), is disregarded due to the foreseeable complexity to integrate the turret with the CWP and the equipment to pump the large amount of water required.

Figure 4.4: Riser Turret Mooring (RTM) System

Hence, two main options remain, namely, external and internal turret mooring systems.

4.2.1 Internal Turret (IT) Mooring

The internal turret mooring system, shown in Figure 4.5, is integrated in the hull structure and permits the vessel to turn around the turret depending on the direction of wind, waves and current. The turret shaft and bearing transfer the vertical and horizontal loads from the mooring lines to the hull.

The turret system can incorporate other systems leaving more space on deck for other equipment. This, however, increases the level of complexity of the structure, while diminishing the flexibility to arrange systems inside the hull. It should be here stressed that, to keep the motions of the CWP within allowable limits the most suitable solution would be to combine the CWP with the turret in a one-piece design. Even though this is normally done in FPSO conversions the large diameter of the CWP poses a great engineering and constructional challenge. An internally mounted turret can be located at the bow or just forward of amidships of the vessel [62].

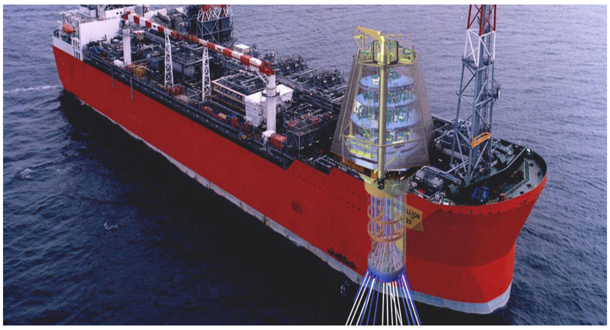


Figure 4.5: Internal Turret (IT) Mooring System

- Bow

  Because the pitch and heave motions at this location are larger than amidships, the system integrating the CWP and the turret will require a more robust arrangement. Nevertheless, the distance from the highly stressed central part of the vessel makes it a preferable design from a structural point of view. More importantly, a bow turret provides natural weathervaning without the need of thrusters.

- Forward of amidships

  The motions and accelerations experienced at this location are lower than at the bow, which would be beneficial for the CWP. However, the structure has to be reinforced to withstand the additional loads combined with the already elevated stresses at the midship section. Furthermore, in contrast with a bow turret, this solution demands for thruster power to control weathervaning, which, in turn, entails extra electricity consumption and a higher cost.

4.2.2 External Turret (ET) Mooring

The external turret (ET) is located at the end of an outrigger structure incorporated to the bow of the hull. It imposes fewer restrictions to the arrangement of plant systems and occupies less space on deck. The downside of the ET is that it cannot be integrated with the CWP, since this will drive the design beyond technical feasibility. As a consequence, there is no potential for the reduction of the motions of the CWP.

Figure 4.6: External Turret (ET) Mooring System

4.3 Spread Mooring (SM) Systems

Passive mooring systems can only be installed in mild to moderate environments that show weather directionality. The consequence of this is that weathervaning capability is not required.

The mooring lines are clustered at the bow and stern of the vessel, as in Figure 4.7 and allow a limited mobility of the platform depending on the number of lines and their tensioning, i.e. offsets are variable. In SM systems the winches, sheaves and other components that make up the complete mooring system are distributed about the deck, which adds congestion to the deck arrangement.

Figure 4.7: Spread Mooring (SM) System

The principal advantages of spread mooring systems are outlined in below:

- CWP, WWP and DWP movements are more restricted, because the platform cannot turn around a fixed point. Nonetheless, the excursion rate of the vessel must be limited and, therefore, special attention must be paid to the design of the offsets.
- Overall, the spread mooring is mechanically simpler than the turret structure and involves less conversion effort. Even more, this simplicity derives in more possible arrangements of the systems on the platform.
- Passive systems are cheaper than turret moorings, which is a substantial issue in the present project, as formerly stated.

For all that, spread mooring systems lack the prime features that make turret systems an attractive alternative. The impossibility to weathervane causes the structure to experience large roll motions when waves are encountered at large angles with respect to the bow, especially in beam seas. As a consequence, the risk of wave slamming or green water washing onto the deck increases, deteriorating the accessibility and operability of the plant. That is to say, passive moorings can withstand lower extreme environmental conditions installations equipped with a turret. Lastly, the number of mooring lines is usually larger for SM systems and their attachment points to the hull require structural modifications.

4.4 Selection of the Mooring System

Although all systems studied present both strengths and weaknesses, as displayed in Table 4.1, spread mooring systems fulfill more effectively the specific requirements of the ship-shaped OTEC plant formulated in Section 4.1. This system entails lower complexity and hull modifications than turret structures, resulting in a cheaper installation and more flexible design solutions.

Table 4.1: Comparison of Mooring Systems

		Advantages	Disadvantages
Spread Mooring (SM)		– Technical simplicity – More flexibility in GA design – Restricted CWP motions	– Only moderate environments – Larger platform motions
Single Point Mooring (SPM)	Internal Turret (IT)	– Compact design – Minimized CWP motions	– Complicated structure and interfaces – Substantial conversion effort – High cost
	External Turret (ET)	– More flexibility in GA design – More space on deck	– Unrestricted CWP motions – Necessary outrigger attachment

The mild environmental conditions described in Subsection 2.2.3 show a high level of directionality regarding wind, waves and current, which enter the South Caribbean Sea from the East and travel westward thereupon. Accordingly, a spread mooring system can be installed in the waters surrounding Curaçao island.

In the future, when the feasibility of the deployment and operation of the platform is demonstrated, the integration of the CWP with a turret system should be revised again. Moreover, the Disconnectable Spread Mooring (DSM) (Figure 4.8), also called Submerged Mooring Pontoon (SMP) ([46],[33]), patented in December 2015, could be considered if detachment is needed in stormy weather. This will be primarily relevant for other locations with a greater risk of typhoons.

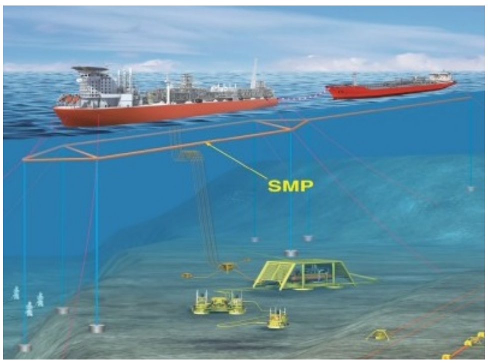


Figure 4.8: Submerged Mooring Pontoon (SMP)

In comparison to the power plant equipment the system selected, i.e. a spread mooring, will not have a significant impact on the general arrangement of the platform. It will, however, require space on deck to install the sheaves and winches that connect the hull to the mooring lines. This, together with the necessary hull characteristics outlined in Section 3.8, will be taken into account for the selection of ship type.

5

Selection of Ship Type

The general arrangement of spaces and components on board the ship will be largely influenced by the structure of the ship to be converted. An exhaustive exploration of the aspects affecting the vessel's selection is, thus, of great importance to develop an optimal integration of systems.

5.1 Vessel Types

The technical complexity of ships as self-sustainable systems has been highlighted beforehand in this report. This inherent complexity varies nonetheless among different ship types. Primarily, the more specific the requirements to be fulfilled by the ship, the higher the level of customization, and therefore, the intricacy of the system. As an example, Offshore Supply Vessels (OSVs) are intended to perform very concrete missions, and entail, accordingly, highly tailored designs. The result is a number of specialized subcategories within this vessel type, e.g. Anchor Handling Tug Vessel (AHTV) or Seismic Vessel.

Customization has, in turn, two further consequences: an elevated associated cost and the limitation on potential standardization processes. These two aspects, in combination with the lack of large, open spaces on board, lead to discarding, at the outset, complex specialized vessels, including dredgers, passenger ships or offshore vessels. The study is, then, narrowed down to the more suitable ship types below:

- General cargo, in particular, Multi-Purpose Vessels (MPVs) and Pure Car and Truck Carriers (PCTC)
- Bulk carrier
- Containership
- Tanker

All of these remaining vessel types are standardized transport ships with open spaces. The comparison of these is not straightforward and requires the execution of a more in-depth analysis.

Prior to the description of the selection criteria, it shall be clarified what is here meant by ship conversion, namely: "major conversion means a conversion of an existing ship: [...] (ii) which changes the type of the ship", as stated in the Annex I of MARPOL [42], the International Convention for the Prevention of Pollution from Ships. The attention is, therefore, focused particularly on this definition, implying an extensive overhaul of the vessel.

Bearing this in mind, the selection criteria can be organized in two distinct groups: purely technical criteria, and commercial or strategic considerations. The first category includes the following: minimum conversion, structural characteristics and volume/weight ratio; while the second covers market availability and cost and previous constructional and operational experience.

Ultimately, the converted ship will perform the functions associated with the hull in Subsection 2.4.3. That is to say, some of the criteria assessed hereunder will not be analyzed in detail in the present work, but it is essential to take them into consideration during the ship selection process.

5.2 Commercial & Strategic Considerations

5.2.1 Previous Constructional and Operational Experience

As of July 1st 2016, there were a total of 169 FPSOs in operation worldwide, 70 % of them from converted second-hand tankers [17].This demonstrates the overall predominance of this kind of conversions over those among other ship types. Since the deployment of the first FPSO in 1977, the life cycle of these installations has been monitored and their design improved throughout the years. That is to say that a large technological know-how is available, both for the construction and the operational phase.

These are, however, not the only cases of ship conversions. Depending on the market situation or newly enforced regulations, shipowners have ordered the adaptation of vessels to different missions.

In 2007, the demand from China spurred the conversion of tankers into bulk carriers [68]. Dozens of single-hull tankers were refurbished as bulkers instead of double-hull crude carriers to meet the safety regulations which entered into force in 2010. Conversely, there has also been the case of a bulk carrier, "Relchem Isha", turned to a chemical tanker by GB Marine [31]. What is more, there are several shipyards and engineering companies that undertake highly complex projects to convert, for instance, a bulk carrier to a pipelaying vessel, or a stern trawler to a seismic vessel. Despite this, these examples remain marginal and are only carried out under very specific circumstances.

5.2.2 Market Value and Availability

The value a vessel can fetch in the market fluctuates greatly over periods of time, following the pace of the global business cycle. In order to maintain coherence throughout the report, the context of the shipping market is taken as of 2017.

According to the International Monetary Fund (IMF), "world growth is expected to rise from 3.1 percent in 2016 to 3.5 percent in 2017 and 3.6 percent in 2018" [39]. The forecasted cyclical recovery of manufacturing, investment and trade is effectively taking place.

The shipping market, largely affected by the shifts in the global economy, is expect to follow the same trend. The growth of the global GDP gives hope to the sector, which has seen its potential hampered since the crash of the financial market in 2008. So, how will the new financial situation influence the shipping market in 2017? "Global GDP growth is currently driven by service sectors and developing/emerging economies which result in a lower "GDP-to-trade multiplier", and thus generate a lower level of shipping demand than we have been accustomed to in the past", says The Baltic and International Maritime Council (BIMCO) [14].

In 2016, all fleet segments experienced a reduction in newbuilding orders, with the exception of ferries and cruise ships. The decline in the investment in the sector has affected the largest shipping markets: containerships, oil tankers and bulk carriers, of which newbuilding orders have decreased by 90.2 %, 81.0 % and 71.0 %, respectively [9].

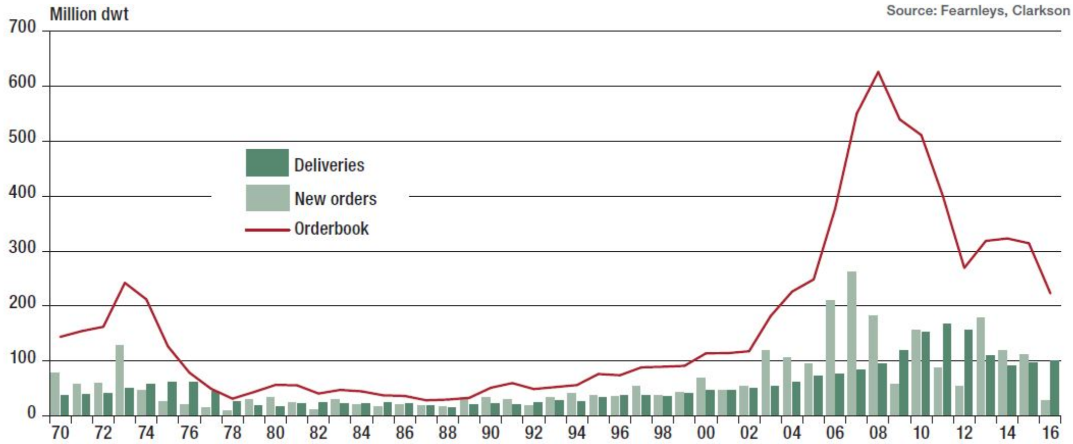


Figure 5.1: World Shipbuilding until 2016 [9]

ISL Bremen indicated that "demolition activity remained high in 2016 with 43 million dwt, a y-o-y increase of 14.8 per cent. The third largest dwt volume ever to be scrapped (after 2009 and 2010)" [45]. The weak market situation triggered the demolition of 841 merchant vessels over the past year. The segment most damaged was the bulk carrier market, with 67 % of the broken up tonnage. Containerships followed closely, reaching a record level of 195 vessels.

Overall, this may suggest that the availability of second-hand ships to be converted will decrease to a large extent in the years to come. However, BIMCO analysts emphasize that "the full restoration of shipping markets will need several years of solid improvements to lift fleet utilization rates. Sector overcapacity almost everywhere must be reduced" [14], leaving room for future second-hand acquisitions.

As of January 2017, the distribution of the world merchant fleet by ship types was as illustrated in Figure 5.2. In terms of GT, bulk carriers account nowadays for a 36.1 % of the merchant fleet, oil tankers for a 21.6 % and containerships for a 18.3 %, without significant changes with respect to the previous year. General cargo ships, on the other hand, lost 333 vessels, thereby reducing their GT by 0.5 %.

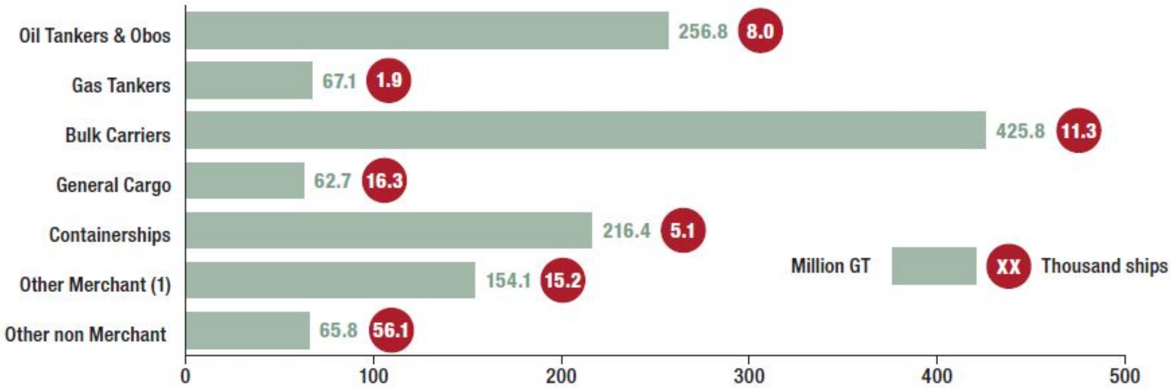


Figure 5.2: World Merchant Fleet by Ship Types, as of January 2017 [57]

It can then be expected that the vessel types which combined account for the biggest market share, i.e bulk carriers, oil tankers and containerships, will also be more available in the second-hand market. Accordingly, these will be the focus of the market study in the following lines.

– Bulkers

Over the past ten years, demand has outstripped supply only in 2007 and 2014. Immediately after, the market situation experienced a sudden reversal, with supply again surpassing demand [75]. 2016 was the worst year on record for the dry bulk sector, reaching an all-time-low of 290 on February the same year 5.3. Since then, the market has experienced a steady recovery, with 2017 showing an improved market situation, despite the high level of volatility. This, however, is not sufficient to regain profitability for dry bulk shipping: heavy demolition activity remains fundamental to restore the necessary balance between supply and demand.

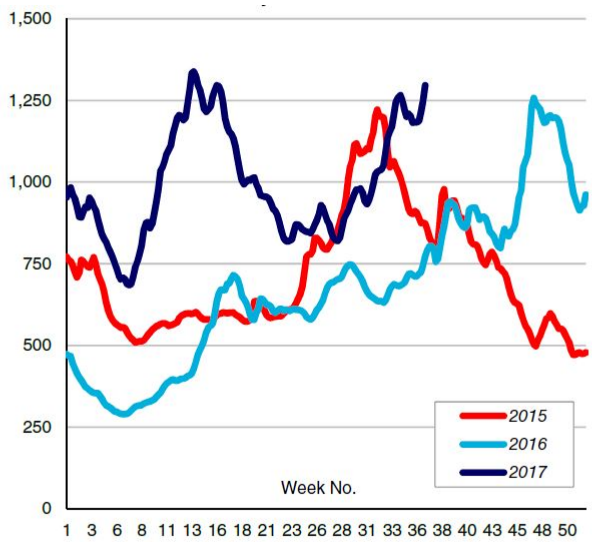


Figure 5.3: Baltic Dry Index Trends [56]

As a result of the market situation, the prices of secondhand bulk carriers have been falling since 2014 for all vessels sizes (Figure 5.4). Capesizes are the only class above 20 M$, while all smaller sizes are available for similar prices, close to 15 M$.

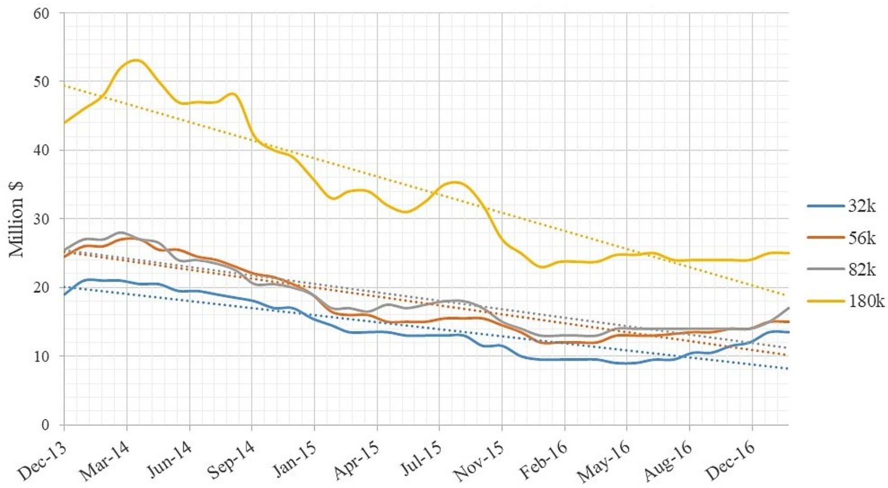


Figure 5.4: Second-hand Bulker Prices (5-year-old) (Own elaboration based on Clarkson Research [56])

– Containers

Container shipping has struggled to adapt to the new global market conditions with lower demand levels than in the past. Nevertheless, thanks to the extensive demolition of vessels in 2016, this year has been the first since 2010 with a fleet growth smaller than demand growth. This elevated number of demolished vessels, with a historical record of 656000 TEU, was backed up by very low newbuilding orders [9]. As a consequence, the idle fleet has been reactivated since the end of 2016 (Figure 5.5. There are, however, still many vessels looking for employement [45].

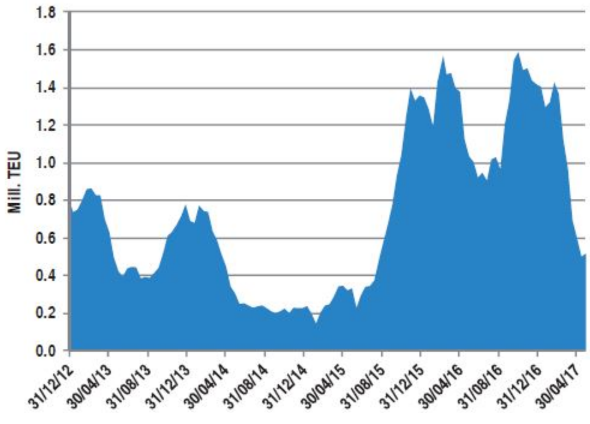


Figure 5.5: Idle container fleet 2013 – 2017 (biweekly) [45])

As a reflection of the overcapacity of the market, second-hand prices were extremely poor in 2016, approaching scrap values. Among the different container sizes, Panamax vessels, outdated after the expansion of the Panama Canal, present a great opportunity. Panamax containers accounted for 47 % of the total demolition in 2016, while Intermediate and feeder containerships represented the 30 % and 23 %, respectively [1]. Figure 5.6 illustrates the downward path followed by second-hand prices in recent times, with all smaller classes falling below 10 M$.

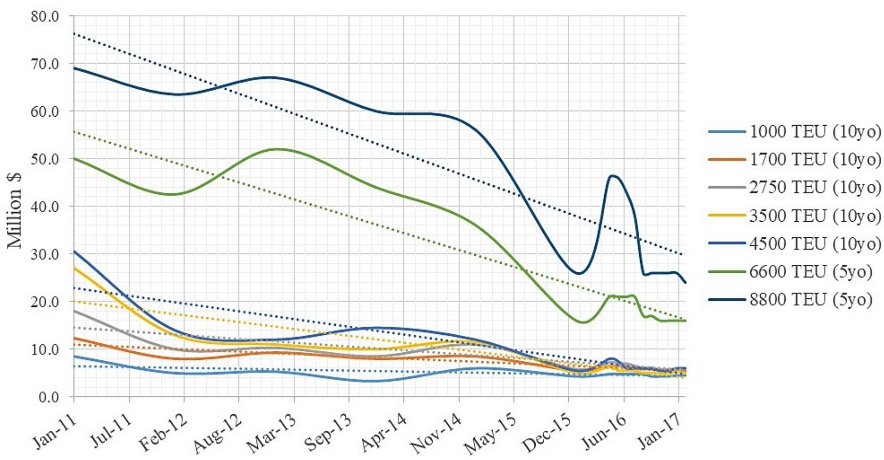


Figure 5.6: Second-hand Liner Vessel Prices (5-year-old) (Own elaboration based on Clarkson Research [56])

– Tankers

In December 2016, OPEC and non-OPEC producers reached their first agreement since 2001 to slow down the production output after two years of low oil prices [82]. As a result, oil tanker shipping, the only money-making market during the past year, is now at risk. On the other hand, both crude oil and oil product's fleets increased in 2016 by 6 %, while the demand eased off, causing freight rates to drop drastically over the past months [35]. Figure 5.7 shows the deliveries and demolitions of tankers since 2014. The graph highlights the remarkably low level of scrapping in comparison to the newbuilds.

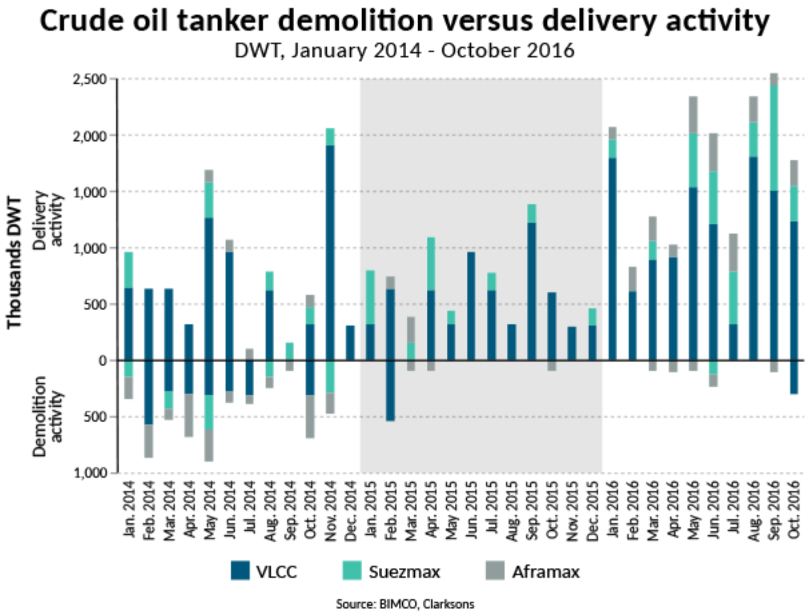


Figure 5.7: Crude Oil Tanker Demolition vs. Delivery Activity [74] [56]

In 2017, the demolition of oil tankers remained again at minimum levels, with only 69 units, amounting to 3 Mdwt [9]. Subsequently, all tanker sizes have seen their values decreased in the past months (Figure 5.8). This notwithstanding, the oil tanker market has been the most stable in comparison to bulk carriers and containerships in recent years.

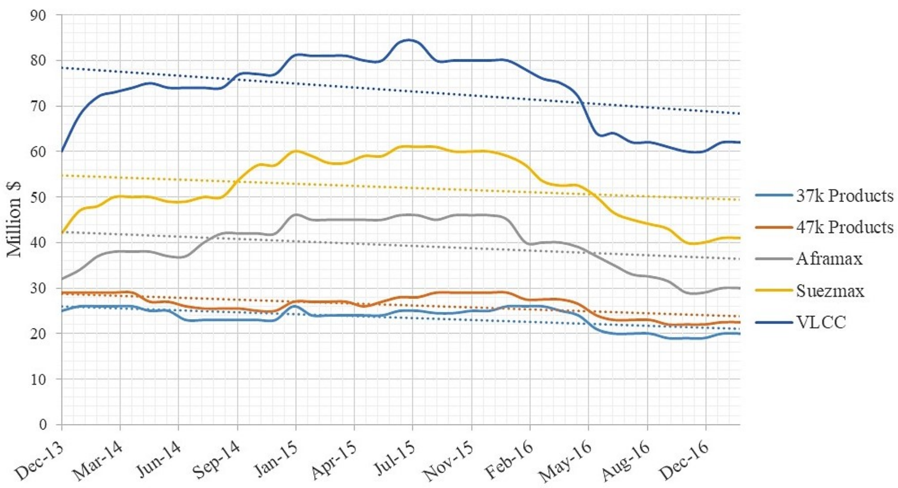


Figure 5.8: Second-hand Tanker Prices (5-year-old) (Own elaboration based on Clarkson Research [56])

– General Cargo Ships

In the first months of 2017, the dwt-share of general cargo ships, including MPVs, PCTCs and conventional cargo ships) stood only at 6.4 %, and yet, roughly one out of three ships is a general cargo vessel. This is an indicator of the small size of these vessel types or, in the case of car carriers, of their volume-based design. Between 2013 and 2017, the growth of conventional cargo vessels stood at 1.9%, while specialized cargo ships and PCTCs grew by 7.7 % and 1.0 % [45].

Breakbulk transportation continues to be necessary for the development of infrastructure, offshore installations, etc. However, general cargo operators have been forced to compete with container lines and Handysize bulk carriers for breakbulk cargoes, due to the low rates in their core markets.

The slight increase in container and bulk rates indicates, nevertheless, an improvement in the market situation of general cargo vessels.

At the same time, the car carrier segment, strongly linked to global economic growth has been greatly affected by the lower demand in recent years. Fleet overcapacity, together with the drop in transport demand has derived in very depressed rates, and ultimately in an escalation of demolitions, which have multiplied by 3.7 in terms of dwt [9].

Overall, it can be expected that the conventional cargo segment will sustain its pace, despite of the difficulties, while the car carriers present a good opportunity for second hand acquisitions, as the demand and supply still need to balance to improve the market situation.

Looking at the complete picture, the sector has suffered from the global economic slowdown since the great recession of 2009. Generally, the shipping industry follows the pace of economy's growth, which is expected to experience very little improvement in the coming years. This affects all ship types considered, hence, leaving a wide variety of available ships to be purchased for conversion at a low price. In particular, bulk carriers and containerships are a good option owing to their bargain prices for different sizes. In Appendix D values for 10-year old bulkers, containerships, as well as tankers are attached. Additionally, Appendix D provides insight into the historical evolution of second-hand prices for a longer period of time.

5.3 Technical Aspects

5.3.1 Hull Shape and Dimensions

Despite being categorized as transport ships, these different vessel types feature rather distinctive hull shapes according to the cargo to be carried. The value and homogeneity of the cargo define the required service speed and the stowage mode of the vessel, which are, in turn, the main drivers of the hull shape and dimensions.

With regards to the current application, the fineness of the lines of the hull affects, among other aspects, the stability of the platform and its seakeeping properties, necessary for the safety of operations. Additionally, the geometric relations between ship dimensions, i.e. length, depth and breadth, have a great influence on the seakeeping characteristics of the platform [22]. Lastly, moderate vessel motions will enable secure access to the installation, either by helicopter or lifeboat.

The conventional shapes used by the offshore industry, as in the case of a standard tanker vessel, have proven to have a good motion performance. The same can be extrapolated to a large extent to bulkers, similarly designed to convey large amounts of cargo in bulk. This been said, it is essential to analyze independently the parameters that govern the stability and seakeeping behavior of the ship types under study:

– Block coefficient (C_b) and Bow Shape

  In general, the higher the block coefficient, the less the motions. The value of the block coefficient is directly related to the bow and stern shape, which are responsible for the bow wave impact and bow slamming, as well as the green water on deck.

  A full rounded bow is expected to experience more green water and bow wave impact than a sharper form. However, it provides maximum buoyancy for minimum steelweight and enough space to allow the installation of a turret. A sharper bow, on the other hand, minimizes bow impact and mooring forces, but limits the available space and buoyancy at the fore end of the vessel.

  Green water may occur also at the stern, when the vessel is pitching down during a wave crest, although the risk, in this case, is not as high as at the bow. [62]

– Length-to-beam ratio (L/B)

Length is the most expensive dimension of a ship and should, therefore, be dimensioned as small as possible. Nevertheless, if a ship is long compared to the wave system, it will pitch and heave to a lesser degree than a smaller one. Another advantage of a long hull is that it attracts lower mooring loads. For this application, the limitation of platform motions takes a priority where it conflicts with the prize. Consequently, a larger length is considered to be beneficial for the ship-shaped plant.

Conversely, a small beam would contribute to avoiding large mooring loads and short rolling periods, which ultimately make the vessel stiff. Thus, beam should be decreased while remaining within allowable values for stability and deck space.

– Beam-to-draft ratio ($^B/_T$)

An increase in draft derives in larger freeboard against green water on deck, while the probability of slamming is diminished. A hull with a high B/D would have a low center of gravity, again deriving in short rolling periods, with the same formerly stated consequences.

Table 5.1: Hull Shape Characteristics of Different Ship Types

	Block coefficient (C_B)	Bow Shape	Length / beam ratio ($^L/_B$)	Beam / draft ratio ($^B/_T$)
Bulk carrier	0.75 - 0.85	round	5.0 - 6.0	2.3 - 2.8
Tanker	0.80 - 0.85	round	5.5 - 6.5	2.3 - 2.8
Containership	0.55 - 0.70	sharp	6.0 - 7.0	3.0 - 4.0
MPV	0.55 - 0.75	neutral	6.0 - 6.5	2.3 - 2.8
PCTC	0.50 - 0.60	sharp	6.0 - 7.0	3.0 - 4.0

The definition of an optimal form of the hull to be converted involves multiple trade-offs that might rule out some characteristics and the advantages related to them. Overall, leaving aside the necessary dimensioning of the structure, the selected hull will preferably feature a high C_B, a round bow shape, a high L/B and a low B/D. Table shows typical dimensions that have performed succesfully in FPSO projects, as provided by the United Kingdom Offshore Operators Association (UKOOA). These values will be used as a reference in this research.

Table 5.2: Hull Shape Characteristics of Different Ship Types

Parameter	Range of Recommended Values
Length	209 - 246 m
Length / beam	5.0 - 6.5
Beam / (max.) draft	2.2 - 2.7

5.3.2 Hull Configuration

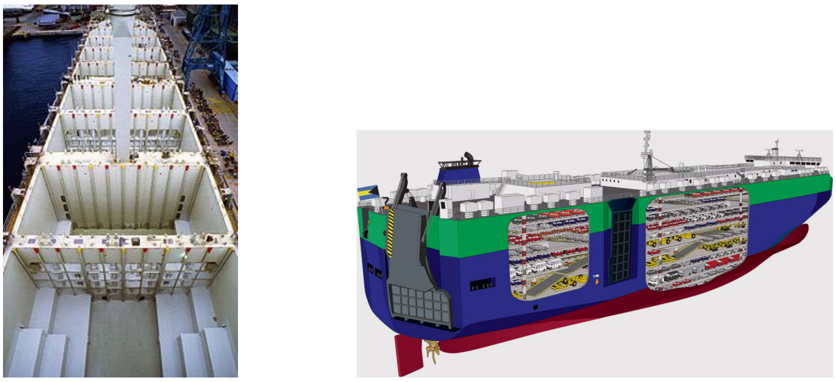
In addition to shaping the hull form, the cargo type and storage method define the arrangement and distribution of systems and spaces inside. According to their value, conveyed goods can be roughly organized in two categories: low density value goods, i.e. bulk cargo; and highly valuable products, like containerized goods, breakbulk or wheeled cargo. The storage method is subsequently designed to achieve the safest and most efficient transportation of cargo.

Bulk cargoes are further divided in liquid and dry bulk, transported by tankers or bulk carriers, respectively. These two vessel types carry the cargo in relatively simple box-like spaces, that is, tanks or holds. It should, nonetheless, be noted that the distribution of spaces greatly diverges from single to double hull tankers, affecting the stability, buoyancy and effort to convert the ship into an OTEC plant. For double hull tankers and bulkers, the bottom and wing tanks provide marginal ballast, which can be used for an improved control of the stability of the structure. For the present application, however, the complexity of a double hull tanker must be avoided.

Cargo holds for containerships, on the contrary, are specially constructed to optimize the number of containers that can be carried inside the hull. From the tank top to the hold opening, the structure is adapted to accommodate TEUs. The stability of containerships is more dependent on the stowage plan of the containers than on the structure of the vessel itself.

General cargo ships, intended to transport a broader range of products, share some of the features of the previous ship types: simple holds inside the hull and a deck fitted to transport breakbulk or containers.

The particular storage method of car carriers determines their unique hull form, optimized to speed up the loading process while accommodating the maximum number of vehicles. In lightship condition ro-ro ships are designed to have a very low center of gravity, as it increases in loaded condition. Though this is beneficial for stability, a low center of gravity reduces the rolling period of the ship, which ultimately makes it a stiff ship. Moreover, in damaged stability the absence of transverse bulkheads will cause the rapid flooding of the vessel, decreasing its buoyancy and initiating a free surface effect.



(a) Holds Inside a Containership

(b) Arrangement of a 6700 CEU PTCT

Figure 5.9: Differences in Spatial Configuration According to Transported Goods

5.3.3 Deck Space

Notwithstanding the yet unknown layout of systems, a need for available space on deck can be expected by the designer. Whether part of the process cycle is installed as topside modules or not, the installation of equipment on deck is a largely extended practice in the offshore industry. Some of its benefits are the accessibility of components for maintenance or replacement, the savings of space in the hull, or the natural ventilation of the cycle components. Even more, the importance of deck space would be enhanced if a spread mooring system is selected, since it requires enough space to support separate winches and sheaves for the forward and aft anchor legs [36].

Besides the value of a wide deck area to avoid spatial congestion, the probable addition of a helideck for platform accessibility will also take up a lot of space. In this regard, a position of the superstructure closer to the aft of the vessel would contribute to a more efficient utilization of space. Likewise, it is essential to take into consideration the outline of the escape plan, as it demands for sufficient space for escape routes from the production areas to the living quarters [62].

The available space on deck is, in the first instance, determined by the length and breadth of the chosen ship. However, the storage and stowage methods based on the type of cargo derive in a diversity of solutions regarding hatches, cargo handling and overall deck arrangement.

Containers as well as general cargo ships, including MPVs, are designed to accommodate the greatest amount of containers or other bulky cargo on deck. As a result, most of the beam is occupied by the hatch covers. By contrast, tankers feature a wide and open deck space, but the lack of hatch openings complicates the installation of plant components. Bulk carriers, on the other hand, combine both available space on deck with big openings to the cargo holds.

Car carriers remain again a special case, due to the integration of the superstructure with the hull. This ship type has an extensive deck area and large equipment could be placed inside the hull using the bow or stern ramps.

5.3.4 Structural Characteristics

The installation of the equipment necessary for the functioning of the plant requires some basic structural characteristics in the selected vessel. These are, primarily:

- Sufficient bottom strength to withstand high point loads from the installation of heavy plant components.
- Capacity to maintain longitudinal strength integrity after conversion.
- Low accumulated fatigue from previous operational profile.

To be able to evaluate the suitability of the structure of the different ship types, a more detailed analysis would be necessary. It is, though, possible to highlight here some of the features that may make these a better or worse choice.

Once again, the storage of the cargo and the corresponding stowage method render a variety of structural configurations. Firstly, a distinction between distributed loads, induced by bulk cargoes, or point loads, from packaged or wheeled products, is defined.

Both bulk carriers and tankers are constructed to bear distributed loads. Even so, while the tank top of tankers would most likely need to be reinforced, bulkers must satisfy the GRAB notation, which results in an increase of the inner bottom and the lower hopper, to counteract the use of heavy loader grabs. Moreover, the inner bottom of bulkers designed to convey high density cargoes is reinforced with longitudinal stiffeners. This is also the case of some general cargo vessels intended to transport high density cargoes [52].

Conversely, ships designed to bear point loads, like containerships or car carriers face other structural challenges. The absence of a continuous deck causes containers to experience torsion loads, which are to be resisted by the torsion box running along the entire length of the beam. Similarly, this happens, to a lesser extent, to ships with big hatch openings, like general cargo, because large hatches reduce considerably the torsional stiffness of the structure [12]. Car carriers, on the other hand, lack transverse bulkheads, thus presenting a lower transversal strength against raking and torsional loads. Furthermore, they are susceptible to racking loads arising from the particular storage method of vehicles inside.

A last remark regarding the structural attributes of the vessel types must be made. When selecting a double bottom bulker or tanker, the additional low weight has to be considered from a stability perspective. If the installation of the systems does not lift the center of gravity, special attention needs to be paid to ballast management, to prevent the vessel from becoming too stable.

Altogether, it can be observed that the above technical criteria are directly connected to the subfunctions to be performed by the hull, as depicted in Subsection 2.4.3. Table 5.3 depicts the relation of these to the technical considerations along the preceding lines, to keep track of the functional requirements of the plant.

Table 5.3: Hull Functions / Technical Criteria

		Hull Shape	**Hull Configuration**	**Deck Space**	**Structure**
Motions Control	Stability				
	Seakeeping				
	Stationkeeping				
Systems Support	Spatial Integration				
	Buoyancy				
	Accessibility				
	Structural Integrity				

1st degree relation 2nd degree relation

5.3.5 Minimum Conversion

By minimum conversion it is meant the necessary engineering and construction effort to change the functionality of the vessel from an ongoing transport ship to a stationary offshore OTEC plant. Minimum conversion has two major implications as a standalone parameter: minimal conversion downtime and cost-effectiveness, both fundamental for the selection of the ship.

Despite being, to a large extent, a consequence of the foregoing criteria, minimum conversion is here considered a separate parameter. This is due mainly to the fact that it can rule out a ship type deemed acceptable by some of the criteria above, or the other way round. In other words, a ship that involves an extensive overhaul could be chosen if the hull configuration is convenient, whereas an easy-to-convert vessel could be disregarded because of structural considerations.

There are many aspects of the ship as a whole directly related to the complexity of its conversion: number and size of hatches, tank subdivision, removal of decks, mandatory installation of transverse bulkheads, structural reinforcements, access to the inside of the hull, amount of equipment to be removed, etc. Most of these aspects have been dealt with in previous sections and can be directly linked to specific vessel types. Others, nonetheless, are not that straightforward, like the amount of components to be withdrawn or the ease of access to them. This depends, among other issues, on the handling or maintenance of the cargo. As an example, tankers integrate specific systems for the transport of oil, such as: cargo oil heating system, cargo tank venting system, overflow control system or overflow control system.

5.4 Multi-Criteria Analysis of Ship Types

Finally, a simplied MCA is used as a decision-making tool to combine all the criteria described in the former sections. Table 5.4 renders bulk carrier as preferable design solution in accordance to the parameters studied in the preceding pages. Therefore, a representative bulk carrier will now be chosen for conversion.

Table 5.4: Multi-Criteria Analysis for the Selection of Ship Type

		Bulk Carrier	Tanker	Containership	MPV	PCTC
Commercial & Strategic	Previous Experience	+	+++	0	0	0
	Market Situation	++	-	+++	−	++
Technical	Hull Shape	+	+	+	++	−
	Hull Configuration	+++	+	0	+++	+
	Deck Space	++	+++	-	- -	+
	Structure	++	+	-	++	-
	Minimum Conversion	+++	+	-	+	- - -
Total		14+	9+	1++	5+	1-

Scale: - - - min. negative score; +++ max. positive score

The result of this analysis follows from the in-depth examination of the aforementioned aspects. The definition of the different considerations affecting the selection of the ship type may vary in accordance to the specific requirements of the project or the experience of the engineer. The parameters included herein have been deemed more relevant for the purpose of this research.

Additionally, it should be noted that, for the purpose of the design, the required ship size drives the price, and not otherwise. Nevertheless, if the market value of large vessels is much lower than that of smaller ones, bigger ship sizes will be taken into consideration. Large vessels might bring other advantages, such as the possibility of upgrading the plant, or reduced platform motions, and have been, thus, contemplated in former studies.

6

Platform Design

With the exception of the power delivery cable, not covered in this work, the main systems of the ship-shaped OTEC plant, in Figure 2.8, have been examined hitherto. The spatial integration of these components requires a floating structure capable of providing sufficient support for the systems on board, as it was defined in Subsection 2.4.1. In Chapter 5, a second hand bulk carrier is selected with this purpose. The specific characteristics of this ship type and the challenges it poses for the design are discussed below.

6.1 Methodology

In Chapter 2 a design process was developed to address one of the main challenges of the ship-shaped OTEC: the shape of the floating structure, as well as the systems of the power plant are fixed and not purposely designed for one another.

The former chapters followed the flowchart in Subsection 2.5. The functions of the plants have been translated into physical solutions: process equipment, mooring system and the selection of the most suitable ship type for conversion. As a result, the TLRs stated in Subsection 2.3 have been partly fulfilled.

The goal of this chapter is to accomplish TLR 2:

To integrate all components necessary to create the specified energy output in the most efficient arrangement possible.

To attain an adequate "Spatial Integration", taking into account the previous findings, several steps need to be completed:

1. Evaluate the actual space available inside the hull.
2. Analyze the impact of the identified design drivers on the general arrangement of the platform.
3. Allocate the equipment and dimension the plant in accordance to 1 and 2.
4. Propose conversion solutions to reduce the size of the platform, if necessary.

This will be further examined in the sections below.

6.2 Availability of Space inside the Hull

Before going into the spatial constraints of a bulker's structure, it is important to note that this assessment is carried out prior to any modification to the hull. Moreover, at this stage the analysis considers a single-deck design, as it was anticipated in Chapter 3.

From outside in, four main restricting characteristics have been identified: hull shape, function of vessel zones, bulkhead subdivision and the internal structure of the cargo holds.

6.2.1 Hull Shape

The first aspect to analyze is the influence of the hull shape on the actual space available inside. In general, a bulker features a long cylindrical body with finer lines predominantly at the bow and aft (Figure 6.1). Consequently, the impact of the hull lines will depend on the position and length of the equipment deck.

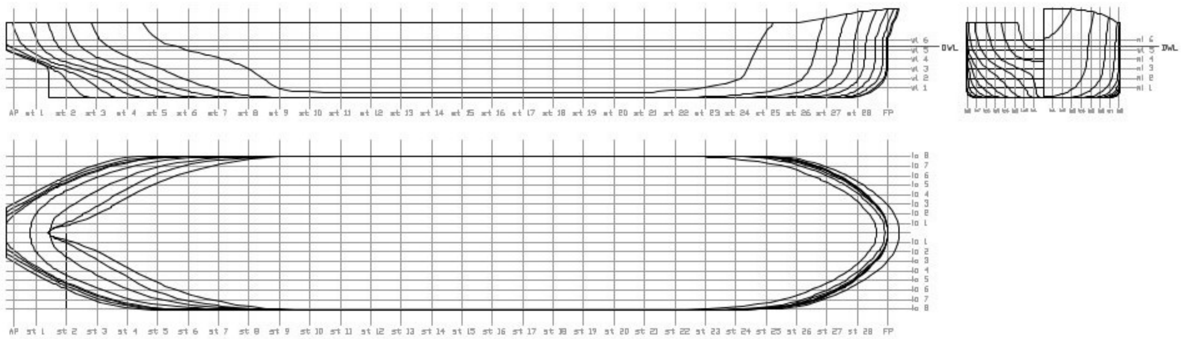


Figure 6.1: Preliminary Hull Lines of Post-Panamax Bulker

6.2.2 Vessel Zones

The spaces inside the bulk carrier will serve a distinct purpose in the new application. In order to better understand the conversion, four parts are differentiated (Figure 6.2):

1. Bridge and accommodation block: the plant is originally designed to be minimally manned. The accommodation areas can retain their initial function with only minor modifications.
2. Engine room: to accommodate electrical equipment, such as switchboards and transformers, to transmit the generated energy to the power delivery cable.
3. Cargo holds: to contain the process cycle equipment.
4. Main deck: to provide space for auxiliary equipment, escape routes and mooring system components.

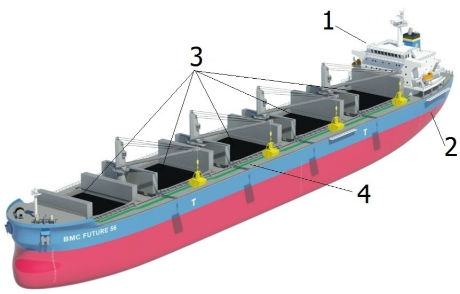


Figure 6.2: Main Parts of a Bulk Carrier

The engineering of the electrical equipment is not within the scope of this thesis, as it was depicted in Table 2.4. Additionally, it is evident that the size of the hull will be primarily determined by the area required for the process equipment. Nevertheless, the turbine generator package could be placed in the engine room or the cargo holds, depending on the space available inside each of them and the original hull configuration. Therefore, the focus of the following lines will be on the space formerly reserved for the cargo.

The length of the cargo holds is, in turn, delimited by the fore peak and the engine room. The prime dimensions of the cargo space, length, beam and depth, will be studied concurrently.

6.2.3 Bulkhead Subdivision

The cargo space is divided in different holds by transverse bulkheads. The bulkheads prevent the ship from sinking if one or two compartments are flooded. This compartmentalization is mandatory according to the rules specific to each ship type, in this case, a bulk carrier. In general, bulkers are designed to withstand the flooding of a single compartment.

The spacing between the bulkheads comes from the definition of floodable length given by SOLAS [41]: “the maximum portion of the length of the ship, having its center at the point in question, which can be flooded [. . .] without the ship being submerged beyond the margin line”. This distance is directly related to the geometry of the ship, and is crucial for its safety. Therefore, they cannot simply be removed or shifted longitudinally to adjust their position to the preferred power plant arrangement.

Currently, there are no rules applying to ship-shaped OTEC plants. On the other hand, FPSOs, as stationary floating platforms, are the most similar installations, but their mission, the production and storage of oil, and the associated risks are substantially different from those of the present concept. For this reason, the original subdivision will be maintained in principle. Later on, if a different bulkhead plan is necessary, the floodable length curve of the ship can be reviewed.

Furthermore, the new function of the vessel will require the installation of pipes and cables as well as the opening of doors for the passage of personnel through the cargo holds. The penetration of watertight bulkheads and decks can be carried out as long as the following provisions from Bureau Veritas (BV) *Rules for the Classification of Steel Ships* [21] are complied with.

- Openings (doors, manhole ventilation ducts, etc.): Pt B, Ch 2, Sec 1 [6]

- Piping systems: Pt C, Ch 1, Sec 10 [5.3 - 5.5]

- Cables: Pt C, Ch 2, Sec 12 [7.5]

Lastly, the space occupied by corrugated bulkheads, including bulkhead stools is not significant relative to the total length of a ship, but they complicate the layout of equipment, especially component group, and, must, consequently, be taken account.

6.2.4 Internal Structure of a Cargo Hold

According to SOLAS IX/1.6 [41], a bulk carrier is "a ship which is constructed generally with single deck, top-side tanks and hopper side tanks in cargo spaces, and is intended primarily to carry dry cargo in bulk, and includes such types as ore carriers and combination carriers". This is illustrated by Figure 6.3, from which a number of constraints can be pointed out:

- Topside and hopper tanks
- Bulkhead stool
- Structural reinforcements at the sides

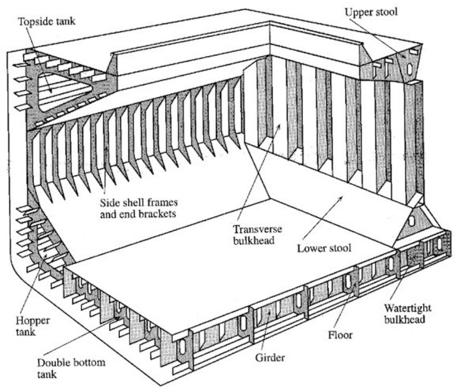


Figure 6.3: Internal Structure of a Cargo Hold

Accommodating the systems on a deck 5.52 m above the tank top, as indicated in Chapter 3, implies that these constraints will have a limited effect on the layout. In particular, and depending on the size of the ship, the bulkhead stools and hopper tanks will most likely have a minimal impact on the space available at that deck level. However, if the remaining distance between the deck and the topside tank is too small, it may make it impossible to locate large components at the sides, thus reducing the usable space on the deck. Lastly, the side shell frames do not take up much space, compared to the previous aspects, but they have to be included in the calculation of the width.

All in all, the above considerations have to be evaluated with respect to the overall beam and depth of the vessel.

6.3 Impact of Design Drivers on Power Plant Layout

In Chapter 3, the main design drivers, i.e. water pipes and heat exchangers, were identified. The design of the platform is, to a large extent, dependent on these component groups. Accordingly, they will be treated separately prior to the development of the general arrangement.

In line with Subsection 3.3.1, a cold water pipe, a warm pipe and a discharge pipe need to be attached to the hull to assure the supply of seawater to the heat exchangers and its subsequent discharge. The result of selecting this solution over a multi-pipe design is that the evaporators and condensers will be clustered, reducing this way the total length of the process piping. This, in turn, has an effect on the pumping power consumption. For instance, if the arrangement of the equipment is such that the discharge of water cannot be done by gravity, extra pumps will have to be installed. The large number of components, without yet including all plant systems, and the different fluids, temperatures and flow rates complicate significantly the layout of the piping.

6.3.1 Water Pipes

The first thing to note is that all water pipes are located along the centerline of the vessel, where they will be less affected by roll angles. By doing so, the connection of the pipes will be easier, since there will be no variation in height or space available far from the vessel's axis, as shown in Figure 6.4. An additional benefit is that the intakes and oulet will be at the same distance of the two power modules, which implies that the symmetry of the plant is maintained.

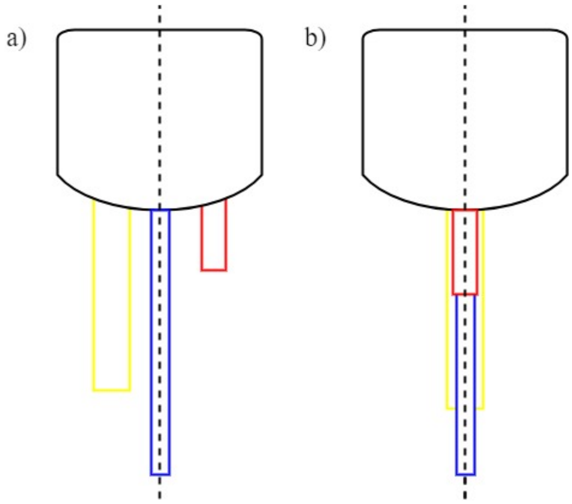


Figure 6.4: Schematic Cross-Sections of the Platform

The subsequent stage is to decide whether all three pipes should be clustered in one location or if they should be placed at a certain distance from the others (Figure 6.5).

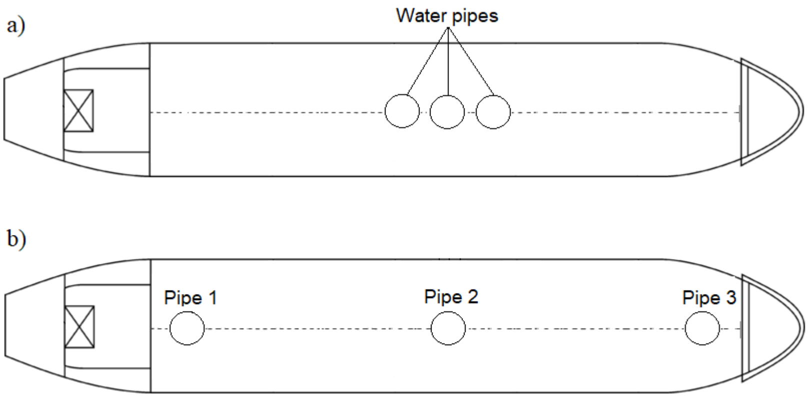


Figure 6.5: Clustered (a) vs. Distanced (b) Water Pipes

The prime advantage of Layout (a) is that amidships is the position with less motions. This is because a spread mooring system was chosen over a turret mooring in Chapter 4. There are, however, several arguments to add distance between the pipes:

- To reduce the hydrodynamic interaction among the pipes. Being clustered, the flow of water around the pipes could induce great motions and accelerations on the platform and ductings.
- To avoid concentrating structural stresses on a specific zone of the vessel. This is particularly important because the presence of the pipes disrupts the structural integrity of the hull.
- If all three pipes are located in the same position, the auxiliar systems in charge of the Seawater Supply (pumps, sumps, piping, etc.) have to be accommodated nearby, which would overcrowd the center of the vessel.

On account of this rationale, the water will be separated and placed along the longitudinal axis of the ship.

Initially, the reduction of platform motions had as prime objective to prevent operational problems derived from the coupling of the CWP and the hull. There are two ways to approach this: installing a restrictive mooring system, examined in Chapter 4, and choosing an optimal pipe location in the hull.

When the advantages of the subsequently selected spread mooring were exposed in Section 4.3, the importance of limiting the movement of all three water pipes was pointed out. The reason is that both the WWP and the DWP, though shorter than the CWP, feature bigger diameters, and can, therefore, experience or induce large loads on the hull. This impedes optimizing the layout only with the original criterion.

The scheme in Figure 6.6 shows the possible locations of the water pipes.

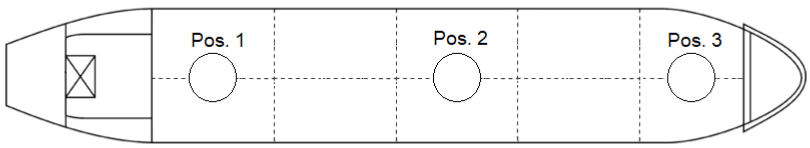


Figure 6.6: Scheme of Possible Positions of the Water Pipes

The three different pipes and locations render 6 potential combinations, illustrated in the matrix in Table 6.1.

Table 6.1: Matrix of Possible Positions of the Water Pipes

	Pos. 1	**Pos. 2**	**Pos. 3**
a	CWP	WWP	DWP
b	CWP	DWP	WWP
c	WWP	CWP	DWP
d	WWP	DWP	CWP
e	DWP	CWP	WWP
f	DWP	WWP	CWP

Before analyzing each of them individually, the next aspects must be taken into account:

- The WWP is the smallest both in length and diameter. As a result, a stronger preference will be given to the CWP and the DWP regarding the amidships position, i.e. Position 2, in which motions are minor compared to the other locations.
- The electrical equipment and the attachment of the power delivery cable will be accommodated in the engine room. The turbines should then be placed at the aft of the cargo space. In addition, looking again into cycle continuity, the ammonia vapor must flow from the evaporators to the turbines. Since the evaporators have to be set near the WW inlet to minimize ducting, this altogether suggests that the WWP should be located after in the first cargo hold.

There are as yet no evident reasons to disregard any of the remaining options, *c* and *d*, in Table 6.1. This will be studied more in-depth in later sections.

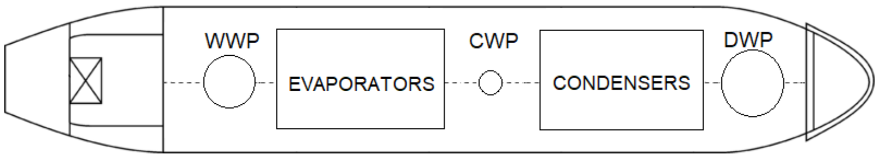
6.3.2 Process Cycle Ductings

The role of the process cycle ductings as design drivers was highlighted in Chapter 3. The piping acts as link between the components of the power modules, ensuring the continuity of the cycle. On their own, however, they are not sufficient motive to discard a certain arrangement. Accordingly, they have to be studied in relation to the water pipes and the heat exchangers.

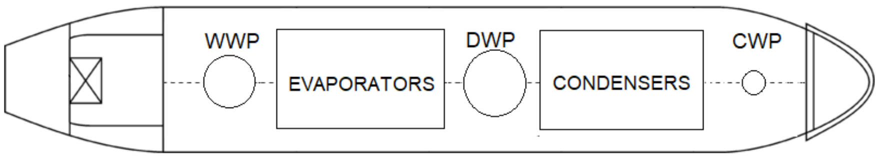
6.3.3 Heat Exchangers

The choice of plate heat exchangers over more compact solutions will make the arrangement fundamentally different from former concepts ([49],[59],[72],[70]), as was already introduced in the P & ID in Section 3.7.

Following from previous sections, evaporators and condensers will be clustered and placed according to the position of the water pipes. In Subsection 6.3.1, the possible pipe configurations were narrowed down to two: { WWP,CWP,DWP } and { WWP,DWP,CWP }. In addition, the evaporators of the two modules have been set in the proximity of the WW intake, i.e. between Position 1 and 2 in Figure 6.7. Likewise, the most logical position for the condensers will be next to the CWP, i.e. between Position 2 and 3 in Figure 6.7.



(a) Option *c*



(b) Option *d*

Figure 6.7: Basic Plant Arrangements, Based on Options in Table 6.1

To assess properly different layouts it is key to bear in mind the size of the equipment to be arranged (Table 6.2).

Table 6.2: Heat Exchanger Dimensions (from Appendix C.2)

L (m)	7.080
W (m)	1.550
H (m)	4.095
MS* (m)	2.35

*Recommended Maintenance Space (MS) at the sides and back

Splitting the power plant into two separated modules poses a major design condition: symmetry. Correspondingly, the heat exchangers will need to be situated either along the centerline or at equal distances from it.

Henceforth, two main arrangement options will be examined: heat exchangers arrayed perperdicular or parallel with respect to the ship's length. To introduce an angle between the components and the vessel's axis is not considered at this stage. This, however, will be looked into if the initial alternatives prove not to be suitable for the fixed L/B ratio.

In line with the previous, Figure 6.8 depicts several configurations of the heat exchangers, including the recommended space for maintenance.

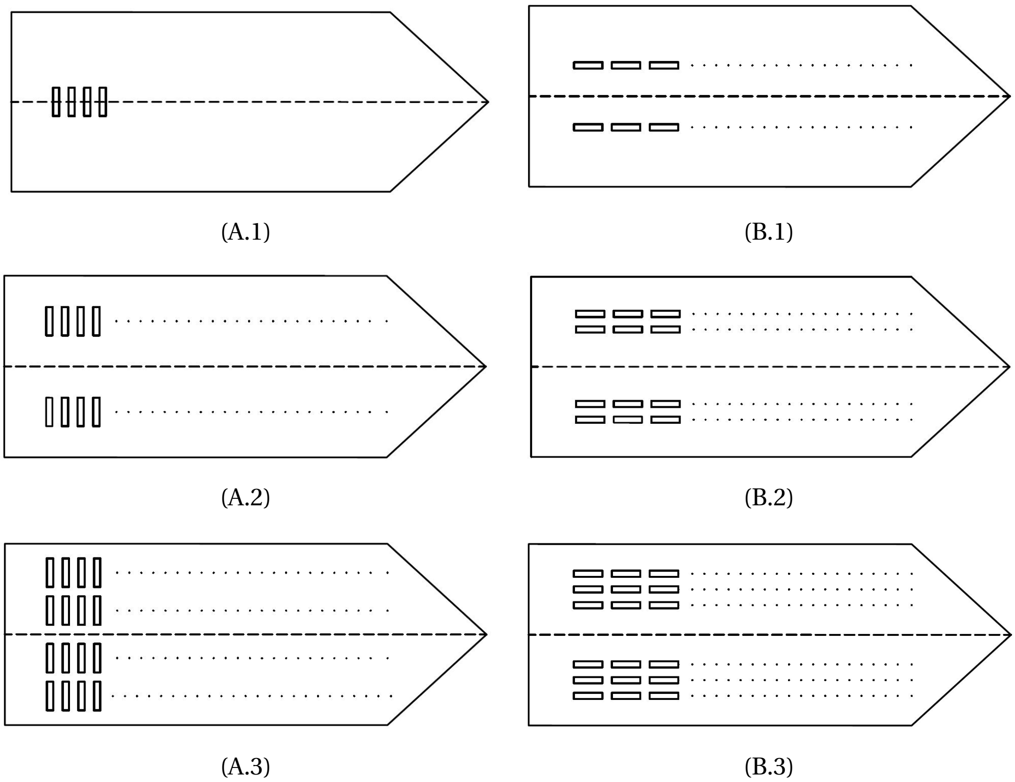


Figure 6.8: Possible Heat Exchanger Configurations

A priori, on the basis of the above, a few of these configurations can be disregarded:

- A.1
  Too space-consuming without entailing any particular benefits other than simplicity. In this configuration, only the heat exchangers would take up to 140 m (36*(1.55+2.35)) of the length corresponding to the cargo holds. The beam of a ship this size can easily reach 29 m, meaning that the approximately 30 % of the available beam is utilized.

- B.1
  Like in the previous configuration, this option demands for 170 m (36*(7.08+2.35)) of cargo space. As in the preceding case, two heat exchangers arrayed in parallel would leave an excessive amount of empty space regarding the beam of the vessel.

- A.3
  The beam required for the heat exchangers would be nearly 38 m (4*7.08+3*37). Arranging the components in 9 consecutive rows would make this solution more suitable for a squared structure.

- B.3
  In a similar way, B.3 would leave too much unused space length-wise, i.e. it is not optimal for the studied L/B ratio. Moreover, since the number of evaporators and condensers is not a multiple of 6, the arrangment would need to be adjusted.

To better illustrate the arrangement of the remaining options, namely A.2 and B.2, the layout of condensers of one of the modules is shown in Figure 6.9 and 6.10. Before proceeding with the analysis, it must be reminded that the dimensions of the water ductings are unknown. Therefore, the dimensions depicted in Figure 6.9 and 6.10 are a rough estimation of the actual required space. The advantages and disadvantages of both are evaluated more thoroughly in the following lines.

– Configuration A.2

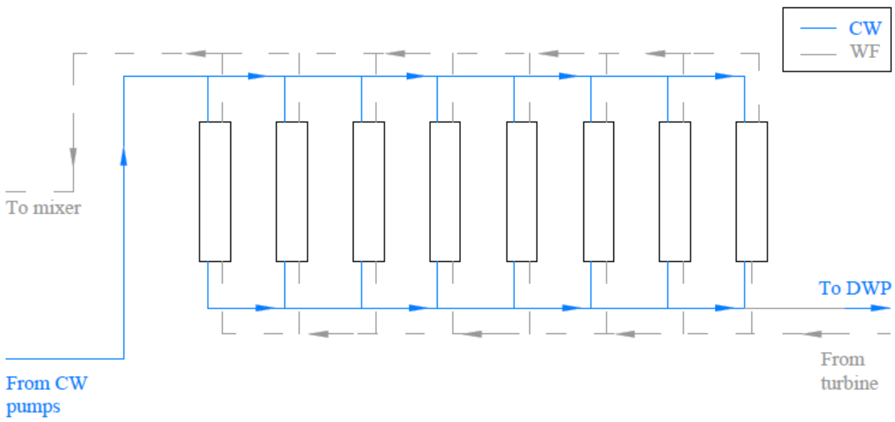


Figure 6.9: Detailed A.2 Configuration

+ Simple arrangement
+ Scalable
+ Uncomplicated ducting layout
+ Easy to accommodate component groups in one cargo hold
– The beam is expected to be quite large, considering the diameter of the ductings, the space for maintenance and the length of the heat exchangers.

– Configuration B.2

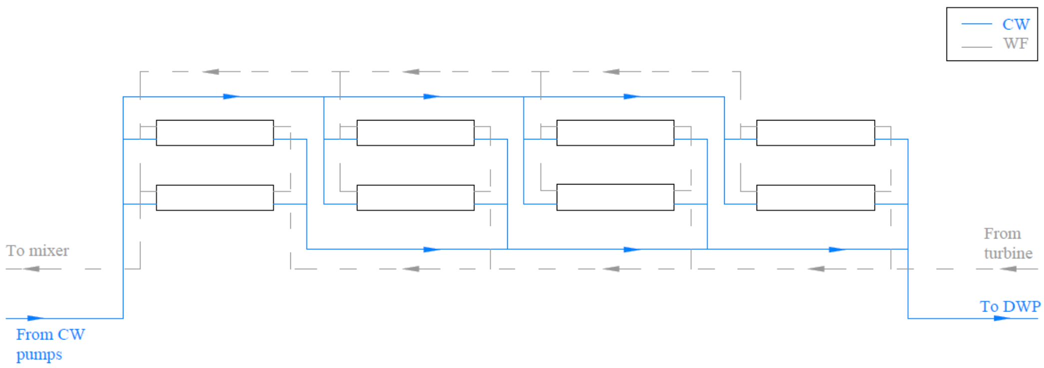


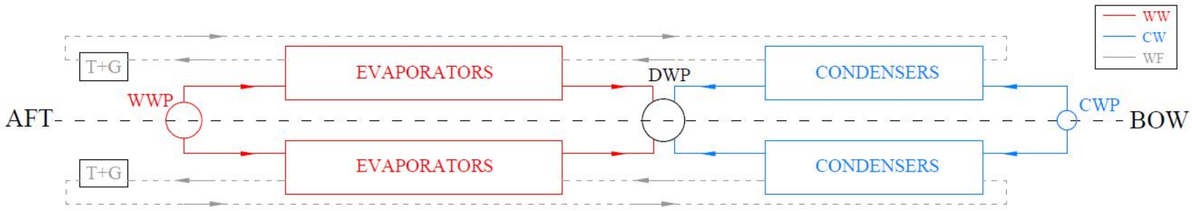
Figure 6.10: Detailed B.2 Configuration

+ Easier to adapt to L/B ratio
+ More design flexibility, i.e. the space at the sides of the heat exchangers could be used to install auxiliar equipment, instead of placing it between groups of components
– Complex component layout in relation to the divisions of the cargo space
– Difficult piping arrangement, featuring more elbows than the previous, which could increase the necessary pumping power

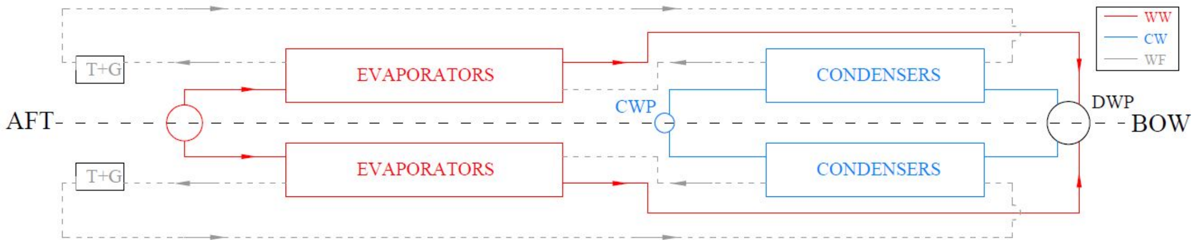
– Length requirements. The opposite from Configuration A.2, a very long vessel will be needed, which, considering the bulkheads, could lead to buying an oversized hull, leaving an excess of unused beam

Having assessed all this, Configuration A.2 is chosen over Configuration B.2. The prime reasons behind this decision are the compactness and simplicity of this solution, especially when it comes to the routing of process piping.

The analysis of the design drivers has rendered the two alternative general arrangements in Figure 6.11. Figure 6.11a and 6.11b differ mainly in the location of the CWP and DWP. This has significant implications for the intricacy of the process piping and, ultimately, for the performance of the platform.



(a) Option *c*



(b) Option *d*

Figure 6.11: Alternative General Arrangements

As previously stressed, the motions of the platform are expected to be affected by the water pipes. The position at which the pipes would experience minimal motions is amidships. Both CWP and DWP feature large dimensions, the first being longer and the latter with a bigger diameter. It follows from this that the selection of one alternative cannot be made without a hydromechanic analysis. On the other hand, Figure 6.11 shows a more complex piping arrangement, possibly entailing a need for discharge water pumps. This, in turn, may derive in a larger width of the ship to be converted.

For the purpose of this research, the size of the platform will be estimated according to Figure 6.11a. This layout will be then assessed regarding space utilization and stability. The general arrangement can be found in Appendix F.

6.4 Vessel Size Estimation

Section 6.2 and 6.3 have examined the space constraints inside a bulker's hull and the key elements determining the layout of the power plant. In the next lines the outcome of the former sections will be integrated in a more comprehensive arrangement of the ship-shaped OTEC platform.

Table 6.3 summarizes the dimensions of the components groups comprising the "Process Equipment" and the "Seawater Supply System", as arrayed in Section 3.3 and 6.3.3. It must be noted that the length and width of the condensers and evaporators refer to the size of the heat exchangers of one module, plus the recommended maintenance space, i.e. 2.35 m. Also, the height of the "Seawater Supply System" is determined by the dimensions of the sumps and the water pumps, without yet knowing the height of the pump motor.

Table 6.3: Summary of the Dimensions

	Seawater Supply			Electricity Generation		
	WW	CW	DW	Evaporators	Condensers	Turbine+Gen
Length (m)	13.95	9.35	13.95	39	31.2	6.35
Width (m)	16.74	11.22	16.74	9.43		3.66
Height (m)	- 5.52 / + 5.01			+ 4.10		+ 3.05

∗ Height below (-) and above (+) the equipment deck

The uncertainties and assumptions involved in this concept design will limit the validity of the vessel's size to be obtained. This is reinforced by the fact that only the largest pieces of equipment are incorporated in the design, and that no database with detailed information on existing bulkcarriers is available for consultation. On this basis, the required dimensions of the second-hand vessel will now be estimated.

6.4.1 Width

The selected layout of heat exchangers, in Figure 6.9, makes them the biggest contributor to the ship's beam. This can be confirmed by comparing the width of the largest water sump in Table 6.3 to the space needed for the heat exchangers of the two modules (2*9.43), in spite of excluding the process ducting around them. Hence, a closer view of Figure 6.9 gives an approximation of the minimum width.

- Maintenance space = 2.35 m
- Length of heat exchanger = 7.08 m
- Piping space (from Section 3.5) = 2.5 m
- Connection distance (assumption) = 0.50 m

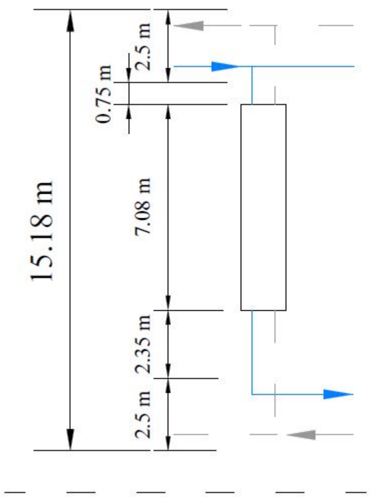


Figure 6.12: Width Estimation, from Figure 6.9 (dashed line: longitudinal axis of the vessel)

Additionally, a distance of 0.5 m has been introduced in between the power modules and at the sides of each of them. If symmetry is now applied to Configuration A.2., the total width is, then, 31.36 m.

Without taking into account the lateral reinforcements, this value is, by a narrow margin, within the range of Panamax vessels.

6.4.2 Length

First, it is worth mentioning that the length of the GA possibilities, {WWP,CWP,DWP} and { WWP, DWP,CWP }, is expected to be very similar, since only the position of the CWP and the DWP have to be reversed.

The space added between component groups is assumed to be 6.5 m, which may be modified when the secondary equipment and pipes are included in the layout.

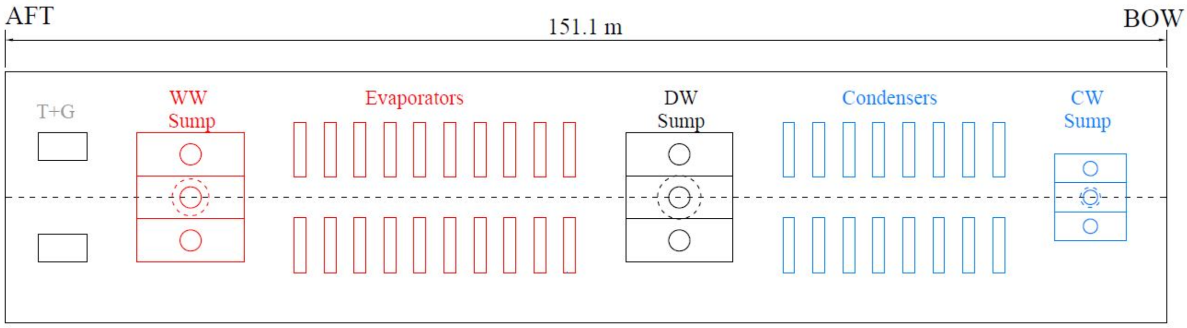


Figure 6.13: Length Estimation, from Figure 6.9 (dashed line: longitudinal axis of the vessel)

The length in Figure 6.13 corresponds to the cargo space, where the power plant systems will be accommodated. It can be observed that the turbine and generator package have been included in this basic layout, resulting in a longer vessel. This will be reviewed later on once a vessel is selected for conversion.

The length of the engine room and the distances of the aft and fore peaks vary from one ship to another. However, to obtain the overall length of the bulker, these spaces have to be taken into account. According to BV Rules Pt B, Ch 2, Sec 1 [21], a minimum value of 10 m will be used as the distance between the collision bulkhead and the forward perpendicular. There are, nonetheless, no specific regulations regarding the length of the engine room and the position of the aft peak bulkhead. A revision of bulk carrier designs, has rendered a range of 20 to 30 m for the engine room, while the distance of the aft peak bulkhead will be taken as the same of the fore peak.

All in all, a vessel of approximately 200 m long, i.e. a Handymax or larger [86], has to be selected. For all that, it must be reminded that this conceptual layout does not consider the spatial constraints that the transverse bulkehads pose for the arrangement.

6.4.3 Height

From Table 6.3 it follows that the maximum height required will be determined by the "Seawater Supply System". The water pumps will be located in the area around the centerline of the vessel, where the height inside the cargo hold is bigger. In addition to the height of the pumps, 4 m are added to account for the unknown height of the motor pump. The heat exchangers, on the other hand, will be closer to the sides of the hull. That is to say, the height needed will be different depending on the distance to the vessel's axis.

Additionally, from BV Rules Pt B, Ch 2, Sec 2 [21], the recommended height of the inner bottom is given as:

$$h = \frac{B}{20} \tag{6.1}$$

Where B has been taken from the width estimated before, rendering a tank top height of 1.6 m.

All these design aspects are presented in Figure 6.14.

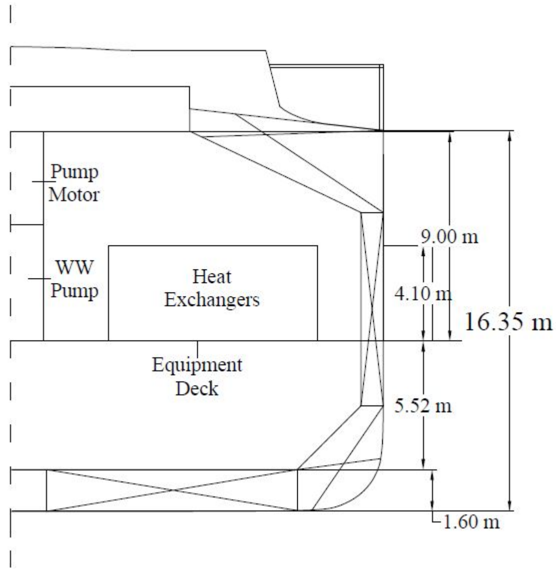


Figure 6.14: Height Estimation

Handymax, and thus also Panamax vessels, feature larger depths than the obtained height. Therefore, it can be said that the resulting height, 16.35 m is less restrictive than the other dimensions.

These dimensions, as well as the deadweight estimated in Section 3.6, determine the size of the bulker. Up to this point, it appears that, due to the slenderness of the hull and the chosen layout, the width is the most constraining dimension. This, however, has to be verified by integrating the arrangement developed before with the structure of a second-hand bulker. The next step is, then, to adjust the layout to the actual bulkhead subdivision of a representative existing Panamax bulk carrier.

6.5 Platform Layout

From Significant Ships of 2004 [80], a Panamax bulk carrier, the PROTEFS, has been selected for the ship-shaped OTEC. Core information of the ship can be found in Appendix E. In Table 6.4, the dimensional requirements of the platform are compared to the dimensions of the chosen vessel. As all these prerequisites are met, the suitability of PROTEFS is verified.

Table 6.4: Requirement Verification

Dimension	Requirements	PROTEFS
L (m)	151.1	177.3
B (m)	31.4	32.2
D (m)	16.4	19.2
DWT (t)	4136.2	63395

∗ The length indicated corresponds to the length of the cargo space

The cargo space of the bulker is divided in seven holds, as shown in the GA in Figure 6.15. This hull configuration will condition the layout of the equipment previously designed.

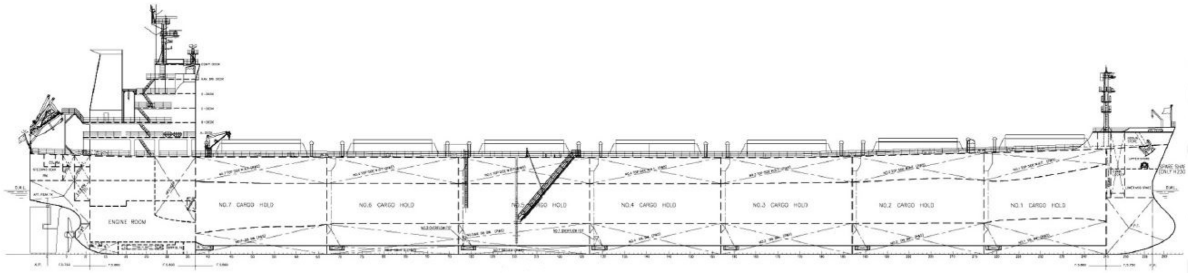


Figure 6.15: General Arrangement of PROTEFS Bulk Carrier

The distance between the transverse bulkheads in the cargo space is 25.3 m. Therefore, it becomes clear that, maintaining the original bulkhead plan, the groups of evaporators and condensers have to be allocated in two different holds, of 19.5 m and 15.6 m, respectively. In consequence, there are, in total, 8 component group that need to be placed in the seven holds: turbine and generator sets, three water sumps, evaporators (2) and condensers (2).

As aforementioned, the turbine and generator packages could be located in the former engine room or in the cargo space. The length of the engine room, 22 m, provides sufficient space for these components and other ancillary electrical equipment. Additionally, in this vessel zone, the space on the tank top can be used to accommodate other systems. All in all, the removal of the turbine and generator sets from the cargo space proves to be a feasible solution to facilitate the integration of the power plant systems with the vessel's hull.

Figure 6.16 depicts a top view of the resulting plaform layout. A more detailed general arrangement is attached in Appendix F.

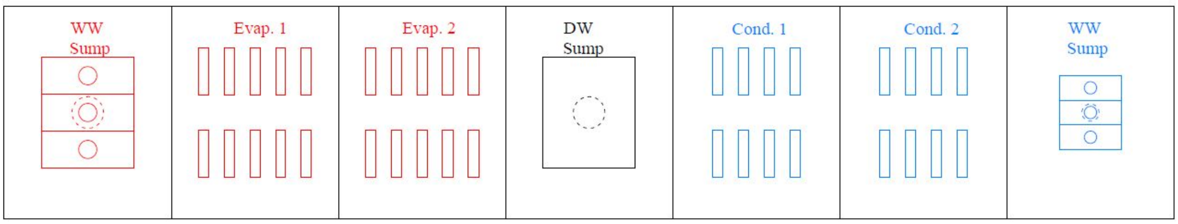


Figure 6.16: Distribution of Systems Inside the Cargo Holds

Note: 3D, side and top views of platform yet to be included in the appendix

6.6 Evaluation of Space Utilization

From a ship design perspective, the accuracy of the fit between the layout of systems and the selected bulker is of great significance. A basic evaluation will look into the ratio of used versus total available space on the equipment deck.

Table 6.5 summarizes the area corresponding to the main equipment, based on the dimensions of the components.

Table 6.5: Area Occupied by Power Plant Systems

		Area (m^2)
Sumps	CW	104.9
	WW	233.5
	DW	233.5
HX		631.9
T+G		46.5
Total		1250.3

The area of the engine room and cargo holds, as obtained from the model in *Rhinoceros*, is 5837.2 m. As a result, the ratio between the used and empty is space is 21.4 %. While this value might initially seem low, it must be reminded that the design does not yet include the following items:

- Water and working fluid ducting
- Electrical equipment: switchboards, transformers, etc.
- Cycle components: separators, absorbers, etc.
- Ancillary equipment: valves, working fluid pumps, etc.

Considering the estimated ratio, it may be the case that a smaller platform would not have sufficient space to accommodate all systems. In addition, the actual benefits of purchasing a smaller vessel for conversion will depend, to a large extent, on the market situation. In the past months, the second-hand value of these vesels appears to have firmed. However, with the expansion of the Panama Canal, the price of second-hand Panamax vessels has moved closer to that of smaller bulkers, which diminishes the economic gain of a size reduction (Figure 5.4). Thus, for the purpose of this thesis, this ratio will not be further maximized.

Nevertheless, once all components are incorporated in the Detailed Design stage, the use of space should be reassessed. If the space utilization remains low, a reduction of the width, being the most restrictive dimension, will be the first step towards decreasing the required ship size. Some of the solutions proposed below are aimed in that direction, whereas others approach the problem from a holistic point of view:

- Reduce the recommended space for maintenance of the heat exchangers.
- Introduce an angle between the heat exchangers and the vessel's axis.
- Choose a different configuration for the heat exchangers from the ones in Figure 6.8, and optimize the design with respect to a different dimension.
- Install smaller components, but a larger number of them. This could be applied to both water pumps and heat exchangers. The first could allow for the construction of a second equipment deck, while the latter could reduce the required width, improving the flexibility of the design.

7

Weight and Stability

In former chapters, the importance of motion minimization has been stressed. The stability is a vital component of the function "Motions Control", and it affects largely the operability and safety of the installation. In particular, the working limits of the power plant equipment and the integrity of the hull-pipe interface call for a initial assesment of the platform's stability. Accordingly, the weight and center of gravity of the platform, comprising the empty hull and the power plant systems, are required inputs for the subsequent stability computations.

7.1 Limitations and Assumptions

The validity of the findings of this section is constrained by the following aspects:

- The hull lines of the PROTEFS are not included in *Significant Ships of 2004* [80]. To solve this, a hull of equal length, beam, depth and block coefficient has been built and imported into *Maxsurf Stability Advanced*.
- The ratio between equipment newly installed and systems removed from the ship is unknown. In this regard, the steel weight of the hull and superstructure, as well as the weight of the process systems are estimated.
- The process piping and other ancillary components have not been considered in this work. The direct consequence of adding these to the design will be a change in the vessel's weight and the center of gravity. For this reason, the computations of the righting lever arm, GZ, are done for a range of displacements from 12000 t to 22000 t.
- The actual loads and impact of the attachment of the water pipes and mooring system to the hull are not taken into account.

Nevertheless, the results of the calculations will point out which aspects of the design need to be improved in first instance. This will be the basis of the subsequent Detail Design.

The stability check begins with the study of the weight and center of gravity of the ship-shaped OTEC.

7.2 Platform Weight and Center of Gravity

The weight of the platform is broken down into its main constituent parts: hull, superstructure and power plant systems. This division is necessary for the estimation of the overall center of gravity.

7.2.1 Ship Structure

The lightship weight of the PROTEFS, 12405 t, can be found in Appendix E, as given in *Significant Ships of 2004* [80]. However, this value includes, not only the structural weight of the bulker, but also the outfit and machinery weight. In order to obtain an estimation of the steel weight of hull and superstructure, the guidelines in [84] and [83] will be followed. Table 7.2 shows the main dimensions of the selected bulk carrier, which will be used to estimate the weight and center of gravity of the ship.

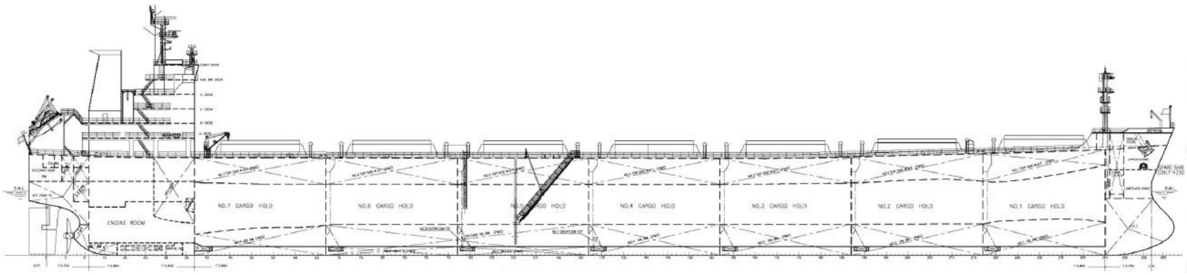


Figure 7.1: General Arrangement of PROTEFS Bulk Carrier (repeated from Figure 2.7)

Table 7.1: Main Dimensions of PROTEFS

LWL (m)	221.2
B (m)	32.2
T (m)	12.5
D (m)	19.2
Cb	0.83

- Weight
  As indicated in [84] and [83], the first step is to obtain the Lloyd's Equipment Number, E, from Equation 7.1. This value will be later used in the weight estimation.

$$E = \underbrace{L(B+T)+0.85L(D-T)}_{\text{hull}} + \underbrace{0.85\sum l_1 h_1 + 0.75\sum l_2 h_2}_{\text{superstructure and deck houses}} = E_{hull} + E_{ss} \tag{7.1}$$

l_1, h_1 = length and height of full width erections (m)
l_2, h_2 = length and height of less than full width erections (m)

In Figure 7.1 it can observed that the superstructure is equal in length to the engine room, while its width is smaller than the beam. The height, on the other hand, is unknown and has to be assumed. A standard height of 2.30 m is given in BV Pt B, Ch 1, Sec 2, [3.20] [21]. The PROTEFS has 6 different deck levels on the superstructure, resulting in a total estimated height, h_2 of 13.8 m.

Table 7.2: Values of E for Hull, Superstructure and Complete Vessel

	E
Hull	11147
Superstructure	417
Structure	11564

Once the E numbers have been found, the total structural weight can be obtained from Equation 7.2.

$$W_s = KE^{1.36}[1 + 0.5(C_b' - 0.7)] \tag{7.2}$$

Where:

W_S = structural weight of ship (t)
K = coefficient, dependent on the ship type, from [83]
E = Lloyd's Equipment Number
C_b' = corrected block coefficient at 0.8D

The value of K, 0.031, for bulkers is taken from Figure 7.2.

Type	K			Range of E	No. of ships in sample
	Mean value		Range		
Tankers	0.032	±	0.003	1500–40000	15
Chemical tankers	0.036	±	0.001	1900–2500	2
Bulk carriers	0.031	±	0.002	3000–15000	13
Container ships	0.036	±	0.003	6000–13000	3
Refrigerated cargo	0.034	±	0.002	4000–6000	6
Coasters	0.030	±	0.002	1000–2000	6
Offshore supply	0.045	±	0.005	800–1300	5
Tugs	0.044	±	0.002	350–450	2
Research ships	0.045	±	0.002	1300–1500	2
Ro-Ro ferries	0.031	±	0.006	2000–5000	7
Passenger ships	0.038	±	0.001	5000–15000	4
Frigates and corvettes	0.023		not known		

Figure 7.2: Value of K [83]

Lastly, the corrected block coefficient is to be obtained from Equation 7.3.

$$C_b' = C_b + (1 - C_b)\left[\frac{0.8D - T}{T}\right] \tag{7.3}$$

The weight, on the other hand, cannot be decomposed in a similar way. Nevertheless, the weights of the hull and superstructure need to be calculated to approximate the center of gravity of the complete structure further on. As a consequence, the weight of the superstructure will be found from Equation 7.4.

$$W_{ss} = W_s(E_{hull} + E_{ss}) - W_{steel}(E_{hull}) \tag{7.4}$$

The results of substituting the given vessel dimensions in Table 7.2 in the previous equations is depicted in Table 7.4.

Table 7.3: Weights of Hull, Superstructure and Complete Vessel

	Weight (t)
Hull	10829
Superstructure	555
Structure	11383

- Center of gravity
  The vertical center of gravity of the vessel is found from Equation 7.5 [53], while the longitudinal center of gravity can be estimated using Equation 7.6, derived from [84].

$$VCG_{hull} = 0.01D\left(46.6 + 0.135(0.81 - C_b)\left(\frac{L}{D}\right)^2\right), \qquad L \geq 120\text{m} \tag{7.5}$$

$$LCG_{hull} = -0.15 + 0.70LCB, \qquad (\%L, +fwd) \tag{7.6}$$

  Where:

  LCG = longitudinal position of center of gravity (% L from amidships, +fwd)
  LCB = longitudinal position of center of buoyancy (% L from amidships, +fwd) (Maxsurf)

  This percentage is then translated to the actual position, from the forward perpendicular, with the Equation 7.7.

$$LCG = \frac{L_{WL}}{2} - \frac{0.15 + 0.70LCB}{100}L_{WL} \tag{7.7}$$

$$L_{\text{WL}} = \text{length in the waterline}$$

  The longitudinal position of the centroid of the superstructure is assumed to be at half of the length of the engine room, from the fore perpendicular, i.e. 196.7 m. In addition, the vertical center of gravity will be located at half of the height of the superstructure, from the baseline of the hull, i.e. 26.1 m.

Table 7.4: Weights of Hull, Superstructure and Complete Vessel

	LCG (m)	**VCG** (m)
Hull	110.9	8.5
Superstructure	196.7	26.1
Structure	**115.1**	**9.7**

7.2.2 Power Plant Systems

The equipment weight was analyzed before in Section 3.6. Therefore, the calculations included hereby are carried out only to estimate the center of gravity of the primary components. Several assumptions has been made to perform this estimation:

- The center of gravity of each piece of equipment is considered to be located in the centroid of the volume box drawn from the dimensions given by the manufacturer.
- The weights of the process ducting and water pipes are unknown. To account for these, an additional margin of 30 % to the initial weight of the equipment.
- Since the size of the motors of the water pumps have not been provided, their center of gravity is assumed to be at the top of the sump tanks.

The contribution of the CWP, estimated in Subsection 3.3.1 needs to be added to the calculation. For the purpose of this work, this weight is included as a point load in the attachment point of the CWP, considered to be the height of the double bottom, i.e. 1.6 m. The longitudinal center, on the other hand, is as drawn in the general arrangement, i.e. 25.1 m.

Table 7.5: Calculation of Center of Gravity of Power Plant Systems

		From FP		Longitudinal	Vertical
COMPONENTS	**Weight (t)**	**lcg (m)**	**vcg (m)**	**Product (t*m)**	
T+G 1	20.9	191.2	8.6	3998.1	180.7
T+G 2	20.9	191.2	8.6	3998.1	180.7
Cond. 1a	22.7	40.5	9.2	919.5	208.0
Cond. 1b	22.7	40.5	9.2	919.5	208.0
Cond. 2a	22.7	44.4	9.2	1008.0	208.0
Cond. 2b	22.7	44.4	9.2	1008.0	208.0
Cond. 3a	22.7	48.3	9.2	1096.4	208.0
Cond. 3b	22.7	48.3	9.2	1096.4	208.0
Cond. 4a	22.7	52.2	9.2	1184.9	208.0
Cond. 4b	22.7	52.2	9.2	1184.9	208.0
Cond. 5a	22.7	65.8	9.2	1493.3	208.0
Cond. 5b	22.7	65.8	9.2	1493.3	208.0
Cond. 6a	22.7	69.7	9.2	1581.7	208.0
Cond. 6b	22.7	69.7	9.2	1581.7	208.0
Cond. 7a	22.7	73.6	9.2	1670.2	208.0
Cond. 7b	22.7	73.6	9.2	1670.2	208.0
Cond. 8a	22.7	77.5	9.2	1758.6	208.0
Cond. 8b	22.7	77.5	9.2	1758.6	208.0
Evap. 1a	22.7	114.5	9.2	2596.3	208.0
Evap. 1b	22.7	114.5	9.2	2596.3	208.0
Evap. 2a	22.7	118.4	9.2	2684.7	208.0
Evap. 2b	22.7	118.4	9.2	2684.7	208.0
Evap. 3a	22.7	122.3	9.2	2773.2	208.0
Evap. 3b	22.7	122.3	9.2	2773.2	208.0
Evap. 4a	22.7	126.2	9.2	2861.6	208.0
Evap. 4b	22.7	126.2	9.2	2861.6	208.0
Evap. 5a	22.7	130.1	9.2	2950.1	208.0
Evap. 5b	22.7	130.1	9.2	2950.1	208.0
Evap. 6a	22.7	139.8	9.2	3170.1	208.0
Evap. 6b	22.7	139.8	9.2	3170.1	208.0
Evap. 7a	22.7	143.7	9.2	3258.5	208.0
Evap. 7b	22.7	143.7	9.2	3258.5	208.0
Evap. 8a	22.7	147.6	9.2	3347.0	208.0
Evap. 8b	22.7	147.6	9.2	3347.0	208.0
Evap. 9a	22.7	151.5	9.2	3435.4	208.0
Evap. 9b	22.7	151.5	9.2	3435.4	208.0
Evap. 10a	22.7	155.4	9.2	3523.9	208.0
Evap. 10b	22.7	155.4	9.2	3523.9	208.0
CW Pump 1	2.8	21.9	5.5	61.3	15.4
CW Pump 2	2.8	21.9	5.5	61.3	15.4
CW Pump 3	2.8	21.9	5.5	61.3	15.4
WW Pump 1	2.8	172.9	5.5	483.4	15.4
WW Pump 2	2.8	172.9	5.5	483.4	15.4
WW Pump 3	2.8	172.9	5.5	483.4	15.4
Total	1170.2			96595.0	8079.2
Center of gravity				**82.5**	**6.9**

The weight and position of the center of gravity of the platform derives from the combination of the previous. Table 7.6 summarizes the results obtained in this section.

Table 7.6: Weight and Center of Gravity of the Platform

	Weight (t)	LCG (m)	VCG (m)
Ship	11383	115.1	9.7
Equipment	1170.2	82.5	6.9
CWP	656	25.1	1.6
Platform	13209	107.7	9.0

7.3 Stability Curves

This section contains the results of the stability calculations performed entering the results from the previous calculations.

7.3.1 Upright Hydrostatics

Table 7.7: Upright Hydrostatics, from 12000 t to 22000 t

Draft Amidships (m)	**2.525**	**3.272**	**4.005**	**4.730**	**5.449**	**4.091**
Displacement t	13000	17250	21500	25750	30000	
Heel deg	0	0	0	0	0	
Draft at FP m	2.525	3.272	4.005	4.73	5.449	
Draft at AP m	2.525	3.272	4.005	4.73	5.449	
Draft at LCF m	2.525	3.272	4.005	4.73	5.449	
Trim (+ve by stern) m	0	0	0	0	0	
WL Length m	216.576	217.379	217.711	217.552	216.896	
Beam max extents on WL m	31.751	31.942	32.002	32.031	32.048	
Wetted Area m2̂	5952.218	6302.094	6635.791	6962.201	7281.466	
Waterpl. Area m2̂	5492.493	5610.012	5688.183	5748.598	5795.313	
Prismatic coeff. (Cp)	0.77	0.775	0.781	0.788	0.796	
Block coeff. (Cb)	0.732	0.742	0.753	0.763	0.774	
Max Sect. area coeff. (Cm)	0.952	0.958	0.964	0.969	0.972	
Waterpl. area coeff. (Cwp)	0.799	0.808	0.816	0.825	0.834	
LCB from fwd perp. (+ve fwd) m	-101.236	-101.116	-101.04	-100.99	-100.959	
LCF from fwd perp. (+ve fwd) m	-100.78	-100.732	-100.731	-100.741	-100.814	
KB m	1.319	1.709	2.09	2.466	2.838	
KG m	9.047	9.047	9.047	9.047	9.047	
BMt m	30.225	23.83	19.678	16.815	14.712	
BML m	1194.827	938.493	774.881	661.071	576.406	
GMt m	22.497	16.492	12.722	10.235	8.503	
GML m	1187.099	931.154	767.924	654.491	570.197	
KMt m	31.544	25.539	21.769	19.281	17.55	
KML m	1196.146	940.201	776.971	663.538	579.244	
Immersion (TPc) tonne/cm	56.298	57.503	58.304	58.923	59.402	
MTc tonne.m	699.094	727.642	747.932	763.461	774.909	

7.3.2 Large Angle Stability

The large volumes of the water sumps have motivated the calculation of the three different load cases in Table 7.8. The percentage of fullness of the tanks accounts for three situations:

- Loadcase 1: empty, not operating plant.
- Loadcase 2: minimum pump submergence, given by the manufacturer, attached in Appendix C.1.
- Loadcase 3: approximate maximum level of tank, from Appendix C.1.

Table 7.8: Loadcases Used for the Calculation of GZ Curves

Item	**Quantity**	M_{unit} (t)	M_{total} (t)	V_{unit} (m^3)	V_{total} (m^3)	**lcg** (m)	**ycg** (m)	**vcg** (m)
LOADCASE 1								
Lightship	1	13209.6	13209.6			113.04	0	9.04
CW Sump	0%	544.567	0	528.706	0	196.126	0	1.6
DW Sump	0%	1294.847	0	1263.266	0	121.076	0	1.6
WW Sump	0%	1210.286	0	1180.767	0	45.176	0	1.6
Total Loadcase			13209.6	2972.738	0	113.04	0	9.04
LOADCASE 2								
Lightship	1	13209.6	13209.6			113.04	0	9.04
CW Sump	10.90%	544.567	59.358	528.706	57.629	196.126	0	1.901
DW Sump	21.40%	1294.847	277.098	1263.266	270.339	121.076	0	2.191
WW Sump	21.40%	1210.286	259.001	1180.767	252.684	45.176	0	2.191
Total Loadcase			13805.056	2972.738	580.652	112.281	0	8.743
LOADCASE 3								
Lightship	1	13209.6	13209.6			113.04	0	9.04
CW Sump	80%	544.567	435.653	528.706	422.964	196.126	0	3.808
DW Sump	80%	1294.847	1035.878	1263.266	1010.613	121.076	0	3.808
WW Sump	80%	1210.286	968.229	1180.767	944.614	45.176	0	3.808
Total Loadcase			15649.36	2972.738	2378.191	111.682	0	8.224

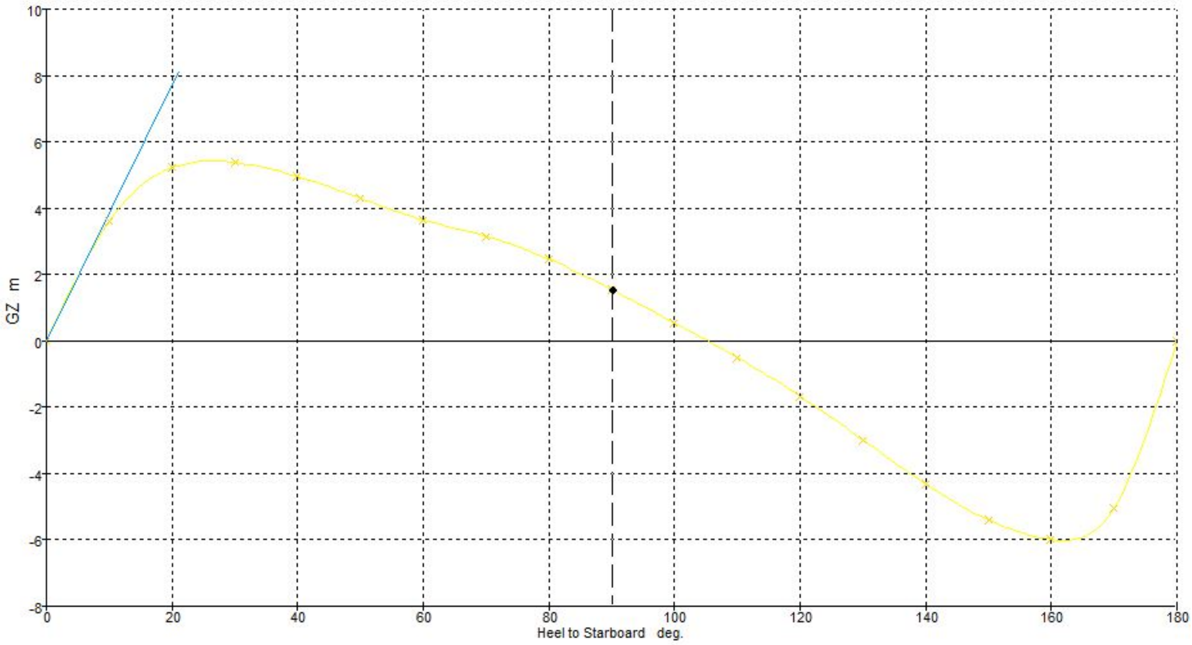


Figure 7.3: GZ Curve of Loadcase 1

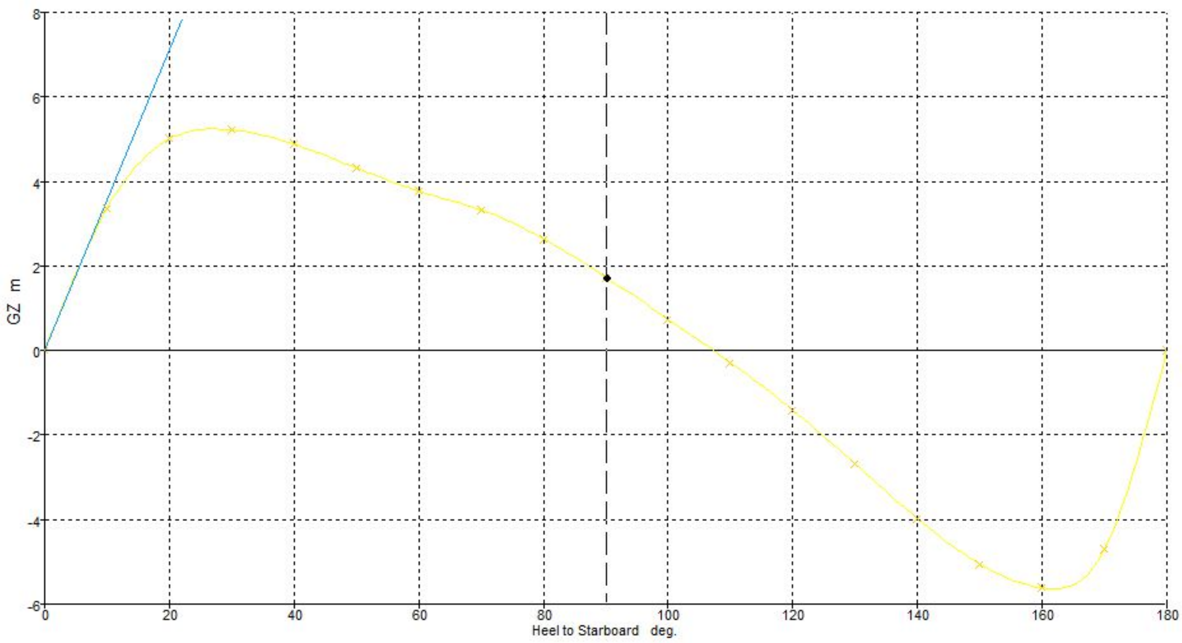


Figure 7.4: GZ Curve of Loadcase 2

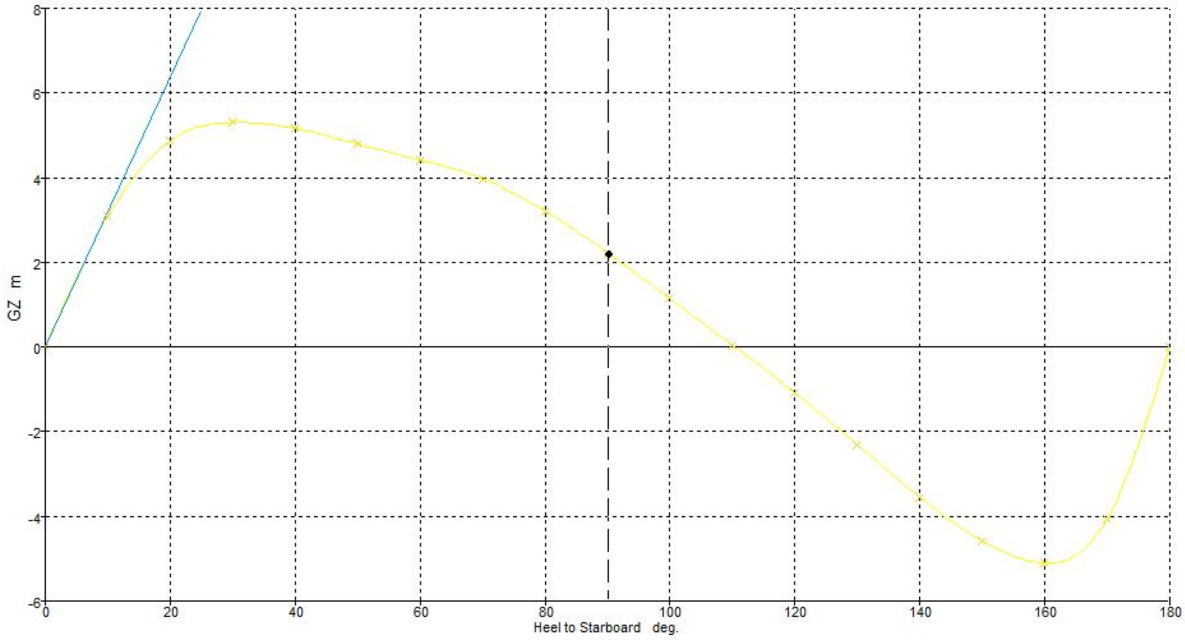


Figure 7.5: GZ Curve of Loadcase 3

7.4 Stability Criteria

At the present time, there are no specific regulations concerning installations like the concept under study. On the other hand, as aforementioned, the mission and inherent risks of FPSOs, despite being the most similar structures, are fundamentally different from those of the ship-shaped OTEC. Bearing this in mind, the stability of the converted bulker is here evaluated in accordance to the IMO Resolution A.749(18), Code of Intact Stability for all Ship Types (1993) [40]. The purpose of the provisions enclosed in this Code is "to recommend stability criteria and measures for ensuring the safe operation of all ships to minimize the risks to such ships, to the personnel and to environment".

The following criteria, taken from the Code, are applicable to a bulk carrier:

1. The area under the righting lever curve (GZ curve) should not be less than 0.055 metre-radian up to $\theta = 30$ ° angle of heel and not less than 0.09 metre-radian up to $\theta = 40$ ° or the angle of flooding θ_f, if this angle is less than 40°. Additionally, the area under the righting lever curve (GZ curve) between the angles of heel of 30° and 40° or between 30° and θ_f, if this angle is less than 40°, should not be less than 0.03 metre-radian.
2. The righting lever GZ should be at least 0.20 m at an angle of heel equal to or greater than 30°.
3. The maximum righting arm should occur at an angle of heel preferably exceeding 30° but not less than 25°.
4. The initial metacentric height GM_o should not be less than 0.15 m.

The requirements enclosed in the Code can be extended once the operability limits of the equipment and the connection of the water pipes are defined. In particular, the maximum heel angle should be determined in relation to the connection to the CWP. The gimbal connection proposed in [73] can withstand angles of ± 20°. These angles can be caused by environmental forces directly on the pipes, or indirect loads arising in the interface between pipe and hull. On the other hand, process equipment, acceptable ranges of motions and accelerations have to be defined and translated into platform motions, and these, in turn, into maximum environmental conditions. For instance, pumps and turbines are the most readily applicable to floating structures [28]. Their operability at sea has been proven up to 0.15G [6].

The compliance of the hydrostatics and GZ curves, performed in Maxsurf, has to be verified with regards to the Code on Intact Stability. Table 7.9 shows the results of the analysis with respect to the selected stability criteria.

Table 7.9: Verification of Compliance with the Code on Intact Stability

Criteria	>= Value	Units	Actual	Status	Margin (%)
LOADCASE 1					
Area 0 to 30	0.055	m.rad	2.0696	Pass	+3662.82
Area 0 to 40	0.09	m.rad	2.9750	Pass	+3205.61
Area 30 to 40	0.03	m.rad	0.9055	Pass	+2918.16
Max GZ at 30 or greater	0.2	m	5.369	Pass	+2584.50
angle of max. GZ	25	deg			
Angle of maximum GZ	25	deg	26.4	Pass	+5.46
Initial GMt	0.15	m	21.979	Pass	+14552.67
LOADCASE 2					
Area 0 to 30	0.055	m.rad	1.9705	Pass	+3482.72
Area 0 to 40	0.09	m.rad	2.8560	Pass	+3073.36
Area 30 to 40	0.03	m.rad	0.8855	Pass	+2851.61
Max GZ at 30 or greater	0.2	m	5.215	Pass	+2507.50
angle of max. GZ	30.0	deg			
Angle of maximum GZ	25	deg	27.3	Pass	9.09
Initial GMt	0.15	m	20.361	Pass	+13474.00
LOADCASE 3					
Area 0 to 30	0.055	m.rad	1.8977	Pass	+3350.40
Area 0 to 40	0.09	m.rad	2.8152	Pass	+3028.00
Area 30 to 40	0.03	m.rad	0.9174	Pass	+2958.10
Max GZ at 30 or greater	0.2	m	5.302	Pass	+2551.00
angle of max. GZ	30.9	deg			
Angle of maximum GZ	25	deg	30.9	Pass	+23.64
Initial GMt	0.15	m	18.307	Pass	+12104.67

Overall, a number of conclusions can be drawn from the preliminary stability study:

- The draft of the vessel, without adding ballast, i.e. in lightship condition is approximately 2.5 m. At this draft, the water level in the sump tanks is not sufficient to enable a proper functioning of the water pumps. This issue requires by itself the installation of ballast.
- The values of the GM_t obtained in Upright Hydrostatics indicate that the vessel is too stable. The large GM_t implies that the ship is "stiff", i.e. it will experience very short roll periods, putting the platform in danger.

7.5 Addition of Ballast Weight

As aforementioned, to this day there no regulations specific to ship-shaped OTEC plants. Moreover, an accurate solution would require the definition of the allowable limits of the process equipment and the water pipes. Nevertheless, the stability of the platform can be further improved based on the following recommendations:

- The process equipment should not be placed above sea level, to avoid an extra hydrostatic head which would increase the power consumption of the water pumps [51]. Therefore, the waterline should be maintained at a height similar to that of the equipment deck, i.e. 5.52 m.
- To prevent undesired resonance, the roll period of the installation should not coincide with the mean and maximum wave periods. In Section 2.2.3 the periods were identified, these being 6 s and 15 s, respectively. The natural period of the platform has to be within that range. Furthermore, the periods between 10 s and 15 s are related to significant wave energy density in a fully developed sea. In conclusion, the goal will be to achieve a roll period of 10 s, thus escaping resonance and high energy waves.

  IMO [43] recommends Equation 7.8 below for the estimation of the roll period of ships:

$$T = \frac{2CB}{\sqrt{GM}} \tag{7.8}$$

  Where:

$$C = 0.373 + 0.023\frac{B}{T} - 0.043\frac{LWL}{100} \tag{7.9}$$

  From Equation 7.8 an optimal metacentric height of 4.7 m is obtained.

As an initial optimization phase, these stability issues will be resolved by adding ballast to the platform. To this end, the original void spaces inside the hull, i.e. top side and hopper tanks and the space confined in the double bottom (Figure 7.6) will be used as well as the aft and fore peaks.

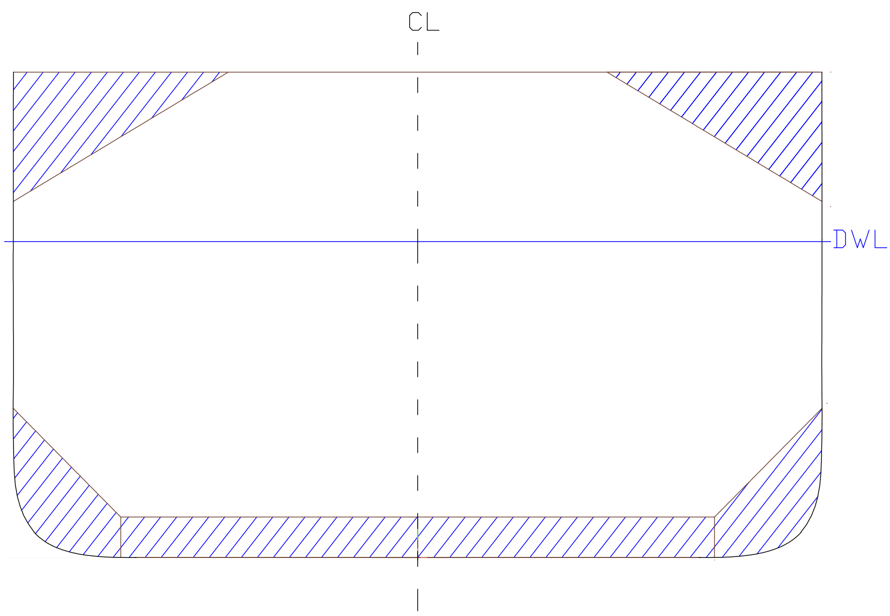


Figure 7.6: Simplified Midship Section, based on Appendix ?? (Ballast tanks in blue)

The original ballast capacity, 32480 m^3, as indicated in Appendix E, considers the possibility of flooding Cargo Hold No. 4. In the present case, however, the layout of components precludes this solution. Table 7.10 shows the actual ballast volume available inside the hull, as found from the Maxsurf Model in Figure 7.7

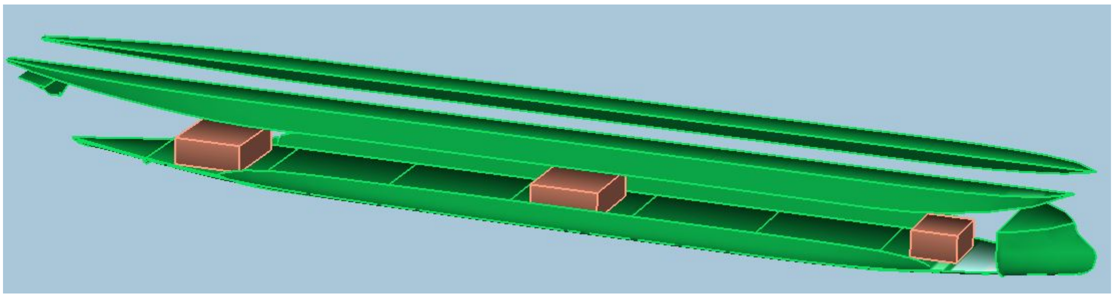


Figure 7.7: Maxsurf Model of Ballast Tanks

Table 7.10: Original Ballast Tanks of PROTEFS Bulker

Item	**Quantity**	M_{unit} (t)	M_{total} (t)	V_{unit} (m^3)	V_{total} (m^3)	**lcg** (m)	**ycg** (m)	**vcg** (m)
Lightship	1	13209.6	13209.6			113.04	0	9.04
Hopper_1	100%	1528.775	1528.775	1491.487	1491.487	125.501	13.967	2.318
Hopper_2	100%	1528.775	1528.775	1491.487	1491.487	125.501	-13.967	2.318
Top Side_1	100%	3942.886	3942.886	3846.718	3846.718	117.646	12.952	17.579
Top Side_2	100%	3942.886	3942.886	3846.718	3846.718	117.646	-12.952	17.579
DB_1	100%	930.893	930.893	908.189	908.189	69.605	0	0.795
DB_2	100%	744.19	744.19	726.039	726.039	45.445	0	0.829
DB_3	100%	770.006	770.006	751.225	751.225	196.286	0	0.838
DB_4	100%	931.408	931.408	908.691	908.691	95.2	0	0.795
DB_5	100%	931.414	931.414	908.696	908.696	120.8	0	0.795
DB_6	100%	931.413	931.413	908.696	908.696	146.4	0	0.795
DB_7	100%	930.687	930.687	907.987	907.987	171.993	0	0.796
DB_CCMM	100%	157.706	157.706	153.86	153.86	25.208	0	0.912
AP	100%	134.397	134.397	131.119	131.119	6.295	0	13.091
FP	100%	1703.931	1703.931	1662.372	1662.372	215.552	0	5.976
Total			34758.729	21616.025	21021.478	120.223	0	8.385

An Equilibrium analysis is performed with all ballast tanks filled completely and the water sumps at 80 % of their capacity, as in *Loadcase 3*. The results are shown in Table 7.11.

Table 7.11: Equilibrium Analysis with Additional Ballast

Draft Amidships m	6.171
Displacement t	34113
Heel deg	0
Draft at FP m	5.785
Draft at AP m	6.557
Draft at LCF m	6.139
Trim (+ve by stern) m	0.772
WL Length m	215.925
Beam max extents on WL m	32.047
Wetted Area m2̂	7591.425
Waterpl. Area m2̂	5838.27
Prismatic coeff. (Cp)	0.799
Block coeff. (Cb)	0.78
Max Sect. area coeff. (Cm)	0.975
Waterpl. area coeff. (Cwp)	0.844
LCB from fwd perp. (+ve fwd) m	-102.747
LCF from fwd perp. (+ve fwd) m	-101.162
KB m	3.197
KG solid m	8.522
BMt m	13.152
BML m	514.585
GMt corrected m	7.464
GML m	508.898
KMt m	16.349
KML m	517.779
Immersion (TPc) tonne/cm	59.842
MTc tonne.m	786.412
RM at 1deg = GMt.Disp.sin(1) tonne.m	4443.955
Max deck inclination deg	0.2003
Trim angle (+ve by stern) deg	0.2003

From Table 7.11 it becomes clear that, while the additional ballast decreases the draft sufficiently, the metacentric height remains higher than the desired value. This Loadcase complies with the requirements established in the Code on Intact Stability, as demonstrated in Table 7.12.

Table 7.12: Verification of Compliance with the Code on Intact Stability after the Addition of Ballast Water

Criteria	>= Value	Units	Actual	Status	Margin (%)
Area 0 to 30	0.055	m.rad	1.0579	Pass	+1823.47
Area 0 to 40	0.09	m.rad	1.8162	Pass	+1918.00
Area 30 to 40	0.03	m.rad	0.7583	Pass	+2427.53
Max GZ at 30 or greater	0.2	m	5.016	Pass	+2408.00
angle of max. GZ	50.9	deg			
Angle of maximum GZ	25	deg	50.9	Pass	+103.64
Initial GMt	0.15	m	7.441	Pass	+4860.67

A GM of 7.441 m is associated with a roll period of 7.96 s. This result, already away from both mean and peak period of the waves, can be further improved to achieve the preferred roll period of 10 s. Several solutions are here proposed:

- To add fixed ballast, e.g. concrete, to the top side tanks, thereby reducing GM and increasing the roll period.
- To flood partially the space below the equipment deck.
- To fill partially a number of ballast tanks, following the opposite principle of antiroll tanks to reduce GM thanks to the free surface effect.
- To install bilge keels, purposely designed to alter the roll period, in this case by increasing it.

These options need to be studied carefully, since their application could endanger the safety of the platform.

Overall, the deadweight of the bulker is, by a large margin, enough to accomodate the main power plant systems, even after adding a margin to account for auxiliar components and process ducting. The stability, on the other hand, presents some complications, derived from the excessive metacentric height of the platform as initially designed. These, however, can be solved in many possible was and, thus, it is not considered a major constraint for the concept. In the Detail Design, a more comprehensive analysis should be performed.

8

Conclusions and Recommendations

The purpose of the research is to provide insight into the feasibility and actual benefits of converting a second-hand vessel into an OTEC plant. To this end, a 10 MW installation to supply electricity for the island of Curaçao has been used as a practical case. The time frame and resources available have limited the study to a conceptual design of the ship-shaped structure. This chapter aims to answer the main research question, as formulated in Chapter 1:

"What is the feasibility of converting a second-hand vessel into a 10 MW ship-shaped OTEC plant?"

8.1 Conclusions

The design of a ship-shaped OTEC plant, as an innovative and complex project, demanded for a clear methodology. Systems Engineering (SE) has provided a firm framework, based on which the operability of the installation was defined and verified along the engineering process.

This report has focused on what constitutes one of the major challenges of the research: the fact that the equipment and structure are not purposely designed for each other. By identifying the prime drivers of the design, it was possible to select bulker carriers as the optimal ship type for conversion.

The selection of a spread mooring system, possible in a moderate environment around Curaçao, was mainly motivated by its low price and simplicity. If a posterior hydromechanic analysis proves that this is, nonetheless, unfeasible a Single Point Mooring (SPM) should be considered instead. This needs to be verified for a specific location.

The final general arrangement has demonstrated that the integration of the power plant equipment and the floating structure can be performed, while minimizing the conversion effort incurred. As a result, the top level requirements established in Chapter 1 have been fulfilled, with the exception of the power delivery and accessibility, not covered in this work. This can be considered an indicator of the feasibility of the concept.

The main benefits of the ship-shaped structure are:

- The expected lower costs of the vessel, when compared to a new construction. This includes not only the cost of the structure itself but also of the engineering process.
- The operational and constructional experience in the conversion of second-hand ships into offshore installations.
- Off-the-shelf offshore systems, namely the chosen spread mooring, can be directly adapted to the platform.
- Bulkers are originally built to withstand environmental loads while carrying heavy cargoes. This robust structure does not need to be redesigned, but instead reviewed with respect to different

criteria.

On the other hand, the hydromechanic performance of the ship-shaped platform has not been analysed in this research. The integrity of the connection between hull and pipes remains a critical issue for offshore OTEC plants, even more for low draft platforms. An exhaustive analysis of the coupling of the water pipes and vessel, including the effect of the mooring system is necessary to demonstrate the feasibility of the concept.

In addition, the dimensions of the selected bulk carrier appears to be, in principle, excessive for a 10 MW. Even though in the current market situation Panamax ships seem to be an advantageous purchase, this size could lead to discarding this option against other floating structures.

In summary, second-hand vessel conversions are not new to the offshore industry. Bearing in mind the different requirements of OTEC plants, it can be expected that the challenges and drawbacks of the studied platform can be overcome. The installation and operation of a ship-shaped OTEC prototype will enable the collection of data and know-how of the actual benefits and challenges of these type of plants.

8.2 Recommendations

First and foremost, the general arrangement developed has considered only the largest components of the process cycle. However, the secondary equipment, e.g. working fluid pumps, transformers, etc. contribute to the intricacy of the layout, as well as to weight and stability of the platform. The same applies to the process ducting, not included in the weight and stability computations.

On another note, in respect of the influence of design drivers on the layout, the relevance of the condensers and evaporators as the biggest space consumers stands out. The use of PHE was suggested by Bluerise due to their optimal heat transfer performance and easy maintenance. Despite these advantages, the great volume occuppied by them suggests that other types of heat exchangers should be considered for ship-shaped plants.

In line with the above, given the large volume taken by the process cycle components, an increase in the requested electrical power, i.e. a large plant capacity might be more suitable for this type of platform. This expansion could also enable the installation of a second equipment deck, or implement multi-pipe solutions. As a consequence, a more space optimized general arrangement could be attained.

Once all the equipment has been incorporated in the design, it is recommended to perform structural and hydromechanic analysis, as aforementioned, to confirm the technical feasibility of the concept. These should take into account, respectively, the loads induced by the power plant systems on the equipment deck, and the motions and loads caused by the interaction of mooring system, hull and water pipes.

It is evident that the economic investment would be a decision driver for the construction of the pilot plant. An economic study of the project should include: the price of the second-hand bulker, actual conversion costs incurred, OTEC systems installed, etc.

Lastly, based on all the information gathered, the concept should then be compared to other possible floating structures.

A

Extreme Wind and Wave Climate in the Caribbean Sea (1987 - 2011)

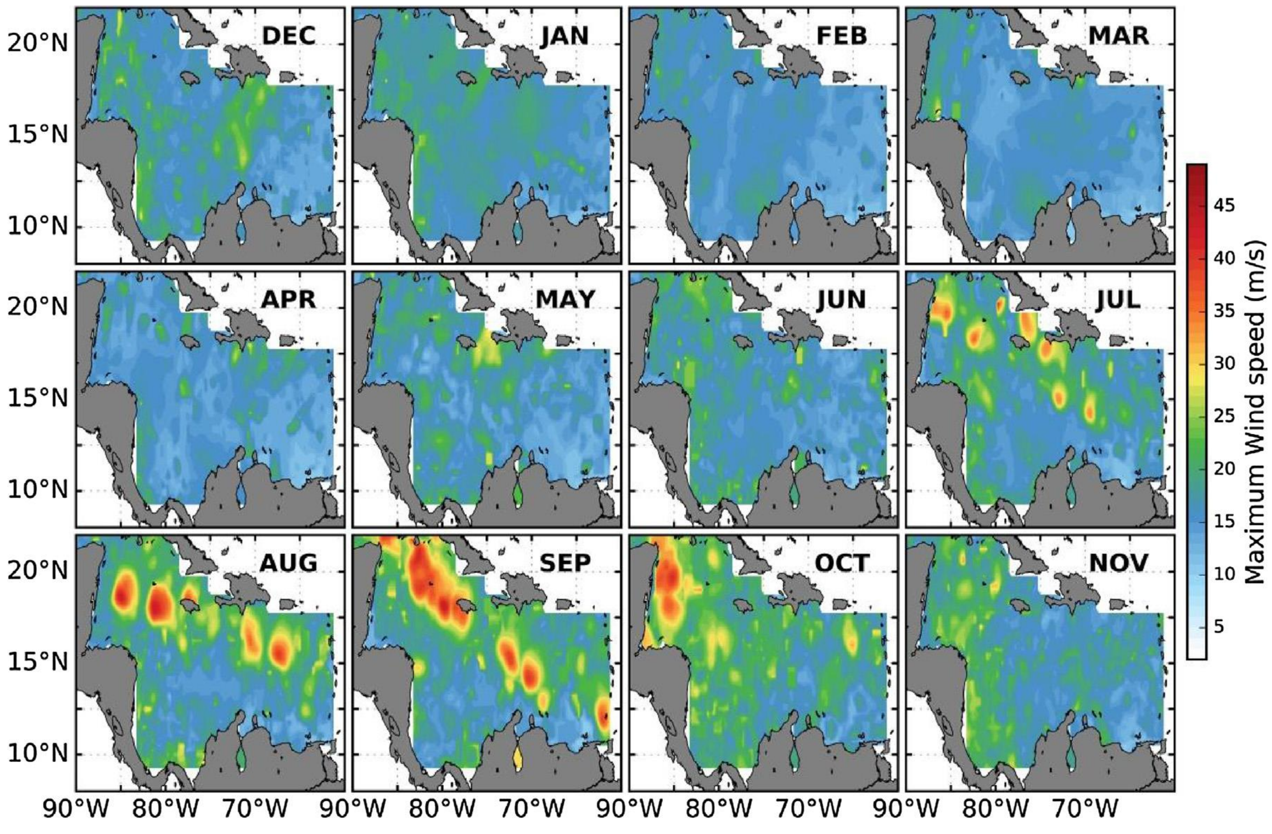


Figure A.1: Extreme Wind Speed $\mathrm{m\,s^{-1}}$ given Each Month in the Caribbean Sea from 1987 to 2011 [29]

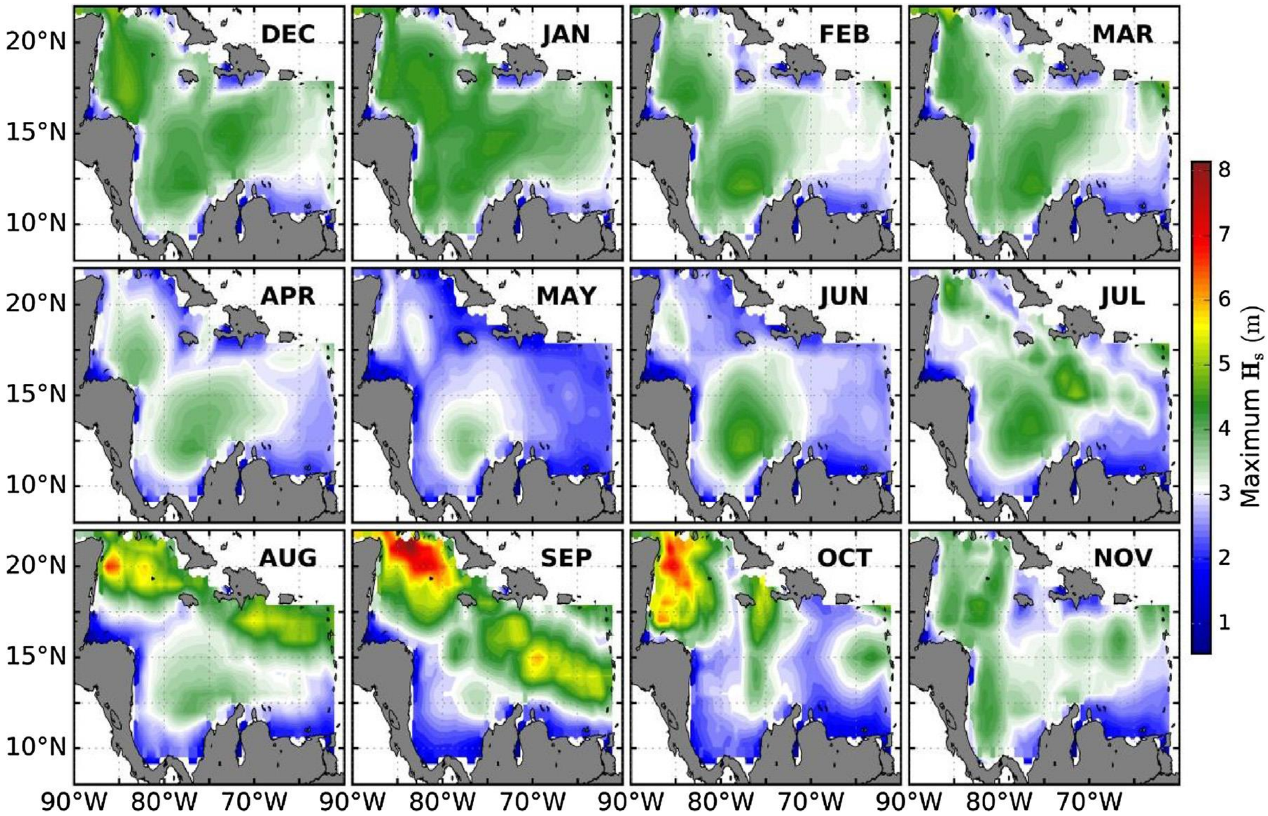


Figure A.2: Extreme Significant Wave height (H_s) given Each Month in the Caribbean Sea from 1979 to 2014 [29]

B

Equipment Calculations

This appendix provides a more comprehensive explanation of the calculations made to define the specifications of the main components of the power plant.

B.1 Head Losses inside Water Pipes

The water pumps have to overcome the head losses of the seawater throughout the cycle. These can be broken down in the components outlined below.

- Inlet and outlet loss
  The dissipative losses at the inlet (cold and warm seawater) and outlet (discharge) of the water pipes are obtained from Equation B.1. The values of the loss coefficients are taken as K=0.2 for the inlet and K=1 for the outlet, which correspond to a beveled ring shape and a discharge to a reservoir, respectively.

$$h_{inlet/outlet} = \frac{KU_f^2}{2g} \tag{B.1}$$

  Where:

$$\begin{aligned} h_{\mathrm{in\ /\ out}} &= \text{head loss at pipe inlet/outlet (m)} \\ K &= \text{inlet loss coefficient} \\ U_{\mathrm{f}} &= \text{velocity of internal seawater } (\mathrm{m\,s^{-1}}) \\ g &= \text{gravitational acceleration } (\mathrm{kg\,m\,s^{-2}}) \end{aligned}$$

Table B.1: Inlet and Outlet Loss

	CWP	**WWP**	**DWP**
K	0.2		1
U_f ($\mathrm{m\,s^{-1}}$)	2	1.5	1.5
$h_{in/out}$	0.0408	0.0229	0.1147

- Friction loss
  The friction loss of seawater inside the pipes is calculated using the Darcy-Weisbach formula in Equation B.2.

$$h_{friction} = f\left(\frac{L}{D}\right)\left(\frac{U_f^2}{2g}\right) \tag{B.2}$$

Where:

h_{friction} = friction head loss inside water pipes (m)
f = Darcy factor
L = length of pipe (m)
D = pipe diameter (m)

The Darcy friction factor is, in turn, found from the Haaland formula in Equation B.3. The wall roughness of the pipe is assumed to be 0.0004, based on HDPE coating [6].

$$\frac{1}{\sqrt{f}} = -1.8\log\left(\frac{\epsilon/D}{3.7}^{1.11} + \frac{6.9}{Re}\right) \tag{B.3}$$

Where:

ϵ/D = relative roughness of the pipe
Re = Reynolds number

For flow in a pipe or tube, the Reynolds number is generally defined as:

$$Re = \frac{\rho U_f D}{\mu} \tag{B.4}$$

ρ = density of seawater, according to its temperature ($\mathrm{kg\,m^{-3}}$)
μ = dynamic viscosity of seawater, according to its temperature ($\mathrm{N\,s\,m^{-2}}$) [79]

Table B.2

Table B.2: Friction Loss of Seawater in Ducting

	CWP (5 °C)	**WWP (27 °C)**	**DWP (20 °C)**
L (m)	1000	20	90
μ ($\mathrm{N\,s\,m^{-2}}$)	0.00092	0.00167	0.00108
U_f ($\mathrm{m\,s^{-1}}$)	2	1.5	1.5
ρ ($\mathrm{kg\,m^{-3}}$)	1030	1025	1025
Re	3.22×10^6	8.01×10^6	8.06×10^6
ϵ/D	1.53×10^{-4}	8.34×10^{-5}	7.06×10^{-5}
f	1.34×10^{-2}	1.18×10^{-2}	1.15×10^{-2}
$h_{friction}$ (m)	1.0443	0.0057	0.0210

– Hydrostatic head
The difference in the density between the cold and warm water contribute to the pumping power consumption. Considering an average vaue for the density of water outside the pipe, the net hydrostatic head is assumed to be 2.3 m.

- Pressure drop
  High flow, and thus pressure gradient, through a heat exchanger will promote turbulence and good thermal contact, but it increases the pumping power required. In this case, no detailed information from the manufacturer is available. Hence, for the purpose of this work, a pressure drop of 3.5 m in the heat exchangers is considered [6].

B.2 Heat Exchangers

The LMTD, in Equation B.5, is a logarithmic average of the temperature difference between the warm and cold side of a heat exchanger. Its calculation is neccesary to obtain the heat exchanged per unit.

$$\Delta T_{LMTD} = \frac{\Delta T_0 - \Delta T_1}{\ln \frac{\Delta T_0}{\Delta T_1}} \tag{B.5}$$

In counter flow heat exchangers, ΔT_0 and ΔT_1 can be found from Equation B.6.

$$\begin{aligned} \Delta T_0 &= T_{w,in} - T_{wf,out} \\ \Delta T_1 &= T_{w,out} - T_{wf,in} \end{aligned} \tag{B.6}$$

Where:

$T_{\text{w,in}}$ = temperature of water at entrance of HX (°C)
$T_{\text{w,out}}$ = temperature of water at exit of HX (°C)
$T_{\text{wf,in}}$ = temperature of working fluid at entrance of HX (°C)
$T_{\text{wf,out}}$ = temperature of working fluid at exit of HX (°C)

In the present case, these temperatures, indicated in Figure B.1, are summarized in Table B.3.

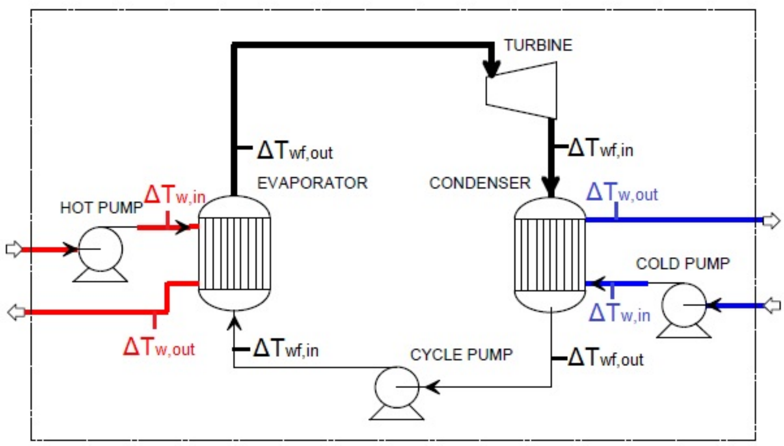


Figure B.1: Simplified OTEC Cycle Diagram, from [50] (modified)

Table B.3: Heat Exchangers

	Evaporators	**Condensers**
$T_{w,in}$ (°C)	27	5
$T_{w,out}$ (°C)	23.9	12.4
$T_{wf,in}$ (°C)	14.3	14.4
$T_{wf,out}$ (°C)	26.1	10.1
ΔT_0 (°C)	0.9	5.1
ΔT_1 (°C)	9.7	2.0
ΔT_{LMTD} (°C)	3.7	3.3

C

Equipment Data

C.1 Water Pumps

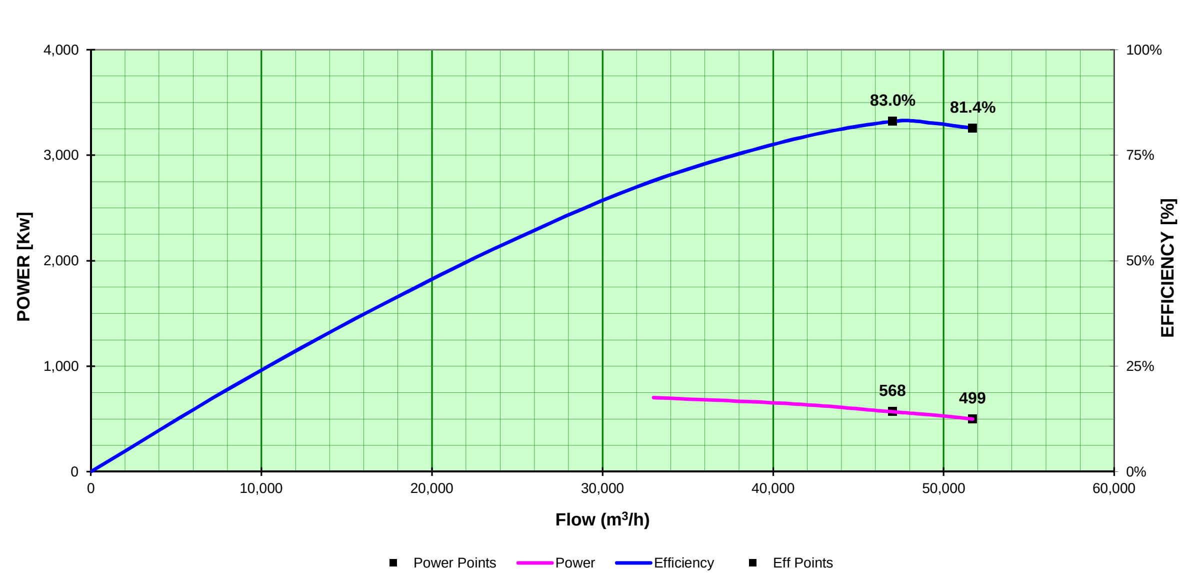
Customer Service OA EMA

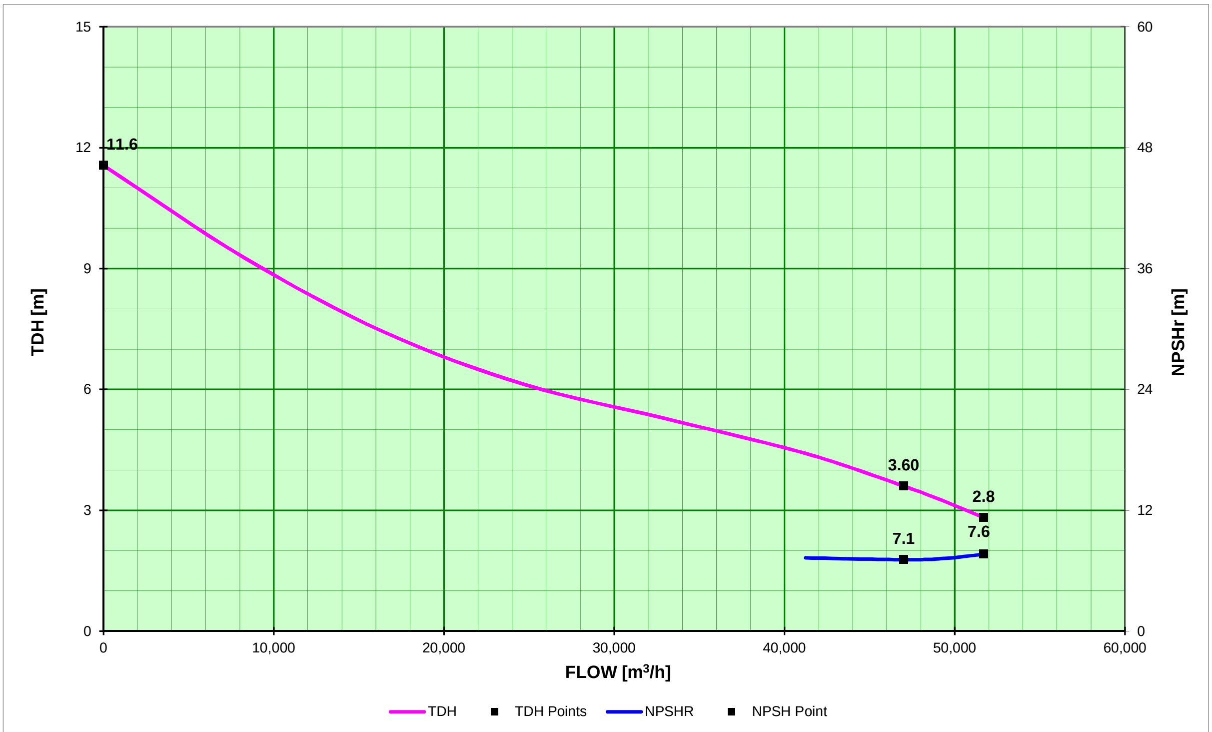
PERFORMANCE CURVES

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**68AFV**	Quantity:	**3**

DESIGN POINT

Discharge flow:	**47000 m³/h**	Design TDH:	**3.6 m**		
RPM:	**196.7**	Curve number:	**68AFV**	Specific Gravity:	**1022 kg/m³**





Curves are approximate. Pump is guaranteed for one set of conditions. Capacity and head guarantees are based on shop test. Efficiency is being calculated at design temperature.

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Customer Service OA EMA

TORQUE vs SPEED CURVE

OPEN VALVE

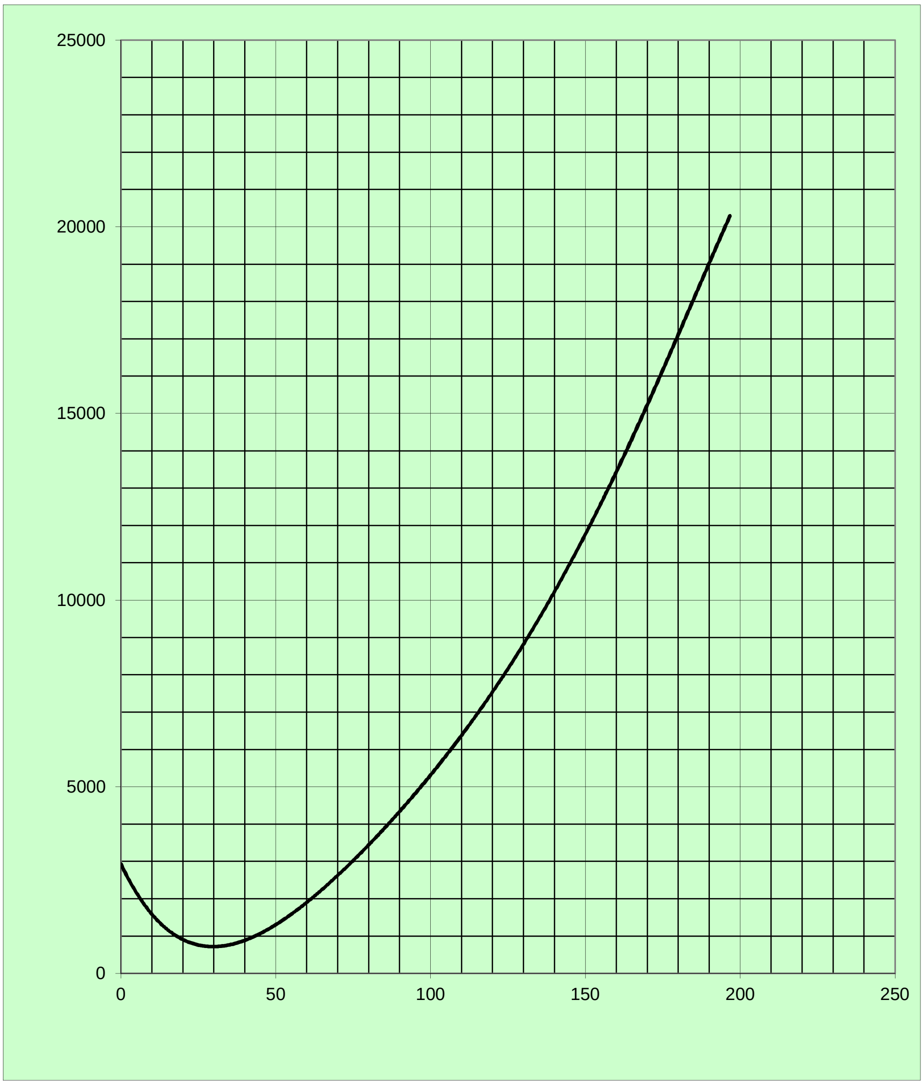
Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**68AFV**	Quantity:	**3**

DESIGN POINT

Discharge flow:	**47000 m³/h**	Design TDH:	**3.6 m**	
RPM:	**197**	Curve number:	**68AFV**	Specific Gravity: **1022 kg/m³**

PUMP 100% SPEED (rpm)	**197**		
PUMP 100% POWER (BHP)	**762**	**568** kW	
PUMP 100% TORQUE (LbFt)	**20326**	**27530** Nm	

UNIT ROTOR INERTIA WR2 (Kg.m2)
PUMP ROTOR **#DIV/0!**



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Customer Service OA EMA

PERFORMANCE DATA SHEET

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**68AFV**	Quantity:	**3**

PUMP

Service		Main sea water intake pump
Liquid		Sea Water
Temperature	ºC	18
Specific gravity		1.022
Number of pumps		3
Design flow	m3/h	47000
TDH	m	3.6
Pump Speed	rpm	197
Efficiency	%	83.0%
Power	kW	568
Min. Submergency at R.O.	m	3.99
NPSHr D.P. (at impeller eye)	m	7.1
Pump model		68AFV
Min. Continous Flow	m3/h	39950
Head @ shut-off	m	11.6
Performance curve		68AFV
Setting	mm	4.587
Pump construcion		Non-Pull Out

MOTOR

Power	kW	625
Speed	rpm	197
Supplied by		FLOWSERVE

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Customer Service OA EMA

MATERIALS DATA SHEET

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**68AFV**	Quantity:	**3**

OPTIONAL MATERIAL: SUPERDUPLEX

Casing	ASTM A240 Ty 2507
Impeller	ASTM A890 Gr 5A
Shaft Sleeves	ASTM A479 UNS S32750
Shaft - Pump Element	ASTM A479 UNS S32750
Shaft - Column and upper	ASTM A479 UNS S32750
Suction Bell	ASTM A240 Ty 2507
Bearings	Rubber / UNS S32750
Shaft Coupling - Between Shafts	ASTM A479 UNS S32750
Shroud	ASTM A240 Ty 2507
Column	ASTM A240 Ty 2507
Discharge head	ASTM A240 Ty 2507
Motor Suport	Carbon Steel
Main Coupling Pum - Electric Motor	Flexible
Shaft Sealing	Soft Packing
Mounting Plate	Structural Steel
Wet Bolts	UNS S32550
Thrust Bearing Location	In pump

Revision: **0** **Date:** **01/08/2017**

FLOWSERVE

Customer Service OA EMA

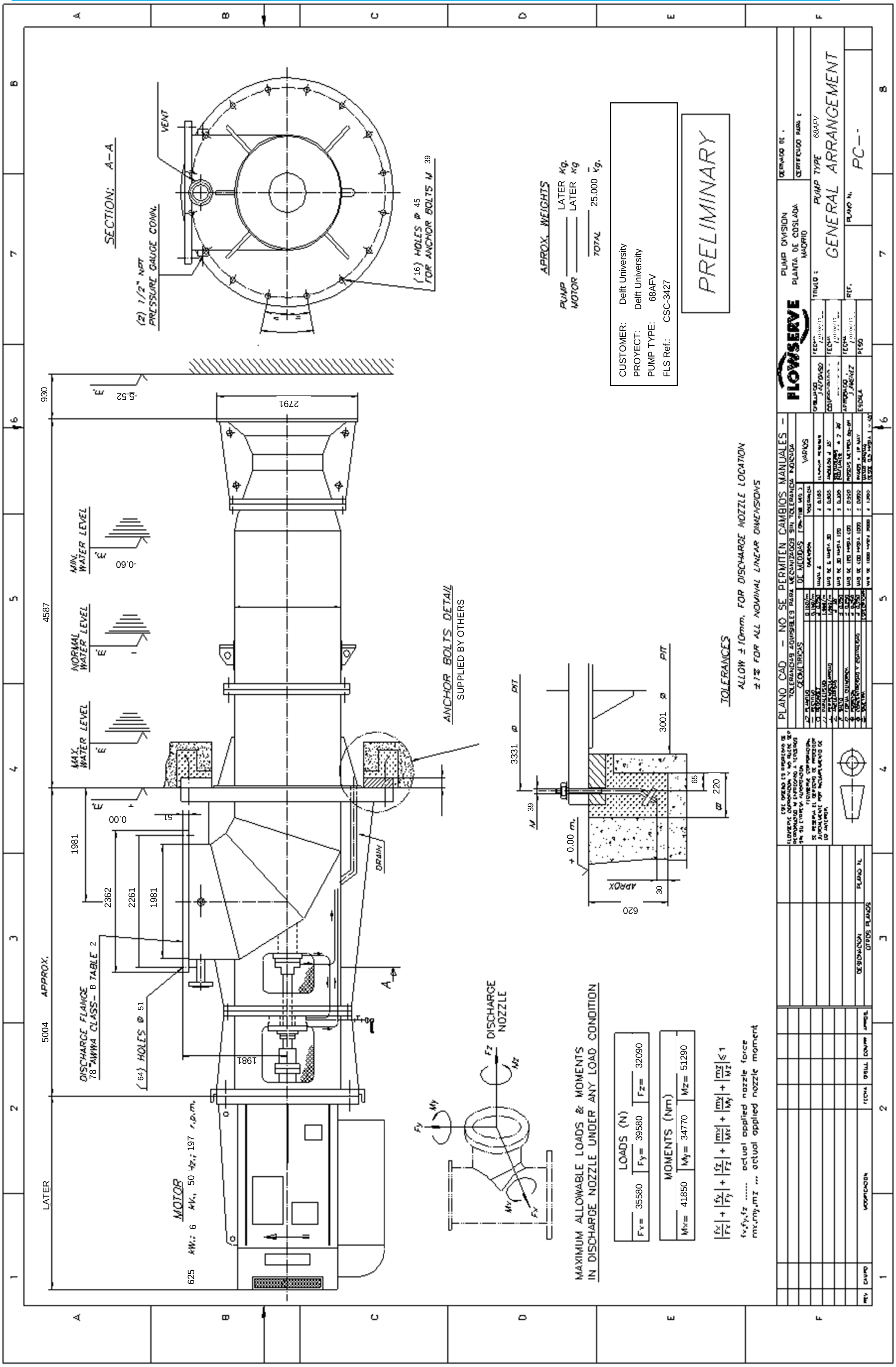
MATERIALS DATA SHEET

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**68AFV**	Quantity:	**3**

BASE MATERIAL: DUPLEX

Casing	ASTM A240 UNS S31803
Impeller	ASTM A890 Gr 1B
Shaft Sleeves	ASTM A276 UNS S31803
Shaft - Pump Element	ASTM A276 UNS S31803
Shaft - Column and upper	ASTM A276 UNS S31803
Suction Bell	ASTM A240 UNS S31803
Bearings	Rubber / UNS S31803
Shaft Coupling - Between Shafts	ASTM A276 UNS S31803
Shroud	ASTM A240 UNS S31803
Outer column	ASTM A240 UNS S31803
Inner Column	ASTM A240 UNS S31803
Discharge head	ASTM A240 UNS S31803
Motor Suport	Carbon Steel
Main Coupling Pum - Electric Motor	Flexible
Shaft Sealing	Soft Packing
Mounting Plate	Structural Steel
Wet Bolts	S31803
Thrust Bearing Location	In pump

Revision: 0 **Date:** 01/08/2017


SECTION: A–A
VENT
(2) 1/2" NPT PRESSURE GAUGE CONN.
(16) HOLES Ø 45 FOR ANCHOR BOLTS M 39
APROX. WEIGHTS
PUMP LATER Kg.
MOTOR LATER Kg.
TOTAL 25.000 Kg.
CUSTOMER: Delft University
PROYECT: Delft University
PUMP TYPE: 68AFV
FLS Ref.: CSC-3427
PRELIMINARY
MAX. WATER LEVEL
NORMAL WATER LEVEL
MIN. WATER LEVEL
ANCHOR BOLTS DETAIL
SUPPLIED BY OTHERS
DISCHARGE FLANGE 78" AWWA CLASS- B TABLE 2
(64) HOLES Ø 51
DRAIN
MOTOR
KW.: 6 KV., 50 Hz.; 197 r.p.m.
DISCHARGE NOZZLE
MAXIMUM ALLOWABLE LOADS & MOMENTS IN DISCHARGE NOZZLE UNDER ANY LOAD CONDITION
LOADS (N)
Fx= 35580 Fy= 39580 Fz= 32090
MOMENTS (Nm)
Mx= 41850 My= 34770 Mz= 51290
fx,fy,fz ...... actual applied nozzle force
mx,my,mz ... actual applied nozzle moment
TOLERANCES
ALLOW ± 10mm. FOR DISCHARGE NOZZLE LOCATION
±1% FOR ALL NOMINAL LINEAR DIMENSIONS
PLANO CAD – NO SE PERMITEN CAMBIOS MANUALES –
PUMP DIVISION
PLANTA DE COSLADA
MADRID
PUMP TYPE 68AFV
GENERAL ARRANGEMENT
PC--

FLOWSERVE

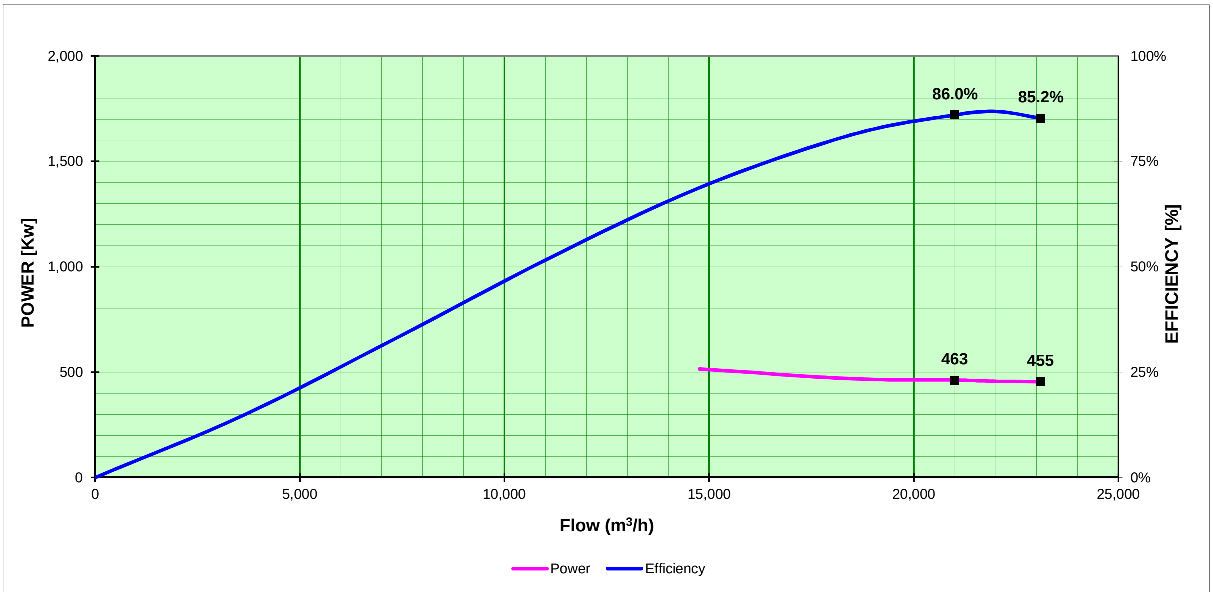
Customer Service OA EMA

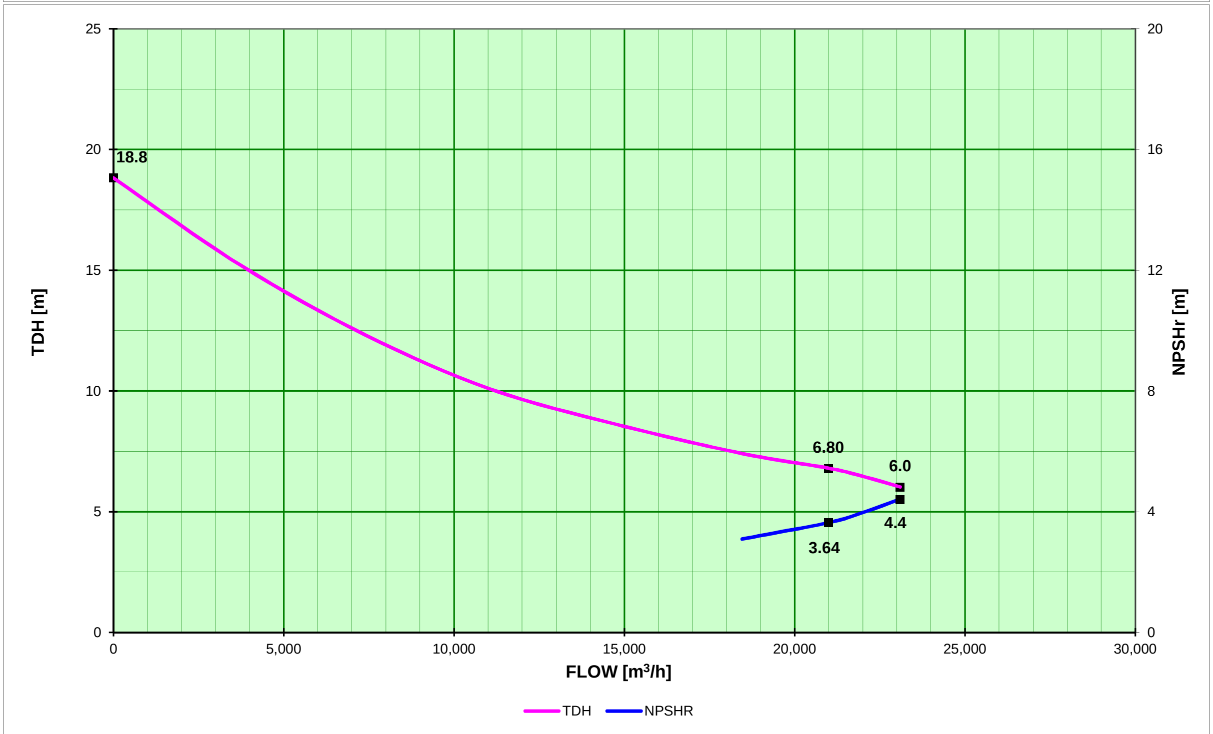
PERFORMANCE CURVES

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**57APM**	Quantity:	**3**

DESIGN POINT

Discharge flow: **21000 m³/h**	Design TDH:	**6.8 m**		
RPM: **295**	Curve number:	**22APM3P**	Specific Gravity:	**1022 kg/m³**





Curves are approximate. Pump is guaranteed for one set of conditions. Capacity and head guarantees are based on shop test. Efficiency is being calculated at design temperature.

Revision: **0** **Date:** **01/08/2017**

Customer Service OA EMA

TORQUE vs SPEED CURVE

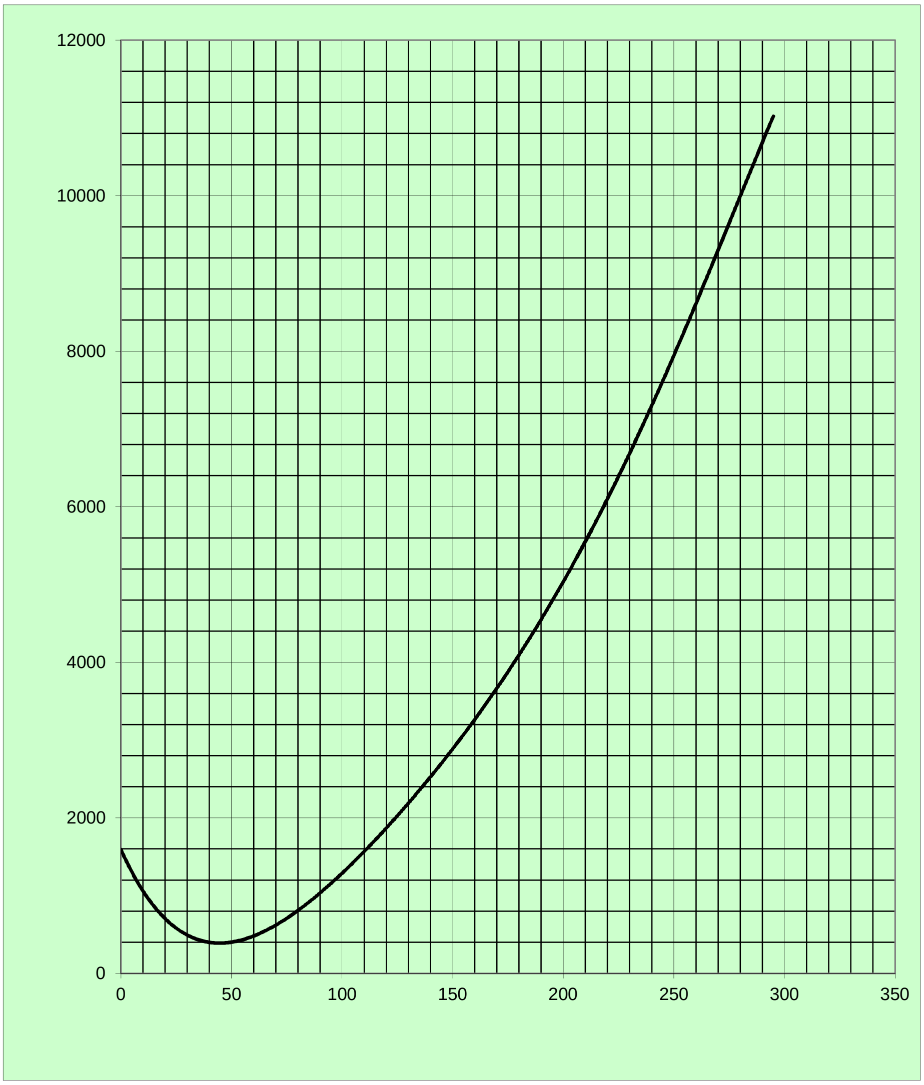
OPEN VALVE

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**57APM**	Quantity:	**3**

DESIGN POINT

Discharge flow:	**21000 m³/h**	Design TDH:	**6.8 m**		
RPM:	**295**	Curve number:	**22APM3P**	Specific Gravity:	**1022 kg/m³**

PUMP 100% SPEED (rpm)	**295**		UNIT ROTOR INERTIA WR2 (Kg.m2)	
PUMP 100% POWER (BHP)	**620**	**463** kW	PUMP ROTOR	**727**
PUMP 100% TORQUE (LbFt)	**11040**	**14953** Nm		



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APPROVED BY

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Customer Service OA EMA

PERFORMANCE DATA SHEET

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**57APM**	Quantity:	**3**

PUMP

Service		Main sea water intake pump
Liquid		Sea Water
Temperature	ºC	18
Specific gravity		1.022
Number of pumps		3
Design flow	m3/h	21000
TDH	m	6.8
Pump Speed	rpm	295
Efficiency	%	86.0%
Power	kW	463
Min. Submergency at R.O.	m	2.91
NPSHr D.P. (at impeller eye)	m	3.6
Pump model		57APM
Min. Continous Flow	m3/h	17850
Head @ shut-off	m	18.8
Performance curve		22APM3P
Setting	mm	4.796
Pump construcion		Non-Pull Out

MOTOR

Power	kW	509
Speed	rpm	295
Supplied by		FLOWSERVE

Revision: **0** **Date:** **01/08/2017**

Customer Service OA EMA

MATERIALS DATA SHEET

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**57APM**	Quantity:	**3**

OPTIONAL MATERIAL: SUPERDUPLEX

Casing	ASTM A240 Ty 2507
Impeller	ASTM A890 Gr 5A
Shaft Sleeves	ASTM A479 UNS S32750
Shaft - Pump Element	ASTM A479 UNS S32750
Shaft - Column and upper	ASTM A479 UNS S32750
Suction Bell	ASTM A240 Ty 2507
Bearings	Rubber / UNS S32750
Shaft Coupling - Between Shafts	ASTM A479 UNS S32750
Shroud	ASTM A240 Ty 2507
Column	ASTM A240 Ty 2507
Discharge head	ASTM A240 Ty 2507
Motor Suport	Carbon Steel
Main Coupling Pum - Electric Motor	Flexible
Shaft Sealing	Soft Packing
Mounting Plate	Structural Steel
Wet Bolts	UNS S32550
Thrust Bearing Location	In pump

Revision: **0** **Date:** **01/08/2017**

FLOWSERVE

Customer Service OA EMA

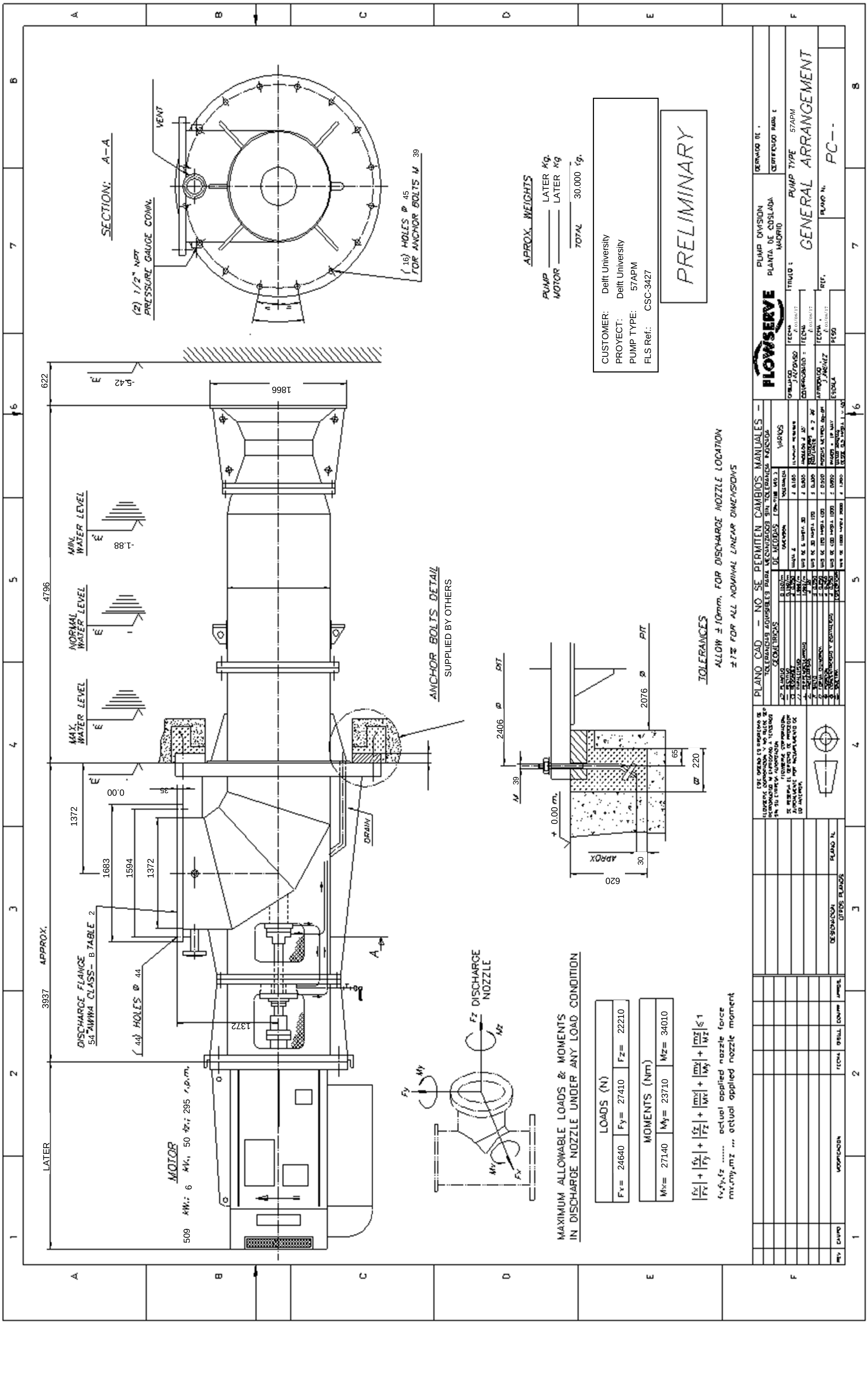
MATERIALS DATA SHEET

Customer:	**Delft University**	Project:	**Delft University**
End User:	**-**	Service:	**Main sea water intake pump**
Location:	**Netherlands**	FLS ref.:	**CSC-3427**
Pump Type:	**57APM**	Quantity:	**3**

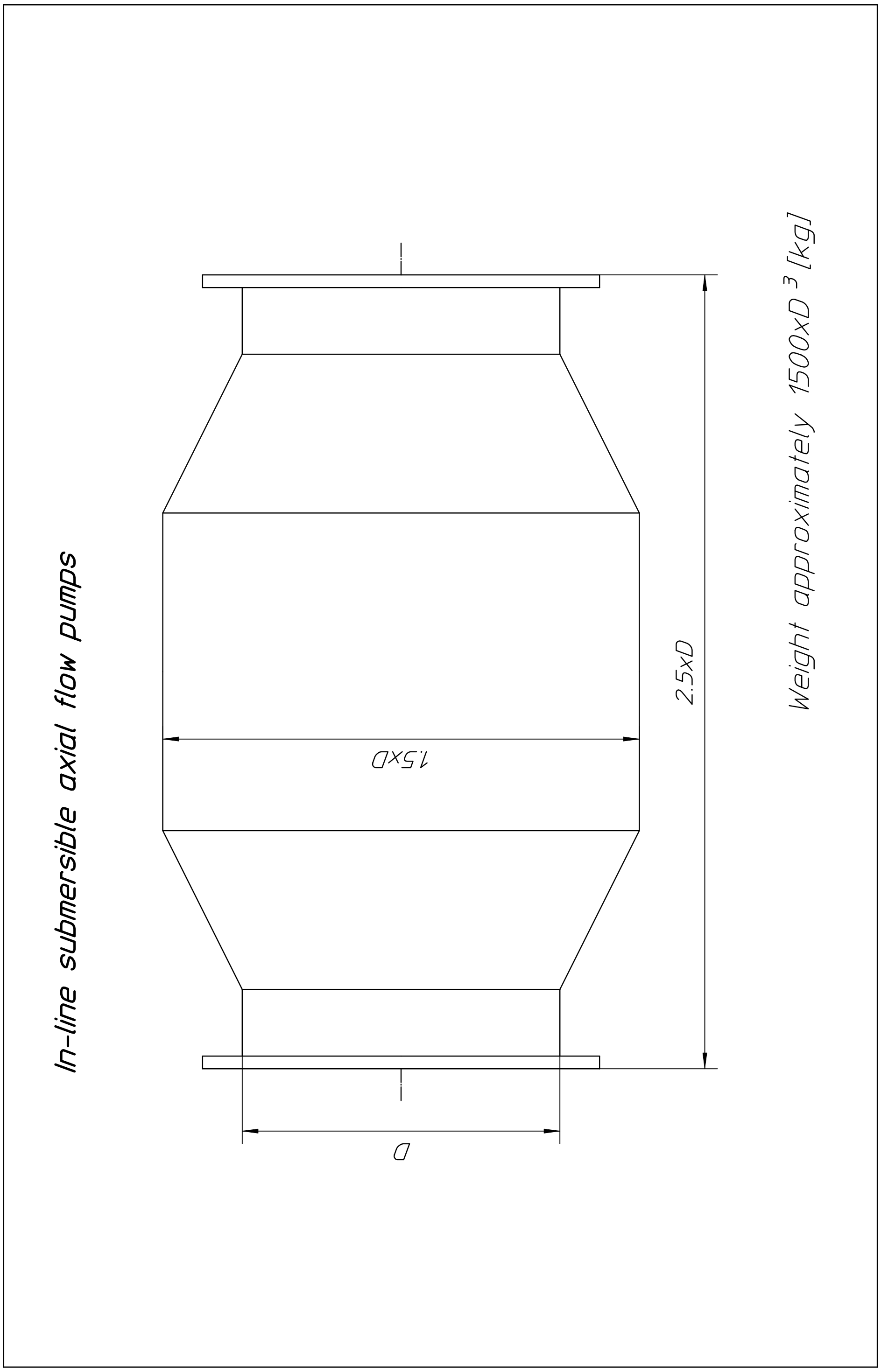
BASE MATERIAL: DUPLEX

Casing	ASTM A240 UNS S31803
Impeller	ASTM A890 Gr 1B
Shaft Sleeves	ASTM A276 UNS S31803
Shaft - Pump Element	ASTM A276 UNS S31803
Shaft - Column and upper	ASTM A276 UNS S31803
Suction Bell	ASTM A240 UNS S31803
Bearings	Rubber / UNS S31803
Shaft Coupling - Between Shafts	ASTM A276 UNS S31803
Shroud	ASTM A240 UNS S31803
Outer column	ASTM A240 UNS S31803
Inner Column	ASTM A240 UNS S31803
Discharge head	ASTM A240 UNS S31803
Motor Suport	Carbon Steel
Main Coupling Pum - Electric Motor	Flexible
Shaft Sealing	Soft Packing
Mounting Plate	Structural Steel
Wet Bolts	S31803
Thrust Bearing Location	In pump

Revision: **0** **Date:** **01/08/2017**



SECTION: A–A
APROX. WEIGHTS
CUSTOMER: Delft University
PROYECT: Delft University
PUMP TYPE: 57APM
FLS Ref.: CSC-3427
PRELIMINARY
ANCHOR BOLTS DETAIL
SUPPLIED BY OTHERS
TOLERANCES
DISCHARGE NOZZLE
MAXIMUM ALLOWABLE LOADS & MOMENTS IN DISCHARGE NOZZLE UNDER ANY LOAD CONDITION
LOADS (N)
Fx= 24640 Fy= 27410 Fz= 22210
MOMENTS (Nm)
Mx= 27140 My= 23710 Mz= 34010
MOTOR
GENERAL ARRANGEMENT
FLOWSERVE


In-line submersible axial flow pumps
Weight approximately 1500xD 3 [kg]
2.5xD
1.5xD
D

C.2 Heat Exchangers

Alfa Laval T50

Gasketed plate-and-frame heat exchanger for a wide range of applications

Alfa Laval Industrial line is a wide product range that is used in virtually all types of industry.

Designed for high throughput, this model delivers excellent thermal performance. A large selection of plate and gasket types is available.

Applications
- Biotech and Pharmaceutical
- Chemicals
- Energy and Utilities
- Food and Beverages
- Home and Personal care
- HVAC and Refrigeration
- Machinery and Manufacturing
- Marine and Transportation
- Mining, Minerals and Pigments
- Pulp and Paper
- Semiconductor and Electronics
- Steel
- Water and Waste treatment

Benefits
- High energy efficiency – low operating cost
- Flexible configuration – heat transfer area can be modified
- Easy to install – compact design
- High serviceability – easy to open for inspection and cleaning and easy to clean by CIP
- Access to Alfa Laval's global service network

Features
Every detail is carefully designed to ensure optimal performance, maximum uptime and easy maintenance. Selection of available features:

- 5-point alignment system
- Reinforced hanger
- Chocolate pattern distribution area
- Glued gasket
- Base-ad gasket
- Leak chamber
- Bearing box
- Fixed bolt head
- Key hole bolt opening
- Lifting lug
- Lining
- Lock washer
- Pressure plate roller
- Tightening bolt cover

Extending performance with Alfa Laval 360° Service Portfolio
Our extensive services ensure top performance from your Alfa Laval equipment throughout its life cycle. The availability of parts and our team's commitment and expertise bring you peace of mind.

Start-up
- Installation
- Installation Supervision
- Commissioning

Maintenance
- Cleaning Services
- Reconditioning
- Repair
- Service Tools
- Spare Parts

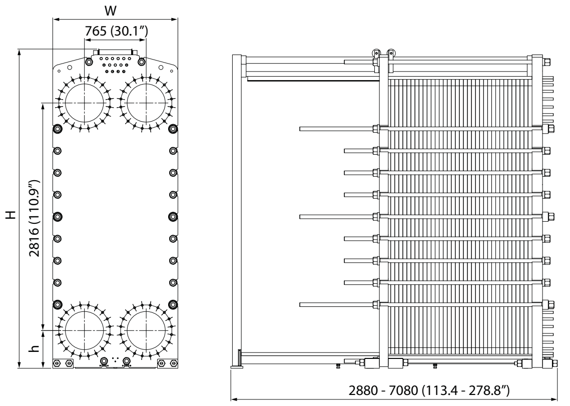
Support
- Exclusive Stock
- Technical Documentation
- Telephone Support
- Training
- Troubleshooting

Improvements
- Equipment Upgrades
- Redesign
- Replacement and Retrofit

Monitoring
- Condition Audit
- Performance Audit

Dimensional drawing

Measurements mm (inches)



Type	H	W	h
T50-FM	4095 (161,2")	1550 (61.0")	467 (18.4")
T50-FG	3951 (155.5")	1550 (61.0")	467 (18.4")
T50-FD	3951 (155.5")	1550 (61.0")	467 (18.4")

The number of tightening bolts may vary depending on pressure rating.

Technical data

Plates

Name	Type	Free channel, mm (inches)
T50-M	Single plate	3.9 (0.15)

Materials

Heat transfer plates	316/316L Ti
Field gaskets	NBR, EPDM
Flange connections	Carbon steel Metal lined: stainless steel, titanium
Frame and pressure plate	Carbon steel, epoxy painted

Other materials may be available on request.

All option combinations may not be configurable.

Operational data

Frame, PV-code	Max. design pressure (barg/psig)	Max. design temperature (°C/°F)
FM, pvcALS	10.0/145	150/302
FG, ASME	10.3/150	177/350
FG, PED	16.0/232	180/356
FD, ASME	20.7/300	177/350
FD, PED	25.0/362	180/356

Extended pressure and temperature rating may be available on request.

Flange connections

FM, pvcALS	EN 1092-1 DN500 PN10 ASME B16.5 Class 150 NPS 20
FG, ASME	ASME B16.5 Class150 NPS 20 ASME B16.5 Class 300 NPS 20
FG, PED	EN 1092-1 DN500 PN10 EN 1092-1 DN500 PN16 ASME B16.5 Class 300 NPS 20
FD, ASME	ASME B16.5 Class 150 NPS 20 ASME B16.5 Class 300 NPS 20
FD, PED	EN 1092-1 DN500 PN25 ASME B16.5 Class 300 NPS 20

Standard EN1092-1 corresponds to GOST 12815-80 and GB/T 9115.

CHE00081EN 2016-04

How to contact Alfa Laval

Contact details for all countries are continually updated on our website. Please visit www.alfalaval.com to access the information direct.

C.3 Turbine & Generator Package

Standard Configurations

Wide choice of Turboexpander-Generator configurations

The majority of applications for Rotoflow Turboexpanders require the expander to be coupled to an electrical generator. There are two basic choices: with the generator mounted directly on the turbine shaft; or connection through speed reducing gears. An integral gearing option provides the additional benefit of multi-staging, allowing multiple expander stages to be mounted on a single gearbox. In most cases the Turboexpander-Generator unit can be completely skid-mounted to simplify transportation and reduce installation costs.

Direct drive

The direct drive option, when feasible, eliminates the need for speed reduction, gear boxes and associated equipment.

External gearbox

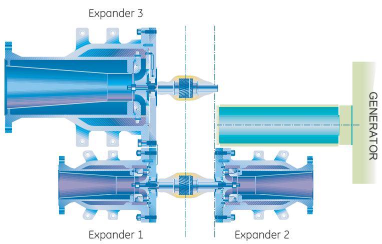
Expanders with an external gearbox feature Rotoflow patented bearings, with a common oil supply system for the complete package. The installed fleet ranges from 50 to 15,000 kW.

Integral gearbox

This arrangement, in use since the early 1970s, mounts the expander wheel directly on the high-speed pinion, eliminating the need for a high-speed coupling. Standard designs are available, up to 15,000 kW.

Multi-stage

High pressure ratios or high flow rates require the multi-stage arrangement. Standard expander-gear designs can accommodate up to four expanders on a common integral gearbox.



Multi-stage expander mounted on a single integral gear.

Expander-generator direct drive.

External gear configuration.

Integral gear configuration.

Multi-stage configuration.

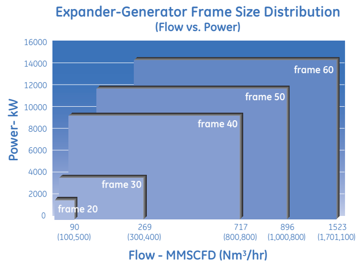
Wide Choice of Turboexpander-Generator Configurations

Wide choice of Turboexpander-Generator configurations

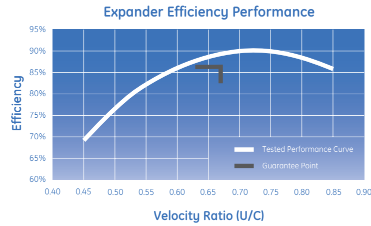
A broad range of expander sizes is available to meet your process requirements for any turboexpander application. Wheels are designed to handle the entire flow range.

Rotoflow Product Range

Pressure	up to 3,000 psia (200 BarA)
Temperature	-450°F to 925°F (-270°C to 500°C)
Power	up to 20,000 hp (15,000 kW) per stage
Expansion ratio	up to 14
Process fluid	All pure or mixed fluids including natural gas, petrochemical products, hydrogen, air, steam, etc.



Frame sizes vary to give the best efficiency to cost ratio.



No negative tolerance on expander efficiency.

Vertically split casings ensure easier installation and assemblly at the job site.

Rotoflow recently modified an axial turbine at a geothermal facility in the western US to a radial turboexpander.

Rotoflow turboexpander-generators are packaged complete and ready for installation.

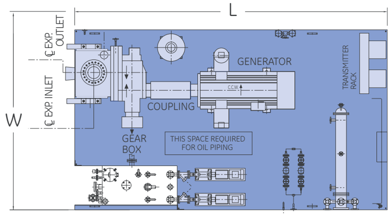
Machinery Layout

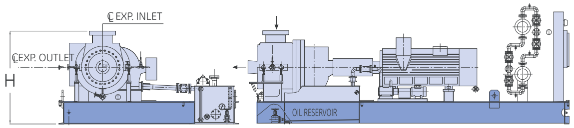
Standard Turboexpander-Generator skid dimensions and weights for our product range are presented here. Custom skid designs are also available.

Size and weights of integral gear equipment

Integral gear footprint of machinery layout for frame sizes 20, 30 and 40

Frame size	L (in/mm)	W (in/mm)	H (in/mm)	Weight (lb/kg)
20	250/6,350	144/3,658	120/3,048	36,000/16,364
30	250/6,350	144/3,658	120/3,048	41,000/18,637
40	250/6,350	144/3,658	120/3,048	46,00/20,910

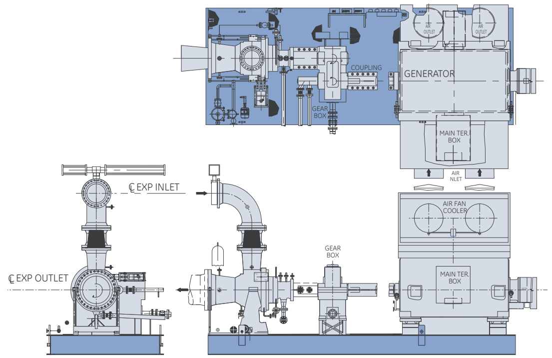




Size and weights of external gear for machinery skid

External gear footprint of machinery layout for frame sizes 50 and 60

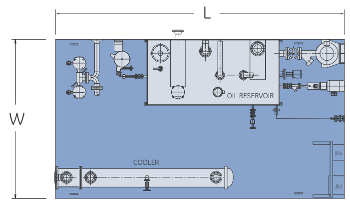
Frame size	L (in/mm)	W (in/mm)	H (in/mm)	Weight (lb/kg)
50	400/10,160	144/3,658	230/5,842	126,000/57,276
60	400/10,160	144/3,658	230/5,842	140,000/63,640

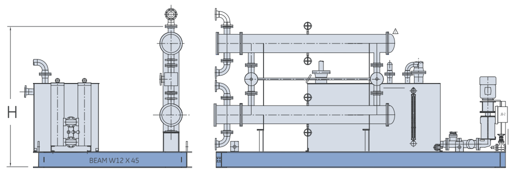


Size and weights of lube console for external gear

Lube console footprint for frame sizes 50 and 60

Frame size	L (in/mm)	W (in/mm)	H (in/mm)	Weight (lb/kg)
50	250/6,350	144/3,658	120/3,050	31,500/14,319
60	250/6,350	144/3,658	120/3,050	35,000/15,910





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Second-hand Vessel Prices from Clarksons Research

SHIPPING REVIEW DATABASE

Table 46 Secondhand Tanker Prices

$ million

End	37k Products 5yrs[1]	37k Products 10yrs[2]	47k Products 5 yrs[3]	Aframax 5yrs	Aframax 10yrs	Suezmax 5yrs[4]	Suezmax 10yrs[5]	VLCC 5yrs[6]	VLCC 10yrs[7]	5-y-o S'hand Index: Index	5-y-o S'hand Index: Yr/Yr change
1994	17.00	11.00	20.00	32.00		34.00	23.38	47.50	31.46	103	-1.3%
1995	18.50	13.00	22.50	32.00		38.00	26.71	54.00	38.51	112	9.0%
1996	22.00	14.50	24.50	34.50		42.50	30.27	58.50	44.66	128	14.3%
1997	23.50	15.50	27.00	36.00		44.00	32.73	65.00	46.06	138	7.2%
1998	16.50	11.50	21.00	23.50		36.50	27.24	50.00	32.73	100	-27.5%
1999	16.00	11.50	20.00	26.00		35.00	24.68	53.00	35.74	99	-0.9%
2000	18.50	14.00	24.50	40.00		49.00	32.41	71.00	45.02	128	29.1%
2001	17.00	11.50	20.50	30.00	24.50	39.00	26.00	58.00	45.00	107	-16.1%
2002	15.50	10.50	19.52	29.00	24.00	38.00	22.00	54.00	40.00	100	-6.7%
2003	24.50	16.00	26.00	36.00	27.00	47.00	22.50	70.00	55.00	131	30.9%
2004	31.00	20.00	39.00	57.50	44.00	75.00	45.00	108.00	85.00	193	47.3%
2005	40.00	21.00	47.00	63.00	54.00	75.00	65.00	117.00	90.00	225	16.9%
2006	44.00	31.00	47.50	66.50	53.00	82.00	71.00	118.00	96.00	239	6.1%
2007	44.50	36.00	52.00	70.00	60.00	92.00	74.00	135.00	110.00	257	7.6%
2008	30.50	23.00	37.00	54.00	43.00	78.00	63.00	104.00	74.00	186	-27.7%
2009	19.50	14.00	23.00	41.50	24.00	56.50	43.00	79.00	59.00	126	-32.3%
2010	24.50	15.50	26.00	40.00	28.00	59.00	40.00	85.00	60.00	136	8.2%
2011	22.00	12.00	26.00	35.00	20.00	47.00	32.00	58.00	36.00	122	-10.3%
2012	22.00	13.00	25.00	27.50	17.00	40.00	26.00	57.00	37.00	112	-8.2%
2013	25.00	15.00	29.00	32.00	22.00	42.00	27.00	60.00	41.00	129	14.8%
2014	23.00	15.00	25.00	42.00	27.00	57.00	37.00	77.00	52.00	131	1.3%
2015	25.00	17.00	29.00	46.00	31.00	60.00	42.00	80.00	55.00	144	10.0%
2016	19.00	14.00	22.00	29.00	18.00	40.00	27.50	60.00	40.00	105	-26.8%
Dec-13	25.00	15.00	29.00	32.00	22.00	42.00	27.00	60.00	41.00	129	14.8%
Jan-14	26.00	16.00	29.00	34.00	24.00	47.00	32.00	68.00	46.00	134	19.0%
Feb-14	26.00	16.00	29.00	37.00	23.00	48.00	33.00	72.00	47.00	136	20.7%
Mar-14	26.00	16.00	29.00	38.00	23.00	50.00	34.00	73.00	48.00	138	23.6%
Apr-14	26.00	16.00	29.00	38.00	23.00	50.00	34.00	74.00	49.00	139	21.8%
May-14	25.00	15.00	27.00	38.00	23.00	50.00	34.00	75.00	50.00	132	14.9%
Jun-14	25.00	15.00	27.00	37.00	23.00	49.00	33.00	74.00	49.00	132	19.2%
Jul-14	23.00	15.00	26.00	37.00	23.00	49.00	33.50	74.00	49.00	128	8.2%
Aug-14	23.00	15.00	25.50	40.00	25.00	50.00	34.00	74.00	48.00	128	7.5%
Sep-14	23.00	15.00	25.50	42.00	27.00	50.00	34.00	74.00	48.00	129	7.9%
Oct-14	23.00	14.00	25.50	42.00	27.00	54.00	35.00	77.00	52.00	130	5.2%
Nov-14	23.00	14.00	25.00	42.00	27.00	57.00	37.00	77.00	52.00	130	3.0%
Dec-14	23.00	15.00	25.00	42.00	27.00	57.00	37.00	77.00	52.00	131	1.3%
Jan-15	26.00	16.00	27.00	46.00	31.00	60.00	42.00	81.00	54.00	142	6.1%
Feb-15	24.00	15.00	27.00	45.00	30.00	59.00	41.00	81.00	52.00	139	2.5%
Mar-15	24.00	16.00	27.00	45.00	30.00	57.50	41.00	81.00	52.00	139	0.2%
Apr-15	24.00	16.00	27.00	45.00	30.00	57.50	41.00	81.00	52.00	137	-1.7%
May-15	24.00	16.00	26.00	45.00	30.00	59.00	40.00	80.00	52.00	137	3.5%
Jun-15	24.00	16.00	27.00	45.00	31.00	59.00	40.00	80.00	55.00	139	5.5%
Jul-15	25.00	17.50	28.00	46.00	32.00	61.00	42.00	84.00	59.00	143	12.0%
Aug-15	25.00	17.50	28.00	46.00	33.00	61.00	42.00	84.00	59.00	144	12.5%
Sep-15	24.50	17.00	29.00	45.00	30.00	61.00	42.00	80.00	55.00	144	12.1%
Oct-15	24.50	17.00	29.00	46.00	31.00	60.00	42.00	80.00	55.00	144	10.6%
Nov-15	25.00	17.00	29.00	46.00	31.00	60.00	42.00	80.00	55.00	144	10.5%
Dec-15	25.00	17.00	29.00	46.00	31.00	60.00	42.00	80.00	55.00	144	10.0%
Jan-16	26.00	18.50	29.00	45.00	30.00	59.00	44.00	80.00	58.00	145	2.5%
Feb-16	26.00	18.50	27.50	40.00	27.00	57.00	42.00	76.00	56.00	139	0.0%
Mar-16	26.00	18.50	27.50	40.00	27.00	53.50	40.00	76.00	56.00	138	-0.3%
Apr-16	25.00	17.50	27.50	40.00	27.00	52.50	40.00	75.00	55.00	137	0.0%
May-16	24.00	16.50	26.50	39.00	26.50	52.50	39.00	72.00	52.00	132	-3.4%
Jun-16	21.00	15.00	24.00	37.00	24.00	50.00	37.00	64.00	44.00	121	-13.0%
Jul-16	20.00	15.00	23.00	35.00	22.00	46.50	34.00	64.00	44.00	116	-19.3%
Aug-16	20.00	15.00	23.00	33.00	21.00	45.00	32.50	62.00	41.00	114	-20.9%
Sep-16	20.00	15.00	23.00	32.50	21.00	44.00	31.50	62.00	41.00	112	-22.3%
Oct-16	19.00	14.00	22.00	31.50	20.00	43.00	30.50	61.00	40.00	108	-25.1%
Nov-16	19.00	14.00	22.00	29.00	18.00	40.00	27.50	60.00	40.00	105	-26.8%
Dec-16	19.00	14.00	22.00	29.00	18.00	40.00	27.50	60.00	40.00	105	-26.8%
Jan-17	20.00	14.00	22.50	30.00	19.00	41.00	28.50	62.00	42.00	109	-25.0%
Feb-17	20.00	14.00	22.50	30.00	19.00	41.00	28.50	62.00	42.00	108	-22.2%

Index: 100 = January 2000. % = Percentage year on year change.

Before February 2010: [1] 30k until May-03, 35k between May-03 and Mar-08, 37k thereafter. [2] 35k dwt until end-2014, then 37k dwt subsequently. [3] 40k until Nov-01, 45k between Nov-01 and Mar-08, 47k thereafter. 4150k dwt then 160k after 2012. 5Data pre-2001 basis interpolation and industry sources, including Platou. 6300k dwt then 310k after 2012. 7250k dwt then 265k after 2012. Data pre-2001 basis interpolation and industry sources, including Platou.

NB Between October 2008 and January 2010, Clarksons Research did not publish benchmark values. This was a period of transition in the Sale and Purchase markets, characterised by spells of rapidly changing price levels, low levels of sales activity and a wide spread of price ideas. During this period, the data should be treated with caution as confidence limits will vary over time, and between sectors.

Table 47 Secondhand Bulkcarrier Prices

$ million

End	32k 5 Yrs"	32k 10 yrs	56k 5 Yrs'	52k 10 yrs~	82k 5 Yrs*	76k 10 Yrs^	180k 5 Yrs	170k 10 Yrs	5-y-o S'hand Index: Index	5-y-o S'hand Index: %
1994	16.00	11.50	20.50	15.75	21.00	15.75	37.33	20.50	**133**	***7.6%***
1995	16.50	12.75	21.00	15.50	21.50	16.50	32.67	19.00	**134**	***0.8%***
1996	13.00	8.75	18.75	13.50	19.50	14.00	28.00	15.00	**108**	***-19.0%***
1997	13.75	10.75	18.00	13.50	22.00	15.75	34.42	18.50	**114**	***5.4%***
1998	9.25	6.00	12.50	8.00	14.00	9.75	25.67	14.50	**78**	***-32.0%***
1999	11.50	8.00	16.00	11.50	16.75	12.00	29.25	17.50	**96**	***23.2%***
2000	12.00	9.00	15.25	12.00	16.00	11.75	30.25	19.00	**95**	***-0.3%***
2001	11.00	7.75	13.25	9.75	14.00	9.50	27.00	16.50	**83**	***-13.3%***
2002	11.25	8.50	14.25	10.50	17.00	11.50	29.00	20.50	**89**	***8.0%***
2003	14.50	10.75	20.00	15.50	28.00	20.00	44.00	32.00	**127**	***42.8%***
2004	21.50	17.00	29.00	22.50	40.00	31.00	64.50	46.00	**186**	***45.8%***
2005	26.00	19.00	25.50	20.50	29.50	24.00	57.00	38.00	**178**	***-4.2%***
2006	28.50	23.00	40.00	32.00	45.50	37.00	81.00	62.00	**239**	***34.2%***
2007	44.00	40.00	75.00	60.00	88.50	72.00	150.00	105.00	**423**	***77.1%***
2008	20.50	16.00	24.50	18.00	26.00	20.00	45.00	31.00	**145**	***-65.7%***
2009	22.00	17.00	27.00	21.50	36.00	27.50	55.00	44.00	**169**	***16.2%***
2010	25.00	21.50	29.00	24.00	36.00	28.00	50.00	38.00	**173**	***2.8%***
2011	21.00	16.00	24.50	17.00	26.50	20.00	36.00	26.50	**139**	***-19.7%***
2012	15.50	12.00	19.50	14.50	18.00	13.00	32.50	21.00	**105**	***-24.6%***
2013	19.00	14.00	24.50	17.50	25.50	18.00	44.00	31.00	**135**	***28.5%***
2014	17.00	12.50	20.50	13.50	20.00	14.50	39.00	27.50	**116**	***-14.3%***
2015	10.00	8.00	13.50	8.00	14.00	8.50	25.00	13.50	**73**	***-36.7%***
2016	12.00	6.75	14.00	9.50	14.00	8.50	24.00	15.00	**79**	***7.1%***
Dec-13	19.00	14.00	24.50	17.50	25.50	18.00	44.00	31.00	**135**	***28.5%***
Jan-14	21.00	16.00	26.00	19.50	27.00	20.50	46.00	33.00	**145**	***39.9%***
Feb-14	21.00	16.00	26.00	20.50	27.00	22.00	48.00	34.00	**146**	***33.9%***
Mar-14	21.00	16.00	27.00	22.00	28.00	22.50	52.00	36.00	**150**	***37.7%***
Apr-14	20.50	15.00	27.00	21.00	27.00	22.50	53.00	38.00	**148**	***33.2%***
May-14	20.50	15.00	25.50	20.00	26.50	21.00	50.00	37.00	**144**	***21.8%***
Jun-14	19.50	14.50	25.50	19.50	24.00	18.50	47.00	34.00	**138**	***14.8%***
Jul-14	19.50	14.50	24.50	18.00	24.00	18.50	47.00	34.00	**136**	***13.6%***
Aug-14	19.00	14.00	24.00	17.00	23.50	17.00	47.00	34.00	**134**	***12.3%***
Sep-14	18.50	13.00	23.00	16.00	22.50	16.50	48.00	35.00	**130**	***4.5%***
Oct-14	18.00	12.50	22.00	14.50	20.50	14.75	42.00	32.00	**123**	***-5.8%***
Nov-14	17.00	12.50	21.50	14.50	20.50	14.75	40.00	30.00	**118**	***-12.5%***
Dec-14	17.00	12.50	20.50	13.50	20.00	14.50	39.00	27.50	**116**	***-14.3%***
Jan-15	15.50	10.00	19.00	13.00	19.00	14.00	36.00	26.00	**107**	***-26.4%***
Feb-15	14.50	10.00	16.50	11.50	17.00	13.00	33.00	21.00	**97**	***-33.5%***
Mar-15	13.50	9.50	16.00	11.00	17.00	13.00	34.00	22.00	**94**	***-37.5%***
Apr-15	13.50	9.50	16.00	11.00	16.50	13.00	34.00	22.00	**93**	***-37.0%***
May-15	13.50	9.00	15.00	11.00	17.50	11.50	32.00	18.00	**92**	***-36.1%***
Jun-15	13.00	9.00	15.00	10.50	17.00	11.00	31.00	18.00	**90**	***-34.7%***
Jul-15	13.00	9.00	15.00	10.50	17.50	11.50	32.50	19.50	**91**	***-33.2%***
Aug-15	13.00	9.50	15.50	10.50	18.00	12.00	35.00	20.00	**93**	***-30.1%***
Sep-15	13.00	9.50	15.50	10.50	18.00	12.00	35.00	20.00	**93**	***-28.3%***
Oct-15	11.50	8.50	15.50	9.00	17.00	10.50	32.00	20.00	**87**	***-29.3%***
Nov-15	11.50	8.50	14.50	8.50	15.00	9.00	27.00	15.00	**81**	***-31.4%***
Dec-15	10.00	8.00	13.50	8.00	14.00	8.50	25.00	13.50	**73**	***-36.7%***
Jan-16	9.50	7.00	12.00	6.00	13.00	8.00	23.00	13.00	**68**	***-36.6%***
Feb-16	9.50	7.00	12.00	6.50	13.00	8.00	23.75	12.00	**68**	***-29.8%***
Mar-16	9.50	7.00	12.00	6.50	13.00	8.00	23.75	12.00	**68**	***-27.5%***
Apr-16	9.50	7.00	12.00	6.50	13.00	8.00	23.75	14.00	**68**	***-27.1%***
May-16	9.00	6.50	13.00	6.50	14.00	8.00	24.75	14.00	**70**	***-24.6%***
Jun-16	9.00	6.75	13.00	7.75	14.00	8.00	24.75	14.00	**70**	***-22.5%***
Jul-16	9.50	6.75	13.00	8.50	14.00	8.00	25.00	14.00	**71**	***-22.0%***
Aug-16	9.50	6.75	13.25	8.50	14.00	8.00	24.00	14.00	**71**	***-24.0%***
Sep-16	10.50	6.75	13.50	9.00	14.00	8.00	24.00	14.00	**74**	***-20.8%***
Oct-16	10.50	6.75	13.50	9.00	14.00	8.00	24.00	14.00	**74**	***-14.7%***
Nov-16	11.50	6.75	14.00	9.50	14.00	8.00	24.00	15.00	**77**	***-4.6%***
Dec-16	12.00	6.75	14.00	9.50	14.00	8.50	24.00	15.00	**79**	***7.1%***
Jan-17	13.50	7.00	15.00	10.50	15.00	9.50	25.00	16.00	**85**	***26.0%***
Feb-17	13.50	7.00	15.00	11.00	17.00	10.00	25.00	16.00	**85**	***25.4%***

Index: 100 = January 2000. % = Percentage year on year change.
Before January 2010: ' 42-45k dwt, 45-48k dwt subsequently, 52k dwt, 56k subsequently; ^73k dwt, 76k until February 2017.
Before January 2012: "28-30k, 32k subsequently; ~45-48k, 52k subsequently.
*Before February 2017: *76k, 82k subsequently.*
NB Between October 2008 and January 2010, Clarksons Research did not publish benchmark values. This was a period of transition in the Sale and Purchase markets characterised by spells of rapidly changing price levels, low levels of sales activity and a wide spread of price ideas. During this period, the data should be treated with caution as confidence limits will vary over time, and between sectors.

SHIPPING REVIEW DATABASE

Table 48 Liner Vessel Charter Rates $/day

Average	6 - 12 Months Timecharter FCC 1,000 TEU	FCC 1,700 TEU	FCC 2,750 TEU	FCC 4,400 TEU	3 Years Timecharter FCC 6,800 TEU	FCC 9,000 TEU	FCC Index	6 - 12 Months Timecharter MPP 17k dwt	MPP 12k dwt	1 Year Timecharter PCTC 6,500 CEU	Ro-Ro 2,000-2,500 lm	Gen. Cargo (€/day) 3,500dwt g'less
2011	7,729	10,142	13,388	19,854			63	9,729		20,875		
2012	5,358	6,292	6,742	9,942	29,857	37,357	43	8,988		23,500		
2013	6,321	7,096	6,829	8,696	27,542	37,625	46	8,358		23,250	9,000	
2014	6,396	7,313	7,425	8,771	24,667	39,125	47	7,488	8,508	25,375	9,521	2,491
2015	7,250	8,842	9,563	11,817	22,750	36,708	53	6,700	7,379	23,333	11,850	2,704
2016	6,550	6,804	6,000	4,979	13,208	24,792	41	7,858	6,767	20,292	13,708	2,779
2017	6,000	6,250	6,225	4,350	12,250	23,250	38	7,500	6,500	15,000	14,667	2,825
Feb-16	6,750	7,000	6,000	5,800	14,000	27,500	42	8,650	7,200	22,500	15,500	2,800
Mar-16	6,850	7,050	6,000	5,400	13,500	26,000	42	8,650	7,200	22,000	15,500	2,800
Apr-16	6,800	7,100	6,000	5,200	13,000	25,500	42	8,000	7,000	22,000	14,500	2,800
May-16	6,800	7,000	6,000	5,200	13,000	25,500	41	8,000	7,000	22,000	13,500	2,800
Jun-16	6,800	7,000	6,000	5,150	13,000	25,500	41	7,500	6,500	22,000	13,000	2,800
Jul-16	6,700	6,900	6,000	5,100	13,000	25,000	41	7,500	6,500	21,500	13,000	2,750
Aug-16	6,700	6,700	5,900	4,700	13,000	25,000	40	7,500	6,500	19,500	13,000	2,650
Sep-16	6,550	6,750	6,000	4,450	13,000	22,000	40	7,500	6,500	19,000	13,000	2,650
Oct-16	6,250	6,650	6,000	4,400	13,000	21,000	39	7,500	6,500	18,000	13,000	2,750
Nov-16	6,100	6,300	6,050	4,250	12,500	22,000	38	7,500	6,500	16,000	13,000	2,800
Dec-16	6,100	6,200	6,050	4,150	12,500	23,500	38	7,500	6,500	16,000	13,000	2,850
Jan-17	6,000	6,300	6,150	4,200	12,000	23,000	38	7,500	6,500	15,000	14,000	2,850
Feb-17	6,000	6,200	6,300	4,500	12,500	23,500	39	7,650	6,550	15,000	15,000	2,800

Monthly data as at end of month. MPP timecharter rate data is based a new vessel specification from start 2014 onwards.

Table 49 Liner Vessel Secondhand Prices $ million

End	1,000 teu 10yo FCC	1,700 teu 10yo FCC	2,750 teu 10yo FCC	3,500 teu* 10yo FCC	4,500 teu* 10yo FCC	6,600 teu 5yo FCC	8,800 teu 5yo FCC	2,500 l.m. 5yo Ro-Ro	2,500 l.m. 10yo Ro-Ro	Index FCC	Yr/Yr % change
2011	8.5	12.3	18.0	27.0	30.5	50.0	69.0	40.1	29.0	65.8	-23.6%
2012	5.0	8.0	10.0	13.0	14.0	42.5	63.5	36.8	27.6	36.6	-43.5%
2013	5.3	9.3	10.3	11.0	12.0	52.0	67.0	34.2	23.3	37.1	3.3%
2014	3.3	8.0	8.5	10.0	14.5	44.0	60.0	29.6	20.9	24.7	-34.6%
2015	6.0	8.5	11.0	11.5	12.0	36.0	56.0	30.5	22.9	35.7	47.9%
2016	4.3	5.5	5.8	5.0	5.5	16.0	26.0	34.6	27.3	24.8	-18.8%
May-16	4.8	6.5	7.3	6.5	8.0	21.0	46.0	37.3	29.4	30.8	-13.2%
Jun-16	4.8	6.5	7.0	5.5	7.5	21.0	46.0	37.1	29.2	30.2	-17.8%
Jul-16	4.8	6.5	7.0	5.5	6.0	21.0	43.0	36.5	28.8	29.5	-19.0%
Aug-16	4.8	6.0	6.5	5.5	6.0	21.0	38.0	37.1	29.2	28.2	-22.1%
Sep-16	4.8	5.8	6.0	5.0	6.0	17.0	26.0	37.1	29.3	25.9	-25.9%
Oct-16	4.3	5.8	6.0	5.0	6.0	17.0	26.0	36.3	28.6	25.4	-23.0%
Nov-16	4.3	5.5	5.8	5.0	5.5	16.0	26.0	35.6	28.1	24.8	-18.8%
Dec-16	4.3	5.5	5.8	5.0	5.5	16.0	26.0	34.6	27.3	24.8	-18.8%
Jan-17	4.5	5.5	5.8	5.3	6.0	16.0	26.0	34.8	27.9	25.2	-10.0%
Feb-17	4.5	5.5	6.0	5.0	6.0	16.0	24.0	35.3	28.3	25.1	-6.1%

*NB Between October 2008 and January 2010, Clarksons Research did not publish benchmark values. This was a period of transition in the Sale and Purchase markets, characterised by spells of rapidly changing price levels, low levels of sales activity and a wide spread of price ideas. During this period, the data should be treated with caution as confidence limits will vary over time, and between sectors. *Panamax vessels. 3,500 teu and 4,500 teu based on a narrow beam (old Panamax) vessel.*

Table 50 Liner Vessel Newbuilding Prices $ million

End	1,700 teu FCC	2,000 teu FCC	2,750 teu FCC	4,800 teu FCC	6,600 teu FCC	8,800 teu* FCC	10,000 teu" FCC	13,000 teu FCC	18,000 teu FCC	Index FCC	Yr/Yr % change
2011	30.5	29.0	38.3	59.0	69.0	92.5	98.5	128.0		91.3	-3.6%
2012	25.0	23.8	30.5	45.0	58.0	76.5	85.5	107.0		72.6	-20.5%
2013	26.0	25.5	31.5	50.5	65.5	85.5	95.5	113.5		79.1	9.0%
2014	27.0	27.0	32.5	53.5	67.8	89.0	99.0	116.0	154.0	82.1	3.7%
2015	25.0	26.3	29.5	49.0	66.5	89.0	99.0	116.0	154.0	77.0	-6.1%
2016	21.8	22.3	27.0	43.0	60.0	83.0	93.0	109.0	145.5	69.0	-10.4%
May-16	24.3	25.3	29.0	45.5	63.3	86.3	96.3	112.5	149.5	74.2	-8.3%
Jun-16	23.5	24.5	28.3	44.0	62.0	84.5	94.5	110.5	147.5	72.4	-9.9%
Jul-16	22.5	23.5	28.0	43.5	60.5	82.5	92.5	108.5	145.0	70.6	-12.1%
Aug-16	22.5	23.0	28.0	43.5	60.5	82.5	92.5	108.5	145.0	70.4	-12.1%
Sep-16	22.5	23.0	28.0	43.5	60.5	83.0	93.0	109.0	145.5	70.5	-11.7%
Oct-16	22.5	23.0	28.0	43.5	60.0	83.0	93.0	109.0	145.5	70.2	-9.7%
Nov-16	22.5	23.0	28.0	43.5	60.0	83.0	93.0	109.0	145.5	70.2	-9.3%
Dec-16	21.8	22.3	27.0	43.0	60.0	83.0	93.0	109.0	145.5	69.0	-10.4%
Jan-17	21.0	21.5	26.0	42.5	60.0	83.0	93.0	109.0	145.5	67.7	-12.1%
Feb-17	21.0	21.5	26.0	42.5	60.0	83.0	93.0	109.0	145.0	67.7	-11.7%

*Newbuilding prices vary as to country of build, delivery and ship specification. Prices here are end month/year and from Jun-08 assume a "European spec", 20/20/20/20/20% payments and "first class competitive yards" quotations, and relate to market contracts where these have taken place and to brokers' best estimates when no contracts have occurred. 4,800 teu based on wide beam vessel. * Based on "single island" design. "Based on "twin island" design.*

E

Representative Panamax Bulker

PROTEFS: fourth-generation Jiangnan Panamax bulker

Shipbuilder: ....Jiangnan Shipyard (Group) Co Ltd, People's Republic of China
Vessel's name: ....*Protefs*
Hull number: H2301
Owner/operator: ....Cypres Enterprises Corp/Diana Shipping Agencies, Greece
Designer: ....Jiangnan Shipyard (Group) Co Ltd, People's Republic of China
Model test establishment used: .... -
Flag: ....Bahamas
Total number of sister ships already completed: 1
Total number of sister ships still on order: 3

JIANGNAN Shipyard is considered to be the oldest shipbuilder in China, however, its Shanghai city premises, which date back nearly 200 years, are now scheduled to be moved to a 'greenfield' site on an island in the River Yangtze where the company plans to set up two shipbuilding divisions. One of these will specialise in building bulk carriers up to Panamax size (the other will build larger vessels), thus continuing an involvement with this class of vessel begun in 1985, and which now totals over 30 units delivered or on order.

Protefs is an example of the fourth-generation (Mk IV) of the design, and is laid out in conventional style with a single-deck, forecastle, and seven cargo holds contained in a single-skin hull arranged with top and bottom wing ballast tanks, the latter being joined with double-bottom tanks centrally divided by a duct keel. Exceptions to this arrangement apply under Nos 5 and 6 holds where the port and starboard double-bottom tanks are separated from the wing tanks, and are designated as Nos 1 and 2 heavy fuel tanks. Additional fuel is carried in side tanks at the fore end of the engineroom, port and starboard. All ballast tanks, including No 4 hold which can be flooded, are coated with bleached tar epoxy. Basic tar epoxy coatings are used in the cargo holds.

Heavy cargoes can be loaded into alternate holds (Nos 1, 3, 5, and 7) in line with Lloyd's Register class notation 'strengthened for heavy cargoes: Nos 2, 4, and 6 holds may be empty', and tanktop, hopper side, and lower stool plating has been strengthened for grab discharge. Transverse bulkheads in the cargo space are of corrugated construction and built on stools at top and bottom of the holds. MacGregor side-rolling, chain-operated hatch covers are fitted, stowing each side of the opening whilst cargo is being worked.

The main propulsion machinery comprises a MAN B&W 5S60MC Mk6 low-speed diesel engine, manufactured under licence by Hudong Heavy Machinery (HHM) and developing 10,200kW at 105rev/min. This is directly coupled to a FP propeller running in an open-water stern frame and producing a service speed at 90% MCR (9198kW/102.80rev/min) and allowing a 15% sea margin of 14.40knots. With no shaft-driven alternator fitted, electrical supply is derived from three Yanmar/Taiyo diesel-driven sets each producing 600kW at 900rev/min. Steam requirements are satisfied from an Aalborg Mission boiler and an exhaust-gas economiser. A Lyngsø DMS-21001 engine remote control system is installed and provides bridge operation of the machinery installation.

Accommodation is arranged in the superstructure aft for eight officers, 15 crew and six 'spare' personnel, with the layout featuring an entire deck given over to the captain's quarters. Leisure facilities include a gymnasium. Traditional lifeboats are not provided, instead a free-fall craft with slipway and recovery davit operates over the stern. That section of the accommodation block housing the single-berth cabins and public rooms is separated from the funnel casing to reduce noise and vibration, but the intervening space is utilised by an athwartship gantry carrying a travelling spares and stores crane.

TECHNICAL PARTICULARS

Length, oa 225.00m
Length, bp..217.00m
Breadth, moulded 32.20m
Depth, moulded to main deck 19.20m
Gross 40,230gt
Displacement, at 14.05m draught 86,035tonnes
Deadweight
- design 63,395dwt
- scantling..73,630dwt

Draught
- design 12.50m
- scantling 14.05m

Speed, service at 90% MCR, 15% sea margin....14.40knots
Cargo capacity...90,624m³
Bunkers
- heavy oil...2680m³
- diesel oil...137m³

Water ballast 32,480m³
Fuel consumption
- main engine only 38.45tonnes/day

Classification ......Lloyd's Register of Shipping +100A1, Bulk Carrier, 'Strengthened for Heavy Cargoes, Holds Number 2, 4, 6 may be Empty, ESN, LI, ESP, UMS, SCM
Percentage of high-tensile steel used in constuction ....approx 45%
Main engine
- Design ....MAN B&W
- Model 5S60MC-Mk6
- Manufacturer ....Hudong Heavy Machinery
- Number 1
- Output 10,200kW/105rev/min

Propeller
- Material ....Nickel-aluminium-bronze
- Manufacturer....................Dalian Marine Propeller Works
- Number 1
- Diameter 6760mm
- Pitch ....Fixed
- Speed..102.8rev/min

Diesel-driven alternators
- Number 3
- Engine make/type Yanmar/6N18AL-EV
- Alternator make/type Taiyo/FE547A-8
- Output 3

Boilers
- Number 1
- Type ....Mission OS1600
- Make ....Aalborg
- Output
  - oil fired...1600kg/h
  - exhaust gas...1300kg/h

Mooring equipment
- Number ....2 x mooring winch/windlass, 2 x mooring winch
- Make...Luzhou-Hatlapa
- Type ....Hydraulic

Hatch covers
- Make...MacGregor
- Type ....Side-rolling

Complement
- Officers...8
- Crew...15
- Spare..6

Ballast control system
- Make ....SSSR
- Type ....Remote air/electric control

Bridge control system
- Make..Lyngsø
- Type DMS-21001

Fire protection system
- Make ....Salwico
- Type ....C316 smoke detector

Fire extinguishing system
- Cargo holds ....CO_2
- Engineroom ....CO_2
- Make...Unitor

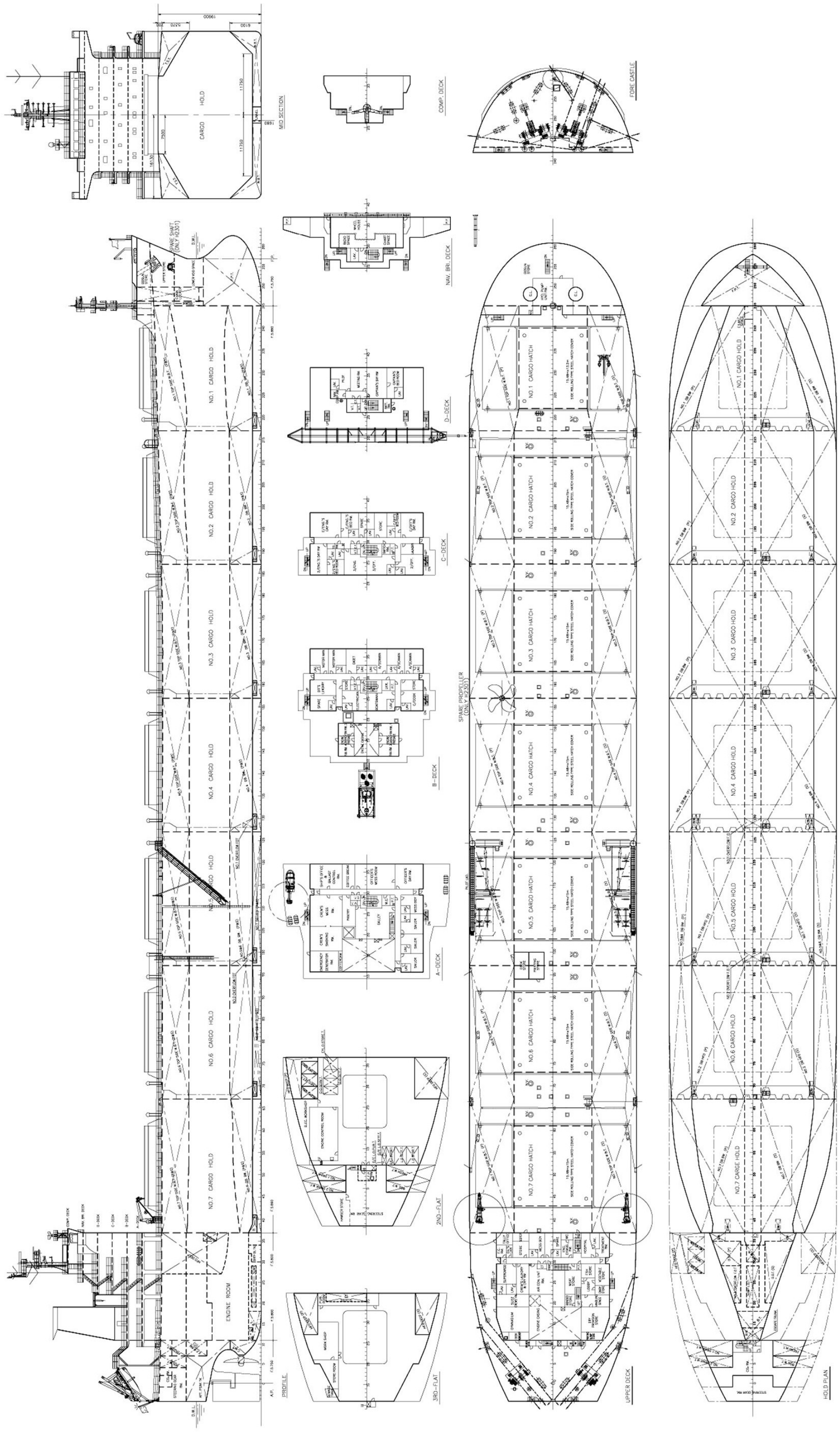
Radar
- Number 2
- Make ....Japan Radio Co
- Models..1 x JMA-9932-SA, 1 x JMA-9922-9XA

Satellite navigation system
- Make..Japan Radio Co
- Models..2 x JLR-7700 MkII

Waste disposal system
- Incinerator
  - Make ....Luzhou
  - Model..OGS-200C

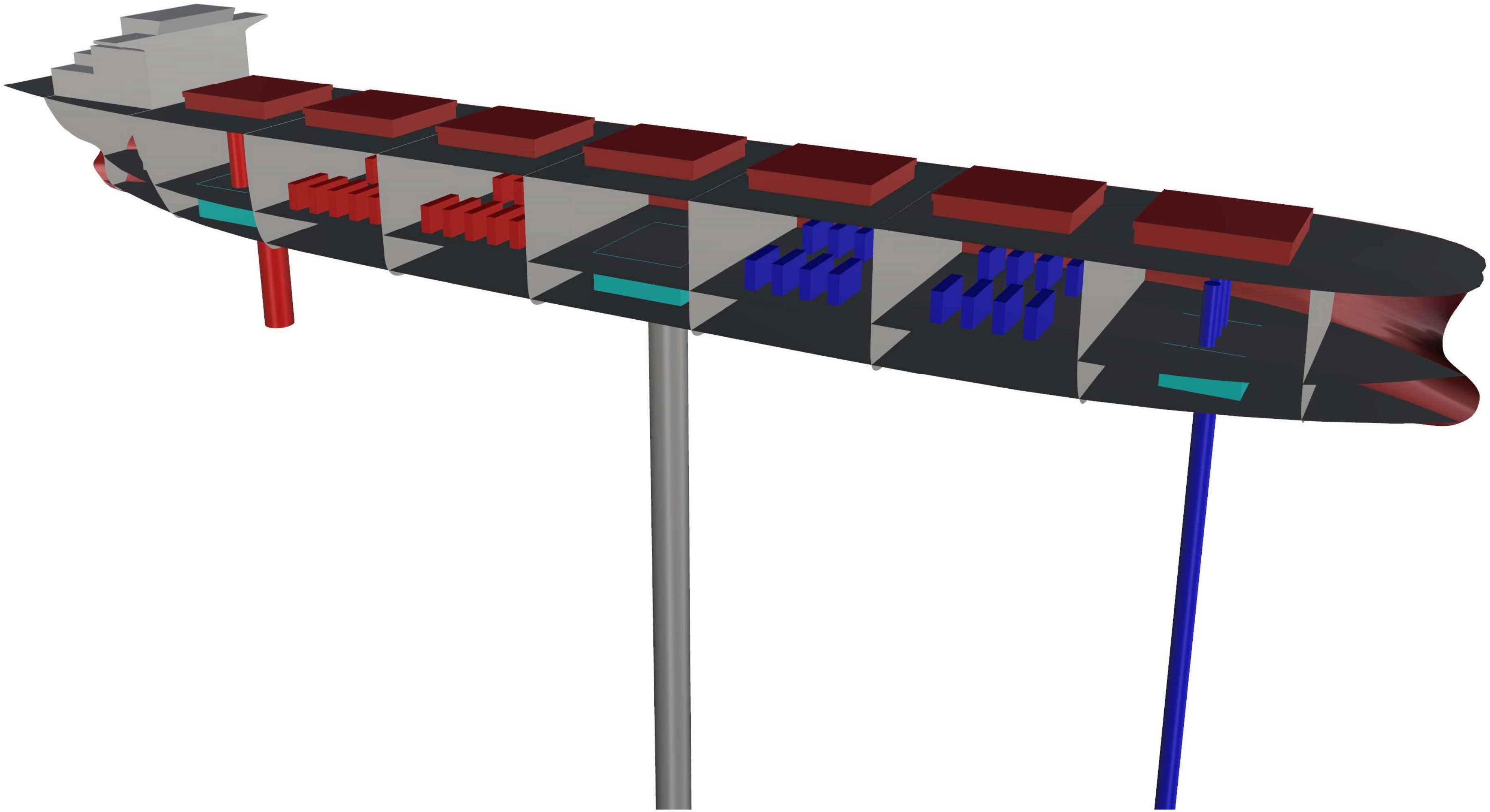
Contract date ....24 December 2002
Launch/float-out date.....................................28 March 2004
Delivery date ....31 August 2004

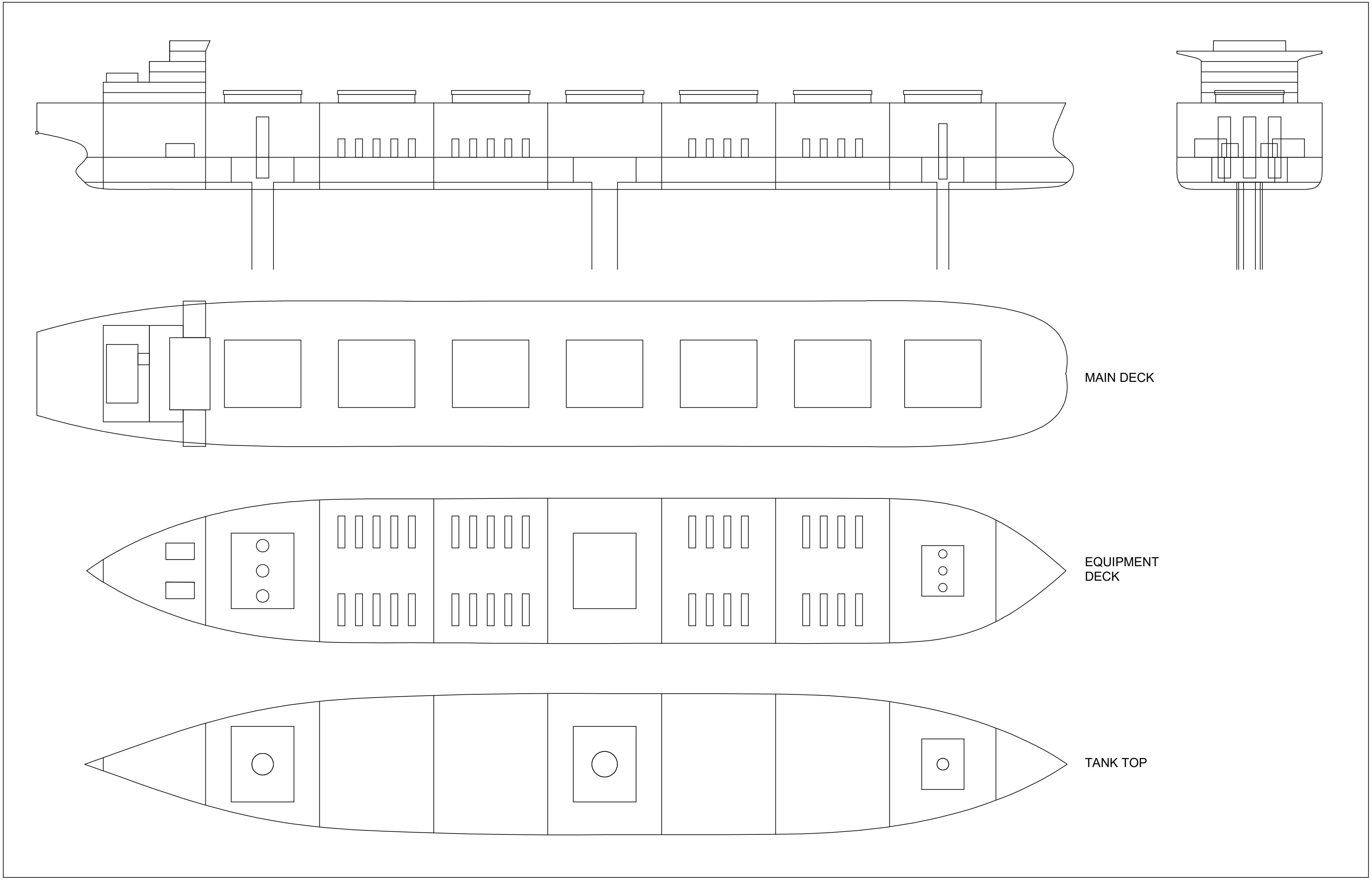
PROTEFS



F

General Arrangement




MAIN DECK
EQUIPMENT DECK
TANK TOP

Bibliography

[1] Alphaliner, The Worldwide Reference in Liner Shipping. URL https://www.alphaliner.com/.

[2] Bluerise - Harnessing the ocean's power. URL http://www.bluerise.nl/.

[3] Sea water temperature. Water surface temperatures on the coast of the oceans and seas. URL http://seatemperature.info/.

[4] Large Turret. *J. Offshore Technol.*, (December):16–19, 2003.

[5] World's Deepest FPSO Heads To Start Production | Offshore Post. 2015. URL http://www.offshorepost.com/worlds-deepest-fpso-heads-to-start-production/.

[6] D Acevedo, R Blokker, P Dinnissen, J Kirkenier, and B J Kleute. General arrangement of a 10MW offshore OTEC plant. Technical report, Bluerise BV, Delft, 2015.

[7] Aziz Ahmed, Khondaker Sakil. Ahmed. *Floating Production Storage and Offloading*. LAP LAMBERT Academic Publishing, 2010. ISBN 978-3-8383-3893-4.

[8] Alfa Laval. Plate Heat Exchanger (PHE) - Operational and Maintenance Manual. Technical report.

[9] ANAVE (Association of Spanish Shipowners). Merchant Marine and Maritime Transport 2016 - 2017. 2017.

[10] Christian Appendini, Alec Torres-Freyermuth, Fernando Oropeza, Paulo Salles, José López, and E. Tonatiuh Mendoza. Wave Climate and Trends for the Gulf of Mexico: A 30-Yr Wave Hindcast. 2013.

[11] C. Avery, W.H. ; Wu. *Renewable Energy from the Ocean: A Guide to OTEC*. Oxford University Press, New York, 1994. ISBN 0195071999.

[12] James P. Bailey. Effect of Size of Hatches on Torsional Strength and Rigidity. pages 31–35, 1959.

[13] C.B. Barrass. *Ship Design and Performance for Masters and Mates*. Elsevier, 2004. ISBN 978-0-7506-6000-6.

[14] BIMCO. The Shipping Market in 2016 and Looking Forward, 2017. URL https://www.bimco.org/news/market{_}analysis/2017/20170103-the-shipping-market-in-2016-and-looking-forward.

[15] Bluerise. Offshore OTEC; Feasibility study of a 10 MW installation. (March):63, 2014.

[16] Bluewater. Mooring systems – Bluewater. URL http://www.bluewater.com/products-technology/mooring-systems/.

[17] Aker Bo, Fluminense Fpso, Song Doc, Pride Mv, Aker Bo, Knarr Field, White Rose, and Modec Avery Laurence. Looking forward. 2016.

[18] Ingrid Bouwer Utne and Magnus Rasmussen. Reliability, Availability, Maintenance and Safety (RAMS) in Design and Operation of Marine Systems. Norwegian University of Science and Technology (NTNU), 2014.

[19] Martin G. Brown. FPSO Technology Applied to OTEC, 1998. URL http://www.clubdesargonautes.org/otec/vol/vol9-2-1.htm.

[20] Bas Buchner. Green Water From the Side of an Weathervaning FPSO. *Proc. OMAE99, 18th Int. Conf.*, pages 1–11, 1999.

[21] Bureau Veritas. Rules for the Classification of Steel Ships. Technical Report 467, 2016.

[22] Subrata Chakrabarti. *Handbook of Offshore Engineering*. Elsevier, 2005. ISBN 978-0-08-044381-2.

[23] Soumya Chakraborty. Understanding Design Of Container Ships, 2016. URL `http://www.marineinsight.com/naval-architecture/understanding-design-of-container-ships/`.

[24] Soumya Chakraborty. The Dangers of Ro-Ro Ship Design: A Naval Architect's Perspective, 2016. URL `http://www.marineinsight.com/naval-architecture/ro-ro-ship-design-dangers/`.

[25] Soumya Chakraborty. Understanding Design Of Bulk Carriers, 2017. URL `http://www.marineinsight.com/naval-architecture/understanding-design-bulk-carriers/`.

[26] Soumya Chakraborty. Understanding Design Of Oil Tanker Ships, 2017. URL `http://www.marineinsight.com/naval-architecture/oil-tanker-ships/`.

[27] P D Contraros and D Whittaker. In Service Container Carriers Converted to General Cargo Vessels Classed to Carry High Density Break Bulk Cargoes Structural and Operational Risks. 277(85): 1–11.

[28] Joseph J. Cunningham, Zachary E. Magdol, and Nancy E. Kinner. Technical Readiness of Ocean Thermal Energy Conversion (OTEC). page 246, 2010. URL `http://coastalmanagement.noaa.gov/otec/docs/otectech1109.pdf`.

[29] A. Devis-Morales, R.A. Montoya-Sánchez, G. Bernal, and A.F. Osorio. Assessment of extreme wind and waves in the Colombian Caribbean Sea for offshore applications. *Appl. Ocean Res.*, 69:10–26, 2017. ISSN 01411187. doi: 10.1016/j.apor.2017.09.012. URL `http://linkinghub.elsevier.com/retrieve/pii/S014111871730250X`.

[30] Henrique M Gaspar, Donna H Rhodes, Adam M Ross, and Stein Ove Erikstad. Journal of Ship Production and Design Addressing Complexity Aspects in Conceptual Ship Design : A Systems Engineering Approach. 28(4):145–159, 2012. ISSN 21582866.

[31] GB Marine. Projects » GB MARINE. URL `http://www.gbmarine.com/projects?tab=4`.

[32] Subhodeep Ghosh. Learn Ship Design: Bulk Carriers (A Detailed Synopsis), 2015. URL `http://lshipdesign.blogspot.nl/2015/04/bulk-carriers-detailed-synopsis.html`.

[33] Haeseung Hitech. New Principle of Disconnectable Spread Mooring (DSM) and Submerged Mooring Pontoon (SMP), 2016. URL `http://www.ekomeri.com/submerged-mooring-pontoon/`.

[34] Helen Knight. 20,000 megawatts under the sea: Oceanic steam engines | New Scientist, 2014. URL `https://www.newscientist.com/article/mg22129580.900-20000-megawatts-under-the-sea-oceanic-steam-engines/`.

[35] Hellenic Shipping News. Tanker shipping: A strong season lifts crude oil tankers before it is expected to hit the fan in 2017, 2017. URL `http://www.hellenicshippingnews.com/tanker-shipping-a-strong-season-lifts-crude-oil-tankers-before-it-is-expected-to-hit-the-fa`

[36] G. Boyd Howell, Arun S. Duggal, Caspar Heyl, and Oise Ihonde. Spread Moored or Turret Moored FPSO's for Deepwater Field Developments. *Procedings Offshore West Africa 2006*, pages 1–21, 2006.

[37] R P M Hudson and D Webster. A Survival Guide For SYSTEMS Engineering in Ship Design Projects. (March), 2012.

[38] Hydraulic Institute. Pump Intake Design, 1998.

[39] IMF. *World Economic Outlook: Gaining Momentum?* Number April. 2017. ISBN 978-1-47559-214-6.

[40] International Maritime Organization (IMO). A.749(18) Ch3 - Design criteria applicable to all ships.pdf.

[41] International Maritime Organization (IMO). *International Convention for the Safety of Life at Sea (SOLAS)*. 1974.

[42] International Maritime Organization (IMO). Annex I of MARPOL 73/78: Regulations for the Prevention of Pollution by Oil. 2000.

[43] International Maritime Organization (IMO). International Code on Intact Stability. 267 (December), 2008.

[44] IRENA(International Renewable Energy). Ocean Energy:Technology Readiness, Patents, Deployment Status and Outlook. *Int. Renew. energy agency*, (August):76, 2014. doi: 10.1007/978-3-540-77932-2. URL http://www.irena.org/DocumentDownloads/Publications/IRENA{_}Ocean{_}Energy{_}report{_}2014.pdf.

[45] ISL Bremen. Shipping Statistics And Market Review 2017. 61(1), 2017.

[46] Tony Youngjoo Jarng. Appratus for Mooring Floater Using Submerged Pontoon, 2015.

[47] Edward H. Ryan Joanna Gyory, Arthur J. Mariano. The Caribbean Current. Ocean Surface Currents. URL http://oceancurrents.rsmas.miami.edu/caribbean/caribbean.html.

[48] Raunek Kantharia. What are Ro-Ro Ships?, 2016. URL http://www.marineinsight.com/types-of-ships/what-are-ro-ro-ships/.

[49] Stephen Kibbee. Ocean Thermal Energy Conversion at SBM. *Proc. 18th Offshore Symp.*, February (Figure 2):3–18, 2013.

[50] J. A. Kirkenier. Techno-economic optimization of Organic Rankine Cycles using working fluid mixtures for 10 to 25 MW OTEC power plants. Technical report, Bluerise BV, Delft, 2014. URL http://www.bluerise.nl/.

[51] B J Kleute. System sensitivity and cold water pipe analysis of a 10 MW offshore OTEC - ocean thermal energy conversion plant. 2014.

[52] J E Kokarakis, V Dimoulas, and E Kariambas. Technical Challenges in Tanker Conversions. pages 1–12, 2008.

[53] L K Kupras. Optimization Method and Parametric Study in Precontracted Ship Design. *Int. Shipbuild. Prog.*

[54] H.M.L. Lems-de Jong. *Ocean current patterns and variability around Curaçao for Ocean Thermal Energy Conversion*. PhD thesis, 2017.

[55] Bob Lightsey. Systems Engineering Fundamentals. *Development*, (January):223, 2001. ISSN 1872-7565. doi: 10.1016/j.cmpb.2010.05.002. URL http://www.dtic.mil/docs/citations/ADA387507.

[56] Clarkson Research Services Limited. Shipping Review Database. 2017.

[57] Lloyds Register-Fairplay. World Fleet Statistics. 2017.

[58] Lockheed Martin. NAVFAC Ocean Thermal Energy Conversion (OTEC) Project. page 274, 2011.

[59] Luis A. Vega and Dominic Michaelis. First Generation 50 MW OTEC Plantship for the Production of Electricity and Desalinated Water. *Offshore Technol. Conf.*, (OTC 20957):1–17, 2010. ISSN 01603663. doi: 10.4043/20957-MS.

[60] A D Pasek M Arifin12. Design of Radial Turbo-Expanders for Small Organic Rankine Cycle System. *IOP Conf. Ser. Mater. Sci. Eng.*, 88:1–10, 2015. ISSN 1757-8981. doi: 10.1088/1757-899X/88/1/012037.

[61] MAN Diesel & Turbo. This Is a Headline Alpha. Technical report, 2013.

[62] S.L. Marshall, R.W.; Smith. UKOOA Fpso Design Guidance Notes for Ukcs Service. Technical Report March, 2002.

[63] Albert Martis, Geert Jan Van Oldenborgh, and Gerrit Burgers. Predicting rainfall in the Dutch Caribbean - More than El Ni??o? *Int. J. Climatol.*, 22(10):1219–1234, 2002. ISSN 08998418. doi: 10.1002/joc.779.

[64] Angus Mather. *FPSO Handbook*. Witherby Seamanship International Ltd, first edit edition, 2009. ISBN 978 1 905331 57 4.

[65] I May. MI L-STD-499A (USAF) Engineering STANDARD Management FSC - Ml SC . *Engineering*, 99, 1969.

[66] J. Melorose, R. Perroy, and S. Careas. World population prospects. *United Nations*, 1(6042):587–92, 2015. ISSN 1098-6596. doi: 10.1017/CBO9781107415324.004. URL `http://www.ncbi.nlm.nih.gov/pubmed/21798940`.

[67] Meteorological Department Curacao. Hurricanes and Tropical Storms in the Dutch Caribbean. 2013.

[68] Nao Nakanishi. Demand from China spurs conversion of tankers into bulk carriers - The New York Times, 2007. URL `http://www.nytimes.com/2007/08/30/business/worldbusiness/30iht-chiships.1.7317779.html`.

[69] NAVIONICS. Navionics Webapp. URL `https://webapp.navionics.com/?lang=es{#}boating@6{&}key={_}io{%}7CHgbuY`.

[70] G. C. Nihous and L. A. Vega. Design of a 100 MW OTEC-hydrogen plantship. *Mar. Struct.*, 6 (2-3):207–221, 1993. ISSN 09518339. doi: 10.1016/0951-8339(93)90020-4.

[71] Leon Osborne, Jeffrey Brummond, Robert Hart, Mohsen Zarean, and Steven Conger. Clarus: Concept of Operations. (October):123, 2005.

[72] Harold D Ramsden. Conversion of the USNS Chepachet to the OTEC- 1 Platform. 20(1):64–70, 1983.

[73] David C. Riggs. Sealed Gimbal for Ocean Thermal Energy Conversion Cold Water Pipe, 2014. ISSN 2005012173. URL `https://doc-08-0k-apps-viewer.googleusercontent.com/viewer/secure/pdf/i4el8f5c2c9lkdr3gltlr53sertjrlka/k5durnj38a4ouqhe6n4b7i5hhc0ufplj/1485767925000/lantern/02038447994795582844/ACFrOgDQeljxTNpdqg0dpsYGlMfhCOelBzEQu35zNEVazj84NKyMpbl{_}DAJjl4s-4CQPR7Vw{_}wpX`.

[74] Peter Sand. Fleet Growth Squeezes Crude Oil Tanker Market, 2016. URL `https://www.bimco.org/news/market{_}analysis/2016/1222{_}fleet{_}growth{_}squeezes{_}crude{_}oil{_}tanker{_}market`.

[75] Peter Sand. Is the Dry Bulk Industry Still on Track for Profitability in 2019?, 2017. URL `https://www.bimco.org/news/market{_}analysis/2017/20170317-dry-bulk-industry`.

[76] Peter N. Smith. Highway stormwater pump station design. *Hydraul. Eng. Circ. No. 24*, (24), 2001.

[77] Ibrahim R Teaima. Application of Modern Technique to Predict and Enhance Flow Problems on Pump Suction Intake for Irrigation Improvement Projects. *Extensive J. Appl. Sci.*, 1(2):31–42, 2013.

[78] Nadia A Tepper. Master's Thesis Exploring the use of Model-Based Systems Engineering (MBSE) to develop Systems Architectures in Naval Ship Design. *Distribution*, 5(831), 2010.

[79] The Engineering Toolbox. Sea Water Properties. URL `http://www.engineeringtoolbox.com/sea-water-properties-d{_}840.html`.

[80] The Royal Institution of Naval Architects. Significant Ships of 2004. 2004.

[81] (UN-DESA) United Nations Department of Economic and Social Affairs. Genuine and durable partnerships for Small Island Developing States: Sustainable Development Knowledge Platform, 2015. URL `https://sustainabledevelopment.un.org/?page=view{&}nr=241{&}type=230{&}menu=2059`.

[82] Vladimir Soldatkin. Rania El Gamal & Alex Lawler. OPEC, non-OPEC agree first global oil pact since 2001 | Reuters, 2016. URL http://www.reuters.com/article/us-opec-meeting-idUSKBN13Z0J8.

[83] D. C. M. Watson and a. W. Gilfillan. Some Ship Design Methods. *Trans. R. Inst. Nav. Archit.*, 119:279–302, 1977.

[84] D.G.M. Watson. *Practical Ship Design*, volume 1. 1998. ISBN 0080429998. URL http://www.amazon.co.uk/Practical-Design-Elsevier-Ocean-Engineering/dp/0080429998.

[85] J Wichers. *Guide to Single Point Moorings*. WMooring, 2013. ISBN 1482649756, 9781482649758.

[86] Wikipedia. Handymax. URL https://en.wikipedia.org/wiki/Handymax.