THE UNIVERSITY OF BRITISH COLUMBIA

HySuCat - Parcel Delivery Vessel

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NAME 591 Computer-Aided Ship Design Project

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Abstract

This report presents the preliminary design of HySuCat, a high-speed hydrofoil-supported catamaran developed as part of the NAME 591 course at the University of British Columbia. Conceived for parcel delivery between Richmond and Squamish, the vessel is engineered to be fast, efficient, and environmentally sustainable—while showcasing innovative hydrofoil integration and advanced marine engineering.

At the heart of the design is a hydrofoil-assisted hard-chine catamaran, optimized to reduce wetted surface area, minimize wave-making resistance, and enable smooth transitions to semi-planing regimes. A key innovation lies in the selection and configuration of the Avion hydrofoil system, consisting of one main forward foil and two aft foils, all embedded within the catamaran tunnel for debris protection and draft control. This arrangement generates up to 40% of the vessel's lift at cruising speed, significantly reducing hull-borne resistance and improving fuel efficiency.

The hydrofoil design was developed using XFOIL-based aerodynamic profiling, spanning eight NACA airfoil series, over 350,000 simulation data points, and Reynolds numbers up to 10 million. Lift, drag, and wave-interaction effects were corrected based on empirical models to reflect real operating conditions. The final selection—an asymmetric NACA 1412 foil—was chosen for its high lift-to-drag ratio, manufacturability, and structural compatibility.

From a construction perspective, the vessel uses marine-grade NV5083 aluminum, balancing lightweight performance with durability in BC's coastal environment. Structural plating and bulkhead design follow DNV HSLC rules, and damage stability was rigorously evaluated with five watertight compartments to ensure survivability.

Powered by a hybrid hydrogen-electric system, HySuCat combines a 340 kW Corvus Pelican fuel cell with a 145 kWh Corvus ORCA battery pack. Propulsion is delivered via twin Evoy electric motors retrofitted to Volvo Penta stern drives with contra-rotating propellers. An integrated thermal imaging debris detection system enhances safety by detecting floating hazards in all weather conditions and low visibility.

HySuCat exemplifies a systems-based design approach, integrating hydrodynamics, structural engineering, and green propulsion into a coherent and forward-looking solution for coastal logistics.

1 Introduction

1.1 Why Does this report exist?

The purpose of this report is to satisfy the University of British Columbia's Naval Architecture and Marine Engineering requirements for NAME 591. This report covers an initial design concept of a high-speed parcel delivery vessel as requested by *Greenline Ferries*[1]. It is meant to portray the knowledge and skills learnt throughout the program applied to a design challenge.

1.2 Statement of Organizational Purpose

The purpose of the organization for this report is to define and cover the requirements of the high-speed parcel delivery vessel design and develop a vessel concept which satisfies the requirements of the course and client.

1.3 Intended Audience

The intended audience of this report is the UBC NAME's teaching and leadership team. This report is intended to provide a technical analysis into the feasibility and performance of a high-speed parcel delivery vessel. It covers only a preliminary design and investigation into the performance of the vessel and is not intended as a final design proposal.

1.4 How is this Report Organized?

This report is broken down into subsections based on the elements of the design process and organized based on when each step was performed, following the recommended process outlined by the design report outline. The initial sections of the report present the design challenge and how the concept of operations (CONOPS) are transformed into key performance parameters (KPPs) and requirements. Following these sections, the report covers the preliminary detailed design of the vessel and how it meets the requirements set forth by the client.

1.5 Who Produced this Report?

This report was produced by the design team who completed this project: John Park, Nithin Badu, Karan Rajani and Jiahao Mei.

1.6 Overview of the ship Design Project

The NAME 591 design project acts as a design exercise to allow students to practice the skills and knowledge learnt throughout the NAME program to a real-life application. In this project the team was challenged with designing a high-speed parcel delivery vessel that incorporates a hydrofoil such that it can be submitted for Mandles' Prize[2].

[1] https://www.greenlineferries.com/
[2] https://foils.org/mandles-prize/

2 Misson - Concept of Operations

Currently, parcels from Metro Vancouver to Howe Sound communities are transported via delivery vans along roads and highways. To alleviate traffic congestion on these routes, the owner plans to introduce a high-speed premium parcel delivery vessel between these regions.

The vessel will initially operate two daily voyages, delivering parcels between River Rock Port in Richmond and Darrel Bay in Squamish. Operating in local waters, the vessel must comply with all applicable Transport Canada domestic commercial vessel requirements or their equivalent.

The vessel is designed to transport small to medium-sized parcels with a carrying capacity of at least 24 cubic meters. Parcels will be handled manually and will primarily consist of high-value or critical items for which customers are willing to pay a premium, such as legal documents, medical samples, and car parts. Additionally, the cargo may include liquids, fragile goods, temperature-sensitive items, and medical samples. The cargo space will be temperature-controlled between 15°C and 25°C. Each parcel can weigh up to 30 kg, with a maximum size of 2m x 2m x 2m. The cargo space will feature a single compartment equipped with shelves or other means to securely store parcels. Loading and unloading operations must be completed within 60 minutes to ensure operational efficiency. The vessel will carry up to 200 kg of parcels per trip, with delivery fees ranging from \$20.00 to \$30.00 CAD per parcel.

The vessel will cruise at a minimum speed of 30 knots to enable high-speed travel and will have a range of at least 70 nautical miles. It will feature a deadweight capacity of up to 2.5 tonnes and a draft of less than 2.4 meters. The vessel must be capable of maneuvering between ports under the weather conditions outlined in Table 1 below.

Conditions	Favourable	Marginal	Unfavourable	Expected Reliability
Wind Speed	Up to 20 knots	20 - 30 knots	Over 30 knots	
Wave Height	Up to 1.25 m	1.25 - 1.75 m	Over 1.75 m	
Summer (North Westerlies)	98.2%	1.8%	0.0%	99.5%
Winter (South Easterlies)	91.7%	7.7%	0.6%	97.5%

Table 1: Environmental Conditions

The vessel will be operated by the minimum crew required under *Transport Canada* regulations. Since the crew will spend 8-hour shifts onboard, a washroom must be included.

To ensure the safety of the vessel and its cargo, a debris detection system is required to be installed. This system must be accurate and reliable at high speeds, capable of identifying small objects that radar may miss, and effective in minimizing "sea clutter." It must function reliably in all weather conditions, sea states, and during both day and night. To control costs, the debris detection system should leverage proven, existing technology.

To minimize ownership and operational costs, the vessel will be electrically propelled, and its hull will be constructed from aluminum, a material known for its durability along the BC coast. The vessel must complete its two daily trips without downtime and is expected to have a service life of 25 years. It must also demonstrate reliability under the specified weather conditions.

3 Parametric Model

3.1 Summary of area and Volume Requirements

Space and volume are critical factors in vessel design. For our project, we adopted the *System-Based Ship Design* approach developed by *Kai Levander*. All required volumes and areas onboard were estimated using data from manufacturer's manuals. In some cases, gathering this data proved challenging and time-consuming. Certain values had to be approximated using assumptions or comparisons with reference vessels, while others were derived directly from technical drawings of ship systems.

Once all space and volume requirements were determined, we calculated the gross area, gross volume, and gross tonnage. Special attention was given to the cargo space, additional margins were included to allow for wall linings and smooth movement of cargo shelves. *Kai Levander*'s compendium was an invaluable reference, especially for estimating volumes that could not be obtained directly.

A system summary of the vessel is presented below. It outlines the vessel's size and capabilities to fulfill its intended mission and shows how space is distributed onboard across various systems and operational needs.

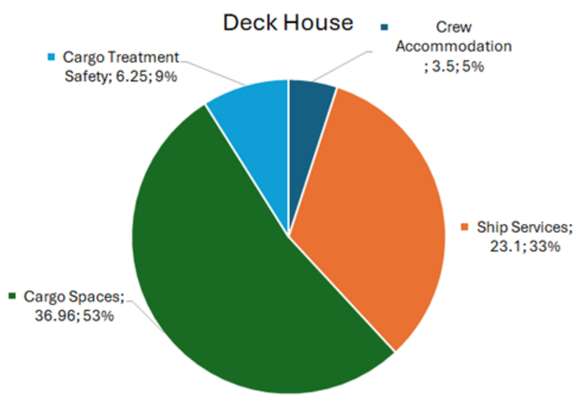

Figure 1: Deck House Volume

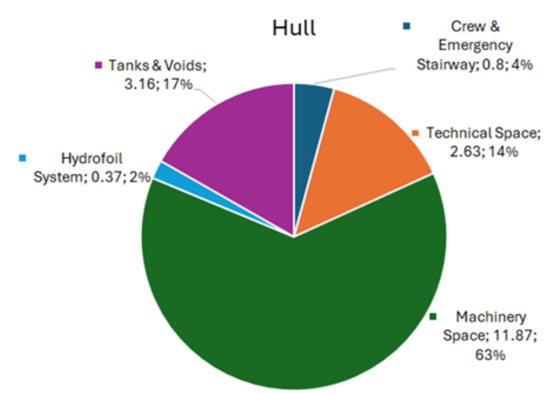

Figure 2: Hull Volume

From Figure 1 and 2, the initial estimated volume of the deckhouse was approximately 70 m^3, while the hull volume was calculated to be around 19 m^3. The relatively low hull volume, when compared to the deckhouse, appeared unusual. This discrepancy might arise from the use of *Kai Levander*'s method, which accounts only for the exact space occupied by machinery and equipment, excluding additional allowances such as clearance between components and access space for operation and maintenance.

To address this, the hull volume was later revised to approximately 61 m^3 based on measurements obtained from the 3D model developed in *Rhino3D*, which more accurately represents the required internal volume.

4 Hull Form and Hydrofoil Selection

4.1 Hull form and Foiling Arrangement Selection

A hydrofoil-supported (or hydrofoil-assisted) catamaran offers two major hydrodynamic advantages: they reduce the hull's wetted surface area, thereby decreasing frictional resistance, and they diminish wave-making resistance by partially lifting the hull out of the water. In addition, the hydrofoils are better protected in this arrangement, typically placed closer to the keel or even within the tunnel of a catamaran hull, reducing the risk of damage from floating debris and aiding in draft control.

Given these advantages, a catamaran hull form with a hydrofoil-supported configuration was selected. This approach satisfies the operational draft requirements, ensures sufficient hydrofoil protection, and provides greater flexibility in the foil arrangement.

In evaluating the hull cross-section, two options were considered: round bilge and hard-chine forms. While a round bilge hull is more efficient in calm water due to smoother flow and reduced frictional resistance, it tends to suffer from adverse pressure distribution around the bilge area. According to *Migeotte*, this low-pressure region creates suction forces that interfere with the hydrofoil's lifting effect, reducing its efficiency. To counteract this issue, a hard-chine hull was selected. Although it introduces higher resistance at low speeds, the hard chine promotes effective flow separation at transitional speeds, minimizes suction effects, and facilitates a quicker and smoother transition to semi-planing conditions — critical for maximizing the benefits of hydrofoil assistance.

Thus, the combination of a hydrofoil-supported symmetrical catamaran with a hard-chine hull form was finalized to meet the design, operational, and competition requirements.

4.2 Hydrofoil

The next challenge was to identify the foiling arrangements among different options, including avion, canards with interceptors, mono-foil, hysuwac and so on.

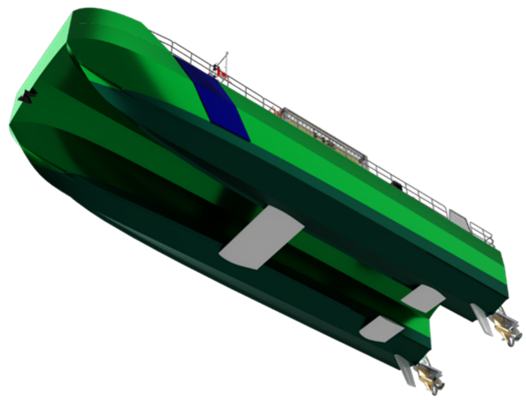

Figure 3: Avion/Hysucat Arrangement

the Avion system shown in Figure 3 has demonstrated widespread success, being deployed on over 200 planning catamarans, primarily ranging from 5 to 20 meters in length (with one known application on a 36-meter vessel). It provides sufficient lift—typically 25–50% of the vessel's displacement—and ensures better protection of the hydrofoils, as they are positioned within the catamaran tunnel at or above the keel level. This placement not only reduces the likelihood of damage from debris but also helps to control the vessel's draft more effectively compared to other systems.

Considering all these factors, the Avion system was chosen for our vessel's design, as it best aligns with our operational requirements and vessel size while ensuring safety, efficiency, and manageable draft characteristics.

4.3 Hull Geometry

4.3.1 Principal Dimensions and Coefficients Table

The principal dimensions and coefficients of form are presented in this section. The principal dimensions of the catamaran and coefficients are outlined in Table 2.

Length overall, LOA	13.50 m	Deck clearance, t	0.40 m
Length waterline, LWL	13.00 m	Displacement	19.32 m3
Demihull beam, b	1.75 m		19.80 t
Demihull beam at waterline, BWL	1.46 m	Block coefficient, Cb	0.72
Beam overal, B	5.00 m	Prismatic coefficient, Cp	1.44
Draft, T	1.30 m	Waterline coefficient, Cw	0.88
Depth, D	2.40 m	Midship Coefficient, Cm	1.46
Free board, F	1.10		

Table 2: Principal Dimension

4.4 Hydrostatics and Stability

To analyze the hydrostatics and stability of the ship, *MAXSURF Stability* is used. The load case considered is fully loaded which is 19.8 tons.

4.4.1 Stability Criteria used

The stability criteria we used is *DNV Rules for high Speed - Light Craft and Naval Surface Craft. Pt. 5. CH. 7. Sec. 5.* To simulate the effects of wind heeling whilst the vessel is rolling in waves, the following requirements should be met.

- The heeling arm should be not greater than six tenths of the maximum righting arm.
- Area 1, which is taken from the equilibrium with the gust wind heeling arm should be 1.4 times larger than Area 2 which is taken from rolled bank angle to the vessel equilibrium angle ignoring the wind heeling arms.

4.4.2 Stability Results Summary Table

Based on the criteria used, a plot of the intact righting arm versus the heeling arm for the case of wind heeling for the fully loaded case can be seen in Figure 4. The fully loaded case presents the worst stability case and therefore it can be assumed that if the stability criteria is met for the fully loaded vessel, it will be met for the lightship condition.

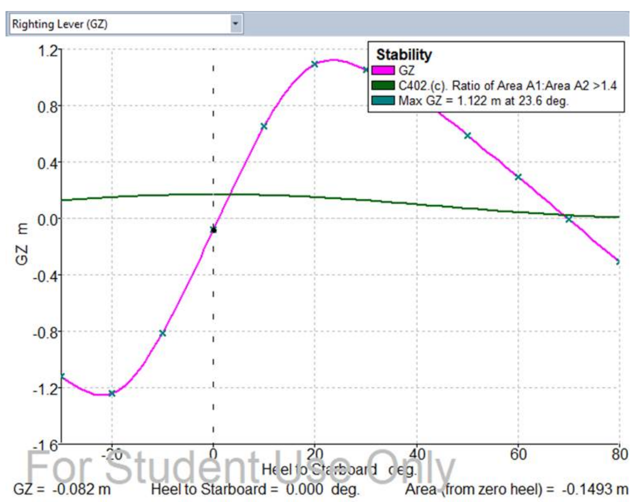

Figure 4: Righting arm plot

From Table 3, we could observe that the ratio of Area 1 to Area 2 is about almost 2, GZ_s is less than 15% of GZ_{max} which meets the requirements.

Area 1	37.25 m deg
Area 2	19.38 m deg
GZ_s	0.166 m
GZ_{max}	1.109 m

Table 3: Requirement Result Table

Bulkhead	Location from AP
1	2.8 m
2	4.8 m
3	7.8 m
4	9.8 m

Table 4: Bulkhead Location

4.4.3 Damaged Stability Cases used

To assess the damage stability of the vessel, a floodable length analysis was carried out using *MAXSURF Stability* until the appropriate number of bulkheads was found. As for the floodable length, the first compartment from the aft ward should not be so long. The floodable length at about midship is very high, which means it could still meet the requirement even if we design a single compartment for mid-section. However, to meet the safety requirements of the machines and engines, we should locate them in individual rooms respectively. So, we decided to divide the hull into at least 5 compartments. The final bulkhead locations are outlined in Table 4.

4.4.4 Damaged Stability Results

For the size and purpose of the vessel, it must be able to withstand one compartment flooding where

- The deck edge is not immersed,
- The maximum angle of heel after flooding does not exceed 10 degrees,
- The minimum transverse GM after flooding does not exceed 1.5 m, and
- The minimum longitudinal GM after flooding does not exceed 0.2 m.

These limits were given to *MAXSURF* before running the analysis. The permeability for different usage is shown in Table 5. The permeability for each of the five compartments was set in *MAXSURF*, the values are outlined in Table 6.

Spaces	Permeability (%)
Cargo and stores	60
Accommodation	95
Machinery	85
Tanks	95

Table 5: Permeability of Spaces

Compartment	Permeability (%)
1	85
2	85
3	85
4	85
5	100

Table 6: Compartment Permeability

Testing the fully loaded displacement of the hull, the floodable length plot for one compartment flooding was generated using *MAXSURF* and can be seen in Figure 5.

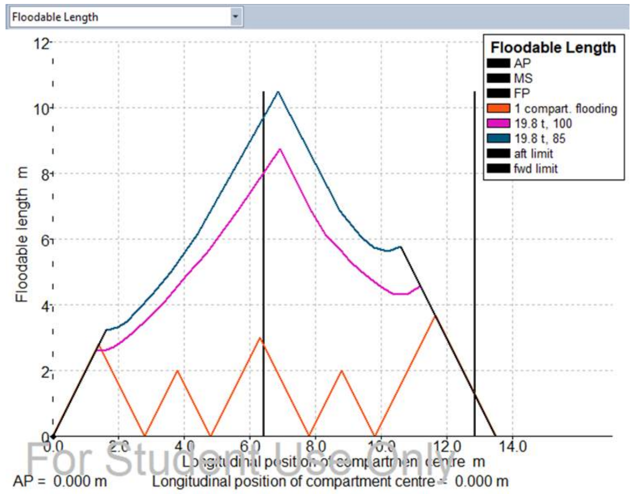

Figure 5: Floodable Length plot for Fully Loaded Condition

4.5 Hydrodynamics

4.5.1 Hydrofoil Selection

It was essential to select/design an appropriate hydrofoil to ensure the balance between the lift and drag force. As discussed before, it was decided to select the Avion system: one main forward hydrofoil and two aft hydrofoils. The preliminary goal of the hydrofoil selection was to have the lift force equivalent to 40 percent of the vessel weight at the cursing speed so that both the hydrofoil system and stern derive system could be submerged.

In order to achieve higher lift-to-drag ratio, NACA asymmetric 4, 5, and 6 series were considered across a wide range of Reynolds numbers between 500,000 and 10,000,000 and the chord length between 0.6 and 1.0 m with the angles of attack between –15 and 15 degrees. *XFOIL* was utilized to obtain corresponding lift and drag coefficients. The parameters were as follows:

- NACA Series: 1412, 2410, 2412, 2414, 2415, 4415, 23012, 632615
- Reynolds Number: 500,000 to 10,000,000 with 50,000 increments
- Angle of Attack: -15°to 15°with 0.25 increments
- Ncrit: 5 and 9

As a result, more than 350,000 data points were collected to calculate the lift and drag of the hydrofoils of 8 different NACA series, 190 different Reynolds numbers, 120 different angles, and 2 Ncrit numbers. In order to achieve the design goal having the lift force of 40 percent of the vessel weight, various combinations of the hydrofoil designs were compared while considering their manufacturing, mounting, and maintaining processes.

Before proceeding further, the lift coefficients of NACA 1412 with the Reynolds number of 3,000,000 from *XFOIL* were compared to the experimental results from *Theory of Wing Sections* by *Albert Edward Von Doenhoff and Ira Abbot*. The two data were found to be well aligned.

Unfortunately, it was infeasible to obtain exact experimental data to estimate uncertainties because the data tables were not included from the source. Therefore, it was assumed that the data at higher Reynolds numbers from *XFOIL* would be accurate and reliable to estimate the lift and drag forces.

4.5.2 Selected Hydrofoil Characteristics

The following hydrofoils without flaps were selected for one main forward and two aft hydrofoils in Table 7.

	FWD	AFT
NACA Series	1412	
Chord, c	0.90 m	0.54 m
Span	1.78 m	0.5 m × 2
Angle of Attack	0.5°	
Immersion Depth, h	0.52 m	
Kinematic Viscosity	1.189E-6 m^2/s	

Table 7: Selected Hydrofoil Parameters

Based on their chord lengths and seawater kinematic viscosity at 15°C, Reynolds numbers would vary up to 8,200,000 at 30 knots.

4.5.3 Lift Estimation

When a hydrofoil is not deeply submerged, the generated lift would be reduced due to wave-making, flow curvature, and a reduction in onset flow speed according to *Marine Rudders Hydrofoils and Control Surfaces* by *Anthony F. Molland*. Therefore, the lift coefficient at a given angle of attack needs to be recalculated to account these effects as shown below:

$$C_L = (\alpha - \alpha_0 - \delta\alpha)\left(\frac{dC_L}{d\alpha}\right)_{2D}$$

Where α is a geometric angle of attack; α_0 is an angle of attack for zero lift; $\delta\alpha$ is a change in angle of attack due to the free surface; and $dC_L/d\alpha$ is the 2D lift curve slope.

Angle of attack for zero lift would be zero if the foil was symmetrical. Since NACA 1412 is asymmetrical, the angle of attacks were obtained based on the lift coefficient plots from *XFOIL*.

Change in angle of attack due to the free surface can be obtained as below:

$$\delta\alpha = \delta\alpha_\infty \cdot \frac{w}{w_\infty}$$

2D lift curve slope was estimated as shown below:

$$\left(\frac{dC_L}{d\alpha}\right)_{2D} = 0.100/\text{degree}$$
$$= 0.2$$

The summarized results between 28 and 30 knots were tabulated in Table 8 and 9 for the forward and aft hydrofoils, respectively.

knots	α [°]	α_0 [°]	$\delta\alpha$ [°]	$dC_L/d\alpha$	C_L	Lift [N]
28	0.5	-1.071	3.098E-2	0.2	3.080E-1	5.243E+4
29	0.5	-1.071	3.098E-2	0.2	3.080E-1	5.624E+4
30	0.5	-1.071	3.098E-2	0.2	3.080E-1	6.018E+4

Table 8: FWD Hydrofoil lift

knots	α [°]	α_0 [°]	$\delta\alpha$ [°]	$dC_L/d\alpha$	C_L	Lift [N]
28	0.5	-1.071	3.183E-2	0.2	3.078E-1	1.769E+4
29	0.5	-1.071	3.183E-2	0.2	3.078E-1	1.989E+4
30	0.5	-1.071	3.183E-2	0.2	3.078E-1	2.031E+4

Table 9: AFT Hydrofoil lift

4.5.4 Resistance Estimation

The resistances from the hydrofoil and bare hull were considered to estimate the total resistance. Similar to the previous section, the resistance/drag generated by the hydrofoil could be calculated as follows:

$$\text{Drag} = \frac{1}{2} C_D \rho S V^2$$
$$C_D = C_{D_0} + C_{D_i} + C_{D_w}$$

Where C_{D_0} is the section profile drag coefficient from *XFOIL*; C_{D_i} is the induced drag coefficient; and C_{D_w} is the wave drag coefficient. It was assumed that the forward hydrofoil would not experience trailing tip vortices, so its C_{D_i} would be negligible.

Section profile drag coefficients could be obtained with the change in angle of attack due to the free surface and lift coefficient as below:

$$C_{D_i} = \delta\alpha \cdot C_L$$

It should be noted that the following assumptions were made:

- The forward hydrofoil would not experience induced drag due to trailing vortexes at each end as it is attached to the hulls.
- The aft hydrofoils would experience induced drag as a single foil as one's end is attached to the hulls.

Wave drag coefficients could be calculated as below:

$$C_{D_w} = \frac{1}{2}(k_0)cC_L^2 e^{-2(k_0)h}$$

The drag summaries were tabulated in Table 10 and 11 for the forward and aft hydrofoils respectively.

knots	C_{D_0}	C_{D_i}	C_{D_w}	C_D	Drag [N]
28	5.090E-3	0	1.920E-3	7.010E-3	1.193E+3
29	5.090E-3	0	1.796E-3	6.886E-3	1.257E+3
30	5.090E-3	0	1.683E-3	6.773E-3	1.324E+3

Table 10: Forward Hydrofoil Drag

knots	C_{D_0}	C_{D_i}	C_{D_w}	C_D	Drag [N]
28	5.110E-3	9.797E-3	1.151E-3	1.605E-3	9.244E+2
29	5.110E-3	9.797E-3	1.076E-3	1.598E-3	9.855E+2
30	5.110E-3	9.797E-3	1.009E-3	1.592E-3	1.050E+3

Table 11: Aft Hydrofoil Drag

The resistances from the bare hull were estimated with *VWS Hard Chine Catamaran Hull Series*, and they could be calculated as shown below:

$$\text{Resistance for Demi-hull} = \frac{1}{2}C_T \rho S V^2$$
$$C_T = (1+k)C_F + C_r + C_a + C_t$$

The resistances here were for the demi-hull, so it should be multiplied by two to have the total resistance from the bare catamaran hull. The summaries were tabulated in Table 12.

knots	$(1+k)C_F$	C_r	C_a	C_t	Resistance [N]
28	1.958E-3	1.969E-3	0.0003	4.227E-3	12.652E+3
29	1.948E-3	1.851E-3	0.0003	4.099E-3	13.103E+3

knots	$(1+k)C_F$	C_r	C_a	C_t	Resistance [N]
30	1.939E-3	1.725E-3	0.0003	3.965E-3	13.501E+3

Table 12: Demi-Hull Resistance

4.5.5 Estimated Lift and Resistance

The total lift and resistances from both the hydrofoils and bare hull were estimated and tabulated in Table 13.

knots	Lift [N]	Foil Resistance [N]	Vessel Resistance [N]	Total Resistance [N]
28	7.012E+4	2.116E+3	25.305E+3	27.42E+3
29	7.522E+4	2.243E+3	26.206E+3	28.45E+3
30	8.049E+4	2.374E+3	27.002E+3	29.38E+3

Table 13: Total Lift and Resistance

4.5.6 Ship Motions and Seakeeping

With the given sea conditions in Table 1, *MAXSURF Motion* was utilized to calculate its motions, and the obtained data was compared to *NordForsk Criteria* to determine its seakeeping performance. The criteria for fast small crafts are as follows:

- RMS of Vertical Acceleration at FPP: less than 0.65 g (6.38 m/s^2)
- RMS of Vertical Acceleration at Bridge: less than 0.275g (2.70 m/s^2)
- RMS of Lateral Acceleration at Bridge: less than 0.1g (0.98 m/s^2)
- RMS of Roll: less than 4°
- Probability on Slamming: less than 0.03
- Probability on Deck Wetness: less than 0.05

As a result, we were able to determine that the vessel would meet all criteria at the favorable conditions, but it did not meet all at the marginal and unfavorable conditions as shown below:

Conditions	Favorable	Marginal	Unfavorable
Vertical Acceleration at Bridge	Pass (less than 2.7 m/s^2)	Up to 3m/s^2 at 180 °	Up to 3.8 m/s^2 at 150°– 180°
Lateral Acceleration at Bridge	Pass (less than 0.98m/s^2)	Up to 1.5 m/s^2 at 30 – 60 °	Up to 1.8 m/s^2 at 30°– 60°

Table 14: Seakeeping Performance

As shown in Table 14, the vertical and lateral acceleration at bridge would exceed the limits by around 1m/s^2 at certain speed and heading conditions. Since these values were relatively small, it would be easily manageable with operational adjustments and/or a shock-absorbing seat.

5 Ship Arrangement

5.1 Summary of Arrangement Driving Considerations / Arrangement Rationale

The general arrangement of the vessel was shaped by several key factors, including cargo storage requirements (minimum 24 m^3), ease of cargo handling, and compliance with relevant regulatory frameworks, specifically *Transport Canada*'s *TP1332E, TP14070E*, and DNV's *Rules for the Classification of High-Speed, Light Craft, and Naval Surface Craft (RU-HSLC)*. As an example, some considerations influencing the layout are summarized below:

- Hydrogen Storage: In accordance with *RU-HSLC Part 6, Chapter 23* of the DNV rules, The hydrogen tanks were installed on the vessel's roof to fully comply with these safety requirements.
- Crew Visibility and Tank Placement: To meet Transport Canada's stipulations for vessels operated by a single crew member, 360-degree visibility from the helm must be maintained. To satisfy this requirement, the hydrogen tanks were divided into two separate sets and strategically positioned to avoid obstructing the crew's line of sight.
- Watertight Integrity and Bulkhead Placement: The number and arrangement of watertight bulkheads were determined based on Transport Canada's stability requirements for *Small Passenger Vessels*. Additionally, the positioning of collision bulkheads followed the guidelines specified in *RU-HSLC Part 3, Chapter 1* of the DNV rules, ensuring compliance with damage stability and survivability standards.
- Fuel Cell and Battery Systems: The location and installation of both the fuel cell system and the battery bank were designed in accordance with the relevant provisions in the DNV *RU-HSLC* rules, ensuring safe, reliable operation and protection from hazards.
- Package Dimensions: The package maximum dimensions of 2m x 2m x 2m has been taken into consideration while designing the doors to the cargo space, the arrangement of shelving units, and cargo loading/unloading ramps.

5.2 Manning Estimate

The vessel has been designed in accordance with Transport Canada regulations for single-crew operation, specifically following the requirements outlined in *TP14070E, TP1332E*, and *SOR/2010-91*. As a result, the vessel can be safely and effectively operated by any able-bodied individual without the need for additional crew.

5.3 Arrangement Description

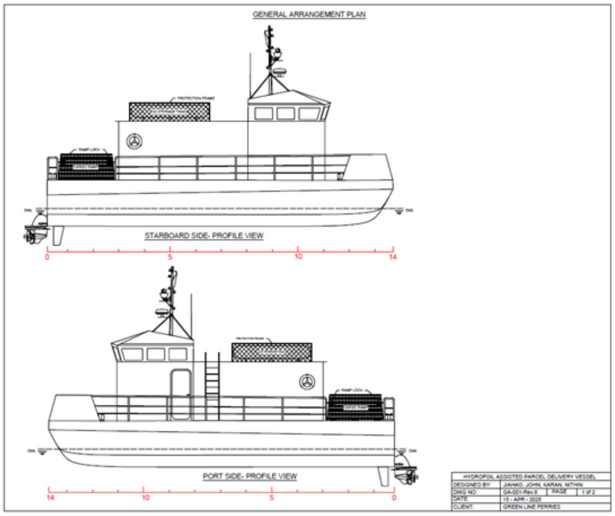

Figure 6: Profile view

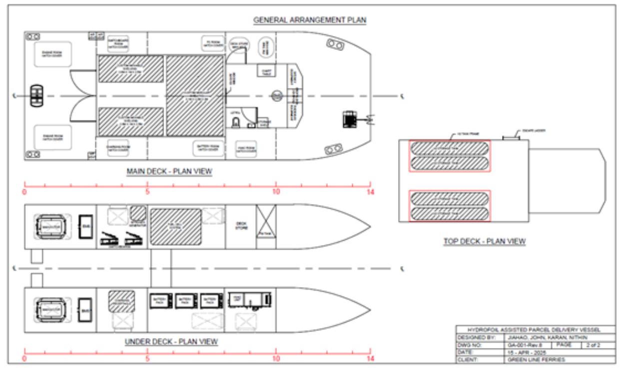

Figure 7: General Arrangement

6 Ship Structure

6.1 Description of Loading Cases or Design Rules / Structural Design Rationale

The ship structural design was based on the DNV-GL Rules for classification: *High speed, light craft and Naval Surface Craft Part 3 Chapter 1 and 3*. For each detail, the rules provide equations for general requirement to help us get the result.

The load and pressure applied to the ship are shown in the Table 15.

Pressures and Forces	Pressure
Slamming pressure	82.2 kN/m^2
Sea Load	Max 13 kN/m^2
Watertight Bulkhead	18 kN/m^2
Dry Cargo & Equipment	20.7 kN/m^2
Deckhouse Front Side	10.7 kN/m^2
Deckhouse Else Side & Deck	5.4 kN/m^2

Table 15: Pressure and Forces Conclusion

For a specific position, the estimated pressure is combination of these kind of pressure:

- Slamming pressure is the pressure exerted on a ship's hull, especially on the bottom or bow areas.
- Sea load usually refers to all the loads that act on a vessel or offshore structure due to the sea environment.
- Pressure for watertight bulkhead means to ensure a general performance of watertight, the lowest pressure on the bulkhead should be.

- Additional pressure may be applied due to the possible wave attacking on the deckhouse and cargo and equipment weight.

With the pressure result above, after discussing, we decided to choose NV5083 aluminum. The minimum thickness of the plate could be determined by three methods, experiencing equation, thickness due to lateral pressure, bottom thickness due to slamming pressure. The minimum thickness for a specific position should be the larger one of these three. The minimum thickness and design our scantling accordingly was calculated and tabulated in Table 16.

	t_{min}/mm	$t_{selected}$/mm
Bottom and side to waterline	4.12	5
Side above waterline	3.53	5
Strength deck forward of amidships	3.2	4
Strength deck aft of amidships	2.6	4
Deck for cargo	4.12	5
Superstructure and deckhouse decks	2.03	4
Watertight bulkheads	3.06	4
Superstructure front	2.93	4
Superstructure sides and aft	2.47	4

Table 16: Minimum and Selected Thickness of Plates

The value of minimum thickness of plates varies from each other a lot, even though we increase them to the nearest larger integer. To make it more practical, we only choose 4 and 5 mm thick plates.

7 Powering System

The vessel's powering system is a hybrid configuration combining a hydrogen fuel cell with battery support. The batteries serve multiple critical functions: they assist the fuel cell during ramp-up and ramp-down phases to handle fluctuating loads, absorb peak loads to extend the operational life of the fuel cell, provide power to ship systems and provide an emergency source of power if required. Both the fuel cell and battery systems are connected in parallel to the vessel's electrical distribution network, which supplies power to the propulsion motors. In this arrangement, the fuel cell exclusively delivers electrical power, while the batteries are capable of both supplying energy to the motors and receiving excess energy from the fuel cell as needed.

7.1 Operation of fuel cell System

Hydrogen for the vessel will be stored at a pressure of 450 bar and a temperature of 25°C. In the Vancouver region, transportation of gaseous hydrogen by truck at 450 bar is currently permitted, aligning with our storage strategy. The required storage capacity was determined based on the vessel's resistance and energy consumption estimates. To achieve a range of 70 nautical miles, it is necessary to carry approximately 40 kilograms of hydrogen on board. This hydrogen is stored in four 292-liter high-pressure tanks.

Before being supplied to the fuel cells, the hydrogen pressure must be reduced from 450 bar down to an operational pressure range of 3.5 to 5 bar. This pressure reduction is critical to ensure safe and efficient operation of the fuel cells.

Hydrogen fuel cells generate electricity through an electrochemical reaction. Each cell comprises two electrodes—a negative anode and a positive cathode—with an electrolyte that transports charged particles between them. A catalyst facilitates the reaction, enhancing efficiency. In this system, hydrogen serves as the primary fuel, while oxygen from the ambient air is also required for the reaction. One of the most significant advantages of hydrogen fuel cells is their minimal environmental impact: the only by-product of the energy generation process is water. When fueled with pure hydrogen, these cells are completely carbon-free.

8 Ship Propulsion

8.1 Machinery Plant Description

8.1.1 Main Engine

The main motor for the vessel will be two inboard motors by *Evoy*. The model selected for our purpose is *Evoy* Hurricane with a nominal power of 300 kW and a peak power of 600 kW. These motors are a robust system designed for 1000+ hours of operation per year.

8.1.2 Stern Drives

The vessel's propulsion system consists of two stern drives powered by a hydrogen fuel cell system. Due to a lack of powerful electric stern drives available on the market, both for recreational and commercial purposes, the water bus is to be equipped with two retrofitted *Volvo Penta Aquamatic Duoprop D6-440 DPI* stern drives. By removing the diesel engine on each stern drive and replacing it with an electric motor, the *Volvo Penta* stern drive will provide the water bus with the required zero- emission propulsive power. The *Volvo Penta* stern drives come as a complete propulsive package, consisting of all the required mechanical components between the engine and propellers. The *Volvo H-series* propellers that are designed for the *Aquamatic Duoprop DPI* series are contra-rotating propellers. Contra-rotating propellers improve system efficiency due to the ability for flow energy recovery from the first propeller to the second.

8.1.3 Fuel cell and Batteries

The fuel cell selected for the vessel is 340 kW *Corvus Pelican* system and a 145 kWh *Corvus ORCA* battery pack is used as a hybrid arrangement. The *Pelican* and *ORCA* systems have inherently safe design and is DNV certified.

8.1.4 Thermal Image Detection Camera

As the vessel is expected to navigate routes with significant logging activity, it was critical to equip it with a reliable debris detection system to ensure the safety of both the vessel and its cargo. The client specifically requested a system capable of operating effectively in all weather conditions, day or night. However, a major constraint was that the solution had to rely on existing, proven technologies, posing a challenge in identifying an appropriate system.

After thorough evaluation, the design team selected the *FLIR M400XR*[3] thermal imaging camera to address this requirement. The *FLIR M400XR* utilizes advanced thermal imaging technology to detect debris by capturing the temperature differences between foreign objects and the surrounding water surface. This allows the system to reliably identify hazards ahead of the vessel even under poor visibility conditions, such as darkness, fog, or heavy rain.

8.2 Machinery Arrangement

As the vessel is propelled by a stern drive system, the main electric motors are strategically positioned in the aft compartment to allow for direct and efficient coupling to the stern drives. This aft space also houses the electric and battery management systems, ensuring compactness and ease of maintenance.

Moving forward, the switchboard room and the charging receptacle room are located adjacent to each other. Positioned between the main motor compartment and the fuel cell/battery compartments, this layout minimizes cable lengths, improving system efficiency and reducing installation complexity.

The fuel cell and battery systems—being the heaviest components of the machinery—are installed near the midship area to maintain proper longitudinal weight distribution between the forward and aft sections of the hull. For additional safety, these systems are housed in dedicated, isolated compartments.

Further forward, the vessel accommodates a packaged air handling unit (AHU), responsible for maintaining temperature-controlled conditions within the cargo space and the bridge area, ensuring optimal operational and environmental conditions.

9 Weight Engineering

9.1 Margins and Allowances

To enhance accuracy and account for real-world variability, we incorporated a 5% contingency buffer in both the weight and volume estimations of fuel. For all other onboard systems and components, the margin was applied solely to volume calculations, under the assumption that they operate at full load or capacity. Additionally, a 5% design margin was included in the lightweight (LWT) estimation to accommodate unforeseen factors such as material tolerances,

[3] https://www.raymarine.com/en-us/our-products/thermal-cameras/flir-thermal-cameras/flir-m400xr

outfitting adjustments, or minor additions during construction. This proactive approach ensures robust and flexible planning, even under the most dynamic operational scenarios.

9.2 Lightweight

To estimate the ship's lightweight, we broke it down into smaller groups: technical equipment, machinery systems, accommodation areas, and the hull structure. A major advantage of our approach is that most of the weight data was sourced directly from equipment manufacturers, ensuring accuracy.

Over 50% of the lightweight is attributed to machinery, with some key contributors being:

- Fuel cell system: 3.7 tonnes
- Batteries (145 kWh backup capacity): 2.04 tonnes

This distribution highlights the efficiency of our hybrid power approach. The batteries are relatively small and only serve as a backup, while the fuel cells provide the primary power for the vessel's entire operational range. This reinforces our belief that opting for a hybrid solution was smarter and more efficient than going fully electric with large batteries.

9.3 Weights Representation of Vessel

Figure 8, 9, and 10 are the images which will show our lightweight, deadweight and the weight taken up by machinery space, respectively, as discussed above in Margins and allowances.

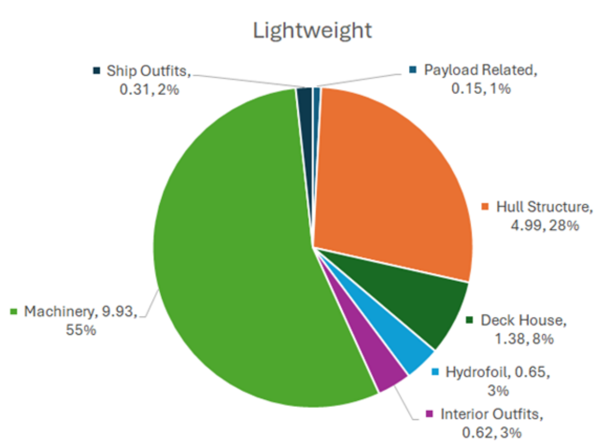

Figure 8: Lightweight Diagram

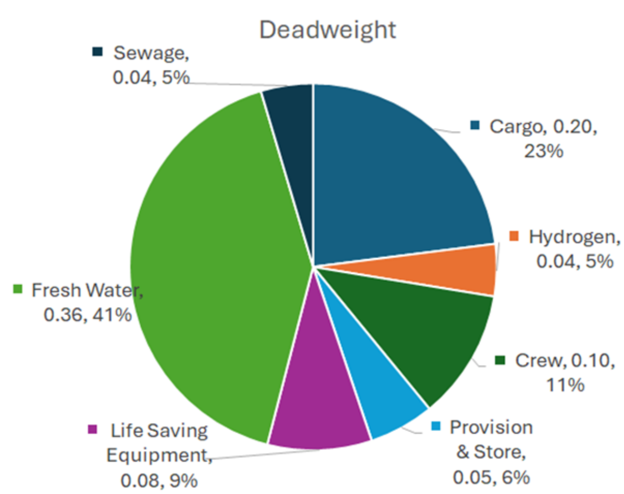

Figure 9: Deadweight Diagram

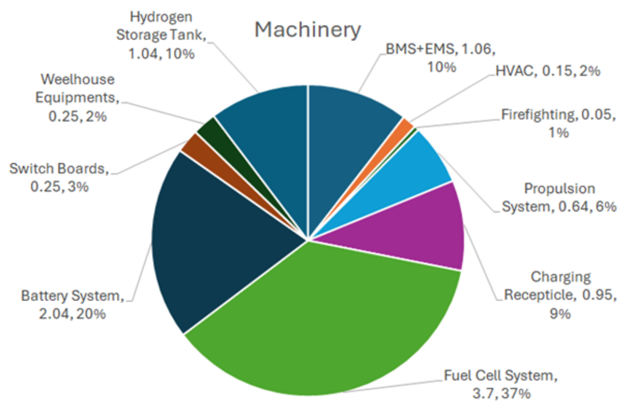

Figure 10: Machinery Weight Diagram

The lightweight and deadweight were found to be 18.93 tonnes and 0.87 tonnes respectively.

10 Cost Estimation

10.1 Methodology and Assumptions

The cost estimation follows a bottom-up approach, breaking down the vessel into its primary components and systems. The cost of each component refers to vendor quotes, historical data from comparable vessels, and engineering judgment. Most of the value of coefficients are from System Based Ship Design by *Kai Levander*. Aluminum constructions usually require more labor hours than steel constructions. So approximately 40% more labor hours rather than steel is applied for hull and deckhouse construction.

10.2 Summary Table

Acquisition cost consists of material cost, labor cost, design cost, building time financial, profit, financing and broker fees. Estimated total acquisition cost is 1.745 million CAD.

PRICE ESTIMATION				Price	Price
	h/LWT	Hours	CAD / h	KCAD	CAD / kg
Design	30.00	568	131	74.57	3.94
Labour + Over Head	102.10	1934	150	290.0	15.32
Material				1213	64.03
Building time financing (Interest × Time/2)	6%	months	4.5	17.74	3.69
Total Production Cost				1,595	86.99
Profit	5			79.74	
Financing, Payment	3			47.84	
Broker fees	1			15.95	
	Final Price			1,738	86.99
BUILDING PRICE	Price / DWT			2,009	kCAD / ton

PRICE ESTIMATION	h/LWT	Hours	CAD / h	Price KCAD	Price CAD / kg

Table 17: Cost Estimation

11 Recommendations for Further Iterations

To enhance the vessel's performance and efficiency, several areas are recommended for future investigation. First, exploring alternative hull shapes—such as slender catamarans or SWATH designs—may yield improved hydrodynamic characteristics and stability. Enhancing seakeeping through hull refinements or the addition of passive stabilization features should be prioritized to reduce motion in rough water. Investigating the damping effect of hydrofoils, particularly under dynamic sea conditions, will also help refine foil performance. Computational Fluid Dynamics (CFD) simulations should be conducted to compare resistance across design variations, enabling data-driven decisions. Finally, the hydrofoil's location, shape, and dimensions should be optimized iteratively to balance lift, drag, and structural integration within the vessel envelope.

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