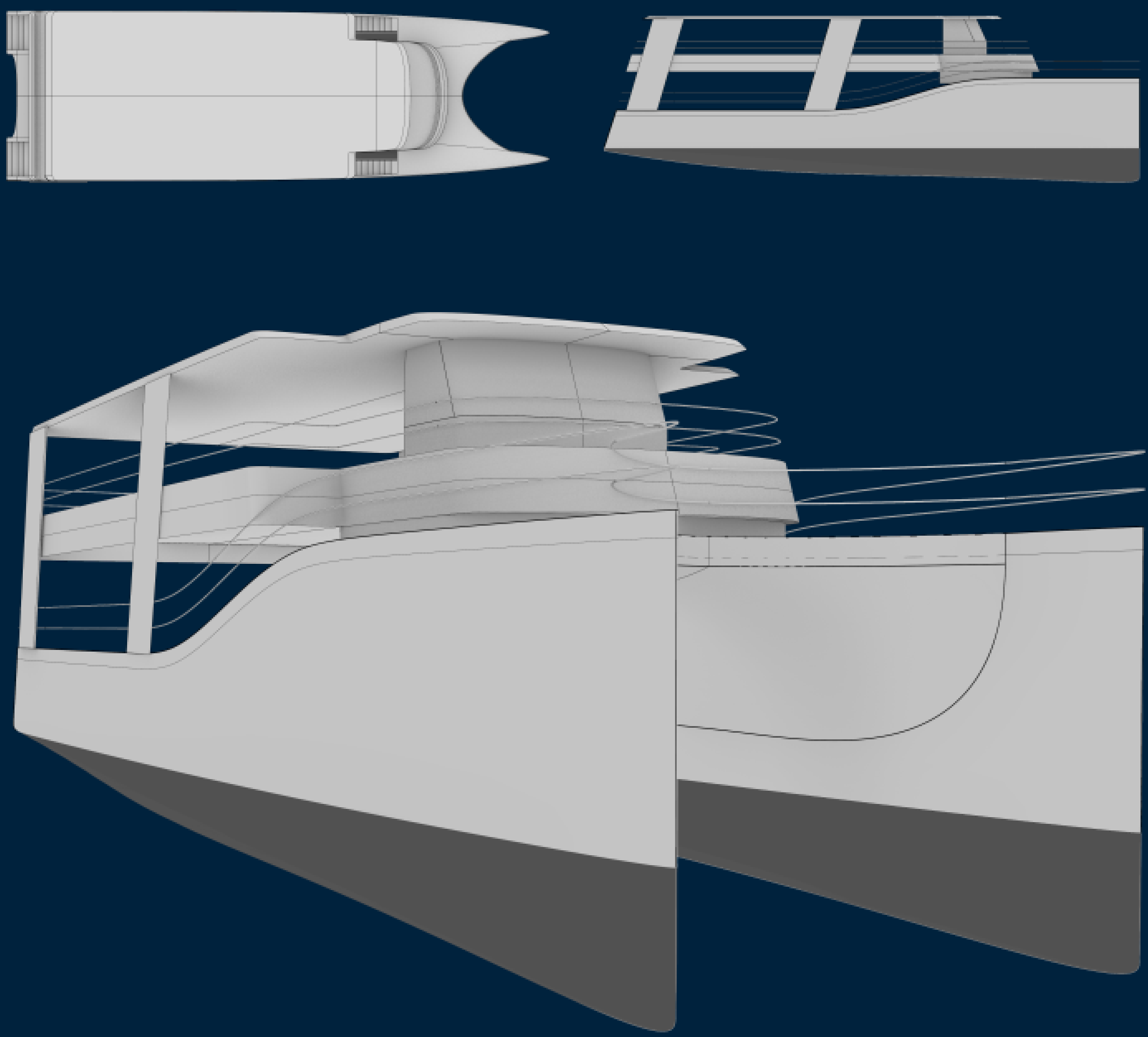
Design of a Hydrodynamically Optimized and Battery-Powered Passenger Vessel for the Caribbean Sea around Curaçao

L.E. Vliex



Delft University of Technology

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Thesis for the degree of MSc in Marine Technology in the specialization of Ship Design

Design of a Hydrodynamically Optimized and Battery-Powered Passenger Vessel for the Caribbean Sea around Curaçao

By

L.E. Vliex

Performed at

Mermaid Boat Trips

This Thesis (MT.25/26.015.M) is classified as confidential in accordance with the general conditions for projects performed by the TUDelft.

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Preface

During the beginning of my masters, I felt I lost touch with what I thought really mattered for my degree. In my hunt for a subject for my thesis, I was looking for a project that was actually going to be built. I think the thought of being able to walk around on something you designed is what sparked my interest in naval architecture in the first place. By picking a subject where I could integrate all aspects of my studies into one final assignment is when everything started to make sense again, I'm honoured for the chance I was given in this master thesis.

It's been great to see my skills develop during this thesis. Complex design assignments such as ship design are surrounded by a lot of noise, conflicting requirements and annoying physical constraints. Surrounding yourself in such an atmosphere can be chaotic at first, but this is necessary to find out what really matters. Perhaps the biggest personal finding was that, before my thesis, the term Naval Architect never made sense to me. I always thought, we are maritime engineers; what does architecture have to do with it? But after a few months of thesis, I finally understood what it meant: a (naval) architect makes decisions because of a multitude of reasons: some technical, some aesthetic, and some choices are made simply because one needs to pick a starting point. I feel this thesis has been a great kickstart for helping me understand Naval Architecture better. To be continued.

First and foremost, I want to thank the company of Mermaid Boat Trips for the opportunity, confidence, and freedom they have given me in this design. For flying me out to Curaçao to get a real feel for the design case and accommodating me on the island. I would also like to thank the Tyler Anthony for his patience in showing me the practical side of a ship operator on Curaçao. And I would like to thank Travis for helping me for all his time and effort supporting this thesis. Providing me with a realistic view on the matter and being a good critic on my work. Due to Travis' effort, I also gained renewed passion for naval architecture and felt a lot more confident in my studies again. I would also like to thank the Anthony family, for adopting me as their third (and possibly favourite) son into their home, and for teaching me what the right end of a screwdriver is.

Sanne van Essen, my hydrodynamical supervisor, I want to thank for arranging Marin software to calculate ship motions, and the subsequent (hours of) guidance that followed. It must've been annoying at times for me to ask unstructured questions all the time. Luckily asking these questions helped structure my thoughts. I would also like to thank her colleagues in providing necessary aid.

Jaap Gelling, my daily supervisor, deserves a lot of credit for his view on the thesis. I would like to thank him for the mountain of ship design knowledge to lean on. A good fall back to test my ideas against and provide a critical view on the thesis as a whole.

I would like to thank the Study Association of the S.G. "William Froude" for being the social safe haven where I had the opportunity to develop various skills alongside my studies. And all the friends this has brought. The skills I learned here will bring a lot of value to the engineer I have become.

I would like to thank my flatmates Sil, Lars & Harm for helping provide a home with good distraction from studies, and also for providing critical feedback on this research. And finally, I would like to thank my family. My parents, Juke & Eugène, for their unconditional support during my thesis and the whole of my studies. My grandmother, Thea Oma, for always burning candles as support. And my sister Inge and cousin Bastiaan.

Thank you.

Loek Vliex
Delft, January 2026

Abstract

This work serves as the initial base for a new vessel for Mermaid Boat Trips, undertaking day trips to the island of Klein Curaçao. The research aims to overcome two significant challenges: 1) Improving seakeeping, as the current vessel has too many passengers who suffer from seasickness, and 2) improving the sustainability of the company's operations through battery propulsion. It is noted that ship design is a complex, multifaceted field with many contradictory and interdependent aspects. A few significant findings could be concluded: 1) A good frame of reference is needed in a project with this many unknowns. 2) A result can be achieved where less seasickness is achieved relative to the baseline, partly by applying slender catamaran hulls with axebows, and in part by reducing the ship speed from 17 to 13 knots (while in rough seas). 3) The ship structures and consequent mass play an essential role in the effectiveness of battery-powered vessels: the vessel needs to be as lightweight as possible to save on battery weight and cost. Further work is required to find true vessel motions (non-linear) and resistance in waves, to proceed to further iterations and detail engineering. It can be said with reasonable confidence that a solid foundation is laid in the design for a new vessel for Mermaid Boat Trips that mitigates the effects of seasickness and operates sustainably.

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Nomenclature

Symbols & Acronyms

$_M$	Subscript denoting this particular at Model Scale
$_S$	Subscript denoting this particular at Full Scale
A	Wetted Surface Area [m^2]
B	Beam overall [m]
b	Beam demihull (beam of óne of the catamaran hulls) [m]
C_f	Frictional Coefficient [-]
C_R	Residuary Resistance Coefficient [-]
C_T	Total Resistance Coefficient [-]
D	Moulded Depth [m]
E	Equipment Numeral [-]
E_S	Structural Numeral [-]
Fn	Froude number [-]
g	Gravitational acceleration (9.81) [m/s^2]
GT	Gross Tonnage [-]
H_L	Wet Deck Height [m]
H_S	Significant Wave Height [m]
I_F	Interference Factor [-]
I_{44}	Moment of inertia for roll [ton·m^2]
I_{55}	Moment of inertia for pitch [ton·m^2]
I_{66}	Moment of inertia for yaw [ton·m^2]
K	Structural Weight Constant [-]
K_{OT}	Outfitting Constant [-]
K_{xx}	Roll radius of gyration [m]
K_{yy}	Pitch radius of gyration [m]
K_{zz}	Yaw radius of gyration [m]
KG	Vertical center of Gravity
L	Waterline Length [m]
L/B	Length-over-Beam ratio [-]
LCB	Longitudinal Centre of Buoyancy [m]
LCG	Longitudinal Centre of Gravity [m]
P_D	Design Power [kW]
P_E	Effective Power [kW]
q_{DH}	Deckhouse Density [kg/m^3]
R	Resistance [N] or [kN]
Re	Reynolds Number [-]
S	Seperation distance between center of the two demihulls [m]
S/L	Separation-over-Length ratio [-]
S_R	Structural Area Numeral [-]
T	Draught [m]
t	Trust deduction factor [-]
T_B	Draught Bow [m]
T_P	Peak period [s]
U	Speed [m/s]
V	Speed [m/s] or [knots]
V_{DH}	Volume Deckhouse [m^3]
w	Wake factor [-]
W_{100}	Hull Structure Weight [tonnes]
W_{150}	Hull Superstructure Weight [tonnes]
W_{OT}	Weight of Outfit [tonnes]

CFD	Computational Fluid Dynamics
COB	Centre of Buoyancy
COG	Centre of Gravity
CW	Calm Water
DOD	Depth of Discharge
EOL	End of Life
FRP	Fibre Reinforced Plastic
ITTC	International Towing Tank Convention
LS	Lightship
MSI	Motion Sickness Incidence
RAO	Response Amplitude Operator
RMS	Root Mean Square
SCV	Small Commercial Vessel Code amended for the Caribbean [40]

Greek Symbols

Δ	Displacement [kg] or [tonnes]
η_D	Propulsive efficiency [-]
η_H	Hull efficiency [-]
η_O	Open water propeller efficiency [-]
η_R	Relative rotative efficiency [-]
λ	Scale Factor [-]
λ	Wave length [-]
∇	Displacement [m^3]
ω	Wave frequency [rad/s]
ρ	Water density [kg/m^3]

1 Introduction

AI Statement

During the preparation of this work, the author used ChatGPT, Perplexity.ai, and Consensus.app to gather and process references and to perform basic grammar and spelling checks. It has also been used as a coding aid for data processing. Grammarly has been used as an in text grammar correction tool. After using these tools, the author reviewed and edited the content as needed and takes full responsibility for the content of the work.

Introduction

Seasickness can ruin your time at sea. In the tourism industry, where passengers are not used to the harsh environment of the open ocean, minimising seasickness should be the top priority. Symptoms can range from mild discomfort to, in the worst case, emesis (vomiting). This seasickness can significantly diminish the client's overall travel experience and the operator's reputation. All ships from five different companies that make the crossing from Curaçao to Klein Curaçao have clients suffering from seasickness. Mermaid Boat Trips, which operates six times per week, aims to reduce seasickness and improve the sustainability of its operations.

This research will provide a preliminary ship design for Mermaids use case. This design can be used by either a shipyard or an engineering firm for the next phase of the design. The research gap in this work lies in the integration of multiple factors into a working system. It's a combination of minimizing seasickness in rough sea states whilst also operating sustainably. Currently, there exists no ferry service between the ABC-islands (Aruba, Bonaire, and Curaçao), due to the rough seas. A ship design that can operate from Curaçao to Klein Curaçao sustainably without seasick passengers can serve as a proof of concept for this broader region of the Caribbean.

The distance the ship must travel to Klein Curaçao is approximately 17 nautical miles. In the morning, the ship faces headwind and waves, and on the way back, these waves come from the stern.

Conditions & Trip Specifications:

- Return trip distance: $\approx$ 34 nautical miles
  - Outbound head wind: 15–25 kts
  - Outbound head waves: $H_s \approx 1.5\,\mathrm{m}$
- Water depth:
  - Max draft of 2 metres allowed in port
  - Water depth at sea 500+ metres
- Speed: 17 kts
- 200 Passengers
- Sailing battery-electric, charging 1x per day

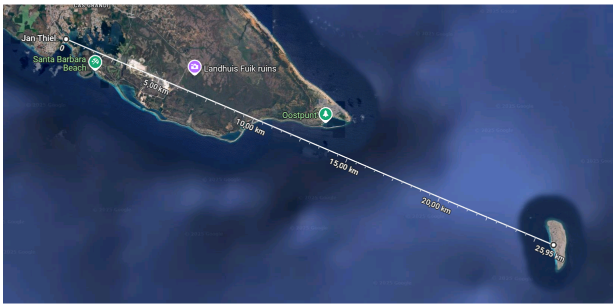


Figure 1.1: Overview of navigational area

This work begins with Chapter 2, which reviews the relevant literature and the preliminary design decisions. After this, the Design Philosophy and Approach will be discussed in Chapter 3, in this chapter a Design Spiral will be introduced which will form the backbone of this research. In the first iteration in Chapter 4 the requirements will be condensed into a first design and the first seakeeping results will be produced. In Chapter 5 some missing information will be filled in to validate the particulars of the design. All this information will finally be condensed into a final ship design in Chapter 6 and tested for seakeeping behaviour. In Chapter 7, the further work and current limitations of the research will be discussed. Finally, in Chapter 8, a general conclusion will be drawn.

2
Review of Relevant Literature & Preliminary Decisions

In this chapter, relevant literature regarding the *Design of a Hydrodynamically Optimised and Battery-Powered Passenger Vessel for the Caribbean Sea around Curaçao* will be reviewed. It includes literature and findings on: Design, Catamarans vs. Monohulls, Seakeeping Hydrodynamics, Ship Resistance, the Wave Spectrum around Curaçao, Powering & Battery Propulsion and the lessons learned from the current fleet. As this work encompasses the complete design of a vessel, the scope is broad and includes some necessary preliminary decisions regarding the ship's design and research methodology.

2.1. Design

Vossen et al. [45] devised that ship design has to be a balance between Available Technology, Operational Requirements, External Requirements (Rules), and Commercial aspects influencing the ship design. To approach this ship design, there are several different methods, each with its own advantages and disadvantages. As Andrews [1] describes, it is a wicked problem that does not comfortably fit within the engineering science framework, and Papanikolaou [35] described it as a holistic problem (i.e., a problem in which the whole system must be considered rather than its individual parts). To tackle this problem, the design spiral was introduced by Evans [11]. The key takeaway from spiral-based design is that a spiral has to be tweaked and changed for the design problem at hand, as it is not a one-size-fits-all solution. In each ship design, the influences of specific requirements differ. There are also models under the name of Multi-Objective-Optimisation (MOM) that can parametrically optimise ship designs in rough iterative states. These models, however, are time intensive to be constructed from scratch. So although spiral-based design might be less locally optimised, this does provide a better overview and understanding of the system as a whole for the designer.

2.2. Hull Shape Decision

An important *Design Decision* is the hull shape, driven by many factors, some more practical, some more academic. The two main options are a monohull and catamaran. Where a relatively long and slender monohull could have favourable seakeeping characteristics, a shorter catamaran on two slender hulls could have more favourable resistance characteristics. The decision was made to **design a catamaran**, as it provides greater surface area for transporting passengers within a given ship length. The space advantages of a catamaran are made visible in Figure 2.1. The "sense of spaciousness" may also reduce the risk of seasickness. There is also a boundary on the ship length that can be built in this design case, one 'softer' boundary driven by regulations (24 metres, SCV code), and a more practical boundary at around 30 metres. A 30-metre monohull is simply less spacious. Furthermore, a catamaran with its two propellers spaced apart can be more manoeuvrable. And as most vessels in this market are catamarans, this gives way to presumably to a better and more developed product.

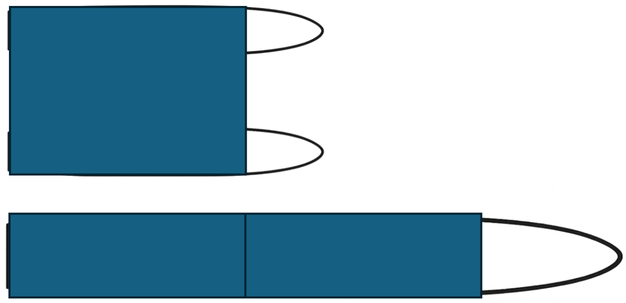


Figure 2.1: Area-based comparison between a mono and multihull (total size of blue areas is equal).

2.3. Hydrodynamics — Motions

Lewis [24] stated that the primary contributor to seasickness is the vertical accelerations a ship perceives. Reducing these accelerations can improve operability and comfort. As the vessel in the design case mainly faces headwaves, research into similar design cases is reviewed. The Enlarged Ship Concept (ESC) by Keuning and Pinkster [23] proved that increasing vessel length resulted in a significant reduction of vertical accelerations as the vessel is "carried" by more waves practically speaking.

A second way to mitigate the extreme effects of head waves is a more forgiving bow shape. This forgiving bow shape took shape in the form of the Axe Bow Concept (ABC) by Keuning [22], which is characterised by a deep forefoot and an almost constant waterline area. As this bow shape reduces acceleration, motion increases: the bow pitches further and deeper into the water, thereby increasing the need for a higher freeboard to prevent green water on board. Based on practical research, the ABC increased operability [14]; therefore, it was a successful design. And its implementation to the design case of this work could potentially give fruitful results.

Motion analysis of a new design has several levels of testing. Rough estimation methods exist based on current vessels. Potential flow provides a robust (but linear) means of calculation. CFD, model testing, and finally full-scale testing are the more accurate design steps. In this research, SEACAL (SEAkeeping CALculations), a potential flow code software [27] will be used as this is deemed to provide enough accuracy at this stage of the design. SEACAL does NOT account for effects such as slamming or viscous effects. SEACAL takes the underwater ship hull geometry, wave frequencies (ω), and ship speeds (U) and computes Response Amplitude Operators (RAOs) in the frequency domain. The applicable solution method is the **Rankine** source method, which provides the advantage of calculating the interaction effects between the two catamaran hulls [27]. With the sea states described in Section 2.3.1 these RAOs can be used to compute RMS accelerations (Root Mean Square, a statistical function to describe how a vessel responds to a certain sea spectrum) which in turn will be used to calculate the MSI values (index for level of motion sickness).

2.3.1. Wave Spectrum around Curaçao

Based on the Copernicus Database received from Marin [26], wave data in the area of operation were derived, which yielded a significant wave height (H_S) of around 1.4 metres and a peak period (T_P) of around 6.2 seconds at the time of year the sea is usually the roughest (May-June). The roughest months were taken as most seasick passenger occur at this time of year and also coincided with the experience of the company and ship crew to be the roughest months. Graphs from which this data was retrieved are available in appendix B.

2.3.2. Motion Sickness Incidence (MSI)

O'Hanlon & McCauley [33] conducted experiments in 1974 with 200 individuals to establish a baseline for when inexperienced individuals become motion-sick. Based on vertical accelerations, they presented a formula with inputs of accelerations, frequencies, and time exposed. These are still used in the industry today [10]. The formula returns the percentage of people who become physically sick (emesis/vomiting) within the specified time period. In STANAG 4154 [32], the MSI has been standardised and made describable by the formulas in appendix C. RMS accelerations calculated by SEACAL at a specific point (e.g. the bridge) can be directly plugged into these formulas.

2.4. Hydrodynamics — Resistance

Another critical factor in a vessel's success is energy consumption, especially for battery-powered vessels, as battery energy density is lower than that of conventional fuels. The problem with Catamaran Resistance is that Holtrop & Mennen isn't directly usable (because this is designed for monohulls). Catamaran hulls are usually more slender, and the interaction between the two hulls can cause an increase in resistance at certain Froude numbers [3] [44]. Strip- or potential-flow-based evaluation methods neglect viscous resistance; however, for a catamaran, this resistance plays a prominent role because of the large hull surface area. Only a few systematic ship series have been tested, such as Insell & Molland [30], but these tests were conducted on too small models, and interpolating between different hull particulars is not really a viable option. So only two possibilities remain – either CFD or model testing – both are too complex and time-intensive for the scope of this work.

In Chapter 4, a reference vessel will be chosen that has been tested both practically and numerically for resistance and motions. Although this first hull might not have the ideal particulars, it provides a good baseline.

2.5. Battery Power

Batteries provide a practical option to improve the sustainability of an operation at sea. The problem is that its energy density heavily impacts the range. The current vessel of Mermaid Boat Trips can sail a whole week on one tank of diesel. The new vessel will have to charge (at least) daily. But in terms of energy efficiency, battery electric propulsion has the lowest number of losses compared to other conventional and sustainable fuels. An electric power train has only 10-15% losses compared to a diesel power train with 40-50% losses. Therefore, íf the electric energy is provided is mostly green, AND it is technically feasible to sail on batteries then battery sailing is the most energy-efficient option.

A typical battery powertrain consists of an Energy Storage System, a DC-DC converter, a Switchboard, a Propulsion Converter (Inverter), and an Electric Motor. Although regulations are still unclear on battery-powered vessels, it is wise to have an emergency backup generator to sail to port at a few knots ship speed. The advantage of batteries is that they come in cells and these weights can be distributed in the ship's hull to change the centre of gravity (COG) and inertia of the vessel favourably.

Because of the favourable efficiency, the Technology Readiness Level of battery propulsion [25], the applicability to this range and the available infrastructure on Curaçao, battery-electric propulsion makes the most sensible sustainable energy carrier for Mermaid Boat Trips.

2.6. Lessons from the Current Fleet

The current fleet sailing to and from Klein Curaçao all perceive the same issues in one way or the other: **Slamming**. This slamming mostly occurs in the roughest months of may and june in headwaves in the morning when sailing toward Klein Curaçao.

For monohulls it is mostly bow slamming above the waterline, with sometimes bottom slamming, resulting in significant accelerations. For the catamarans, it is a mix of bottom slamming and wet deck slamming (the wet deck is the underside of the deck between the two catamaran hulls). These slamming issues are perceived to be one of the large factors contributing to sea sickness among passengers, roughly 10% has emisis on rough days. And these slamming effects also strain the vessel's structural elements.

On the way back, the vessels perceive a mix of following and stern following waves. This results in heavy rolling for the monohull vessels, but because of low accelerations, this hardly causes any seasickness. The catamarans on their return trip should have stiffer motions (and higher accelerations) due to higher transverse stability, but this also results in a stable ride and no seasick passengers. Less tangible or quantifiable factors influencing seasickness include the ability to see the horizon, the availability of fresh air, the ability to lay flat, and whether other passengers have fallen ill.

2.7. Regulations

Regulations applicable in this size and capacity range are the Small Commercial Vessel (SCV) Code amended for the Caribbean. Where the two most important regulations at this state of the design are the rules on vessel length and allowable number of passengers, as they directly influence the main particulars of the vessel. These regulations limit passenger capacity to 150 passengers and length to 24 metres (96% of Waterline Length at 0.8 Depth). If vessels are larger or carry more passengers, they *"may operate under the provisions of this Code to the satisfaction of the Administration."*. The administration in this case is the Flag State authority: Maritime Authority Curaçao (MAC). Because of exemption from this authority, it opens the option to also investigate the effectiveness of longer vessel length. This authority certifies the vessels to ensure safe operation. In the case of a newly built ship, they usually authorise a (local) classification society to certify the vessel and the production process on their behalf [34].

3
Design Philosophy and Approach

The first approach in designing a vessel of this scale is having a rough, but accurate enough, estimate for each of the vessel's particulars. It doesn't make sense to optimise one aspect of the vessel in full detail whilst other elements are still completely unknown, as these are all tied together. First, this chapter proposes a design spiral in Section 3.1 which will be followed throughout the entire work, to provide a point of reference. Consequently, the depth will be mapped in Section 3.2 and this will also say something about the approach and calculational method used. And finally in Section 3.3 the approach regarding vessel length (and the uncertainty involved) will be presented.

3.1. Proposed Design Spiral

In Figure 3.1 a first indicative design spiral is given that properly grasps the scope of this work. Here, the light blue elements are only performed in the first iteration. The first iteration will be centred around "rough" estimation methods, with the main goal to find a good "realistic" baseline to compare future iterations to. Without a means to compare vessels, it is impossible to know whether a good ship has been designed. This underscores the need for a good baseline vessel.

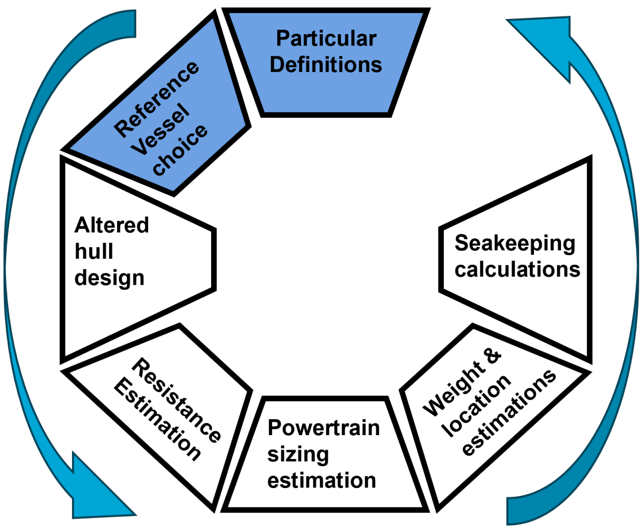


Figure 3.1: Indicative Design Spiral

This baseline vessel must have published resistance values and ideally published seakeeping properties. When this baseline is defined, the hull can be fitted with an axebow, and the implications on resistance can be calculated and translated to true vessel size. Which in turn will determine battery and powertrain size. This finally will determine ship mass and can (together with the hull shape) be used in potential flow solver SEACAL to retrieve the seakeeping properties.

3.2. Scope of the Work

The scope for this work is defined in the Figure 3.2, where calculational methods that are currently out of scope, are also the depicted. These depth "elements" range from rough estimation methods to full-scale measurements/testing. Seakeeping has the highest calculational depth, as this is considered an important factor in this design to get right. The reason potential flow is being used for seakeeping but not so much for resistance has to do with the fact that catamarans have a lot of viscous drag and interference effects which simply cannot be calculated with potential flow methods as these are non-viscous methods.

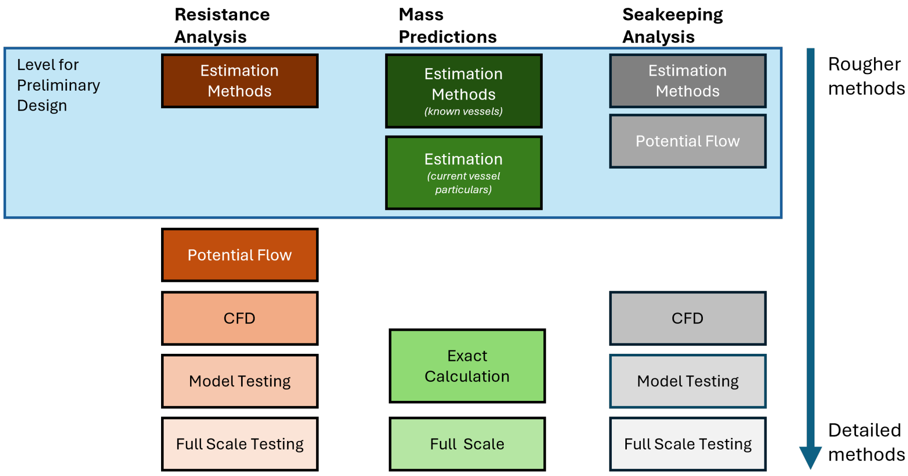


Figure 3.2: Research depth levels for this work

What remains essential during this research is incremental validation of values across references of other academic work and existing vessels. To ensure that an efficient, but more importantly, a realistic vessel is designed.

3.3. Vessel Length Optimization

The importance of vessel length was heavily underscored by its influence on vertical accelerations in head waves, as discussed in Section 2.3 of the literary work. Although permissions were acquired by local Curaçao authorities to go beyond the initial 24 metres stated by the SCV code [40], it is important to investigate the influence of these extra metres. Staying within code can have some commercial advantages. The company of Mermaid Boat Trips aspires to produce more vessels for other islands in the Caribbean, and other islands may be more stringent with these rules. Therefore, 24-metre vessel length could become a requirement. To make an informed decision regarding vessel length, the choice was made to perform the first iteration twice, for both 24- and a 30-metre vessel length. This 24-metre waterline length was chosen as this is the maximum length allowed by the rules, and the 30-metre length in cooperation with Mermaid, as this was deemed the maximum comfortable length for berthing and manoeuvring around the narrow ports and harbours. Based on the results of the first iteration, an informed decision can be made regarding what the right vessel length will be.

4

First Iteration

The chosen design method for this vessel is spiral-based design, as introduced in Chapter 3. The starting point must be a good reference from which to improve upon. Shipyards in the industry typically know what works and what doesn't work because of earlier built ships, allowing them to optimise within their domain. For this, design from scratch, it is necessary to find a similar field of reference, which makes sense within the academic scope. From here, improvements can be made, and the new vessel can be designed. This chapter **follows the structure of the design spiral** from Figure 3.1.

4.1. Particular Definitions & Assumptions

The chosen reference vessel will be scaled to the respective lengths of **24 and 30 metres.** This reference vessel will therefore be guiding in terms of other dimensions such as beam and draught for the vessel in this first iteration. For a passenger vessel, the main driver for sizing is surface area, to fit all passengers. It is deemed that for this instance, both the 24- and 30-metre vessel will provide sufficient area for the 200 required passengers.

A central **assumption** in the comparison between the two vessels is that they are both scaled up directly from the reference vessel, to keep as many variables as consistent as possible. To accurately apply Froude / ITTC scaling laws, **and to have a clear overview on *what basis* the vessels get compared.**

4.2. Reference Vessel Choice

In Chapter 2, the choice was made for a catamaran hull. The broadly tested round bilge catamaran hull form that was chosen as the frame of reference is the **Delft 372 Catamaran,** first researched by Van't Veer [44] and later by Broglia et al. [3].

Although there are broader series for round bilge catamaran hulls, these all have their downsides, as discussed in Chapter 2. Most of these works primarily focus on resistance only. In contrast, this research focuses on *resistance* and *motions*, offering a good baseline hull to improve upon. The lines plan and the specific particulars of this vessel can be seen in Figure 4.1 and Figure 4.2.

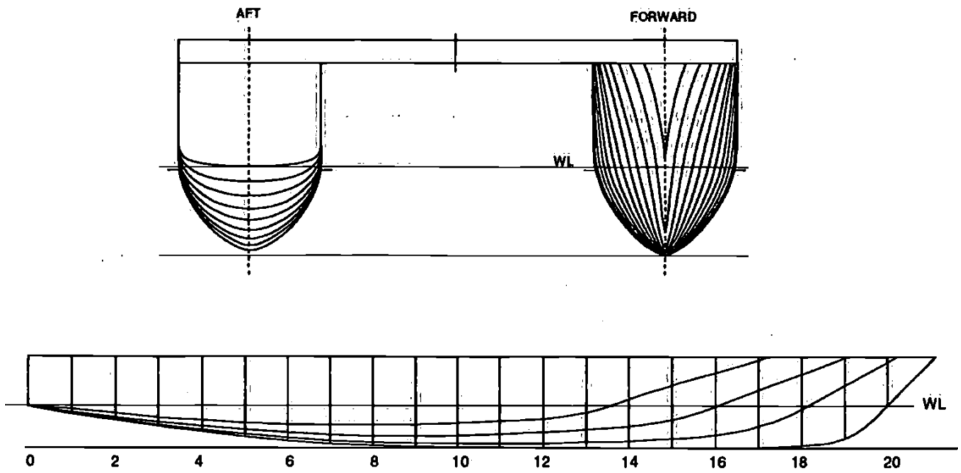


Figure 4.1: Delft 372 Catamaran linesplan from the work of Van't Veer 1998 [44]

In Section 4.3, the particulars of the reference model will be scaled to respective sizes of 24 and 30 metres. And in Section 4.4, the resistance values will also be scaled using Froude scaling to end up with a proper estimate for the resistance values of this vessel. The S/L ratio of this vessel is 0.23. It is known from other research that this hull spacing may be too narrow. But for the first iteration, these ratios were kept, as they again provide a baseline. In subsequent research on the *Delft 372* catamaran, it was found that higher S/L ratios reduce resistance [3]. This will be kept in mind for subsequent iterations if resistance reduction (or increased vessel space) is required.

Main Particulars	Symbol	Value	
Length over all	L_{OA}	3.11	m
Length between perpendiculars	L	3.00	m
Beam overall	B	0.94	m
Beam demihull	b	0.24	m
Distance between center of hulls	H	0.70	m
Draught	T	0.15	m
Displacement	Δ	87.07	kg
Draught, AP	T_{AP}	0.15	m
Draught, FP	T_{FP}	0.15	m
Vertical center of gravity	KG	0.34	m
Longitudinal center of gravity	LCG	1.41	m
Pitch radius of gyration	k_{yy}	0.782	m
Moment of inertia for pitch	I_{55}	53.245	kg m^2
Length over beam ratio	L/b	12.5	
Length over draught ration	L/T	20.0	
Block coefficient	C_B	0.403	
Mass during oscillation tests	M	62.00	kg
Moment of inertia, osc. tests	I_{55}	13.454	kg m^2

Figure 4.2: Main particulars of the Delft 372 Catamaran by Van't Veer 1998 [44]

4.3. Altered Hull Design

This linesplan from Figure 4.1 was transformed into a 3D model in CAD software (Rhinoceros), with a few important alterations to fit the design requirements:

- Bow shape changed into an axe shape to give more forgiving accelerations.
- Mid ship draught slightly reduced to keep displacement equal to the 372 catamaran.
- High increase in freeboard to counter expected increase in motions.

The first iteration of this hull, which will be equal for the 24- and 30-metre case, can be seen in Figure 4.3. The hulls in Rhinoceros are formed with the use of the *"loose loft"* function. Where the surface is formed based on polynomial curves, these curves are in turn influenced by point objects. The focus is put on as few curves and points as possible, as this yields the smoothest surfaces in this software.

Axe Bow Shape

A so-called S-keel is visible in the shape of the vessel. In later designs of axe bow vessels, the keel is usually a straight line. This is easier for docking, but to keep the displacement similar to the Delft 372 catamaran, the choice was made to have an S-shaped keel. The draught of the forefoot for this first iteration is estimated by referencing existing axe vessels, but it is known that in reality this is based off the wave height in the operational area and subsequent ship motions.

All these aspects together result in the particulars shown in Table 4.1. Take note that the information on the wetted surface area from the original DUT-372 model on wasn't available (marked N/A); therefore this had to be redetermined from the original lines plan. The Block Coefficient is underscored, as it differs for the axe bow vessel because the deep forefoot strongly influences this value. The displacement of the axe vessels differ 0.2% percent of their scaled counterparts. This forms the base for future steps where the Axe versus DUT-372 catamarans can be compared on their performance.

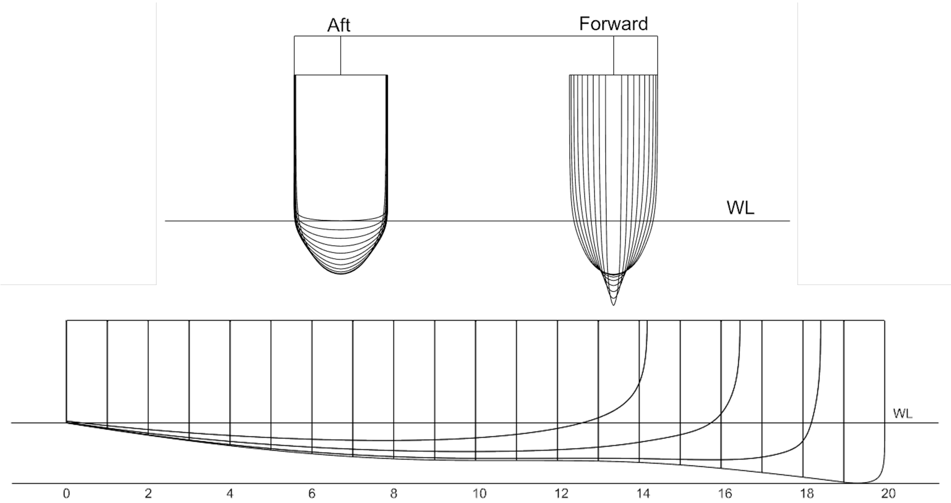


Figure 4.3: Delft 372 refitted with an Axe Bow.

	Symbol	Delft 372 Model	Delft 372	Delft 372 AXE	Delft 372	Delft 372 AXE	unit
notes			*scaled values*		*scaled values*		
Length	L	3	24	24	30	30	[m]
Beam overall	B	0.94	7.52	7.52	9.4	9.4	[m]
Beam demihull	b	0.24	1.92	1.92	2.4	2.4	[m]
Distance between centre of hulls	S (or H)	0.7	5.60	5.60	7.00	7.00	[m]
Draught (Midship)	T	0.15	1.2	1.10	1.5	1.375	[m]
Draught Bow	Tb	0.15	1.2	1.77	1.5	2.21	[m]
Displacement	∇	0.087	44.58	44.52	87.07	86.94	[m^3]
Displacement	Δ	0.087	45.69	45.63	89.25	89.11	[tonnes]
Wetted Surface Area	A	N/A (1.95)	N/A (124.59)	132.56	N/A (194.67)	207.13	[m^2]
Scale Factor	λ	1	8		10		[-]
Length over (demihull) beam ratio	L/b	12.5	12.5	12.5	12.5	12.5	[-]
Separation over Length ratio	S/L	0.23	0.23	0.23	0.23	0.23	[-]
Block Coefficient	Cb	0.403	0.403	0.274	0.403	0.274	[-]
Water density	ρ	1000	1025	1025	1025	1025	[kg/m^3]

Table 4.1: Particulars of the Delft 372 catamaran model vs. its full-scale equivalent and the modified axe bow versions. Wetted surfaces area's marked N/A when these were not available from the reference and had to be redetermined from the linesplan.

4.4. Resistance Estimation

Although this section is named an "estimation method", as determined in Figure 3.2, the usage of the model tests by Van't Veer [44] makes it a more accurate than other estimation methods. This will partially compensate for the loss of accuracy that results from not using more advanced computational methods for the vessel's resistance.

This section will discuss the process of converting the ship resistance from model tests to the 2 full-scale cases of 24 and 30 metres. But first, the impact of an axe bow on resistance will be discussed.

4.4.1. Axebow Influence on Calm Water Resistance

Axebows generally have more wetted surface area due to the deep fore foot of the vessel. Comparing the created axe design from Figure 4.3 to the reference vessel yields an increase in wetted surface area of around 6%. This increase results in higher frictional resistance for the axe vessels. The assumption in this iteration is made that the wave-making resistance *does not change* regarding the baseline vessel. This is because the following parameters remain constant.

- Froude Number
- Slenderness ratio
- Transom Immersion
- Demihull Separation

There were multiple model tests done by Keuning and Walree [22] comparing a more conventional bow to an axe bow. In these tests, around Froude numbers of 0.6, the calm water resistance of the axebow is generally lower. The ship resistance *in waves* is even lower with an axe bow. But because these hulls have a lower L/B ratio (the hulls are wider), and these are planing vessels, it is assumed that these results are incomparable.

4.4.2. Scaling the Resistance

The calm water resistance results of the Delft 372 catamaran can be seen below. Using the conventions by the ITTC [20], these have been scaled to full scale.

Fn	U [m/sec]	R [N]	sinkage [mm]	trim [deg]
0.184	0.999	3.31	-1.77	-0.035
0.240	1.300	6.60	-3.09	-0.048
0.300	1.627	12.77	-5.88	-0.091
0.348	1.890	16.14	-8.06	-0.050
0.400	2.170	26.23	-12.57	-0.48
0.450	2.439	41.75	-15.58	-1.39
0.499	2.709	52.79	-15.16	-1.99
0.549	2.978	59.08	-12.36	-2.18
0.601	3.258	63.50	-7.89	-2.09
0.650	3.527	66.00	-4.35	-2.01
0.700	3.797	70.60	-2.39	-1.90
0.748	4.058	74.71	+2.57	-1.35

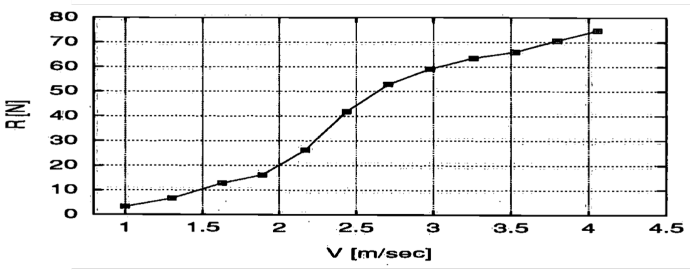


Figure 4.4: Results on the resistance measurements from the Delft 372 Catamaran [44]

A brief step-by-step overview of how to retrieve the resistance values at the desired full scale can be seen below, these steps can be performed analogously for the 24 and 30 metre variants. Important to note is because the Froude numbers are different (same speed different ship length) different model speeds need to be used for each of the vessels to determine model scale resistance.

Step 1. Determine the Froude number, and use this to determine model speed at scale.

$$Fn = \frac{V_S}{L_S} \tag{4.1}$$

$$V_M = \frac{Fn}{L_M} \tag{4.2}$$

Step 2. Determine the Reynolds number, at both model and full scale, kinematic viscosity (ν) can be used from ITTC tables.

$$Re = \frac{V \cdot L}{\nu} \tag{4.3}$$

Step 3. Calculate the frictional resistance coefficient C_f at both model and full scale.

$$C_f = \frac{0.075}{(\log_{10} Rn - 2)^2} \tag{4.4}$$

Step 4. Calculate the total resistance coefficient of the model C_{TM}, with R_{TM} linearly interpolated to have the resistance at the right model speed.

$$C_{TM} = \frac{R_{TM}}{0.5 \rho_M A_M V_M^2} \tag{4.5}$$

Step 5. Calculate the residuary resistance coefficient and from this the full scale C_T (assumed equal at model and full ship scale.

$$C_R = C_{TM} - C_{FM} \tag{4.6}$$

$$C_{TS} = C_R + C_{FS} \tag{4.7}$$

Step 6. Finally this can be converted to ship resistance at full scale.

$$R_{TS} = 0.5 \rho_S V_S^2 A_S C_{TS} \tag{4.8}$$

Here the A_S is the wetted surface area at full scale. Which for the axe version of the vessel is 6% larger, as mentioned previously. Resulting in the larger overall resistance explained in Section 4.4.1. Results of these resistance calculations are visible in Table 4.2.

4.4.3. Resistance Estimation Results

		At 24 metres			At 30 metres			Unit
	Symbol	Delft 372 model	Delft 372	24m Axe	Delft 372 model	Delft 372	30m Axe	
Length	L	3.00	24.00	24.00	3.00	30.00	30.00	[m]
Speed (knots)			17.00	17.00		17.00	17.00	[knots]
Speed (m/s)	V	3.09	8.75	8.75	2.77	8.75	8.75	[m/s]
Froude number	Fn	0.57	0.57	0.57	0.51	0.51	0.51	[-]
Kinematic Viscocity	V	1.0034E-06	1.0508E-06	1.0508E-06	1.0034E-06	1.0508E-06	1.0508E-06	[m^2/s]
Gravitational Acceleration	g	9.81	9.81	9.81	9.81	9.81	9.81	[m/s^2]
Wetted Surface Area	A	1.95	124.59	132.56	1.95	194.67	207.13	[m^2]
Reynolds Number	Re	9.2446E+06	1.9975E+08	1.9975E+08	8.2686E+06	2.4968E+08	2.4968E+08	[-]
Frictional Coëfficient	Cf	3.0414E-03	1.8894E-03	1.8894E-03	3.1016E-03	1.8325E-03	1.8325E-03	[-]
Residuary Coefficient	CR	3.50E-03	equal	equal	4.17E-03	equal	equal	[-]
Total Resistance Coëfficient	CT	6.540E-03	5.388E-03	5.388E-03	7.273E-03	5.331E-03	5.331E-03	[-]
Total Resistance	RT	60.86	26314.34	27998.84	54.15	40682.67	43286.95	Newton
			26.31	**28.00**		**40.68**	**43.29**	**kN**

Table 4.2: Overview on scaled resistance values for the Delft 372 catamaran

The total ship resistance of the axe fitted vessels (marked in orange) is therefore around 6% higher, in line with the increase in wetted surface area. It can be concluded that this resistance estimation method yields a realistic result that can serve as a baseline for further iterations. The 6% penalty of an axebow is deemed worth the expected seakeeping improvements it provides.

4.5. Powertrain Sizing Estimation

The battery capacity estimation and sizing of the electric motors will follow the approach used for the widely studied Stavanger Demonstration or Medstraum vessel by Boulougouris [2]. To calculate this capacity, first the effective power (P_E) needs to be derived, from the resistance (R) and vessel speed (U). Then to achieve the required (battery) power the losses between the batteries and the effective power need to be known.

$$P_E\,[kW] = R\,[kN] \cdot U\,[m/s] \tag{4.9}$$

Propulsive loss

Propulsive efficiency determines the propulsive loss, which is built up of the following parts: hull efficiency, open water propeller efficiency and relative rotative efficiency.

$$\eta_D = \eta_H \cdot \eta_O \cdot \eta_R \tag{4.10}$$

In reality, the constituent parts of these efficiencies are based on assumptions. Therefore, it helps that the Medstraum has full-scale measurements of its vessel as seen in Figures 4.5 and 4.6, and will be used as a reference. The propulsive efficiency (η_D) of the Medstraum is taken at 17 knots and therefore assumed to be around 75%. In reality, it is known that this is a highly optimized vessel where a lot of investment has been put in the calm water resistance optimization. Therefore, some margin will be taken, and the propulsive efficiency η_D will be set at 65%. Which results in a propulsive loss of 35% for the 24- and 30 metre vessels.

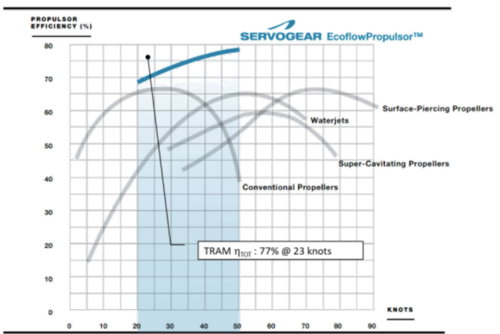


Figure 4.5: Medstraum propulsive efficiency compared to other propulsors [46]

V	Fn	t	w	ETAH	ETAO	ETAR	ETAD	Notes
kn	-	-	-	-	-	-	-	
8.00	0.243	0.003	0.002	0.999	0.782	0.982	0.762	
10.00	0.334	0.021	0.006	0.985	0.785	0.974	0.753	
13.00	0.394	0.030	0.032	1.002	0.764	0.989	0.757	
15.00	0.455	0.015	0.016	1.001	0.744	0.994	0.741	
17.00	0.516	0.011	0.000	0.989	0.753	1.000	0.745	
19.00	0.576	0.017	−0.003	0.980	0.765	1.006	0.755	ETAR> 1.000
21.00	0.637	0.025	0.007	0.982	0.773	1.013	0.770	ETAR > 1.000
23.00	0.698	0.036	0.020	0.983	0.777	1.024	0.782	ETAR > 1.000 ETAD > ETAO!
25.00	0.758	0.045	0.029	0.983	0.779	1.031	0.789	ETAR > 1.000 ETAD > ETAO!
27.00	0.819	0.053	0.038	0.985	0.780	1.039	0.798	ETAR > 1.000 ETAD > ETAO!

Figure 4.6: Full scall losses of the Medstraum vessel by Papanikalou [38]

Other Losses
The other losses: Discharge loss (EOL), Converter loss, Motor loss and Gear loss were assumed equal to the losses from the Medstraum reference.

The overview of losses is presented below. Take note: the losses don't add up linearly, each efficiency (100%-loss%) multiplied together equates to the effective power of 57% as displayed in Figure 4.7.

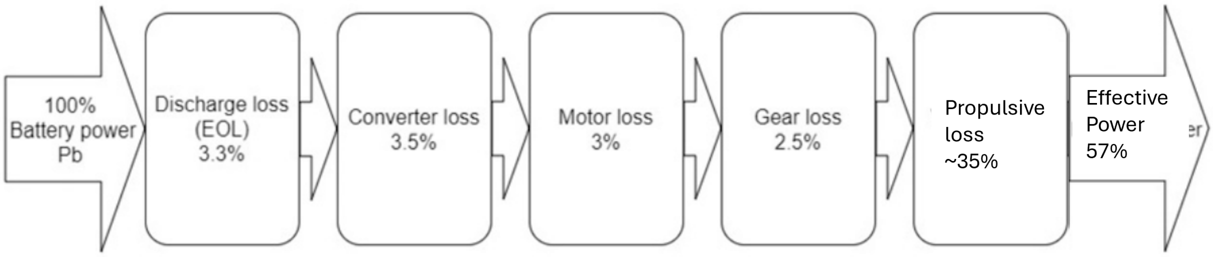


Figure 4.7: Power train of a battery-powered system [2], adapted to fit the design case.

4.6. Weights, Centres of Gravity (COGs), and Inertias

As mentioned in the previous section, the goal of this first iteration is to retain many of the particulars that affect the motions of the Delft 372 catamaran, so we can study the effects of the changed bow shape and length differences independently. The weights, their centroids, and the inertia are necessary inputs for the seakeeping calculations in the next section.

What this means for the Weights, COGs, and Inertias for *this* iteration is that this will be the scaled version of those of the Delft 372 Catamaran. From the second paper of Van't Veer [43], report 1130, a complete overview of these particulars is given. In Table 4.3, these masses, COGs, and inertias are scaled. It is important to note that the LCG (Longitudinal Centre of Gravity) is NOT scaled because the axebow design has a different LCB (Longitudinal Centre of Buoyancy). In order for the vessel to float horizontally on its designed waterline, the LCG and LCB need to be equal. The LCB was determined by calculating the volume centroid of the displaced water volume with Rhinoceros (CAD). The used LCG therefore has been underlined in the table.

Particulars	Symbol	Model	Full Scale 24m	Full Scale 30m	Unit
Length	L	3	24	30	[m]
Displacement	Δ	0.087	45.63	89.11	[tonnes]
Vertical Centre of Gravity	KG	0.34	2.72	3.4	[m]
Longitudinal Centre of Gravity (scaled)	LCG	1.41	11.28	14.1	[m]
Longitudinal Centre of Gravity (used)	LCG	[-]	12.05	15.05	[m]
Roll radius of gyration	K_{xx}	0.389	3.11	3.89	[m]
Moment of inertia for roll	I_{44}	[-]	442.53	1350.49	[ton m2]
Pitch radius of gyration	K_{yy}	0.81	6.48	8.1	[m]
Moment of inertia for pitch	I_{55}	[-]	1918.72	5855.48	[ton m2]
Yaw radius of gyration	K_{zz}	0.93	7.44	9.3	[m]
Moment of inertia for yaw	I_{66}	[-]	2529.35	7718.95	[ton m2]

Table 4.3: Overview of the Masses, Inertia's and COGs for model and full scale of the tested vessel. The Full Scale values are calculated. The used LCG values are not scaled, therefore underlined.

4.7. Motion Analysis

To assess whether altering the bow of the Delft 372 catamaran improves seakeeping, Marin's software SEACAL [27] will be used: *"**SEACAL** is a 3D potential flow panel code for zero- and forward-speed seakeeping calculations in the frequency domain."* The validity of testing this catamaran model of this size is underscored in the theory manual, as the original *Delft 372 Catamaran* and its measurement results have been used to validate the SEACAL software [27] [28].

In this section, two separate effects will be studied:

- The effect of adding an axebow to the DUT-372 catamaran to reduce accelerations.
  - Differences between experimental and numerical DUT-372 results will also be portrayed.
- The influence of vessel length on the vertical accelerations of the 24- vs. the 30-metre axe vessel.

The focus is put on headwaves and thus heave and pitch RAOs and the consequent accelerations. These results will form the foundation for the further steps in this research.

4.7.1. Description of Practical and Numerical Experiments

Van't Veer [44] tested for the following conditions: *Heave and Pitch motions registration in head waves, $Fn =$ 0.30, 0.45, 0.60, 0.75.* With the total testing regime depicted in Figure 4.8.

run	Fn	λ/L	run	Fn	λ/L	run	Fn	λ/L	run	Fn	λ/L
26	0.30	0.601	50	0.45	0.601	72	0.60	0.798	98	0.75	0.998
28	0.30	0.803	52	0.45	0.799	74	0.60	0.897	100	0.75	1.100
30	0.30	0.899	54	0.45	0.899	76	0.60	0.999	102	0.75	1.204
32	0.30	0.965	56	0.45	1.001	78	0.60	1.098	104	0.75	1.297
34	0.30	1.007	58	0.45	1.097	80	0.60	1.200	106	0.75	1.394
36	0.30	1.101	60	0.45	1.201	82	0.60	1.292	108	0.75	1.491
38	0.30	1.204	62	0.45	1.295	84	0.60	1.393	110	0.75	1.596
40	0.30	1.301	64	0.45	1.396	86	0.60	1.490	112	0.75	1.806
42	0.30	1.395	66	0.45	1.595	88	0.60	1.600	114	0.75	1.985
44	0.30	1.596	68	0.45	1.803	90	0.60	1.795	116	0.75	1.694
46	0.30	1.803	70	0.45	1.991	96	0.60	1.977	118	0.75	1.906
48	0.30	1.978				122	0.60	1.741	120	0.75	2.012
126	0.30	0.699				124	0.60	1.816	128	0.75	1.983

Figure 4.8: Test run overview of the heave and pitch motion measurement program of the Delft 372 Catamaran [44].

These testing regimes have been translated to the 3-, 24, and 30-metre cases in SEACAL. Where the Froude numbers have been converted to speed in knots (V) for each respective ship length. The 3-metre case in this instance is the **original DUT-372** vessel. This has been included to show the inherent differences between SEACAL and model testing. The results at 3 metre ship length (response amplitudes and wavelengths) have been translated to 24 metre ship length to properly study the differences independent of vessel length. The exact methodisation is described in Appendix A.1.

The λ/L relation — wave length non-dimensionalised against ship length — from Figure 4.8 has been used to find the approximate testing range for the 24- and 30-metre vessels in [rad/s], as these are the required inputs for SEACAL. The overall testing plan, with a somewhat similar range and step size, is given in Table 4.4.

Vessel	3m	24m	30m	unit
Ship Speed Testing				
Speed min (V)	3.17	8.95	10.00	[knots]
Speed max (V)	7.91	22.37	25.01	[knots]
Stepsize	1.575	4.47	5.00	[knots]
Nr. of steps	4	4	4	[-]
Wave Frequencies Testing				
Wave frequency (ω) min	2.26	0.8	0.8	[rad/s]
Wave frequency (ω) max	5.66	2	2	[rad/s]
Stepsize	0.14	0.05	0.05	[rad/s]
Nr. of steps	25	25	25	[-]

Table 4.4: Testing overview for SEACAL

4.7.2. Results

Before assessing the results, it is important to note that the main goal of the axe hull is to *reduce accelerations* as these are the main contributors to motion sickness [33]. The MSI index introduced in Chapter 2 will be used to quantify this, where the index (in %) corresponds to the proportion of people who are physically ill (emesis) within a given period. A time of 1 hour is taken, and on the current fleet sailing to Klein Curaçao, approximately 10% of people have emesis on rough days.

In this section, only the results at $Fn = 0.6$ are presented for the axe vs. no axe comparison. To consequently study the difference between the 24- and 30-metre vessels, the results at 17 knots are presented. The complete results across all tested speeds are given in Appendix A.1. The wave bands correspond to the peak frequency of $T_P = 6.2$ seconds determined in Section 2, which corresponds to a radial frequency of around 1 radian per second. The significant wave height (H_S) used for the Johnswap spectrum to acquire the RMS (Root Mean Square) values is 1.4 metres, as determined in Chapter 2.

Throughout this section and in the appendix, the black dots represent the experiments (scaled to 24m), the blue line represents the SEACAL calculations on the 3m DUT-372 catamaran (scaled to 24m), the orange line represents the 24m axe results (not scaled), and the green line represents the 30m axe results (not scaled).

Real Wave Frequencies & Periods versus Encounter Frequencies & Periods

Within all the results the *real* wave frequencies and periods have been depicted, i.e. frequencies and periods compared to a stationary object at sea. But as the vessel is sailing at forward speeds, the ship perceives different frequencies: the encounter frequencies. In Figures 4.9 and 4.10 the relation between the frequencies and periods regarding their encounter versions at forward speed of 17 knots have been depicted, and ranges typical throughout these results. In later chapters, RAOs at different speeds will be evaluated. Therefore, the choice was made to plot the *real* wave frequencies and periods throughout this work. The formulas describing the relation are visible below.

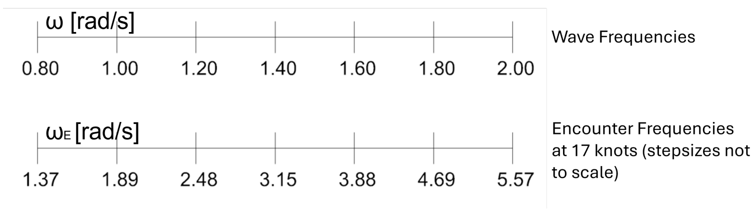


Figure 4.9: Radial wave frequency to encounter frequency at 17 knots

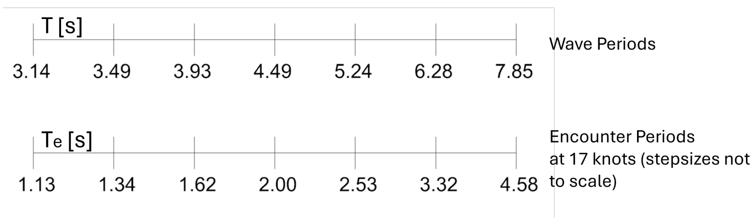


Figure 4.10: Wave period to encounter period at 17 knots

Relational formula's (for headwaves):

$$\omega_e = \omega + kU,\, k = \frac{\omega^2}{g},\, T = \frac{2\pi}{\omega} \tag{4.11}$$

4.7.3. Axe bow influence on Heave and Pitch RAOs at $Fn = 0.60$

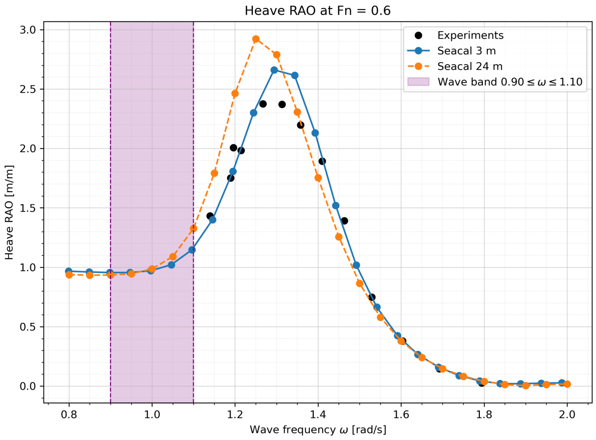


Figure 4.11: Heave RAO at $Fn = 0.60$

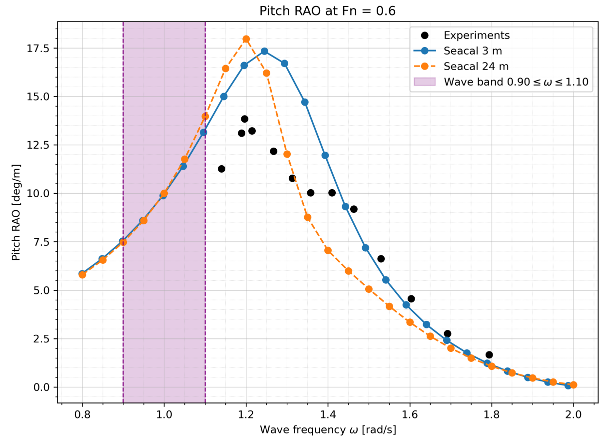


Figure 4.12: Pitch RAO at $Fn = 0.60$

4.7.4. Axebow influence on accelerations at $Fn = 0.60$

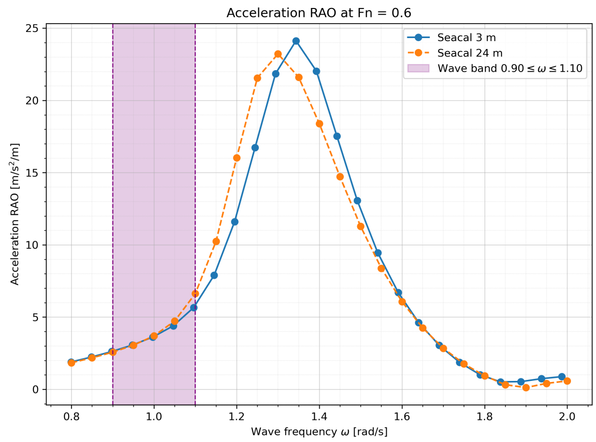


Figure 4.13: Acceleration RAO at $Fn = 0.60$

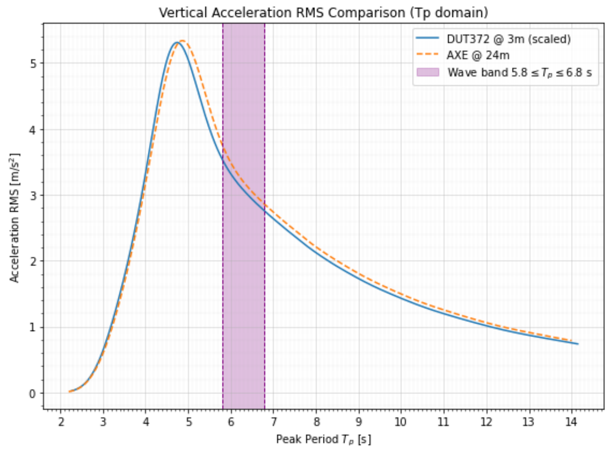


Figure 4.14: RMS of acceleration at $Fn = 0.60$

The effect of the axebow in SEACAL results in more heave response and slightly more pitch response, but barely less accelerational response. In pitch, the peak is slightly narrower, which is positive for operability, the shift towards the design Tp of 6.2 seconds ($\omega = 1$ [rad/s]) is not beneficial. The alignment of the results with those of the original DUT-372 indicates that the procedure and method used in SEACAL are correct. It is noted that, in reality, the heave and pitch responses will be lower for all SEACAL-calculated results at this speed due to the nonlinear effects of forward motion and viscous damping. In the appendix, it is visible that at slower speeds, SEACAL and practical experiments align even better.

The calculated MSI value based on one hour at the COG is approximately 6.9% for both vessels. Which can be acceptable, but on the bridge or the bow the ship will move a lot more, these areas will be studied in the vessel comparison between the 24- and 30-metre axebows.

Axebow Conclusion

The influence of an axebow on RAO's (Figures 4.11 to 4.13) follow expectations, although not as pronounced as expected. The heave and pitch (Motion) RAO's are slightly bigger, and the accelerational RAO is somewhat smaller, which follows the theorisation that an axebow has less acceleration at the sacrifice of more heave and pitch. The peak shift towards the peak wave period is less favourable.

However, when the accelerations are converted to RMS values using the Johnswap spectrum (Figure 4.14), the differences become negligible. From this, we can conclude the following:

- In comparison to an already slender catamaran hull, an axebow is less effective.
- From the results, it appears that an axebow is less effective than expected. However, an axebow is designed for non-linear effects, which is something potential flow can't model.

As disclosed in the results, particularly in the acceleration results, the effectiveness of the axebow on this vessel at this stage is therefore disputed.

4.7.5. 24-metre versus 30-metre vessel RAO's & RMS values at 17 knots

At the design speed of 17 knots the RAO's for both vessels for heave and pitch can be seen below.

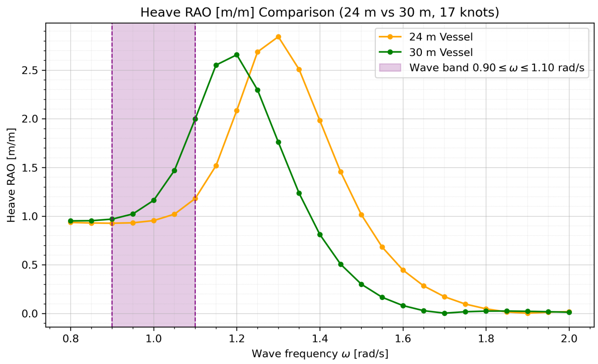


Figure 4.15: Heave RAOs at 17 knots

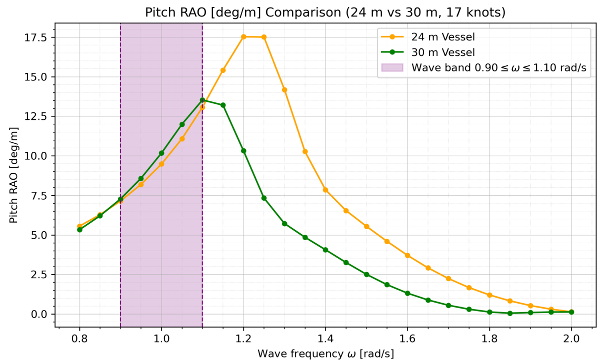


Figure 4.16: Pitch RAOs at 17 knots

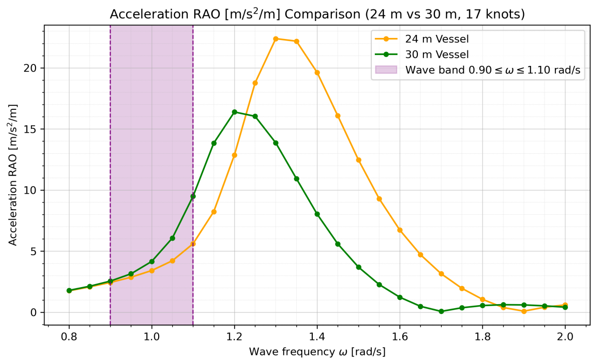


Figure 4.17: Acceleration RAOs at 17 knots at the COG

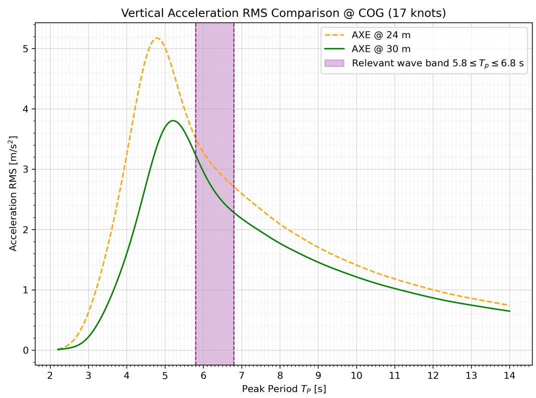


Figure 4.18: RMS accelerations at 17 knots at the COG

The shift of the peak leftward for the longer ship is unfavourable when looking at the heave RAO's. But the reduction in pitch and mainly in accelerational RAO's compensates for this, which, with the spectrum included, results in overall lower RMS acceleration for the 30m vessel at the COG. In terms of MSI values at the COG this comes down to 5.4% for the 24m vessel and 3.4% for the 30 meter vessel.

If we calculate the values at the bridge for both vessels: assumed at 1/3 ship length from the bow and 5 metres high on the centerline of the ship, the MSI values will be 20% for the 24m vessel and 16% for the 30 m vessel.

4.7.6. Non-linear Effects

Because of the inherent nature of potential flow code such as SEACAL there are some (non-linear) effects that unaccounted for:

- SEACAL only uses the part of the hull that's *under the waterline*, therefore neglecting one of the key differences between the 2 hulls. The DUT-372 has a changing waterline area, whilst the axe design has an almost constant waterline area. Which in practice would add more acceleration to the DUT-372 catamaran, thus favouring the axe design.
- There is also additional pitch and heave damping to be expected for *both vessels,* as potential flow code *neglects viscous effects.* Therefore, both RAOs are larger than expected.

- *Slamming effects are neglected* in SEACAL. Meaning that the acceleration peaks (at which an axebow is better at) should be higher for the DUT-372 Catamaran. A deeper forefoot that the axebow has which is depicted in Figure 4.3, in theory lengthens the time the hull stays in contact with the water and thus reduces the chances of hull-bottom slamming. In literature on monohull axebows this effect is significant [22].

4.8. Hull Choice, Conclusion, Reflection, Improvements

So, what is the right hull to pick? From what is known about vessel length & ship motions, the choice leans heavily towards the 30-metrehull shape. However, we know that the 30-metre hull consumes 50% more energy for only 6 metres of additional ship length (as shown in Table 4.2). At the end of this chapter, there is simply *not enough information* to make the right choice. For instance, it is not known if the displacement of the 24- (or 30-) metre vessels make sense in comparison to other ships of that size. And it is also not known what the masses are of the constituent components of the ship, and how these scale with relation to vessel length. So, a brief detour from the Design Spiral is needed, and this will be performed in the following chapter.

5
Complete Ship Mass Estimation

Before diving into the 2nd iteration, some necessary calculations need to be completed. This is mainly dependent on the factors that determine the vessel's final weight, as they influence displacement. One issue is that the DUT-372 catamaran is quite a light vessel at 24 metres in length. This sidestep will dive into more detail, specifically on ship mass, describe the relationships between ship particulars and the masses and will end with a decision matrix to fully map *what* the right ship length is and *why*.

5.1. Maximum length according to the Rules

The original philosophy behind the 24 and 30-metre division was to have two vessels to quantify and compare. At this stage of the design it is known that every possible metre of ship length is absolutely necessary (both for seakeeping and displacement advantages). The maximum length (L) the Caribbean SCV Rules [40] allow for is 24 metres at 96% of the Waterline Length at 85% of the moulded depth (D), see the red line for clarification in Figure 5.1. As this is at 96% of the waterline, the full allowable waterline length (assuming a straight bow) is therefore $24/0.96 = 25$ [m]. 30-metre ship length is allowed under exception only. In the rest of this chapter, the 25 and 30 metre vessels will therefore be compared, and the implications for the earlier seakeeping calculations will be discussed in Chapter 6.

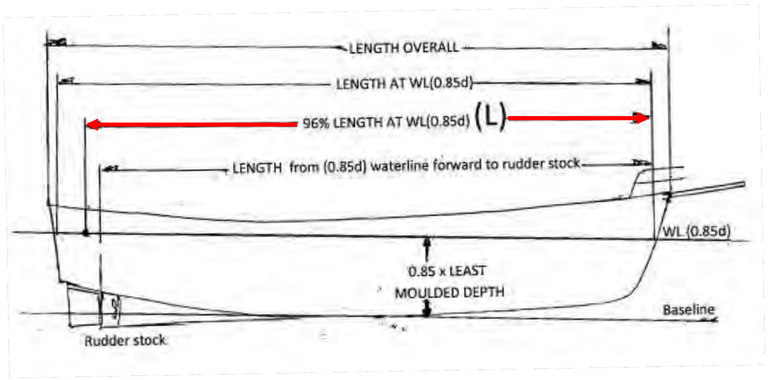
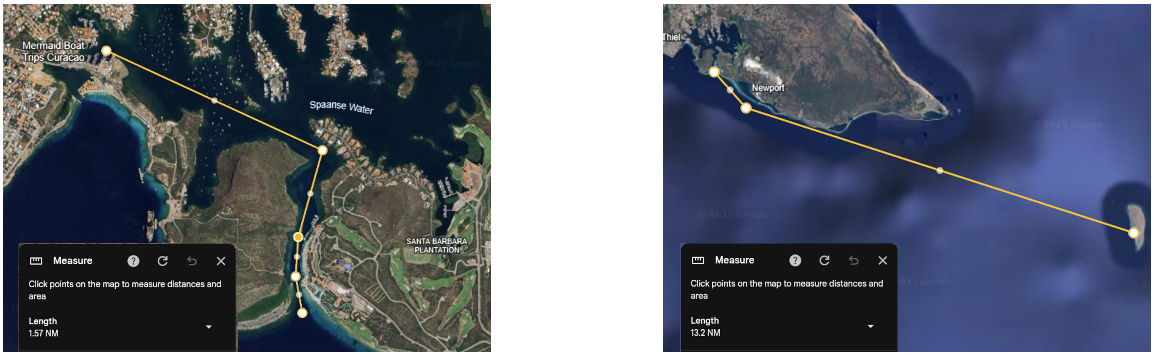


Figure 5.1: Definition of maximum attainable ship length according to the Rules [40], with the red line marking the applicable length dimension (L).

5.2. Realistic Operational Profile

A heavy component of an electric vessel are the batteries, as this energy carrier is much heavier than conventional fuels. A realistic operational profile helps estimate the correct battery weight, in addition to the losses related to powering and propulsion stated in Section 4.5. There are also several other losses to be calculated to determine the final required battery capacity. In this section, a more detailed operational profile will also be provided to estimate the required battery capacity more accurately. The accurate port and open sea distances were determined with Google Maps as seen in Figure 5.2. The detailed profile is given in Table 5.1 for the 25-metre vessel, and the results for 30-metre vessel are added in Table 5.2.



(a) Protected waters operational area (7 knots)

(b) Open sea operational area (17 knots)

Figure 5.2: Overview of total operational area and route length

In the figures of 5.2, it is visible that the vessel sails around 1.6 NM in protected waters, and about 13.2 NM in open sea. The overview of distances, time, and energy usage for these operational profiles is presented in Table 5.1.

	Outbound		Return		*Unit*	Relation
	Spanish Waters	Open Sea	Open Sea	Spanish Waters		
Distance (nautical miles)	1.6	13.2	13.2	1.6	[*Nautical Miles*]	
Speed	7	17	17	7	[*Knots*]	
Speed	3.60	8.75	8.75	3.60	[*m/s*]	
CW resistance at that speed	4.5	31.2	31.2	4.5	[*kN*]	
Effective Power	16.31	273.08	273.08	16.31	[*kW*]	$Pe = R \cdot V_S$
Pd (=57%)	28.43	476.05	476.05	28.43	[*kW*]	$Pd = P_E/0.57$
Time at that speed	0.23	0.78	0.78	0.23	[*hours*]	
Calm Water Energy Required	6.50	369.64	369.64	6.50	[*kWh*]	$E = P_D \cdot t$
Sea Margin		20%			[-]	
Total Energy (25m Axe)	**6.50**	**443.56**	**369.64**	**6.50**	**[*kWh*]**	

Table 5.1: Trip overview and energy consumption for the 25 metre vessel.

	Spanish Waters	Open Sea	Open Sea	Spanish Waters		
CW resistance at that speed	2.9	48.8	48.8	2.9	[*kN*]	
Effective Power	10.49	426.57	426.57	10.49	[*kW*]	Pe = R* vs
Pd (=57%)	18.29	743.61	743.61	18.29	[*kW*]	Pd = Pe/0.57
Calm Water Energy Required	4.18	577.39	577.39	4.18	[*kWh*]	E = Pd*t
Sea Margin		20%			[-]	
Total Energy (30m Axe)	**4.18**	**692.87**	**577.39**	**4.18**	**[*KWh*]**	

Table 5.2: Energy consumption for the 30 metre vessel.

The total energy consumption for a return trip for the 25-metre vessel is therefore **826 [kWh]** and for the 30-metre vessel this is **1279 [kWh]**, a 55% increase for 5-metre (20%) extra vessel length. However, the 30-metre vessel has 95% more displacement, so in load-bearing terms it's a more fuel-efficient vessel.

5.2.1. Possibility of changing the S/L Relation

It is known that some resistance advantages can be won by changing the separation distance between the hulls. The question is *how much* resistance advantage this gives. And if this space and resistance advantage is worth the extra penalty in structural mass that needs to be carried by the vessel. An added effect of this extra beam is the increased surface area available to passengers. Increasing the sense of spaciousness on board, possibly helping reduce seasickness.

The continuation of the research on the DUT-372 Catamaran by Van't Veer was done by Broglia et al. [4]. Here, the authors experimentally varied the model on separation distances to research the effect of interference resistance on this specific catamaran. In Figure 5.3, the resistance graphs can be seen for different separation ratios. At the design Froude numbers for the 25 metres and 30 metres ship lengths at 17 knots, the possible C_T reductions are depicted for wider separation ratio's. The original S/L (or H/L) ratio was 0.233, it is compared to an increased S/L of 0.3. (The possible C_T reduction for the slow speed of 7 knots are currently disregarded, as the net energy savings are considered negligible).

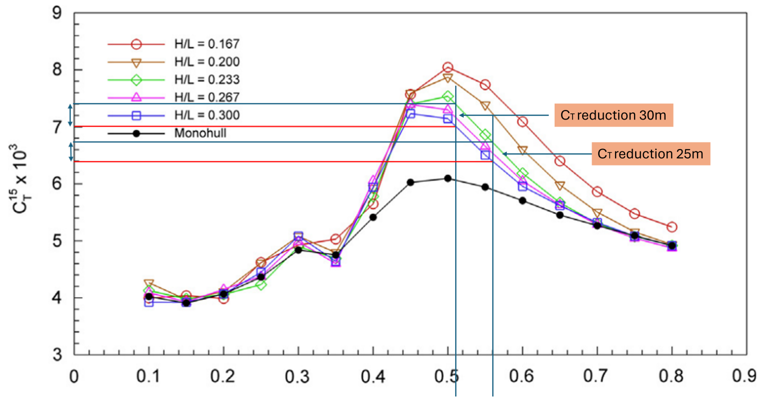


Figure 5.3: Influence off hull seperation distance on the resistance (C_T) by Broglia et al. [4] from experimental tests, with annonations on C_T reductions for 24 and 30 metre vessels.

This C_T reduction means, in practice, that the 25- and 30-metre vessels have 2.1 and 3.2 kN less resistance. The percentage reductions are comparable, but in absolute terms, the increase in beam is more pronounced for the 30 metre vessel. The results are depicted in Table 5.3. Whether this extra space and/or energy reduction is worth the increase in structural weight will be discussed in the second iteration. For now, (structural) mass estimation continues with the original hull seperation.

S/L Resistance Savings	25m vessel	30m vessel	Unit
Beam increase (from $\rightarrow$ to)	7.83 $\rightarrow$ 9.5	9.4 $\rightarrow$ 11.4	[m] $\rightarrow$ [m]
R (S/L = 0.233)	29.5	48.8	[kN]
R (S/L = 0.3)	27.4	45.6	[kN]
% reduction	-7.3%	-6.9%	[-]

Table 5.3: S/L increase & Calm Water Resistance Savings.

5.2.2. Battery Margins

Losses or margins inherent to a battery-powered vessel primarily focus on preserving long-term battery capacity, as charge and discharge limits and degradation must be accounted for to maintain battery quality. Ship design around battery-powered vessels also asks for a balance. As there are different battery types, some can handle more cycles, some can discharge further, some just have preferable gravimetric or volumetric density. Luckily this is already a quite mature research topic but the key takeaways is knowing how many cycles the batteries need to handle.

Depth of Discharge (DOD) & End of Life (EOL)

Assuming the vessel will operate the trip to Klein Curaçao 6 days a week, and the vessel will charge once for each trip, the number of yearly cycles will be:

$$\text{No. of Cycles} = (6/7) \cdot 365 = 313 \text{ charge cycles a year} \tag{5.1}$$

For an initial lifespan of 10 years, the batteries must withstand **3130 cycles**. Battery provider XMP states that at 80% Depth of Discharge (DOD) – i.e. (dis)charging between 10% to 90% each cycle – the battery pack can withstand more than 4000 cycles [13]. Where at the end of life (EOL) the battery pack still attains 80% of its original capacity. To account for both DOD and EOL, only 60% of the capacity can be used. This means that $\frac{1}{0.6} = 1.667 \equiv$ 66.7% increase in the required battery capacity. Research on the Selfa Fast Ferry [41] came to the same conclusion at 8500 cycles.

Another factor in battery degradation is discharge speed (C-rate), where at 1C the battery charges its entire rated capacity in 1 hour. This battery will discharge (at end of life capacity) 80% in 2 hours. Equivalent to a C-rate of 0.4C. This is well within recommended levels for modern battery capabilities. A battery pack of interest by manufacturer Corvus for this design has a recommended continuous C-rate for charging and discharging of 0.5C [5].

5.2.3. Hotel Load

The electrical elements on board the vessel not used for propulsion are considered the hotel load. The main contributors to this hotel load are the ice-freezers, the water pumps for fresh water consumption, and navigational equipment. The time away from port on an average day is 12 hours, and the rest of the time the ship is connected to shore power. The bow thruster is included in the hotel load here for ease of overview.

No lights are included as the ships sail during daytime, and no AC is considered as the vessel will have an open plan layout.

User	Average Time Used [hrs]	Power [Watt]	Required Energy [Wh]	
Ice-Freezer 70% continuous load	12	469	5628	
Water Pump	2	720	1440	
Bow Thruster	0.083	5000	417	
Navigational Equipment	12	130	1560	
Total Hotel Load Required			9045	
Total with 100% Margin			**18.1**	**kWh**

Table 5.4: Hotel Load elements from [17] [47].

5.2.4. Total Required Capacity & Spare "Emergency" Range

It is wise to include a safety margin in the design and battery capacity to ensure the vessel can operate within its calculated range. But this is also a pitfall, as too many margins will make the ship only heavier and more expensive. A lot of battery capacity will then have to be carried around that won't be used. And what will become apparent in Section 5.2.5, is that there is already a backup for an emergency. Therefore, the *Design Choice* is to carry only 20% extra battery capacity on top of the aforementioned levels. This will come down to the following required capacities for both the 25- and 30-metre vessels respectively. (Note this is **Without S/L increase**).

Vessel Type	25m	30m	Unit
Bare Energy Required per Return Trip	844	1297	kWh
20% Margin on bare trip	169	259	kWh
66.7% to account for EOL & DOD	563	864	kWh
Total Required Battery Capacity	**1576**	**2421**	**kWh**

Table 5.5: Total battery capacities for 25- and 30-metre vessels

5.2.5. Emergency Generators

Although it might be possible to do the whole return trip on battery power, emergencies must always be accounted for, and redundancy is always desirable. Two emergency generators (one per hull), providing sufficient capacity to sail to port at 5-7 knots, are deemed necessary for a full single-trip distance of 14.8 nautical miles. This distance is the maximum distance the vessel will be from port.

From the calm water resistance results from Van't Veer [44], the required power is $P_D = 24.6$ [kW] at 7 knots, 25 metre ship length. Two gensets, one for each hull, are then selected to provide this emergency power. The diesel gensets from Genesal Energy have been used as a reference. And at $24.6/2 = 12.3$ [kW] required power, *GEN22KC-IN* [15] provides sufficient emergency power, weighing 690 [kg] per generator and consuming around 4 [l/h] at 75% of PRP (Prime Rated Power). A diesel tank of 200L will be used, so the total weight (including a margin) of the emergency power system will be around 2 tonnes. This mass will be important for the following sections.

5.3. Mass Estimation

The previous section provided one of the first tangible masses for the vessel's design. This section will continue to provide masses for all elements of the vessel. To check if the displacement is equal to all the masses aboard the vessel.

5.3.1. Electronic / Battery Weights

The parts present in the powertrain of an electric vessel are visible in Figure 5.4, as derived by M. Francis [12].

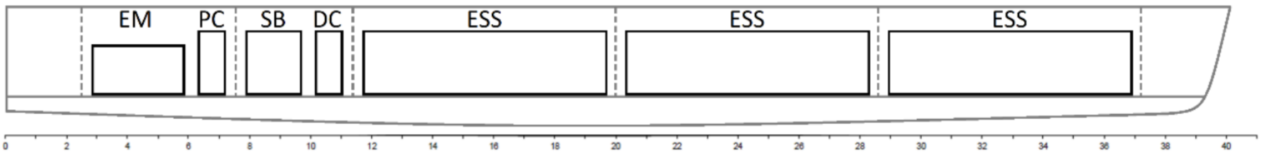


Figure 5.4: Side view of a demi-hull with an overview of the battery powered propulsion system: electric motor (EM), propulsion converter (PC) [*inverter*], switchboard (SB), DC-DC converter (DC) and energy storage system (ESS) by Francis [12]

In the design, the focus is put on the electric battery systems by Corvus, as their technology is mature, and they are market leaders with over 1300+ vessels installed [6]. Their systems are air cooled. The decision was made to select their top-of-the-line battery, the *Corvus Dolphin NxtGenESS – Energy* [5], as it is the lightest possible battery to ensure the displacement isn't (overly) exceeded.

Battery Weight Based on String Weight

The maximum energy for the Corvus battery per string is 168 Wh/kg [5]. For the 25- and 30 metre vessels this results in **9.38** and **14.42 tonnes** in battery mass for each of the vessels.

Remaining Electric Weights

Table 5.6 shows the individual masses of the remaining parts of the power train. For the motor and propulsion converter, Danfoss parts were selected [8] [7]. For the DC-DC converters and Switchboard mass estimation, Francis' method was used [12] to scale the components from the Selfa Battery Ferry [41].

	25-metre vessel			30-metre vessel		
	Amount	Mass [kg]	Combined [kg]	Amount	Mass [kg]	Combined [kg]
Electric Motor	2	450	900	2	720	1440
Propulsion Converter	2	20	40	2	40	80
DC-DC Converter	2	168	335	2	262	524
Switchoards	2	223	447	2	349	698
Total Electrical Mass			1722			2742
In Tonnes			**1.72**			**2.74**

Table 5.6: Electrical Masses based on Danfoss components [8] [7] and estimation methods [12] [41]

5.3.2. Structural Mass

Hull Material Choice; hull material, from the client's perspective, gravitated from the beginning towards FRP (Fibre-Reinforced-Plastic). As the current aluminium vessel has significant galvanic corrosion issues. FRP is also the most common shipbuilding material on Curaçao, so when repairs need to be made, this is the most practical and cheapest material to work with.

In this section, a brief estimate of the vessel's structural mass will be provided. Two estimation methods will be used and compared in their final bare hull structure weights.

Method Karayannis

The first method is based on the Method by Karayannis et al. [21] as implemented by Moraes et al. [31] based on 50, 75 and 100 metre aluminium cats (36 vessels were analysed). Although material and sizing differ, this work can tell something about the underlying physical relations. Equipment numeral "E" uses the parameters of the vessel.

$$E = 2L(b+T) + 0.85L(D-T) + 1.6L(B-2b) \tag{5.2}$$
$$B = S + b \tag{5.3}$$
$$D = 4 + 0.44B \tag{5.4}$$

And if $E < 3025$, which is the case:

$$P_{struct(t)} = 0.00064E^{1.7} \tag{5.5}$$

The next step in this aluminium method is to determine how much lighter an FRP vessel actually is. A structural mass analysis by Hertzberg [18] comparing a 10-ton high-speed monohull on different structural materials according to code yielded a structural weight of 72% of that of FRP compared to aluminium (carbon was 57%). FRP has lower production costs [18] and the aforementioned workability for repairs on Curaçao is a big plus.

Method Grubisic

A second method recommended by industry specialist Papanikalao [36] (involved in the design by the Medstraum vessel) is the method from Grubisic [16]. Although it is a method developed for FRP monohulls, it can give us valuable insights and reveal underlying interdependencies in structures. This method splits the superstructure weight and the structural hull weight. This method has been developed and verified by the authors using a database of 61 FRP vessels, ranging in length from 10 to 60 metres.

The full method is attached in appendix D.1. Essentially superstructure weight (W_{150}) is based on the volume, Grubisic determines densities for an FRP deckhouse to be approximately $q_{DH} = 21.7$ [kg/m^3]. This is for a fully enclosed deckhouse with glass windows. As in reality, the vessel will have a more open-plan layout, a 25% mass reduction will be applied to the superstructure weight.

The hull structure weight (W_{100}) is based on the effective surface area of the hull plating (S_R), split up into bottom, sides and deck. Where similar characteristic parameters have been used as in Karayannis' method. Grubrisic's method has been implemented in two ways.

- No alteration apart from a 25% lighter superstructure – This would assume structure monohull mass = catamaran structure mass under the same LxBxH.
- Calculating the separate demihull weight and adding regular superstructure weight on top.

$$W_{100} = K \cdot E_S^{1.33} \tag{5.6}$$

Where the structural numeral E_S is based on the structural area numeral S_R with a few coefficients, and the structural weight constant K is determined by vessel type and area of operation.

The generalised inputs are displayed in Table 5.7. Where most dimensions were derived from the scaled designs from the design of Figure 4.3 from the first iteration. Draft was taken a midship as the axebow only extends below that draft for a portion of the vessel. Depth was averaged, as there is a higher freeboard at the front than aft. The height of the deckhouse was based on 2 levels with a combined height of 4.4 metres, approximately 2/3 of the vessel's length and spanning the full beam of the vessel.

Notation	Name	Used in	25m	30m	Unit
L	Length	Karayannis + Grubisic	25.00	30.00	[m]
T	Draft	Karayannis + Grubisic	1.15	1.38	[m]
D	Depth	Grubisic	3.25	3.48	[m]
b	Demihull beam	Karayannis	2.00	2.40	[m]
S	Seperation	Karayannis	5.83	7.00	[m]
B	Total Beam	Karayannis + Grubisic	7.83	9.40	[m]
V_{DH}	Volume Deckhouse	Grubisic	623.13	747.76	[m^3]

Table 5.7: Generalised inputs for the weight estimation methods

Method Results

The results of the structural mass estimation methods of both 25 and 30 metre vessels can be seen in Table 5.8.

Method		Karayannis		Grubisic - Monohull		Grubisic - Altered		
Vessel Type		25m	30m	25m	30m	25m	30m	Unit
W100	Hull Structure Mass	14.6	25.7	11.7	18.3	9.3	14.0	[t]
W150	Superstructure Mass	[N/A]	[N/A]	10.4	14.1	13.8	18.8	[t]
W100+W150 / Pstruct	**Total Mass**	**14.6**	**25.7**	**22.1**	**32.4**	**23.1**	**32.7**	[t]

Table 5.8: Structural mass method results

Both Grubisic methods are in agreement with one another. The Karayannis method, however, is lighter. Both methods have their downsides, the Karayannis method is designed for larger catamarans (50m+) and the other method focuses on monohulls. A lot can be said for either of the methods in their advantage or disadvantage. In this stage of the design, mass estimation remains a “rough uncertainty”. It is assumed that the average result of both methods is sufficiently accurate. Resulting in bare structure weights of roughly **18.5 and 29 tonnes** for the 25- and 30 metre vessels, respectively.

5.3.3. Remaining Lightship Weights & Deadweight

The Grubisic method [16] for structural mass also provides an estimation method for the remaining machinery weights, primarily based on either the vessel’s propulsion power or LxBxD dimensions (for monohulls). Grubisics Electrical machinery and Electronic Equipment may be correlated with the electrical machinery from Section 5.3.1. But to adhere to a margin as an electric vessel has a lot of electrical mass, both are included in the final calculation. One element that has been altered is the correlation formulas that use the beam (B); they have been adjusted to half-beam, as catamarans are disproportionally wide compared to monohulls.

Weight of outfit is determined by Papanikolaou’s ship design correlations [37], using the following formula, with K_{OT} being between 0.036-0.039 for passenger vessels. Resulting in Table 5.9.

$$W_{OT} = K_{OT} \cdot GT = 0.038 \cdot 202 = 7.67 \text{ t} \tag{5.7}$$

	Grubisic Name	relation	25m vessel	30m vessel	Unit
Electrical Machinery	W300	L, 0.5B, D	2.14	3.66	[t]
Electronic Equipment	W400	L	0.75	1.13	[t]
Auxiliary machinery	W500	L, 0.5B	2.75	5.27	[t]
Weight of Outfit	-	L	7.41	8.37	[t]
Total Remaining Weights			**13.05**	**18.43**	**[t]**

Table 5.9: Remaining lightship weights based on correlation formulas by Grubisic [16] and Papanikolaou [37].

Deadweight

There are four final contributors to the deadweight of the vessel, which can be seen in Table 5.10, where these masses are assumed equal for both the 25- and 30-metre vessels.

	Amount	Total Weight	Unit
Passenger + Crew	210	17325	[kg]
Diesel Fuel	200 litres	166	[kg]
Fresh Water Supply	3000 litres	3000	[kg]
Provisioning	500kg of Food	500	[kg]
Ice Cubes	500 litres	500	[kg]
TOTAL		**21.5**	**[t]**

Table 5.10: Deadweight contributors for the 25- and 30-metre vessels

5.3.4. Total Masses

	25m vessel	30m vessel	unit
Structural	18.5	29.1	[t]
Batteries + Electrical	11.1	17.2	[t]
Genset	1.5	2.3	[t]
Remaining LS Weights	13.0	18.4	[t]
Deadweight contributors	21.5	21.5	[t]
Total	**65.6**	**88.5**	[t]
Original Displacement	**51.7**	**89.1**	[t]
Percentage overweight	27%	1%	[-]

Table 5.11: Total Ship masses

It can be concluded that the original reference vessel of the DUT-372 catamaran was relatively lightweight (i.e. not enough displacement) when scaled to a length of 25 metres, but better at a length of 30 metres *for an electric ferry*. A large portion of the mass in both vessels is attributable to battery weight.

Even without batteries, the 25-metre vessel remains overweight. This can be attributed to the fact that the 372 catamaran was either not designed to this scale nor designed to be electric. Another factor is that the weight correlation formulas are based on ships that sail on conventional fuels, where mass and fuel consumption are less critical than for electric ships, therefore being heavier. And this might result in an overestimated weight.

Two possible reference vessels can point in the right direction of these ship masses. The first is the Ika Rere Ferry [29], a 19-metre, 20-knot carbon, electric ferry. With a displacement of 28 tonnes and designed for 150 passengers. Scaling this displacement to 25 metres ship length, the ferry would weigh: $28 \cdot (25/19)^3 = 64$ tonnes. Another vessel is the earlier-mentioned 24.5 metre Selfa Fast Ferry [41]. This vessel is carbon hulled, diesel powered, sails at 30 knots and designed for 130 passengers, with a displacement of 63 tonnes. Both vessels with an FRP- instead of a carbon hull would be estimated to be around 4 tonnes heavier based on Hertzberg [18]. And an extra 60 passengers would also weigh another 4 tonnes. Both the Ika Rere ferry and the Selfa Fast Ferry have glass-panelled superstructures, but the superstructure of the Mermaid case will be open-plan. potentially saving weight. And lastly, the axe vessel must withstand higher sea states. This mass estimation is a rough method, but an effort was made to keep the numbers conservative. Therefore, the expectation is that in further iterations the true vessel mass will be lighter. All these aspects together validate the ship mass calculations for the 25-metre vessel as they correlate with the 2 reference vessels.

The conclusion can be drawn that, from a ship's mass perspective, the 25-metre vessel needs **more displacement**, which, in turn, affects the ship's mass and resistance again. In the 2nd iteration, the hull will be redesigned to satisfy the displacement requirement, and the impact on ship weight and resistance will also be discussed.

5.4. Conclusion on Vessel Size

Throughout this and the previous chapter, two ship lengths have been compared: the 24/25- and 30-metre axe vessels, which find their basis in the DUT-372 Catamaran. Both vessel lengths have their advantages and disadvantages. In Table 5.12 the performance of both vessels have been compared.

	25m vessel	Fixable?	30m vessel	Fixable?
Displacement	Problematic	Yes, but with impact on Resistance, Demihull Beam, Battery Weight	Good	[-]
MSI	Questionable	Questionable	Quantifiably better	[-]
Resistance	Relativily Low	*Will increase with increased displacement*	High but better relatively	[-]
Rules	Complient	[-]	Not Complient outside Curaçao	Not for 30m
Expected production price	Baseline	[-]	Structures + batteries; both +55% heavier	Not fixable
Passenger space	Sufficient	[-]	More spacious	[-]

Table 5.12: Comparative matrix on the (performance of) the 25- vs. 30 metre axe vessels.

The displacement of the 25-metre axe vessel is the most problematic, the vessel will be about 15 tonnes heavier than its designed displacement. With this option displacement needs to increase, which in turn affects resistance which in turn affects ship mass and thus required displacement. The upside is, however, that a 15 cm increase in draught could increase the existing displacement by 10 tonnes already. And there are still some weight saving methods available such as decreasing vessel speed in rough weather. This speed reduction could also significantly benefit seakeeping, as current MSI levels at 17 knots are around 20%.

The major downside of the 30-metre vessel is that, if scaled proportionally, 5 metre ship length extra causes an increase in 55% of the structure weight and 55% of the required battery capacity. Impacting the production price significantly. In relative terms, the 30-metre vessel has a better battery mass-to-displacement ratio. It needs significantly less increase in displacement (relatively) to carry all the required ship weights, as seen in Table 5.11. But this does not outweigh the downside of the expected increase in production costs.

In terms of rules and regulations, although Mermaid has the exemption from the authority to build a vessel longer than 25 metres, this is an insecure exemption as in the future the authority *may* decide to adhere more strictly to the Rules. Or when Mermaid may decide to expand operations to other islands in the Caribbean where the SCV code is applicable, the ship might not be certifiable.

These factors drive the decision to continue this work with the **25-metre vessel**; this decision was made in collaboration with Mermaid. This choice, however, will have implications for the prediction methods for resistance, as they are based on the 372 model tests and altering the hull too much from this reference vessel will make this method invalid. In the next chapter, the vessel will be redesigned with a more compliant displacement, and the impact on vessel mass, power prediction and battery weights will be discussed.

6 Second Iteration

6.1. Revisiting the Design Spiral

In this chapter, spiral-based design will be revisited. Below, a brief overview will be given as to 1) what this chapter will do at each step and 2) how it is different (or the same) to the previous work done. At some stages, the method will be the same, but the results may differ.

- The **main particulars** are already mostly defined; a 25-metre vessel length, some parameters comparable to that of the Delft-372 catamaran, others different.
- The **reference vessel** for comparison at this stage will be the 24/25 metre vessels from last iteration.
- The **altered hull design** is where more details will get defined, such as the Wet Deck Design and deck layout of the vessel.
- The **resistance estimation** will become even more of an estimate. Because with the increase in vessel displacement, the original method will be more detached from 372 model tests. Therefore, a new method will have to be derived.
- The **powertrain sizing method** will be largely the same, some margins will be tweaked.
- The **weight and location estimations** will also be largely the same.
- The **seakeeping calculations** will be largely the same, but the effect of slower ship speed will be investigated. As well as the effect of the waves coming from different angles, and finally the wet deck height will be investigated and the chance of slamming will be discussed.

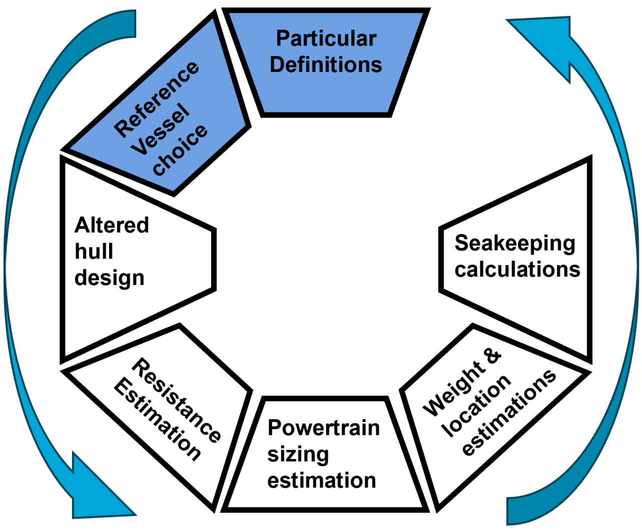


Figure 6.1: Indicative Design Spiral

6.2. Review of the Requirements

With the knowledge gained from the first iteration and the consequent mass estimation, the original requirements can be reviewed. Mermaid Boat Trips accepts a speed penalty in rough conditions; therefore, during the months of May and June, the speed is reduced to 13 knots, which reduces battery weight and MSI index while increasing travel time to 1 hour and 20 minutes.

Furthermore, for the client, the 200 pax requirement isn't as strict, which gives some design freedom in the deck layout without overly increasing the vessel's dimensions. The client prefers that part of the passengers lie in beds rather than seats, as this could reduce the effect of seasickness.

The first iteration and the sidestep dictated the necessity of an increase in vessel displacement. It is important to tread with caution, as this can initiate a negative spiral where the increased displacement causes (too much) extra resistance, significantly impacting battery weight and thus the displacement. These interdependencies are shown in Figure 6.2. (Structural mass will also increase with more displacement, but not as significantly.)

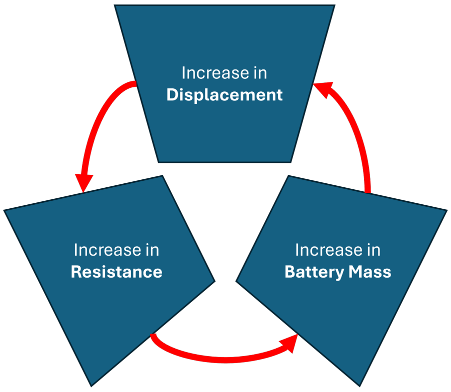


Figure 6.2: Effects of increased displacement.

	Symbol	Axe IT1	Axe IT2	unit
Length	L	25.00	25.00	[m]
Beam OA	B	7.83	7.83	[m]
Beam demihull	b	2.00	2.00	[m]
Distance between centre of hulls	S (or H)	5.83	5.83	[m]
Draught midship	T	1.15	1.26	[m]
Draught Bow	Tb	1.82	1.63	[m]
Displacement	∇	51.57	63.76	[tonnes]
Displacement	Δ	50.31	62.20	[m^3]
Wetted Surface Area	A	143.84	155.00	[m^2]
Longitudinal Centre of Buoyancy	LCB	12.55	12.78	[m]
Separation over Length ratio	S/L	0.23	0.23	[-]
Length over (demihull) beam ratio	L/b	12.50	12.50	[-]

Table 6.1: Iteration I vs. Iteration II ship particulars.

6.3. Hull Changes to Meet the Displacement Requirements

To meet the displacement requirements set in Section 5.3 the vessel particulars need to be adjusted. An overview of particulars and their changes is presented in Table 6.1. More displacement volume was aquired by changing the following aspects of the hull:

- Lowering the transom, resulting in a draft of 15cm at the transom (this was 0).
  $\rightarrow$ This is also useful for compensating for a slightly more forward COB because of the axebows.
- Lowering the midship draft by 10cm.
- Making the bow more voluminous.
  $\rightarrow$ This is also beneficial for seakeeping, as it provides greater volume forward.
- Reducing axe bow draft to reduce overall wetted surface area. $\rightarrow$ Axe bow effectiveness is also disputed as discussed in Iteration 1 Section 4.7.4.

6.3.1. S/L Relation

In Section 5.2.1, the effect of increasing the separation-over-length ratio was discussed. For the 25-metre vessel, this would reduce the calm water resistance by 7.3%. Translated to battery weight, including all margins, this would result in a mass reduction of approximately 1 tonne. However, structural mass would increase by 4 tonnes using the method from Section 5.3.2. Therefore, it is not effective to increase the S/L ratio, as the ship's displacement will already need to be increased and it is necessary to spare as much weight as possible.

6.3.2. Wet Deck Design & Height

Imperative to operability and comfort is wet deck design. A well-designed wet deck reduces the chance of slamming and mitigates the harsh accelerations if slamming does occur. This reduces the stresses on the structures and seasickness for the people on board.

Height

The biggest parameter influencing wet deck slamming is the wet deck height. This height provides a harsh boundary on the wave heights a vessel can comfortably operate in. The DNV guideline (reported by Shahraki [39]) for minimal tunnel clearance to avoid slamming is given by equation 6.2.

$$H_L = 0.22L(0.3 - \frac{0.8}{1000}L) \tag{6.1}$$

$$= 1.54 \text{ metres} \tag{6.2}$$

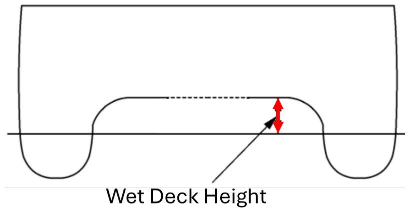


Figure 6.3: Wet Deck Height Definition [39]. Otherwise also known as tunnel clearance or air gap.

With a significant wave height of $H_S = 1.4$ metres, this clearance dictated by DNVGL is assumed to be rather low. As this formula is related to ship length, it is assumed this formula is designed for rather large vessels. From Dubrovsky's work [9] (see Figure 6.4), a more realistic guideline can be obtained.

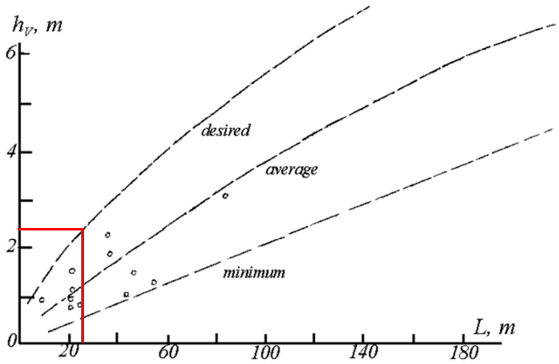


Figure 6.4: "Recommended zero approximation of vertical clearance of multi-hulls with conventional hull shape; from top – desired, average, minimal values." [9], ship length and corresponding Clearance Height marked in red.

Here, a desirable wet deck height for this ship length is around 2.4 metres. To account for a structure that reduces slamming effects and the typical roughness of the sea around Curaçao, the wet-deck height at the fore side of the vessel is set at **2.9 metres**. The downside of such a high freeboard at the bow of the vessel is that it becomes more difficult for the crew to tie down the mooring lines. *A raised dock is recommended.*

Design

A good practice to avoid slamming is moving the wet deck further aft. As this surface area isn't needed for passengers, this can easily be attained. The maximum distance the wet deck was placed aft was four metres to keep some structural integrity and to have enough space forward for the deckhouse.

Although the deck height and location can be optimised, not all slams can be prevented, as waves are statistical; this means that in the future, the vessel WILL encounter a wave that WILL slam against the wet deck. The next step is to realise a design that mitigates this shock pressure.

Commonly seen in wet deck design is a smaller, third "hull" above the waterline. Another option is the "cathedral" shape that is used in the DAMEN 2710 Catamaran, derived from the twin axe concept [14]. Given Mermaid's preference, the second option was selected. This has been implemented in the design and is visible in Figure 6.5.

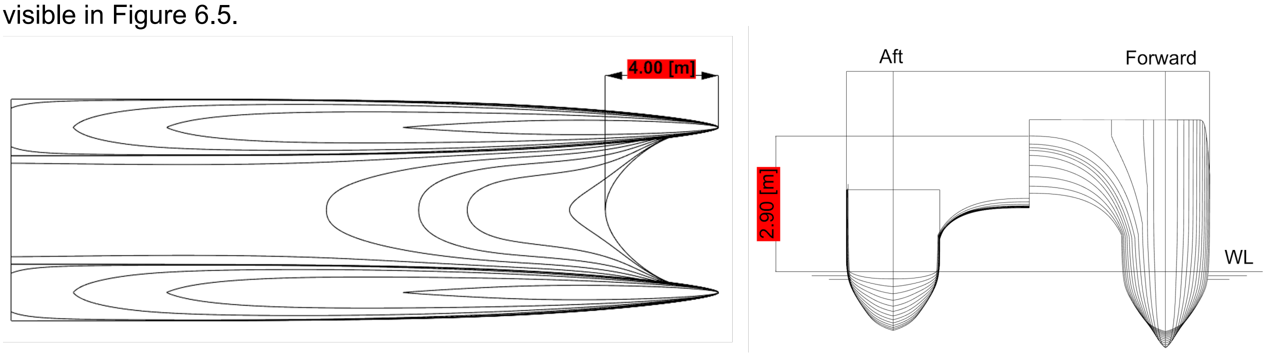


Figure 6.5: Implemented wet deck design with characteristic distances marked in red, based on the design of the twin axe concept [14].

6.4. Expected Resistance Increase

Because the vessels' displacement was increased at a ship length of 25 metres, it is safe to assume ship resistance will also increase. The method used in the first iteration was based on scaling the DUT-372 model resistance using the IITC rules and Froude's scaling law, plus adding the effect of increased wetted surface area for the axebow. Because in this iteration too many variables change compared to this original baseline, a new method needs to be derived.

As the option of CFD for resistance at this stage is not feasible a *rough* method is derived using the widely used Holtrop & Mennen method [19] [42] to calculate demihull resistance. Subsequently, the interference factors for the 372 catamaran from Broglia et al. [3] were used to find interference resistance. Formula 6.4 has been rewritten to formula 6.4 to achieve complete catamaran resistance. Where the superscript (C) denotes Catamaran and (M) denotes monohull (demihull) resistance.

$$I_F = \frac{C_T^{(C)} - C_T^{(M)}}{C_T^{(M)}} = \frac{R_T^{(C)} - 2R_T^{(M)}}{2R_T^{(M)}} \tag{6.3}$$

$$R_T^{(C)} = I_F \cdot 2R_T^{(M)} + 2R_T^{(M)} \tag{6.4}$$
$$= 15.5\% \cdot 2 \cdot 18.66 + 2 \cdot 18.66 \tag{6.5}$$
$$= 37.3 \text{ [kN]} \tag{6.6}$$

This method differed by 11% from the design speed of 17 knots with the original DUT-372. This method thus underestimates resistance; this 11% margin is therefore added to the total resistance in order to properly dimension batteries and engines. The trip overview and the total energy consumption is visible in Table 6.2.

Single Trip	Spanish Waters	Open Sea	Open Sea	Spanish Waters	
Distance (nautical miles)	1.6	13.2	13.2	1.6	Nautical Miles
Speed	7	13	17	7	Knots
CW resistance at that speed	5.63	26.62	41.57	5.63	kN
Pe	20.26	232.79	363.55	20.26	kW
Pd	35.32	405.80	633.76	35.32	kW
Time at that speed	0.23	1.02	0.78	0.23	hours
Calm Water Energy Required	8.07	412.05	492.10	8.07	kWh
Sea Margin		20%			%
Incl. margin	**8.07**	**494.46**	**492.10**	**8.07**	**kWh**

Table 6.2: Iteration II energy consumption

This results in a total required installed battery capacity of **1803 kWh** *(The original ITI capacity was 1575 kWh)*.

6.5. Design of Iteration II

A schematic version of the final design is visible in Figure 6.6. This can be considered a preliminary design to make the integration of the high forward hulls on the passengers ship. Logically, the freeboard aft is lower, as the effect of high waves are less noticeable here. And this allows for the second deck to be lower as well. The downside is that the effective surface area of the lower deck is not that big as there is a lot of space given to the wet deck tunnel.

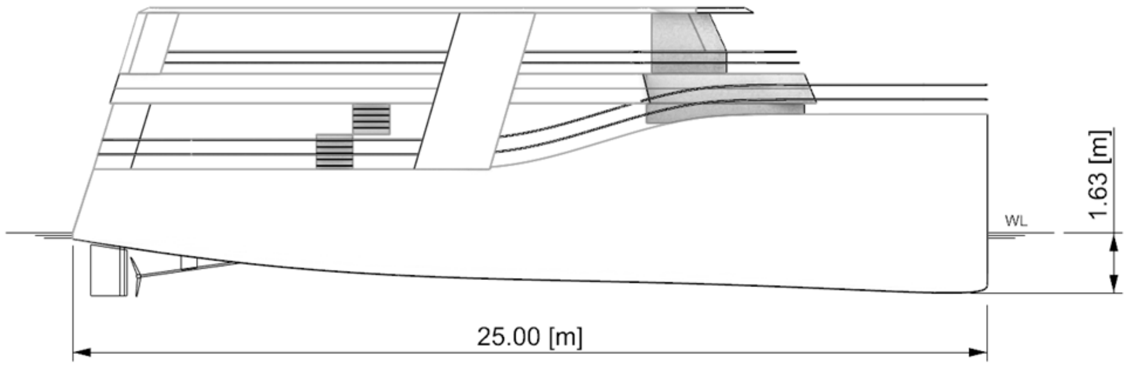


Figure 6.6: Schematic Side view

The Full General Arrangement of the 2 passenger decks and the inside of the vessel hulls is provided in Appendix E. In the current vessel's configuration, there is space for **184** passengers, with lower-deck passengers seated on benches and upper-deck passengers mostly in beds (as this is assumed to mitigate some effects of seasickness).

6.5.1. Ship Masses

The ship masses for this new hull are recalculated using the described method in Section 5.3. The results are visible in Table 6.3. The masses come within a 2 percent margin of the displacement, which is deemed acceptable for this stage of the design.

Structural	19.11	[t]
Batteries+Electrical	12.46	[t]
Genset	1.50	[t]
Remaining LS Weights	11.73	[t]
Deadweight Contributors	20.25	[t]
Total	**65.04**	[t]
Displacement	63.76	[t]
Difference	2%	[-]
Draft change	16	mm

Table 6.3: Iteration II mass overview

An analysis of the centre of gravity has not been performed in this work, as the batteries (and gensets) effectively serve as ballast, with a total weight of 13 tonnes. Their combined volume is *only* 9 cubic meters, which can be easily placed in the hulls at any point to ensure the vessel floats at neutral trim.

6.6. Seakeeping Results

In the first iteration, the 24- and 30-metre seakeeping results were compared. In this iteration, the lighter variant of the 25-metre vessel is compared to the 24-metre vessel. It is assumed that the motions and accelerations will not change significantly with the approximately 25% increase in displacement. This assumption had to be made due to calculational reasons with SEACAL.

Lewis [24] discussed these effects in the form of length-to-displacement ratios and natural period-to-length ratios. And concluded that the natural pitch period of a vessel decreases with increasing displacement. In this design case, the resonant peaks in Figures 6.9 to 6.12 would be further from the purple wave band – this shift would be about half a second (0.1 rad/s) away from the purple wave band. It can therefore be concluded that the effect of increased displacement is only preferable for ship motions (in head waves).

First, the RAOs for the 24- and 25-metre vessels are presented at 17 knots to demonstrate agreement between the two results, in the same graph the 25-metre vessel at 13 knots is plotted to display the positive effect of reducing forward speed. Secondly, for the 25-metre vessels, RMS graphs are plotted at 13 and 17 knots, and MSI values are calculated at the bridge and at COG. Lastly, the RAO's and RMS values are plotted for the oblique waves (at $215°$).

Real Wave Frequencies & Periods versus Encounter Frequencies & Periods

Again, in these seakeeping results the *real* frequencies and periods have been used, the relation to the encounter frequency period at 13 knots is given the Figures 6.7 and 6.8. The relation at 17 knots is depicted in Figures 4.9 and 4.10 of Iteration 1.

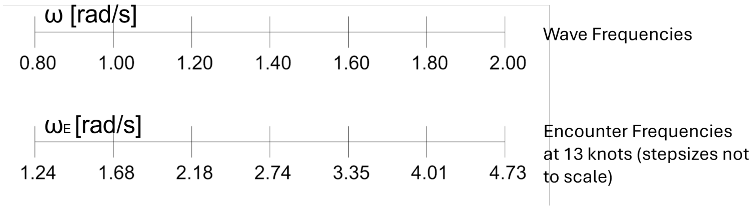


Figure 6.7: Radial wave frequency to encounter frequency at 13 knots

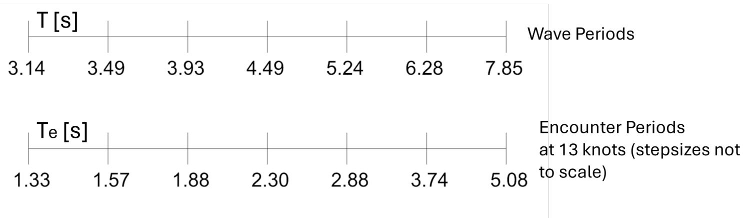


Figure 6.8: Wave period to encounter period at 13 knots

6.6.1. RAO, RMS, and MSI

In Figures 6.9 to 6.12 it is visible that the 25 metre vessel only performs slightly better in headwaves compared to the 24 metre vessel, as to be expected. And that reducing ship speed has a very positive effect on responses and accelerations. As the height of the peak is reduced and there is a significant peak shift away from the purple wave band. The MSI values between these two ship speeds change in the following manner:

- Bridge: 16% $\rightarrow$ 2.5%
- COG: 7.1% $\rightarrow$ 0.2%

This means a quantifiably better result at these ship speeds, but it remains difficult to assess how the nonlinear effects will affect these accelerations, as discussed in the first iteration.

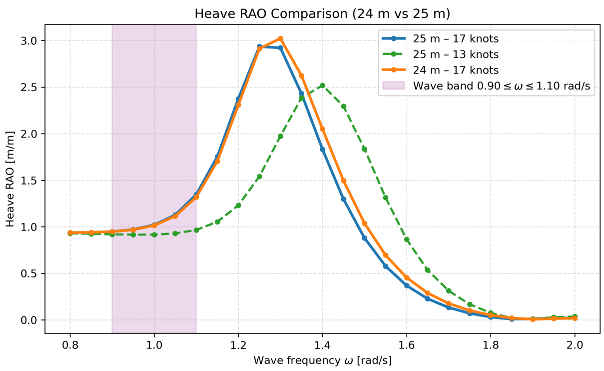


Figure 6.9: Heave RAOs at 13 and 17 knots for two vessel lengths calculated in the COG

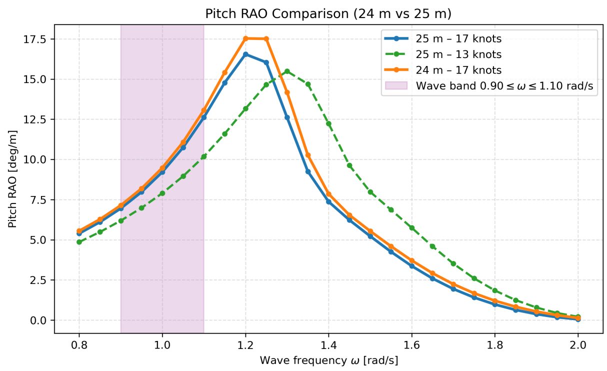


Figure 6.10: Pitch RAOs at 13 and 17 knots for two vessel lengths calculated in the COG

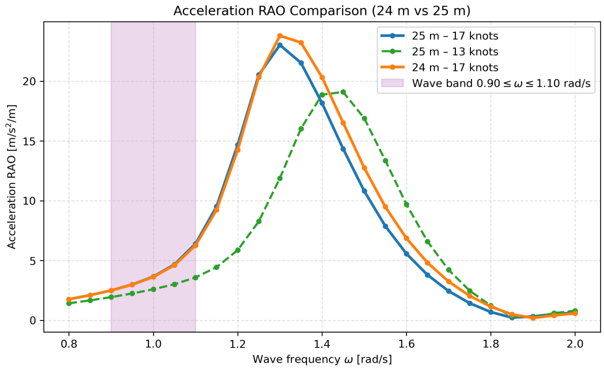


Figure 6.11: Acceleration RAOs at 13 and 17 knots for two vessel lengths calculated in the COG

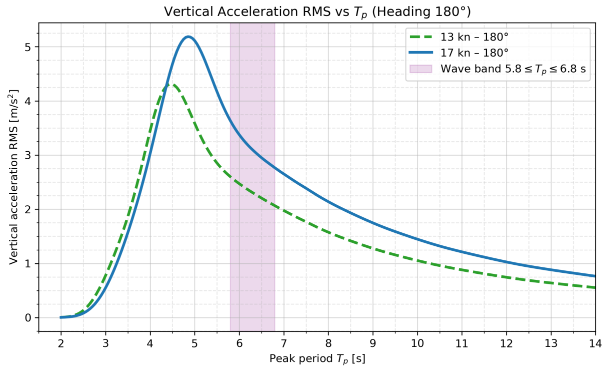


Figure 6.12: RMS graphs at 13 and 17 knots for 25 meter vessel length calculated in the COG

6.6.2. Oblique Waves

In Figure 6.13, the RMS results for oblique waves at 13 knots are plotted (at the COG on the port side hull). The individual RAOs follow this reduced trend aswell. This was analysed as the primary wave direction on the trip to Klein Curaçao is at around 215 degrees (with 180 degrees being headwaves) at a slight angle. See the analysis in appendix B. It can be concluded that this slight angle is preferable for the accelerations and thus for seasickness effects. This conclusion is in line with some current catamaran captains on Curaçao who opt for taking waves at a slight angle rather than headwaves to ”cut through” the waves more smoothly.

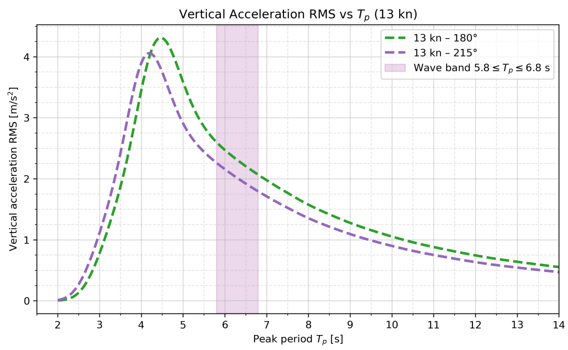


Figure 6.13: RMS graphs of two different headings at 13 knots calculated in the COG

6.6.3. Stern Following Waves

On the trip back from Klein Curaçao, there is hardly any seasickness on all vessels, even though the monohulls perceive large rolling motions, and the catamarans move a bit “stiffer” because of their higher transverse stability. Therefore, it is interesting to calculate the return trip accelerations as well to get a feeling for what happens. However, the resulting RMS accelerations in stern quartering waves with an angle of $\mu = 45$ degrees came down to practically 0 m/s^2 for any wave period. Therefore, these findings align with the findings from practice, and it is reasonable to accept that the return journey will also not cause any seasickness for the new vessel.

6.6.4. Wet Deck Evaluation and Green Water

SEACAL has also been used to evaluate if the wetdeck designed in Section 6.3.2 is sufficiently high. Although wet deck slamming is a non-linear effect, the local relative wave height can say something about the probability of slamming. Also, the chance of green water can be evaluated. Green water is water seawater that flows onto the deck and hits the superstructure. When local relative wave height on a location of the vessel is higher than the freeboard, there is a chance of green water. The calculations in this section have also been tested at a forward speed of 13 knots with head waves with a significant wave height of 1.4 metres. The 3 locations evaluated are all on the waterline and are labelled A, B, and C.

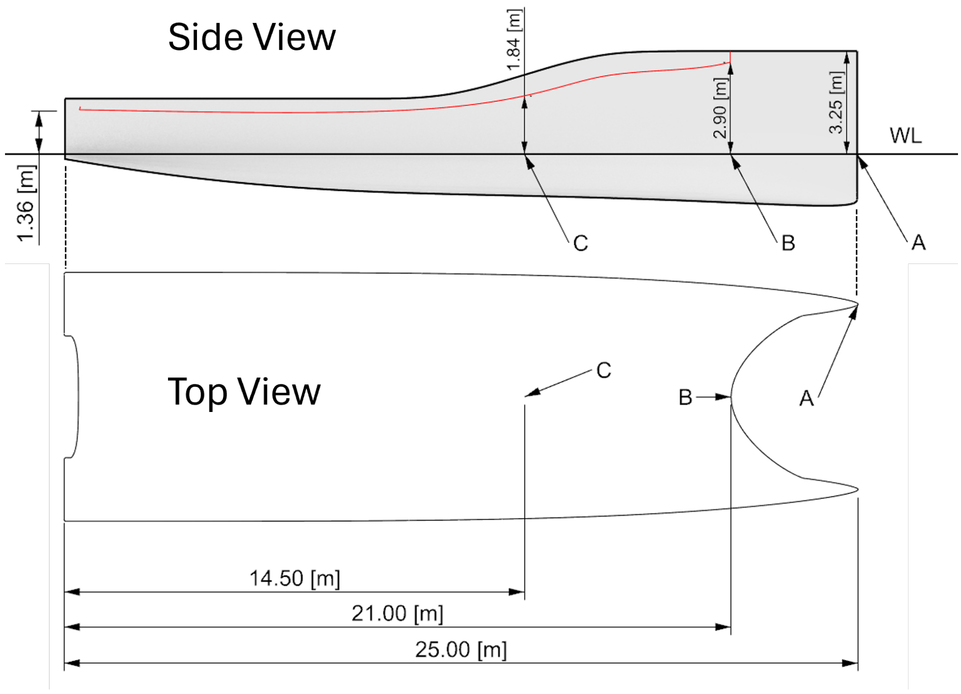


Figure 6.14: Locations for evaluation of wave height for wet deck validation

- A – Bow, to evaluate if hull pitches into the water and if there is a chance of green water.
- B – Middle and start of the wet deck, to evaluate height and location.
- C – Location where wet deck height transitions from high (2.9 metres) to low (1.35 metres). The point is chosen along this transition, where waves are expected to strike the wetdeck.

The locations have all been depicted in Figure 6.14 and the height of the wetdeck on the centreline has been marked in red. All points were evaluated at waterline height in order to see what the predicted wave height will be with respect to the waterline. The RMS results from these SEACAL simulations are shown in Figure 6.15.

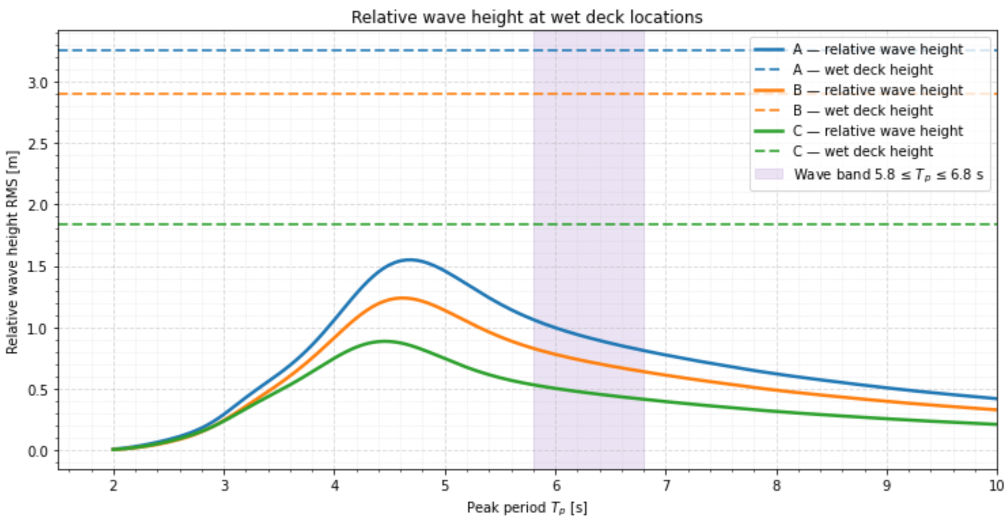


Figure 6.15: RMS relative wave heights at locations A, B and C and their maximum allowable upper limits.

It can be concluded that according to SEACAL, the freeboard and wetdeck heights are sufficient in order to prevent wet deck slamming and green water. The non-linear effects that SEACAL does not predict are more difficult to judge. Under the assumption that non-linear effects add extra damping, the motions and relative heights could be even less, reducing the chances of wet deck slamming even more. These non-linear effects, however, could also induce a phase shift, which result in the hull ending up in an unfavourable position. In this position, a second incoming wave could increase the chance of wet deck slamming. More work is needed to adequately judge what the non-linear effects will do to vessel motions, but with the design margins taken in Section 6.3.2 it could be assumed these margins are safe enough.

6.7. Conclusion

The aim of this second iteration was to define certain elements of the vessel in more detail. The displacement was increased to have displacement equal ship mass. A proper wet deck design was made according to literature and design practices, imperative for a successful catamaran. The possible effects of increased displacements on resistance have been studied. These ship masses have been recalculated and matched to the displacement. The seakeeping calculations have been redone for a 25-metre vessel, and vessel speed has been reduced in rough periods in an effort to bring down the MSI values. And finally, the wet deck design has also been validated by evaluation local relative wave height with SEACAL.

7
Further Work & Current Limitations

In this work, two full iterations of the design spiral (Figure 3.1) have been performed. The complex process of designing a full ship will always require further work right until the ship will be actually built. In this section, the “main uncertainties” will be discussed, that are eligible for to be further work.

From the main design tasks, such as Resistance Analysis, Mass Predictions and Motion Analyis the figure from the design philosophy and approach is revisited in Figure 7.1.

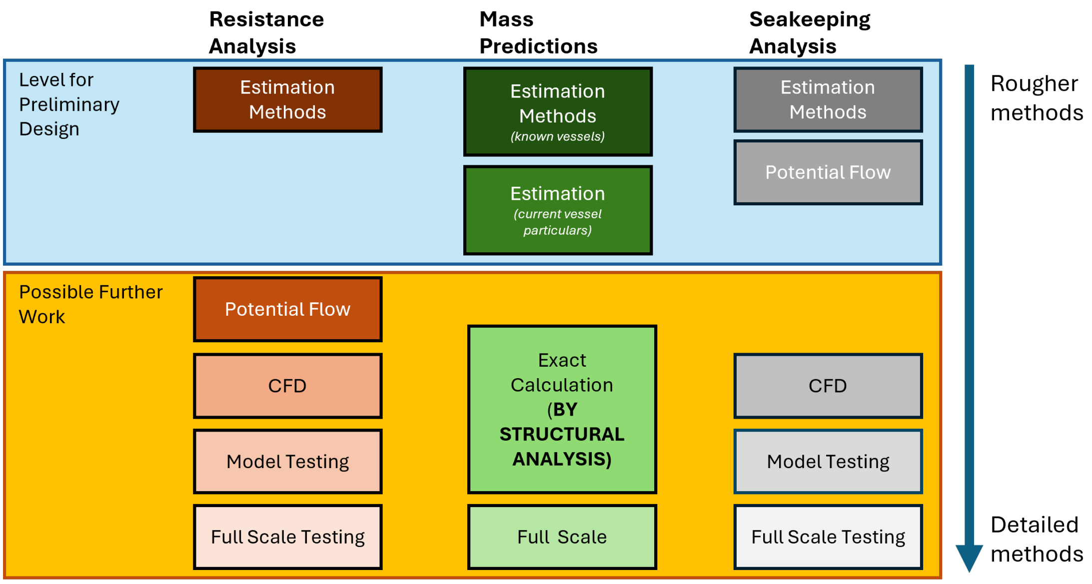


Figure 7.1: Research depth levels for this and further work

This figure grasps the full possibilities of further work for the design of this vessel. It also shows where current limitations lie.

Calm Water Resistance currently is limited as it is based on an altered ship model + a Holtrop & Mennen modification. *CFD is recommended*.

Structural analysis is required to calculate the interaction between the ship and its structures, enabling the complete prediction of ship masses. Where SEACAL can be used to calculate the structural loads.

Slamming is considered a significant element in the seasickness of passengers, as this contributes to a lot of vertical accelerations. This slamming, however, is not predicted by SEACAL and remains a difficult effect to model. This design has implemented the best design practices (axebows and a high wet deck) to mitigate these effects. However, it cannot be guaranteed that there will be no slamming.

To fully calculate the calm water resistance as well as the nonlinear seakeeping behaviour, model testing would be advised. This also provides the advantage of the ability to calculate added resistance in waves, to more accurately judge the applicable sea margin.

7.1. Other Further Work

Apart from these 3 main elements, there are of course plenty of other topics that require further work. A small list has been added in order to map the possibilities.

- Investigate the effects of a high continuous superstructure to provide strength.
- When true (structural) ship mass is known, investigate the advantage of increasing S/L ratio.
- Hydrodynamically investigate the seakeeping behaviour of an axebow in stern quartering waves, as axe bows are known to cause some lateral lift, resulting in extra yawing motions, aft skegs could possibly counteract this.
- Investigate the possibility of increasing the pitch radius of gyration to reduce accelerations, this could be relatively easily achieved by placing batteries in the hulls further away from each other.
- Investigate other battery types, their cost advantages and the feasibility of their increased mass, when structural mass is more accurately known.
- Mitigating green water effects, although high freeboard design aims to reduce the probability of green water on board. A 'rogue' wave even could cause green water to splash through the decks, a good deckhouse design can mitigate these effects.
- Operability is of high importance for Mermaid, investigate accessible engine room hatches in order to do battery or engine swaps within one day.

8 Conclusion

In conclusion, this report tried to cover all aspects governing *the Design and Hydrodynamic Optimisation of a Battery-Powered Passenger Vessel for the Caribbean Sea around Curaçao*. The main design goals were to provide a preliminary design for Mermaid Boat Trips that is optimised to minimise vertical acceleration to mitigate seasickness, and to provide a design suitable for full-electric sailing to improve the sustainability of its operations. Following the set design spiral, two complete iterations have been performed, focusing on resistance, mass predictions and most importantly seakeeping calculations.

The ship resistance was first based on model tests of the DUT-372 reference vessel, and later on an adjusted Holtrop & Mennen method. It is recommended that these results are verified with CFD, as catamaran resistance is a comprehensive subject where viscous resistance and the interference effects of wave making resistance complicate the use of traditional prediction methods.

Ship masses were retrieved using estimation methods, and the results were consistent with the displacements of similar electric vessels. However, a comprehensive structural analysis is required to determine the true vessel mass. It must be noted that some aspects may be over-dimensionalised, as the estimation methods originate from vessels in which ship mass is less critical. Due to the low gravimetric power density of batteries, battery-powered ship design initiates an intricate design loop: increased resistance requires more battery mass, which in turn increases displacement, and the cycle repeats. Therefore, electric vessels must be as lightweight as possible.

The ship motions were analysed with MARINs SEACAL potential-flow software. Based on these results, the decision was made to reduce ship speed to 13 knots in rough weather. This resulted in satisfactory MSI values of around 2.5% at the bridge. Slamming effects require further investigation, as these are not covered by SEACAL. Best design practices, such as axebows and high wet decks, have been applied to mitigate slamming effects and green water. Wet deck and freeboard heights have been verified with SEACAL and proved to be sufficient.

Under the used calculational methods within this research, a converging ship design was reached, performing well within the given constraints. This means a realistic vessel:

- On which enough passengers fit;
- That has low predicted seasickness;
- That reaches Klein Curaçao on battery power within 1 hour and 20 minutes on rough days (within an hour on normal days);

However, further work is required in order to more accurately predict the three key research topics within this ship design: Resistance, Ship Mass and Seakeeping analysis. Where more accurate results for resistance and ship mass can be achieved through CFD and structural analysis, non-linear seakeeping remains a difficult subject. Because CFD for seakeeping is not advanced enough yet to properly calculate the slamming effects the vessel will encounter. Good design practices such as high freeboards, slender hulls, axebows and high wet decks can mitigate these effects but to know the true effects model testing needs to be performed.

Overall, this vessel design provides a solid baseline for future iterations to enhance passenger comfort and sustainability for Mermaid Boat Trips.

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A

SEACAL Method and Validation

As mentioned in the first iteration, a reasonable frame of reference is essential in order asses the results that SEACAL calculates. The aforementioned DUT-372 by Van't Veer [44] is the frame of reference for the design in this research. In earlier work SEACAL has been validated with the practical experiments of the DUT-372 catamaran [28]. In the figures in Section A.1 the following ships have been assessed:

- Original Experimental results of the DUT-372 (scaled from Non-Dimensional to 24m ship length)
- SEACAL calculations of the DUT-372 at 3m scaled to 24m ship length
- SEACAL calculations of the Axe Design Catamaran at 24m ship length

Differences

What important to note is that, as apposed to normal conventions, the RAO's for the Axe Design Catamaran is **NOT in the COG**, but in the old COG acquired from the scaled model of the DUT-372 catamaran (see LCG scaled in Table 4.3). This had to be done to provide an equal frame of refence to analyze the RAO's in the exact same point on each vessel. The reason that the axe catamaran has a different LCG is due to the LCB being further forward due to the deep forefoot of the vessel. And LCG needs to equal LCB to float upright. If these RAOs hadn't been analysed at the same point, the Heave RAO of the axe vessel would have been bigger in heave due to an extra pitch component being integrated in the heave RAO.

Scale factors

Because results (RAOs) of several ship lengths and wave frequencies are scaled to the initial 24-metre case, several scale factors apply. These scale factors are depicted in Table A.1.

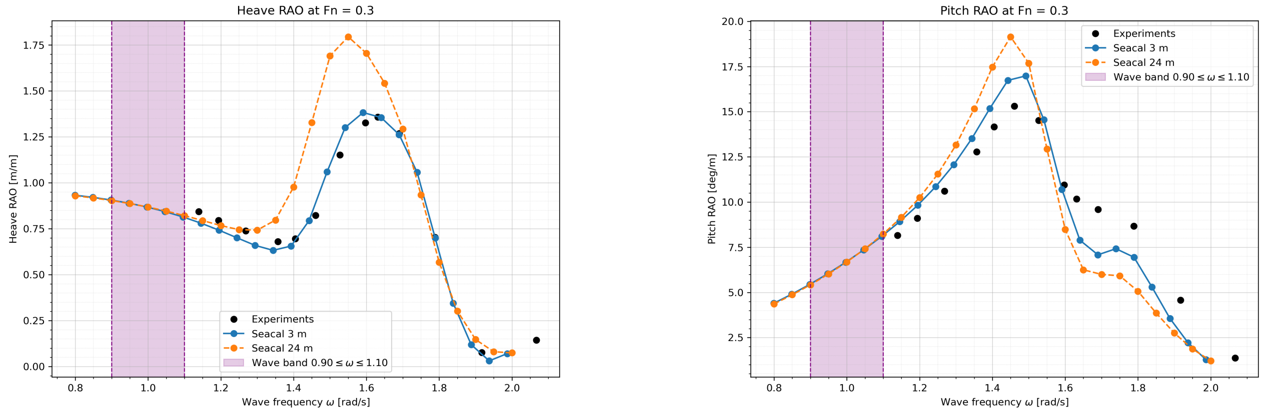
Quantity	Scaling Relation	Scaling factor (3m $\rightarrow$ 24m)
Linear dimension	λ	8
Heave RAO $[m/m]$	[-]	[-]
Pitch RAO $[deg/m]$	$1/\lambda$	0.125
Accelerational RAO $[m/s^2/m]$	$1/\lambda$	0.125

Table A.1: RAO scaling relations relevant for scaling experimental results by Van't Veer [44], and numerical results of the DUT-372 catamaran at 3-metre ship length.

Result interpretation

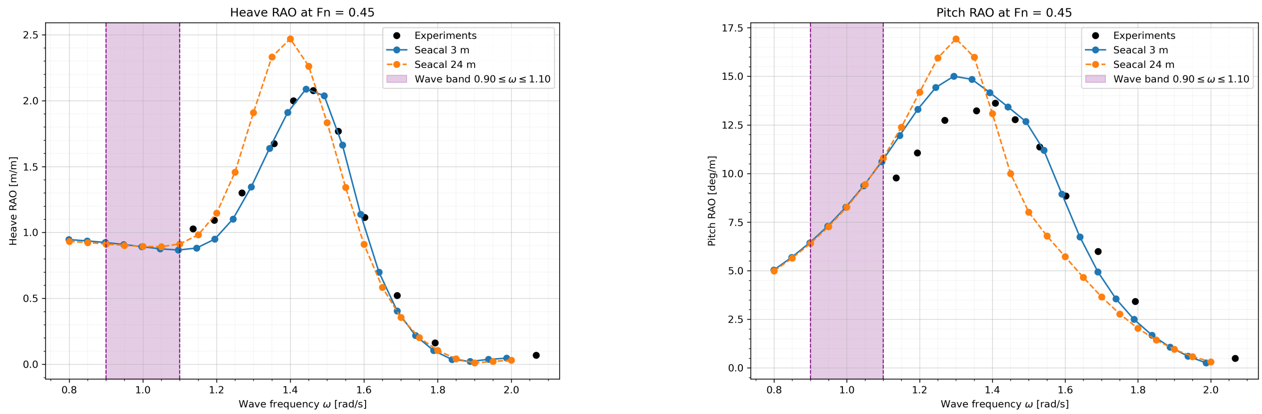
As was known from MARIN [28], at higher Froude numbers (0.6 and up), the SEACAL and Experimental results start to differ (in the range of around 30+%). It is assumed that this is largely attributed to the lack of viscous damping and other non-linearities not covered by SEACAL. What can be seen is that the frequency peaks of all three vessels align and thus SEACAL can give a good indication on where the resonant wave frequencies lie and in what order they can help predict ship motions.

A.1. Iteration 1 Seakeeping Comparison



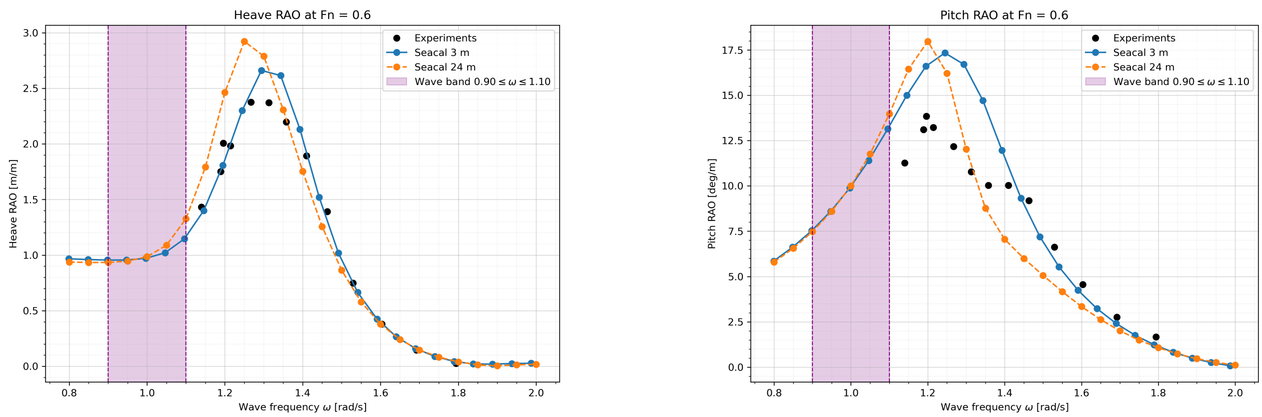
(a) Heave RAO at $Fn = 0.30$ **(b)** Pitch RAO at $Fn = 0.30$

Figure A.1: RAO comparison at $Fn = 0.30$



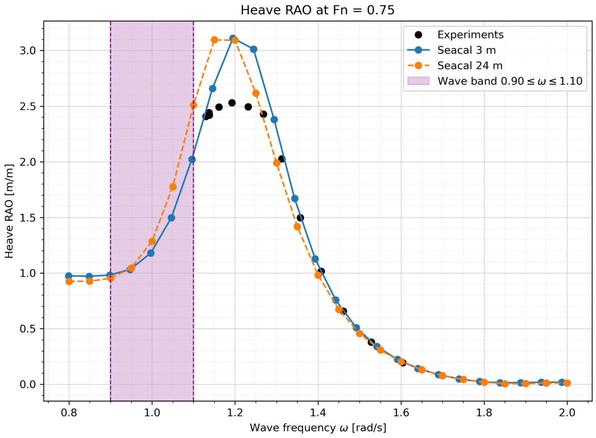
(a) Heave RAO at $Fn = 0.45$ **(b)** Pitch RAO at $Fn = 0.45$

Figure A.2: RAO comparison at $Fn = 0.45$

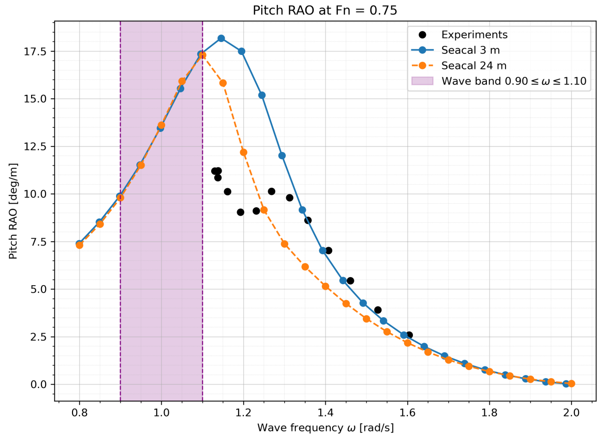


(a) Heave RAO at $Fn = 0.60$ **(b)** Pitch RAO at $Fn = 0.60$

Figure A.3: RAO comparison at $Fn = 0.60$



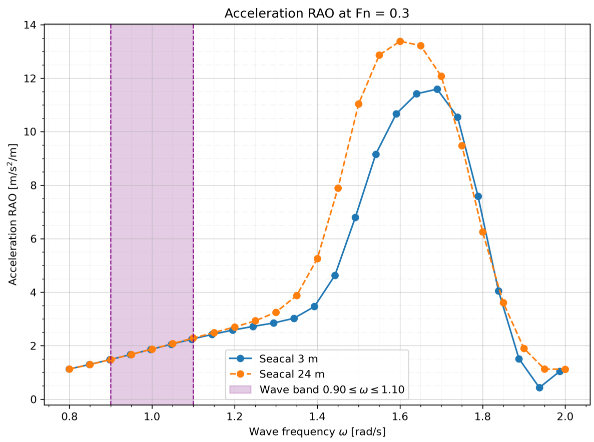
(a) Heave RAO at $Fn = 0.75$



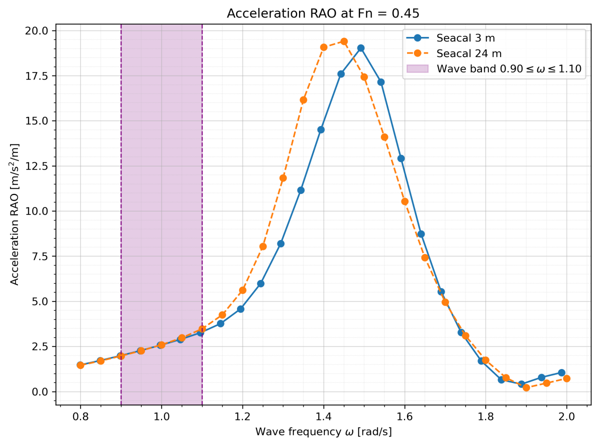
(b) Pitch RAO at $Fn = 0.75$

Figure A.4: RAO comparison at $Fn = 0.75$

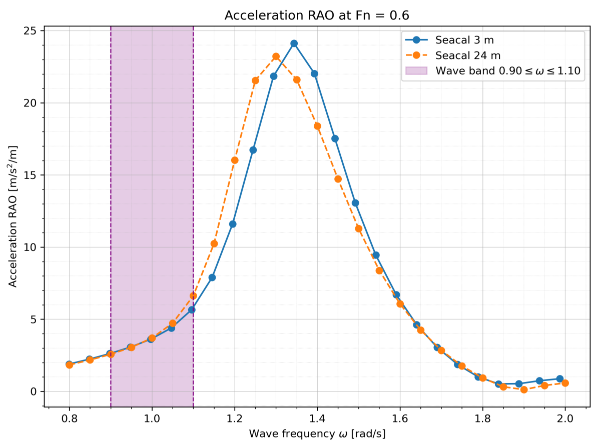
A.2. Acceleraltional Comparisons



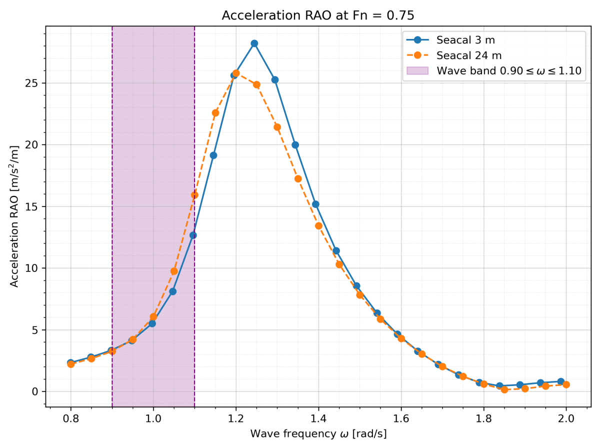
(a) Acceleration RAO at $Fn = 0.30$



(b) Acceleration RAO at $Fn = 0.45$



(a) Acceleration RAO at $Fn = 0.60$



(b) Acceleration RAO at $Fn = 0.75$

B
Curaçao Wave Spectrum

Using the Marin database (Copernicus [26]), Curaçao wave data were analysed. As May and June are considered the ”roughest” months for mermaid, the significant wave height and peak period from these time periods were taken.

This resulted in: $H_S = 1.4$ [m], $T_P = 6.2$ [s]

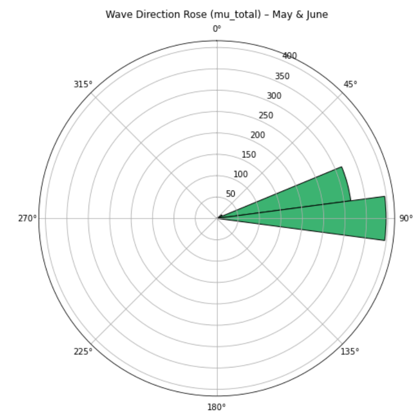


Figure B.1: Wave Direction Rose (May–June)

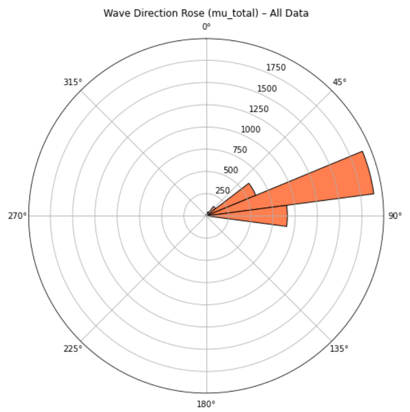


Figure B.2: Wave Direction Rose (All Data)

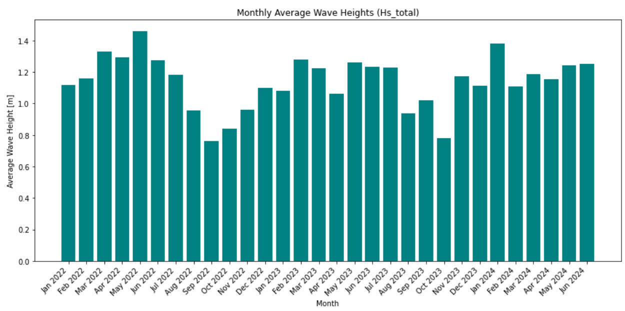


Figure B.3: Monthly Average Significant Wave Height

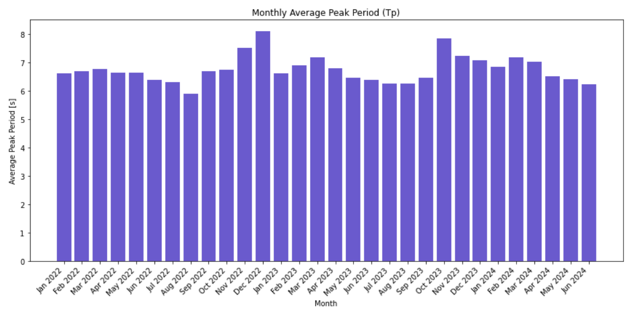


Figure B.4: Monthly Average Peak Period

C
MSI Formulas

$$\text{MSI} = 100\,\Phi(z_a)\,\Phi(z_t') \quad \text{in \%} \tag{C.1}$$

$$z_a = 2.128\log\left(\frac{a}{g}\right) - 9.277\log(f) - 5.809\,[\log(f)]^2 - 1.851 \tag{C.2}$$

$$z_t' = 1.134\,z_a + 1.989\log(t) - 2.904 \tag{C.3}$$

$$\Phi(z) = \frac{1}{\sqrt{2\pi}}\int_{-\infty}^{z} e^{-\chi^2/2}\,d\chi \tag{C.4}$$

$$a\,[\text{m/s}^2] = \text{RMS value of generalised vertical acceleration estimator } G_{\text{av}} \tag{C.5}$$

$$g\,[\text{m/s}^2] = \text{acceleration of gravity } (g = 9.81) \tag{C.6}$$

$$f\,[\text{Hz}] = \text{peak frequency of the generalised vertical acceleration spectrum } S_{G_{\text{av}}} \tag{C.7}$$

D

Grubisic -- Mass Estimation Methods

D.1. Structural Weight Estimation Method

Grubisic [16] bases hull weight off of "the plating area of four main components, *i.e. bottom, sides, deck and bulkheads*". This means the method focuses on the **Surface Area's of the parts of the hull**, which get plugged into a weighted formula for total surface area S_R.

$$\text{Bottom:} \quad S_1 = 2{,}825 \cdot \sqrt{\Delta_{FL}} \cdot L_P \tag{D.1}$$

$$\text{Sides:} \quad S_2 = 1.09 \cdot (2 \cdot L_{OA} + B_M) \cdot (D_X - T_X) \tag{D.2}$$

$$\text{Deck:} \quad S_3 = 0{,}823 \cdot \left(\frac{L_{OA} + L_{WL}}{2}\right) \cdot B_M \tag{D.3}$$

$$\text{Bulkheads:} \quad S_4 = 0.6 \cdot N_{WTB} \cdot B_M \cdot D_X \tag{D.4}$$

With Δ_{FL} denoting full load displacement. L, D, T, and B are equal to Length (L), Depth (D), Draught (T), and Beam (B) used throughout the report. L_{OA} and L_{WL} for the axebow design are all equal.

N_{WTB} denotes the number of bulkheads. Which is determined in accordance with the rules of Regulation **III/20** of the SCV code [40]. Which describes the maximum spacing between bulkheads. The forward crash bulkhead must be between 5% and 15% of the forward particular. The other spacings are either taken as one-third of the Bulkhead Deck Length (L) or obtained from the table of floadable length factors in the rules, whichever is less. In the case of the 25- and 30-metre vessels, the former rule takes effect. So a maximum spacing of either 8.333 metres (25m vessel) or 10 metres (30m vessel). Requiring both vessels to have **4 watertight bulkheads**.

The Surface Area's from Grubisic have been calculated in 2 separate ways. First, the dimensions have been plugged in so the area S_R for an overly wide monohull was calculated. The second method consists of calculating the area numeral using Grubisic just for one demihull, and then adding the full superstructure weight on top.

The complete area numeral comes down to:

$$S_R = S_1 + 0.73 \cdot S_2 + 0.69 \cdot S_3 + 0.65 \cdot S_4 \tag{D.5}$$

The final effective surface area is determined by E_S; several numerals account for minor corrections.

$$\Delta_{LR} = 0.125 \cdot \left(L_{LR}^2 - 15.8\right)\ t \tag{22}$$

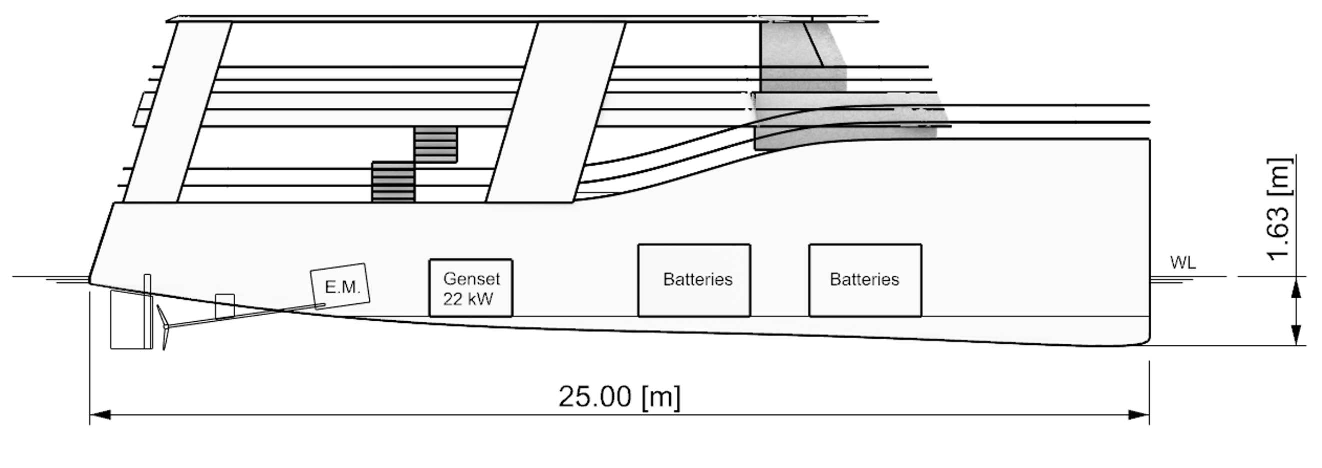
$$f_{DIS} = 0.7 + 2.4 \cdot \frac{\nabla}{L_{WL}^2 - 15.8} \tag{23}$$

$$C_{T/D} = 1.144 \cdot \left(\frac{T_X}{D_X}\right)^{0.244} \tag{24}$$

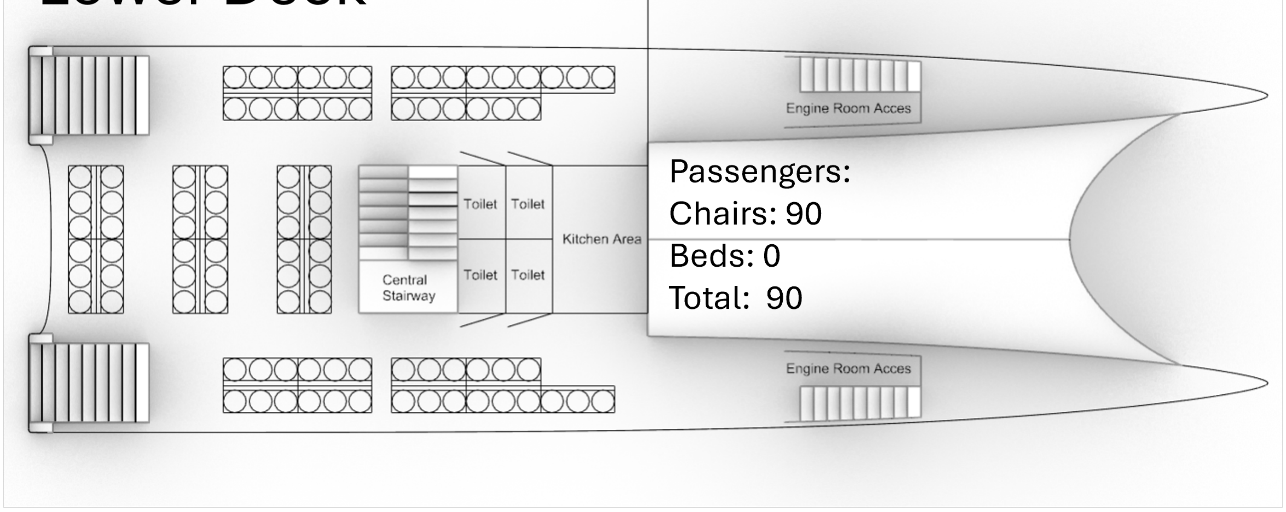
$$E_S = f_{DIS} \cdot C_{T/D} \cdot S_R \quad [\mathrm{m}^2] \tag{25}$$

E General Arrangement IT2

Side View



Lower Deck



Upper Deck