Design of a Trailer Capable, Open Ocean Sailing Yacht

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Abstract

A design is developed for a small sailing yacht capable of being towed, launched, and recovered with a standard-sized truck or sport utility vehicle, while retaining capability for extended, open ocean transits. A review of factors affecting small yacht seaworthiness is presented, and relevant design parameters are proposed. Design requirements pertaining to trailer capability, seaworthiness, and vessel intended use are developed, and a multi-criteria decision-making method is employed to down-select to preferred options in key functional areas of the design. From there, an iterative point-based design approach is employed to converge on a design that satisfied requirements. Major design work encompassed developing a suitable hull form; keel and rudder design; selection and validation of appropriate scantlings; designing a composite mast and spars; determining a sail plan and rigging schema; engine selection, propeller design, and off-design propulsion analysis; arrangements layout; detailed weights and stability assessments; and sailing performance predictions. The design meets or exceeds all developed requirements, including exceeding International Standards Organization (ISO) stability and buoyancy requirements on Stability Index (S.I.) and Righting Energy for the highest design category classification, which pertains to vessels expected to experience significant wave heights up to 7 m and up to Force 10 winds. A 1:7 scale model of the hull was constructed with a fused deposition modeling 3D printer and used to measure upright resistance of the yacht in towing tank experiments, for comparison to resistance predictions generated from the Delft Systematic Yacht Hull Series.

Thesis Supervisor: Paul Sclavounos, Professor of Mechanical Engineering and Naval Architecture

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Acronyms

ABS American Bureau of Shipping.

AR Aspect Ratio.

CAD Computer Aided Design.

CSM Chopped-Strand Mat.

DWL Design Waterline.

FBD Beam Displacement Factor.

FDF Downflooding Factor.

FDL Displacement Length Factor.

FDM Fused Deposition Modeling.

FDS Dynamic Stability Factor.

FIR Inversion Recovery Factor.

FKR Knockdown Recovery Factor.

FRP Fiber Reinforced Plastic.

FWM Wind Moment Factor.

HP Horsepower.

ISO International Standards Organization.

ITTC International Towing Tank Conference.

LBS Base Length Factor.

LCB lateral center of buoyancy.

LCF Lateral Center of Flotation.

LCG Longitudinal Center of Gravity.

MT Metric ton.

NURBS Non-Uniform Rational B-spline Surface.

PLA Polylactic Acid plastic.

RPM Revolutions per Minute.

STIX Stability Index.

VCG Vertical Center of Gravity.

VPP Velocity Prediction Program.

WR Woven Roving.

List of Symbols

Δ	Displacement
Δ_{FL}	Full Load Displacement
Δ_{MO}	Minimum Operating Displacement
∇	Submerged Volume
ϕ_{DF}	Downflooding Angle
ϕ_V	Angle of Vanishing Stability
C_M	Midship Section Coefficient
C_P	Prismatic Coefficient
C_{WP}	Waterplane Coefficient
B_{OA}	Beam Overall
B_{WL}	Beam at Waterline
BM	Metacentric Radius
D	Depth
D_M	midships depth
DA	Dellenbaugh Angle
F_m	Freeboard Midships
Fn	Froude number
GM	Metacentric Height
GM_T	Transverse Metacentric Height
GZ	Righting Arm
H_D	Downflooding Height
HA	Heeling Arm
KB	Distance from Keel to Center of Buoyancy
KG	Distnace from Keel to Center of Gravity
L_H	Hull Length
L_{OA}	Length Overall
L_{WL}	Waterline Length
RE	Righting Energy
RM	Righting Moment
SA	Sail Area
Sn	Scantling Number
T	Draft

1 Introduction

1.1 Background

"By waves and wind I'm tossed and driven.
But still my little ship outbraves
The blust'ring wind and stormy waves"
[1]

In June of 1898, Captain Joshua Slocum sailed into Newport harbor aboard his 37 foot sloop *Spray*, concluding a voyage of some 46,000 miles and three years.[1] He was the first sailor to circumnavigate the world single-handed. Since this momentous voyage, many intrepid sailors have crossed oceans or completed circumnavigations in small sailing vessels, some considerably smaller than *Spray*.[2]

This work presents the design of a sailing vessel. The overarching goal was to develop a cruising yacht of sufficient outfit and seaworthiness for safe ocean crossings, yet was capable of transport, launch, and recovery from a trailer, with a standard pick-up truck or sport utility vehicle. This design attempts to balance these conflicting aims: for while "the size of a boat is, beyond any doubt, an important design factor in a survival situation"[3], a boat that must be trailered is necessarily constrained in length, beam, and displacement.

Seaworthiness is a basic necessity of any vessel intended for offshore sailing, for the first charge of a ship is to carry her crew safely to their destination. Since the present yacht must be capable of extended voyages far from the possibility of aid, this characteristic was of central importance in her design, and informed many of the decisions throughout. And while there exist numerous successful, seaworthy small sailing yachts[4], there are fewer examples of such vessels that are also suitable for trailering. Trailer capability in a sailboat confers many advantages. Ownership costs can be significantly lower, since they do not require recurring marina slip or mooring fees. Maintenance is also simpler, and less expensive: there are no crane services or haul out fees, and no fees for boatyard space. If properly stored when not in use, maintenance requirements are reduced relative to a yacht left in a slip or on a mooring. Finally, the ability to trailer a yacht vastly increases the cruising areas accessible to sailors whose employment or other obligations lamentably restrict their available sailing time.

This design is an attempt to capture the benefits of a trailer capable boat, while retaining the yacht's ability to safely carry her crew on offshore voyages. The end result, displayed in figure 1, is a mono-hull sailboat, with a Scheel type shoal keel, constructed of fiberglass and rigged with a junk main on a freestanding mast. Yacht particulars are displayed in table 1. Appendices A, B and C contain renderings, lines drawings, and offsets of the yacht.

Yacht Particulars	
L_{OA}	7.55 m
L_H	7.20 m
L_{WL}	6.78 m
B_{OA}	2.44 m
B_{WL}	2.09 m
Δ_{MO}	2935 kg
SA	33.7 m^2

Table 1: Yacht Key Parameters

Figure 1: Quartering View of Yacht

1.2 Design Requirements

Design requirements for the yacht are displayed in table 2, and explained in greater detail below.

Parameter	Threshold	Objective
Stability Index	23	32
Righting Energy	57000 kg-m-°	172000 kg-m-°
Minimum Crew	1	-
Passage Speed	4 knots	7 knots
Accomodations	2 adults	3 adults
Stores	21 days	30 days
Weight, yacht and trailer	3855 kg	2720 kg
Length, on trailer	10.67 m	9.14 m
Width, on trailer	2.59 m	2.44 m
Height, on trailer	4.11 m	3.96 m
Launch/Recover	2 crew, No crane	1 crew, No crane
Structural Strength	Suitable for Open Ocean	-

Table 2: Design Requirements

The main criteria used to quantify seaworthiness requirements in this design are the Righting Energy (RE) and Stability Index (STIX) computed from International Standards Organization (ISO) 12217-2. This index is a function of the yacht's main dimensions and righting moment curve, and is used to help 'score' a vessel's seaworthiness, by assessing her ability to withstand and recover from knockdown and inversion. [5], [6] The ISO establishes four design categories for sailing yachts of 6 to 24 meter lengths: Design category A indicates fitness for extended ocean crossings, while category B indicates fitness for offshore voyages. Category C vessels are fit for inshore waters, and category D vessels are suited only for sheltered waters. Vessels must possess a STIX of 32 or greater to qualify for category A, and a minimum STIX of 23 to qualify for category B. [6] These values form the objective (desired) and threshold (minimum acceptable) design requirements. The RE is an indication of work required to heel a yacht to her angle of vanishing stability, ϕ_V, at which angle the righting arm is zero and the yacht no longer has positive stability. iso 12217-2 establishes minimum righting energies for each design category: 57000 and 172000 kg-m-degrees for categories B and A, respectively.

The vessel must be capable of being managed by one person, as it designed to be capable of single-handed voyages. Even on voyages with two or three crew, however, this is a necessary attribute of a small ocean-going vessel: in part so that the crew may rotate watches, and in part for reasons of safety at sea: if one of two crewmembers should fall overboard, become injured or ill, the remaining fit crewmember should be capable of sailing the vessel unaided. This implies that all sail handling, steering, reefing, and anchor systems be within the physical capabilities of one person to operate.

As this is not a racing yacht, there is no need for 'round-the-buoys' speed to windward; other elements of the design, namely seaworthiness and trailer capability, take priority. However, a good cruising vessel should make passages in respectable times relative to her size. Obviously, the speed obtainable on a sailing yacht depends on both the strength of the wind and its direction relative to the intended course, as well as the vessel herself: her hull form, appendages, sail plan, and displacement. To set a requirement on speed, it is necessary to first define the conditions in which the vessel is expected operate. Established cruising routes for small sailing vessels exist for all the world's oceans, determined largely by seasonal weather patterns and prevailing winds and surface currents. [7]. Judicious route planning and timing enables the small boat skipper to take advantage of regular wind patterns, so that she may encounter, on average, favorable conditions. Favorable conditions are here considered to be winds abeam to astern, of speeds between roughly 10 to 20 knots. Such wind speeds are typical of the trade wind systems and the northern hemisphere prevailing westerlies [8], of which numerous cruising routes make use [7]. Under such conditions, a threshold speed of 4 knots, and an objective speed of 7 knots are set as design requirements.

In addition to standing up to the conditions of the sea, a cruising sailboat must provide accommodations and stores capacity for her crew. Given the size limits inherent to trailer-able boats, the requirement on accommodations was set to a threshold of two adults, with an objective of three. Stores capacity sets the limiting leg of a voyage by fixing the longest permissible travel time between ports. Combined with the yacht's average speed made good on a given leg, this constrains the distance between ports for voyage planning purposes. For North Atlantic crossings, the limiting leg of a crossing could be the 1800 NM passage from Bermuda to the Azores (eastern transit), or the 2020 nm passage from Cape Verde to Barbados (western transit). In the Pacific, Hawaii provides a jumping off point for multiple destinations, and is a 2020 nm passage from San Francisco. All the above routes have favorable prevailing conditions, if sailed in the correct season. [7] If the yacht can average 100 nm per day, equivalent to an average speed made good of just over 4 knots, the limiting passages described above can be conducted with 21 days of provisions. If 30 days of provisions are stored, additional flexibility in route selection results, as well as additional margin for shorter passages.

Requirements on trailering dimensions (length, width, and height on trailer) are based on legal limits on public roads in various states within the US. According to data compiled by the American

Boating Association, trailers 35 feet in length and less may be towed without permit in the majority of US states. Most states allow un-permitted towing of trailers less than 13.5 feet in height and 8.5 ft in width [9]. Towed weight requirements, the allowable weight of the yacht and trailer combined, were set by a review of advertised towing capacities of select standard sized pick up trucks and sport utility vehicles for sale in the US market. For the trucks, towing capacity ranged from 7,500 lbs to 13,200 lbs.[10]. For sport utility vehicles, the values ranged from 7,716 to 9,300 lbs [11]. These ranges were used to set the threshold and objective values for combined weight of boat and trailer, to 8,500 lbs and 7,500 lbs respectively. Weights and dimensions above are given in English units, consistent with the sources from which they are derived, but are converted to metric units below.

In order to capture the full benefits of a trailer capable yacht, one or two crew members should be capable of rigging and launching the vessel without the use of crane services or boatyard facilities, excepting of course a decent launch ramp of sufficient length and depth. This requires developing a mechanism for stepping the mast, or engineering it to be sufficiently light to allow manually stepping it. This also places an implicit constraint on draft, discussed further below.

Finally, the hull, deck and rig must be of suitable structural strength to withstand the loads imposed on them during open ocean sailing. This requirement is intimately connected to seaworthiness, which requires a vessel to "be able to defend herself against the incursion and perils of the sea."[3]

1.3 Assumptions and Derived Requirements

Several assumptions were necessary to further define requirements for a "trailerable" yacht. The first assumption pertains to the limit on draft for trailer launch and recovery. It is assumed that a vessel with a draft of 1.22 m or less can be launched from a suitably designed ramp or gently sloping beach, with the assistance of a trailer extension bar or towing straps. This is based in part on the ramp design recommendations from several US states [12][13] and a private firm [14], indicating a ramp toe depth of 1.22 meters below the low water height, and a ramp grade of 12-15 percent. A reasonable tidal swing should ensure the boat can be floated without backing the trailer wheels off the ramp. In regions without tidal influence, it may be necessary to back the trailer off the ramp, in which case the sill of the ramp should rest on gravel, firm sand, or similar bottom type. Alternatively, a firm sand or gravel beach of appropriate grade could be employed. This assumption was further validated by the documented experiences of other sailors.[15][16][17][18][19]

The second assumption relates to the weight and dimensions of the trailer, which is complicated by the fact that sailboat trailers are often custom designed for a given class. In order to convert the combined weight requirement listed above into a weight requirement for the vessel alone, an estimate of the trailer weight is needed. Based on vendor data, it is assumed that for the weight capacity range relevant to this design, the yacht's trailer can be manufactured with a weight equal to 20 percent of its rated weight capacity. [20][21] The trailer is assumed to require a minimum of 1.55 m of length beyond the yacht overall length, and a clearance of 0.5 m from the yacht keel to the ground.

Some additional assumptions regarding the weights of stores, crew, and their effects are needed to derive a maximum load weight capacity for the yacht. For stores, it is assumed that 0.5 gallons of freshwater per crewmember, per day is sufficient. [2] Food needs are assumed to be 2 kg per person, per day. This is an average of recommendations in the literature [22], [2], and a calculation performed using US Department of Agriculture (USDA) nutrition data.[23] Crew weight is assumed 75 kg per person, with 20 kg per person allotted for personal effects. [6]. A fuel tankage weight of 25.4 kg is assumed, corresponding to an 8 gallon tank of diesel. The load weight capacity is the maximum weight that can be added to the craft, starting from the minimum operating condition. The minimum operating condition is defined as the weight of the yacht, its rigging and sails, all standard equipment and gear normally carried aboard, and the weight of the minimum crew.

[6]. The load capacity is then calculated as the sum of additional crew/passengers, crew personal effects, edible stores, and fuel and water tankage. Table 3 displays the necessary load capacities for the ranges of stores and crew sizes established as design requirements.

Stores	**Crew Size**	
	2 Crew	3 Crew
21 Days	318	482
30 Days	375	586

Table 3: Load Capacities (kg)

Finally, it is assumed that adequate structural strength can be assured through proper application of a suitable scantling rule. Scantling rules are empirical or semi-empirical in nature, reliant on engineering analysis and validated by comparison to successful vessel designs. Such rules have been successfully employed by designers and builders stretching back over a hundred years, and have been incorporated into class societies, such as Lloyds and the American Bureau of Shipping (ABS). However, it is essential that the rule selected is applicable for the size, type, and material of the vessel under consideration. [24]

The above assumptions result in additional derived requirements, displayed in table 4. The "trailer condition" refers to the weight and draft of the yacht in the condition in which she will be towed. This weight assumes that no crew is on the yacht, the fuel and water tanks are empty, and non-installed tools and spare parts have been removed. There is no objective value on the draft, as draft has a significant interplay with speed, stability, and seaworthiness. However, a smaller draft makes launch and recovery simpler. The draft should be as small as practical, consistent with achieving the desired characteristics in these other areas of the design.

Parameter	**Threshold**	**Objective**
Weight, yacht in trailer condition	3210 kg	2265 kg
Draft, yacht in trailer condition	1.22 m	-
Maximum Load Capacity	318 kg	586 kg

Table 4: Derived Requirements

1.4 Design Philosophy

As all vessel designs represent a balance of compromises, a design philosophy was established to guide the author's decisions. This philosophy was informed by the intended purpose of the yacht: to be capable of crossing oceans under sail, and to be capable of transport, launch, and recovery with a trailer.

The most important attribute of the design is seaworthiness, which encompasses all those elements that make the yacht fit and safe for ocean sailing. However, the design is also heavily influenced by the requirements on trailer capability. Beyond these two facets, the design should emphasize reliability, lend itself well to singlehanded sailing, and prioritize decent passagemaking times over race performance. This last point bears emphasizing: no element of the design is influenced by the rating rules that govern yacht racing.

It is acknowledged here that, for many sailors, a vessel with a draft close to a meter may not fit their conception of "trailerable." Such a vessel requires some forethought in selecting an adequate launch site, and some additional work to launch or recover: extension bars or towing straps are necessary to prevent submerging the rear of the towing vehicle. However, the yacht is not conceived as a "day sailor," intended to be launched with little effort, sailed for a few hours, then recovered and driven home before dinner. Rather, she is conceived as a portable cruising vessel, which confers the advantages enumerated above without unduly limiting her potential for open ocean voyaging. The intent is that it be trailered to a desired cruising ground, launched, and sailed for a period of several days to weeks or longer. Alternatively, some sailors might prefer to leave the yacht in a slip seasonally, only trailering it home when their cruising season ends, and launching at the start of the next season. In such circumstances, some sailors may prefer to use a boatyard crane to launch and recover, using the trailer only to transport the vessel. However, the design requires that launch and recovery from the trailer be possible, preferably with minimal additional effort, to prevent constraining the launch and recovery locations to solely those boatyards with adequate crane services. The ”trailer capable” nature of the design is viewed primarily as a set of constraints that must be satisfied. Where possible, consideration is given to simplifying the launch and recovery, though not at the expense of seaworthiness.

The design philosophy may be summarized:

1. *Seaworthiness first.*
2. *Trailer capability is a constraint.*
3. *Design for reliability: minimize failure points and the probability of catastrophic failures.*
4. *Design for the single-handed skipper.*
5. *Passagemaking matters, ”Round the bouy” times do not.*

2 Review of Literature

A review of design data on vessels of similar size and purpose was conducted as a starting point for the point based design approach adopted for this yacht. Additionally, a variety of references were consulted to gather information on general yacht design processes, seaworthiness and stability, structural design, mast and rig design, and ship model testing. These works were used extensively to assist the author in developing and assessing the design.

2.1 Review of Similar Vessels

In order to establish feasible initial values for key design parameters of the yacht, data was compiled on twenty small sailboats that were designed or had been successfully employed for open ocean voyaging. Many of the vessels selected for the set are described in [4], which reviews small, ocean-capable yachts. The vessels selected for comparison possess a variety of sail plans, including masthead and fractional sloops, cutters, ketches and junk rigs. They also vary in type of keel, including traditional full keels, modified full keels, high aspect ratio fins, wing keels and bilge keels. The vessel dimensions and non-dimensional parameters from this set were used to gain insight into feasible values to target at the outset of the design process. Key parameters are summarized in table 5, and the complete data set is presented in appendix D.

Parameter	**Minimum**	**Maximum**	**Mean**	**Median**
L_{OA} (m)	6.10	9.75	8.02	7.92
L_{WL} (m)	5.49	7.77	6.57	6.51
B_{OA} (m)	2.13	3.18	2.55	2.47
T (m)	0.91	1.68	1.19	1.17
Δ (kg)	884	6757	3093	3095
L_{OA}/L_{WL}	1.06	1.33	1.22	1.23
L_{WL}/B_{OA}	2.27	3.13	2.59	2.55
L_{WL}/T	4.18	7.25	5.62	5.52
$L_{WL}/\nabla^{1/3}$	4.00	5.76	4.66	4.56
$SA/\nabla^{2/3}$	12.11	19.99	15.28	15.13
Ballast Ratio	0.29	0.54	0.42	0.41
HP/MT	2.1	9.5	4.4	3.7

Table 5: Design Parameters of Similar Vessels

The vessel review guided reasonable initial selections for displacement, ballast ratio, principal dimensions, sail area, SA, and auxiliary engine Horsepower (HP).

2.2 Seaworthiness

2.2.1 Definitions and Attributes

One definition of seaworthiness is a vessel's ability to to "defend herself against the incursion and perils of the sea." [3] This definition, while succinct, says nothing about what attributes confer seaworthiness to a given vessel. Seaworthiness remains a challenging concept to strictly define or quantify, in part because it encompasses a multitude of interacting elements that impact a vessel's response to the ocean environment. It is most clearly defined in terms of these attributes, so that one might characterize a vessel as seaworthy if she possessed "strong durable and watertight construction, structurally sound rig, [and] good survival characteristics in extreme weather conditions." Because "anything which floats on the air/water interface...may be destroyed by sea forces," the question of assessing seaworthiness becomes one of determining whether a vessel has a relatively high or low probability of survival in extreme conditions. [3]. Even this more specific definition requires considerable expansion on what constitutes 'good survival characteristics.' Obviously, a vessel's stability is a necessary condition: her ability both to resist capsize, and to recover quickly, without serious damage, should she capsize. Also important is her ability to maneuver clear of dangers, such as a lee shore, even in extreme weather. [2] Finally, the seaworthy cruising vessel should exhibit seakindliness, typified by vessel motions that are soft enough to permit the crew to work and rest. [5] A vessel whose motions are "comparatively slow, small and easy...in spite of rough sea and weather," will cause less fatigue to her crew and be less likely to render them incapacitated - improving the vessel's probability of surviving severe conditions. [3]

To summarize, a seaworthy vessel possesses the following attributes:

1. Strength: Structural design suitable for the loads experienced in heavy weather, to include the possibility of capsize.
2. Stability: the ability to resist capsize, and recover quickly if a capsize does occur.
3. Coursekeeping: the ability to keep course or maneuver clear of danger in heavy weather.
4. Seakindliness: comparatively soft motions in a seaway.

2.2.2 Design Factors

Having determined required attributes for a seaworthy vessel, it is necessary to examine how design elements contribute to or detract from these attributes. Structural strength may be treated separately and is discussed in a subsequent section. The remaining attributes may be approached in a dynamic analysis of vessel response in heavy seas.[3] The goal of the analysis is to determine the forces and mechanisms that cause a boat to capsize, and the design characteristics a boat must possess to increase her probability of surviving extreme conditions. Three fundamental components of a vessel's response to wind and wave forces are identified in reference [3]: the amount and distribution of weight, the damping experienced by the vessel, and the vessel stability. The vessel response depends on "the relative magnitudes of these components, and the way they are blended into a unified whole" within a given design. [3] Of course, these components depend on a number of design parameters, including displacement, principal dimensions, freeboard, hull shape, and keel design. These and other factors influencing a seaworthy design are examined below.

Size The size of a vessel, both in terms of her length and displacement, has an indisputable impact on her seaworthiness: "other things being equal, the sea is less kind to smaller boats."[3] In part this is a matter of the greater inertia conferred by a larger displacement, which acts to reduce the accelerations imposed on the vessel by the sea. This results in more seakindly motions and also reduces the likelihood of capsize due to wave action.[3] It is also because the length of the vessel forms a scale with which to measure the size of the waves: a smaller yacht faces relatively larger waves in a given seaway. [5].

The Fastnet Race of 1979 serves as a grim case study in small yacht behavior under extreme conditions. The Fastnet Race consists of a 605 mile course across the Celtic sea. The 303 vessels that participated in the 1979 race encountered an unexpected gale, which resulted in extreme conditions during the race. Winds of up to Force 11 and wave heights of more than 40 feet were observed. [25] Fifteen lives were taken by the storm, 24 yachts were abandoned (5 of which were never recovered), and 136 crewmembers were rescued by outside agencies. [25],[3] The yachts raced in classes based on their 'rated length', with the smallest three classes (Class III-V) of rated length between 21 and 29 feet. The rated length is a measure of "effective sailing length," so that the smallest yacht that might qualify for entry would possess a L_{OA} of around 28 feet. [25] The results of the storm were "catastrophic to small boats": the number of race retirements, abandoned boats, and knockdowns all demonstrated a diminishing tendency with increasing boat size. [3] All crew lost were from the three smallest classes, as were 23 of the 24 abandoned yachts. [25].

The foregoing discussion may seem to indicate that small vessels are inherently unseaworthy. However, size alone, as measured by either displacement or length, is not the sole determinant of a vessel's ability to survive in extreme conditions. In particular, criticism has been levied against small, light displacement vessels of broad beam and shallow hull form, which have been encouraged by ratings rules. [3],[25] One vessel of the smallest class in the 1979 race did finish: a Contessa 32, whose characteristics are described in the following section. Of the other vessels in this class, most retired undamaged and unaided. [25] Twelve other yachts in Class III and IV also finished. [25] In addition to overall displacement, both the manner in which the weight is distributed, and the form of the hull are crucial factors, so that boats of similar size but different shapes "may vary considerably in their behavior in the same heavy sea conditions." In particular, the weight distribution and the hull shape impact a vessel's stability characteristics and the motions she experiences in a seaway, which are vital parameters in determining her dynamic response to the environment. [3] As a lower bound on size, reference [2] suggests that certain boats of 20 feet L_{OA} or more have proven themselves seaworthy enough to sail around the world. This claim is supported in some measure by documented ocean passages in craft such as the Flicka 20 and Cal 20. [4]

Stability The stability of a yacht is a measure of her ability to resist inclining moments along the pitch (longitudinal stability) or roll (transverse stability) axes. As a vessel is inclined along one of these axes, a righting arm, GZ, develops that acts to restore her to the upright condition. The magnitude of GZ and the corresponding righting moment, $RM = \Delta * GZ$ may be used to quantify static stability at all angles of inclination. [26] Because this moment depends in part on the waterplane moment of interia, longitudinal stability is much greater than transverse stability for typical vessels in normal conditions. The discussion that follows focuses on transverse stability.

Ship static stability depends on the location, relative to the keel (K), of three centers: the center of gravity (G), the center of buoyancy (B), and the metacenter (M). The metacenter is defined as the intersection of verticles drawn through the upright and inclined centers of buoyancy, as the inclination angle approaches zero. [26] For small angles of inclination, it may be assumed to be fixed. Referencing the centers to the keel of the ship, the following relations between center distances hold [26]:

$$KM = KB + BM \tag{1}$$

$$GM = KM - KG \tag{2}$$

Here, KB is the distance from the keel to the centroid of the displaced volume of the hull, and so depends on the shape of the submerged hull form. KG is the distance from keel to center of gravity and depends only on the weight distribution of the yacht structures, payload, and crew. The metacentric radius, BM, is the distance from the center of bouyancy to the metacenter, and the metacentric height, GM, is the distance between the center of gravity and the metacenter. When the ship is inclined, the center of buoyancy shifts to the new centroid of the displaced volume. Because the equal and opposing forces of gravity and buoyancy are now offset by some distance, they form a couple which generates a moment acting to restore the ship to the upright condition. The offset distance is the aforementioned righting arm. Figure 2, reproduced from [5], displays these centers, distances and righting arm.

An analysis of ship stability is divided into initial stability and ultimate stability. Initial stability refers to the vessel behavior at small angles of heel and is largely a function of form, in particular beam.[2] Below about 25-30 degrees of heel, this "form stability" is the dominant factor in determining the yachts stiffness, or resistance to heeling. [3] Initial stability is an important parameter in determining the 'normal' operating condition for a yacht, which is designed to heel as she carries sail. Ultimate stability refers to the vessels resistance to inclining moments over the entire range of possible angles of inclination, and is heavily dependant on weight distribution [2], in particular the location of the vertical center of gravity. Beyond about 30 degrees, the vessel stability is increasingly determined by this "weight stability". [3] Ultimate stability is a vital parameter in assessing a yacht's response to extreme conditions, where capsize is possible.

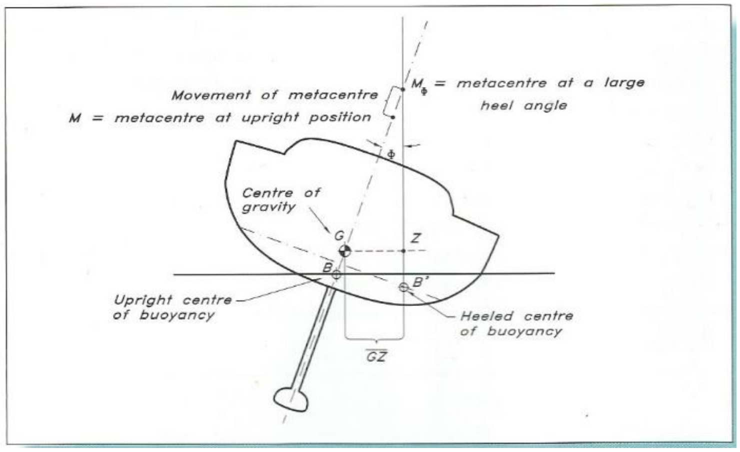


Figure 2: Static Stability Diagram

Simple relations exist to calculate initial stability. At small angles of heel, it may be shown that the metacentric radius, BM, depends only on the waterplane moment of inertia and the submerged volume.[26]

$$BM = I/\nabla \tag{3}$$

If the hull geometry and weight distribution are known, the GM may be determined from equations (1) through (3). GM in turn is directly related to the righting arm and righting moment developed at a given heel angle, θ. For small heel angles [26]

$$GZ = GM * sin(\theta) \tag{4}$$

$$RM = \Delta * GM * sin(\theta) \tag{5}$$

Metacentric height thus serves as a key parameter of initial stability, with a larger GM corresponding to larger righting moments at small angles of heel. It can thus be used to compare initial stability across different designs, either on its own or in combination with SA, Δ, and heeling arm, HA, to compute the Dellenbaugh Angle, DA. [5]

$$DA = 279 * \frac{SA * HA}{\Delta * GM} \tag{6}$$

The DA is an indicator of sail carrying capability arising from initial stability, with lower angles corresponding to stiffer yachts of larger initial stability, and higher angles corresponding to more tender yachts. Statistics on DA ranges for modern yachts are presented in reference [5], along with statistics on yacht GM's, which range from 10-15% of the L_{WL} for modern yachts.

It is immediately apparent from equations 4 and 5 that initial stability can be increased by increasing GM. Because GM is the distance between the the vertical center of gravity and the metacenter, this can be accomplished by lowering the center of gravity, raising the metacenter, or both. The position of the metacenter is determined by hull geometry, in particular the ratio of a vessel's beam to her depth. A higher beam-to-depth ratio raises the metacenter, improving initial stability. [3] Modern yachts display a trend towards beamy, shallow hulls that possess high initial stability from their form. [3] This enables carrying large sail areas without incurring a "penalty" for increased ballast weight. Such yachts tend to have larger GM's than more traditional yacht designs of narrower beam and deeper hull. [5] However, this does not indicate that the modern yachts are more seaworthy than their traditional forebears, because form stability is dominant only at small angles. While GM must be sufficiently high to enable the yacht to stand up to her sails and operate well under normal conditions, it is the yacht's ultimate stability that governs her ability to withstand and recover from the severe inclinations, including knockdown or capsize, that can occur in extreme heavy weather.

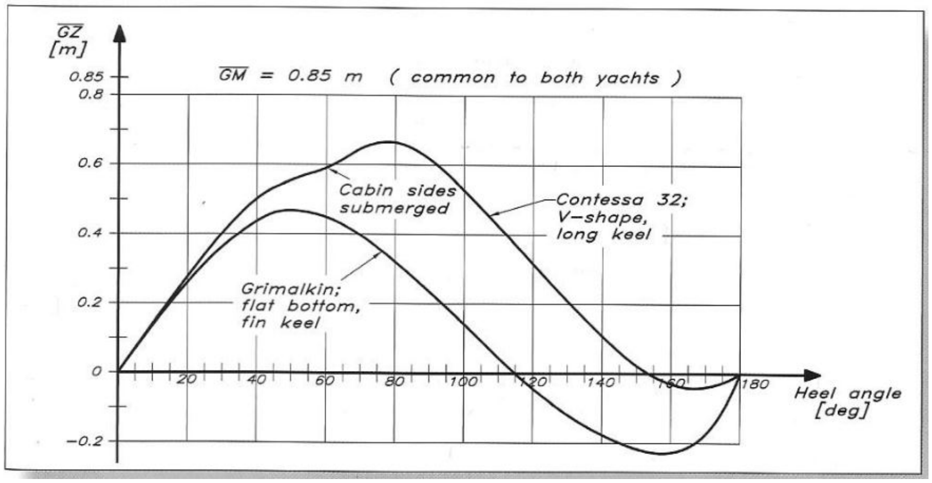


Figure 3: Righting Arm Curves for the Contessa 32 and the Grimalkin

In contrast to initial stability, no simple formulae exist for determining GZ at large heel angles. As the yacht heels to larger angles, the waterplane shape changes significantly, and the yacht trims in proportion to the different volumes submerged by forward and after sections. An iterative approach is required to address the unknown heeled waterline. [5] Although approximate methods exist [5], this computation in practice this is accomplished by computer software. The result is a plot of righting arm over a range of heel angles from 0 to 180 degrees. Two such curves are displayed in figure 3, reproduced from reference [5]. The curves correspond to two vessels of similar size, with identical GM = 0.85 m. Both vessels competed in the disastrous Fastnet Race of 1979, and evidence the marked difference that can exist in ultimate stability between vessels with the same initial stability. The Contessa 32 is a more traditional hull form, with a deeper, V-shaped hull and a longer fin keel. She was the only yacht in the smallest class to finish the race. The Grimalkin is more typical of modern yachts, with a beamy, shallow hull, lower displacement/length ratio, and short fin keel. Racing in the same class as the Contessa 32, she experienced a capsize that left her inverted for at least 5 minutes, and resulted in the deaths of three of her crew. [3] A cursory examination of the curves indicates much greater stability characteristics in the more traditional Contessa 32. [3]

The overall stability curve of a design can be used to calculate the righting moment at any heel angle, though there are several specific elements of the curve that bear particular importance to a seaworthy yacht. In addition to the initial stability already discussed, these elements are the maximum righting arm and corresponding angle, the angle of vanishing stability, the righting energy, and the area of negative stability. If the yacht experiences a heeling moment greater than the maximum righting moment, which occurs at the angle ϕ_{MAX}, she will capsize if the moment persists long enough. [5] However, positive righting moments are still generated beyond the maximum, up to the angle of vanishing stability, ϕ_v, meaning that if a passing wave knocks a yacht to a heel angle beyond ϕ_{MAX}, she will still be capable of recovering upright without capsize. The range of angles from upright to ϕ_v form the range of positive stability. The area under the GZ curve up to a given angle is related to the work required, by wind and waves, to heel a yacht to that angle. [5] Thus, the total righting energy, RE, equal to the work required to heel the yacht to ϕ_v, may be expressed in kg-m-degress, as [6]

$$RE = \Delta A_{GZ} \tag{7}$$

Where Δ is the displacement in kg and A_{GZ}, the area under the curve, is given by:

$$A_{GZ} = \int_0^{\phi_v} GZ(\phi)\, d\phi \tag{8}$$

and may be computed by numeric methods, such as the trapezoidal rule, given an overall stability curve. A large range of positive stability, a maximum righting arm that occurs at a relatively high ϕ_{max}, and a large RE are all important to a seaworthy design.

Beyond ϕ_v, where the righting arm becomes negative, the yacht is stable in the inverted position. The range of inverted stability, from ϕ_v to 180 degrees, and the area enclosed by the GZ curve in this region both pertain to the difficulty a yacht encounters when recovering from a capsize. [5] A larger range of inverted stability and a greater area mean that the external forces of waves and wind must do more work to heel the inverted yacht back to the region of positive stability, where she can right herself. The larger this region, the longer a yacht will remain capsized, and the greater the risk to the crew. Minimizing this region improves the seaworthiness of the design.

Key design factors that influence overall stability GZ curve are the position of the center of gravity, the vessel beam and depth, the hull shape, and the freeboard. A lower center of gravity improves the stability curve over the entire range. As discussed above, lowering the center of gravity improves initial stability. However, unlike form effects, the improvement is not limited to small angles, but persists through 180 degrees of heel. It results in a larger ϕ_{MAX}, a larger ϕ_v, and a correspondingly greater range of positive stability and smaller range of negative stability. It is then "clearly better, from a safety standpoint, to have as low a practical value of G as possible." [3] More traditional hulls, similar to the Contessa 32, exhibit comparatively narrow and deep forms and tend to have G located below the waterline, relying on the weight stability that a low G confers. More modern yachts, such as the Grimalkin, tend to have G located above the waterline, relying instead on their form stability which arises from a shallow hull of wide beam. [3] This trend results in inherently less stable, less seaworthy boats, and is an unfortunate and unintended consequence of the influence of yacht ratings rules on design. [3]

The beam of a vessel has a significant impact on her overall stability characteristics. As discussed above, a relatively large beam in combination with a low immersed depth raises the metacenter, providing high initial stability. At larger angles of heel, however, excessive beam becomes dangerous, since it promotes greater inverted stability. Beam thus represents a tradeoff between initial and ultimate stability, with a wider beam raising the maximum righting moment, but also reducing the range of positive stability. [3] Experiments conducted on yacht capsize behavior in waves have indicated that beam is the most significant factor impacting capsize, with wider beam resulting in greater knockdown angles, greater range of inverted stability, and less downwave control. [3] Obviously, the characterization of 'wide' or 'narrow' beam depends on the overall size of the vessel, and some minimum beam is required to enable adequate initial stability for sail carrying ability. Scaling a yacht up in length, stability increases with the fourth power of L_{WL}, while heeling moment increases with the third power of L_{WL}. [5] Thus, vessels of larger L_{WL} can have proportionally smaller beams, with larger L_{WL}/B ratios. For small sailboats, a maximum L_{WL}/B ratio is around 3; any higher and they encounter difficulty standing up to their sails in moderate wind. [2]. A ratio closer to 3 would be considered a 'narrow' beam for the size of sailing vessels considered here, while a ratio closer to 2 would be considered relatively beamy.

Hull shape also impacts the stability characteristics of the vessel. A flared hull increases transverse stability, by providing increasing buoyancy as the boat heels. [2] The increase in stability grows with increasing flare angle. [3] Tumblehome hulls improve longitudinal stiffness [2], and may help reduce topsides weight, which is beneficial for maintaining a low G. [2],[5]. However, excessive tumblehome results in a reduction stability at large heel angles.[27]

The impact of freeboard on stability is somewhat mixed. While a higher freeboard may increase the range of positive stability, as stated in references [27] and [2], several studies summarized in reference [3] on the capsize of yachts in waves have indicated that increased freeboard shows little to no impact on capsize probability, or even slightly reduces the range of positive stability.[28] The freeboard experiment detailed in reference [28], which indicated a reduction in range of positive stability with greater freeboard, consisted of replacing the coachroof and deck of the parent model with a flush deck at a higher freeboard. This change in geometry may explain why a slightly greater range of negative stability was observed in the higher freeboard model. In the present work, the author's variance of freeboard as a design parameter and subsequent computations has indicated that, at least for vessels of similar characteristics to the proposed design, higher freeboard does result in improved static stability characteristics, provided the VCG is held fixed. This is discussed further in subsequent sections, but it is noted here that this relationship is developed under hydrostatic analysis, and does not include the effect of wave impacts of the type considered in the above-mentioned studies. Because the energy imparted by a breaking wave to a yacht depends on the area of the yacht subjected to the wave jet,[3] it is possible that excessive freeboard may make a yacht more vulnerable to these forces. Furthermore, higher freeboard in a light, shallow hull can raise the center of gravity appreciably, actually decreasing stability through this effect, unless it is compensated for with additional ballast.[27]

Finally, a vessel's stability characteristics are impacted by the action of waves. The righting moment of the vessel is influenced by the centrifugal forces of water particles near the surface, which travel in orbits as the wave passes. On the wave crest, the centrifugal forces imparted to the hull oppose the force of gravity experienced by the hull, resulting in a virtual decrease in hull weight and a corresponding reduction in righting moment.[5] This is particular dangerous, given that it is on the wave crest that the yacht is exposed to the full force of the winds.[3] The opposite occurs in a trough, with a virtual rise in displacement and increase in stability. In severe conditions, such as those experienced in the 1979 Fastnet Race, such effects can result in a reduction of nearly 50% to the hydrostatic GZ curve of a yacht such as Grimalkin.[5] This may argue in favor of 'overdesigning' the static stability characteristics of a small yacht, since it is known in advance they will be reduced in severe weather. However, hull shape also impacts the degree of wave effect on stability. Hull sections with V-shapes, rather than U-shapes, experience less vertical accelerations from waves in the speed ranges of interest to sailing yachts, and therefore exhibit a smaller reduction in stability. The V shaped sections have the added benefit of reducing slamming. Section depth provides a similar benefit, with a deeper hull draft associated with improved dynamic stability in waves. The mean hull depth of a traditional yacht may be on the order of 5 percent of her waterline length, while a 'modern' yacht influenced by ratings rules may be half that.[3] The impact of waves on stability seems to argue in favor of the traditional yacht design, of comparatively deep hull and V-shaped sections.

The Stability Index The stability characteristics of a given vessel may be used, together with her principal dimensions, to compute the Stability Index (STIX) defined by ISO 12217-2. The STIX is a measure of a yacht's ability to withstand and recover from knockdown and inversion. It thus forms a means of assessing the overall safety of a vessel's stability and buoyancy characteristics, and it is used, in conjunction with other design information, to assign the vessel to a design category that indicates the limiting environmental conditions for which she is deemed suitable. [6] Vessels of design category A are suitable for extended open ocean sailing. The conditions associated with design category B might occur on "offshore voyages of sufficient length," while those of design category C may be encountered in coastal sailing or on exposed inland waters. Conditions associated with design category D are expected on sheltered inland waters. [6]. Design category information is summarized in table 6.

Parameter	**Design Category**			
	A	B	C	D
Beaufort Wind Force	<10	≤8	≤6	≤4
Max Wind Gusts (m/s)	32	27	18	12
Wind Speed (m/s, 10min avg)	24.4	20.7	13.8	7.9
Significant Wave Height (m)	7	4	2	0.3

Information from [6].

Table 6: ISO 12217-2 Design Categories and Environmental Limits

Different elements of the design that are important from a seakeeping and safety perspective are included as factors in the STIX calculation. [5] Formulae for each factor are included in appendix I These factors are:

1. **Base Length Factor (LBS)**: A weighted average of the L_{OA} and L_{WL}, this accounts for the fact that the size of a yacht defines the relative size of the waves she encounters: the smaller the yacht, the larger the relative size of the wave. It is heavily weighted in the STIX computation, reflecting size as "the single most important parameter when assessing safety at sea." [5]

2. **Displacement Length Factor (FDL)**: Depends on the displacement of the yacht relative to her length, and penalizes lighter yachts, as a larger displacement/length ratio is judged to improve resistance to capsize. [6]

3. **Beam Displacement Factor (FBD)**: Large beam relative to a yacht's displacement increases susceptibility to wave induced capsize. Large beam also increases the range of inverted stability, reducing the yacht's ability to self-right. [5] However, excessively narrow beam reduces initial stability and results in excessive heel angles under normal sailing conditions, potentially increasing the risk of downflooding, the inrush of water at recess or cabin openings. This factor depends on both B_{WL} and B_{OA}, and is computed differently based on if the yacht's beam relative to her displacement is considered narrow or wide relative to a 'normal' yacht. [5]

4. **Knockdown Recovery Factor (FKR)**:Characterizes the yacht's ability to spill water from the sails and recover upright after a knockdown, and depends on the ratio of the righting moment at 90 degrees heel and the wind heeling moment on the sails. [5]

5. **Inversion Recovery Factor (FIR)**: Depends on the angle of vanishing stability, ϕ_v, and characterizes the yacht's ability to recover unaided from an inversion. [6]

6. **Dynamic Stability Factor (FDS)**: Related to the work required by wind and waves to heel the yacht to ϕ_v, and depends on A_{GZ}, the area under the GZ curve in the range of positive stability.[5]

7. **Wind Moment Factor (FWM)**: Represents the risk that a sudden gust of wind could cause downflooding, by heeling the unreefed yacht to the downflooding angle, ϕ_D. Depends on ϕ_D and the apparent wind speed needed to heel the vessel to this angle.[5]

8. **Downflooding Factor (FDF)**: This factor depends on the downflooding angle, and is related to the risk of downlfooding during a knockdown.

The above factors are combined to yield the STIX:

$$STIX = (7 + 2.25 * L_{BS}) * (FDL * FBD * FKR * FIR * FDS * FWM * FDF)^{0.5} \tag{9}$$

In addition to the STIX, other design parameters influence the design category a vessel receives. These include structural strength (discussed in a subsequent section), the ability to detect and remove water, the required safety gear aboard, and prescribed minimums for righting energy (as computed from equation 7), angle of vanishing stability, downflooding angle, and downflooding height, H_D. Select requirements for categories A and B that influenced the design process are presented in table 7.

Parameter	Design Category	
	A	**B**
STIX	32	23
RE (kg-m-deg)	172,000	57,000
ϕ_v (deg)	$130 - 0.002m$ or 100	$130 - 0.005m$ or 90
H_D (m)	0.5	0.4
ϕ_D (deg)	40	40

[1] All tabulated values are required minimums. Minimum angle of vanishing stability is determined by the maximum of a fixed angle (100 or 90) and a computed angle that accounts for vessel's mass, m.

Table 7: Select Requirements for Design Categories A and B, ISO 12217-2

Motions In addition to her static stability characteristics, a vessel's dynamic response in waves and wind influences her seaworthiness. The motions induced on the yacht by the environment influence both her seakindliness and susceptibility to capsize. The former is relevant not only to crew comfort, but also to the crew's ability to operate the vessel in severe conditions: excessive, violent motions increase the likelihood that fatigue or sickness will incapacitate the crew. In designing a seaworthy yacht, it is necessary to examine what design factors contribute to motions that are seakindly and minimize the probability of capsize.

As discussed in connection with dynamic stability, the vertical accelerations of a hull in waves are influenced by the section shape and depth. Hulls of deeper, V-shaped sections are subject to significantly lower accelerations. A 'deep' section may be consider to have depth on the order of 6% of L_{WL}, while a 'shallow' section may have around half this depth. [3]. Pitching motions are also affected by the hull flare, particularly in the forebody. Flared forward sections act to damp pitching motion, and, together with high freeboard forward, help keep the decks drier from spray and green water.[5] Hull overhangs, which act to increase a yacht's sailing length as she heels, can also influence pitching motion. A long forward overhang will likely increase pitching in head seas, as waves hitting the overhang generate large pitching moments. The same may occur with long aft overhangs in stern seas, but is unlikely to be as problematic due to the reduced encounter frequency; the stern overhang may actually help damp the pitching motion arising from head seas.[5] However, long after overhangs may pose a threat in quartering seas, by providing a lever to slew the yacht into a broaching position.[2] Additionally, long overhangs on either end can increase slamming in a seaway.[2]

Beyond the hull shape affects, the motions a yacht experiences in seas depend on her inertia, stability, and damping. The most significant motions are pitch, roll, and heave, meaning the most important inertial quantities are the moments of inertia about the transverse and longitudinal axes, as well as the mass of the yacht itself. Roll is the most significant motion from a safety perspective, while pitch and heave influence the added resistance a yacht encounters in waves.[5] As discussed above, a lighter yacht will experience more severe motions in waves, all else equal; similarly, larger moments of inertia about the key axes will tend to reduce the roll and pitch motions forced by a given sea state.[5] A key contributor to both moments is the weight of the mast, rig and sails, due to its distance from the axes. Taller or heavier rigs thus act to limit rotational motions induced by waves. However, an excessive inertia of the rig can also produce the undesired effect of 'hobbyhorsing,' an impeded forward motion when attempting to sail to windward in head seas.[2] Additionally, heavier or taller rigs can raise the vertical center of gravity, negatively impacting ultimate stability.

The total inertia of a yacht, which contributes to her response to wave forcing, is a function not only of her own weight and its distribution, but also of the added mass of the water entrained and carried along by her underwater profile and appendages. This mass of water must move with the hull, and thus contributes to her response. In particular, the rolling energy, E_r, transferred to a vessel by a wave may be expressed[3]:

$$E_r = \frac{(\int_0^{\Delta t} M_r \, dt)^2}{2 * (I_r + I_a)} \tag{10}$$

where M_r is the roll moment induced by the wave slope, I_r is the yacht's roll moment of inertia, and I_a is the added mass roll inertia. Larger lateral area on the hull underbody and appendages act to increase the added mass inertia, improving a yachts motion in waves from a seakeeping perspective.[3]

The natural roll period, T_n of a yacht depends on both GM and the roll gyradius (k), a measure of the mass distribution related to the moment of inertia: [3]

$$T_n = 2\pi \frac{k}{\sqrt{g * GM}} \tag{11}$$

where g is the acceleration due to gravity. Thus, yachts of higher initial stability, that is higher metacentric height, experience shorter rolling periods and correspondingly higher accelerations.[3] The factors in the preceding discussion on stability then apply to roll motion; in particular, vessels of wide beam and shallow hull form may be expected to experience higher roll induced accelerations. This results not only in less seakindly motions, but also increases the likelihood of capsize from wave induced rolling. It has been shown that the relationship between T_n and the wave period of encounter, T_e, determines whether a a yacht will lean toward or away from the wave crest: if T_n is greater than T_e, the yacht tends to lean toward the crest as she rides over the wave, an inherently safer attribute in a storm.[3] Additionally, the roll moment of equation 10 is proportional to the righting arm, GZ, in addition to wave slope, α, and the duration during which the boat is exposed to the wave[3]:

$$M_r \propto \alpha GZ \Delta t \tag{12}$$

Thus, boats of higher initial stability actually experience greater roll moments and more violent motions from waves. As previously discussed, beam is the key design parameter in determining the initial stability. Experimental studies have corroborated the negative effect of comparatively wide beam on capsize in breaking waves, indicating it is a factor of primary importance, with displacement and position of vertical center of gravity a secondary factor.[28]

In addition to her inertia and stability, the damping characteristics of a vessel are of vital importance in mitigating her response to waves. A vessel's damping reduces the extreme heel angles and knockdown probability from waves, by dissipating a portion of the energy imparted by the exciting force.[3]. It is a critical parameter from a safety perspective, as it limits the rapid magnification of roll angle that occurs in the resonance condition, where the wave encounter frequency matches the natural frequency of the yacht.[3],[5] While encounter frequency may be altered through a course or speed change, it is possible to achieve dangerous roll conditions after the passage of only 2 or 3 synchronous waves, which could occur at any time.[3] Additionally, certain downwind sailing conditions can result in aerodynamically induced rolling, even in the absence of waves. With insufficient damping, such conditions can lead to broaching and knockdown.[3] Roll damping characteristics thus form an essential component of a seaworthy yacht.

Vessel damping is caused by friction between the yacht and water, generation of waves by the yacht's motion, and generation of vortices from the hull underbody, keel, rudder, and sails. The latter source is the dominant effect in sailing yachts.[5] The damping efficiency of the yacht depends on her section shapes, with V shaped sections providing considerably better damping than U or semi-circular sections. Wineglass sections with soft bilges provide better damping than even the V shaped sections.[3] Damping also depends on both hull depth and lateral area, with a greater lateral area and deeper hull improving damping characteristics. This of course comes at the cost of increased resistance from skin friction, which is the dominant component of resistance at lower speeds. Finally, the impact of the keel is highly significant, with damping increasing with the length of the keel. The most efficient keel, from a damping perspective, is the "longest one which can be mounted on the hull centerline."[3]

Hull balance is another design factor linking stability and dynamic motions in a seaway. A balanced hull is one that does not experience substantial changes to trim or heading when she heels or rolls. [3]. Modern racing yachts exhibit wide, full stern sections and narrow, fine bow sections to reduce resistance and improve surfing abilities. Such hull forms experience bow down trim and large yawing moments when they heel or roll, due to the dramatically different volumes immersed in forward and after sections.[5] A practical means of ensuring balance in a yacht is to design the hull to displace roughly similar volumes forward and aft when heeling. This prevents the large and reversing yaw moments that can occur when the vessel rolls in heavy seas, which could lead to broaching and capsize.[3]

Coursekeeping and Maneuverability In addition to the hull balance discussed above, the design of the keel and rudder are important factors determining a yacht's coursekeeping and steering abilities. Also of significance is the balance between the aerodynamic forces on the rig and the hydrodynamic forces on the keel and hull, which affects the degree and direction of rudder necessary to maintain course in a given set of conditions. A distinction is made between coursekeeping and maneuverability: the former pertains to what may be termed 'directional stability,' the propensity of the yacht to return to her original course after disturbance from wind or wave forces.[3] This relates as well to the yacht's self-steering ability, an important consideration for singlehanded or short-crewed yachts. Maneuverability, by contrast, refers to the control of the yacht, that is, how lively and easily steered she is. This directly conflicts with the coursekeeping ability of the yacht, such that steering characteristics are ultimately a trade off between maneuverability and directional stability.[3] While lively steering may certainly be an asset in navigating crowded harbors or in racing, it is the coursekeeping abilities of a yacht that pertain most to her seaworthiness, particularly in heavy weather conditions.

A primary function of the keel is to balance the aerodynamic side force of the sails, minimizing leeway. Some leeway will always be required to provide a sufficient angle of attack that enables the keel to develop adequate lift to resist the sideforce from the sails. However, "drag is always a drag,"[29], and ought to be kept as minimal as practical, consistent with the considerations discussed below. The main components of drag on a foil such as a keel or rudder are the viscous drag, dependant on wetted area and Reynolds number, and the induced drag, discussed below. The rudder's function is to impart moments to the ship to control her heading, and depends not only on the rudder design but also its location. Rudders placed well aft deliver greater steering moments than a similar rudder located more inboard.

One of the most significant parameters of the keel or rudder is its Aspect Ratio (AR). The geometric aspect ratio depends on the ratio of the span, S (depth) of the keel and the mean chord length, C; and may also be expressed in terms of planform area, A:

$$AR_g = \frac{S}{C} = \frac{S^2}{A} \tag{13}$$

The effective aspect ratio accounts for the mirroring effect of a solid boundary, such as the hull bottom, and also may be used to account for increases in drag and reduction in lift that occurs for force distributions that depart from the ideal elliptical distribution.[5] For a foil of elliptical distribution mounted on a relatively flat hull, $AR_e \approx 2AR_g$. Treating the keel and rudder as foils, lifting line theory predicts that a higher AR results in a better Lift/Drag ratio and more efficient foil. In particular, the following relationships hold for non-dimensional lift and induced drag coefficients, C_L and C_{Di} of a foil at angle of attack α:

$$C_L = \frac{C_{L,2D,1^\circ}}{1 + \frac{2}{AR}} * \alpha \tag{14}$$

$$C_{Di} = \frac{C_L^2}{\pi * AR} \tag{15}$$

where $C_{L,2D,1^\circ}$ is the lift coefficient of the 2 dimensional foil cross section at 1 degree angle of attack, and is a function of section geometry. For symmetric foils, 0.10 is a good approximation.[5] The coefficients are related to their dimensional counterparts lift, L, and induced drag, Di, by planform area, A, speed, V, and fluid density ρ:

$$C_L = \frac{L}{\frac{1}{2}A\rho V^2} \tag{16}$$

$$C_{Di} = \frac{Di}{\frac{1}{2}A\rho V^2} \tag{17}$$

Thus, solely from a speed perspective, the high aspect ratio, low wetted area fin keel and separated rudder common on modern racing yachts appear justified.[3] Fin keels of modern yachts commonly have effective AR's of around 3, while more traditional long keels have AR's less than 1.[5] However, the former arrangement has distinct drawbacks from a seaworthiness perspective. One such drawback was discussed in connection with roll damping: the shorter, high AR fin keels are considerably less effective at damping roll motion.[5]. Other drawbacks include lower stall angles and their behavior in rough seas, which departs considerably from the steady flow assumptions of lifting line theory.

The stall angle of a foil decreases as its aspect ratio increases.[5],[3] The stall angle is the angle at which flow separates from the foil surface, causing a marked increase in drag coefficient and decrease in lift coefficient. Because of their small areas, equation 16 implies that this can be problematic for high aspect ratio fins at sufficiently low speeds, even in calm waters. When such fins stall, the yacht has a tendency to move sideways through the water.[5] Foils of small aspect ratios, less than around 1.5, demonstrate significantly larger stall angles as a result of the increased impact of the 3 dimensional flow effects of the foil tip vortex.[3] In rough seas, the angle of attack seen by the foil is significantly impacted by the orbital velocities of the water particles, and also the roll motion of the boat. In such conditions, the high aspect ratio foils common on modern fin keels may easily encounter angles of attack beyond their stall angles, resulting in degradation of coursekeeping capability. By contrast, low aspect ratio keels are unlikely ever to operate beyond their much higher stall angles.[3]

The impact of unsteady flow effects reveals further benefits of a low aspect ratio keel. There is a time lag, or hysteresis effect, in establishing steady flow conditions after the angle of attack is abruptly changed, as occurs frequently to a keel in waves or on a rolling boat. This time lag mean the new steady state lift force doesn't appear instantaneously, but develops on a time scale related to the chord length. This means that a long keel confers steadiness, by providing an averaging effect and effectively filtering out relatively high frequency variations. In this sense, a long keel helps act as a 'yaw-damper'.[3] Additionally, the side forces (and resulting yaw moments) that develop on higher AR foils in unsteady conditions are larger, both because of the time lag effect and because of the higher C_L that they experience relative to low AR foils. This requires larger rudder forces to compensate, which may result in overload in severe conditions. In a seaway, larger keel aspect ratio results in a more heavily loaded rudder.[3]

As in keels, the aspect ratio of the rudder is a key parameter, with higher aspect ratios providing improved lift-to-drag characteristics at the expense of lower stall angles. From equation 16, the area of the planform is also significant in determining the force a rudder can deliver. Because the rudder is used to impart yaw moments, its location relative to the hull's yaw axis is also relevant. A rudder placed well aft will deliver a greater turning moment than the same rudder mounted inboard,[2] at least in calm water. However, this mounting configuration is susceptible to reduced effectiveness in waves, as the stern lifts and pulls some of the rudder from the water, reducing the effective rudder area.[3] There is also the risk of ventilation, in which the low pressure suction side of the rudder entrains air. This effect may be particularly acute for high aspect ratio spade rudders that develop large suction peaks at the leading edge, and also for rudders whose aft corner is near the edge of an immersed transom.[3]. The portion of the rudder near the surface is also more heavily impacted by the orbital velocities of the water particles in waves, which can reduce the apparent velocity seen by the foil and so reduce the rudder force. Finally, the stall angle of a rudder can be considerably increased by mounting the rudder on a fixed skeg.[3] This benefit is in addition to the added structural security provided by such a configuration. Deeply immersed rudders of comparatively low aspect ratio and high areas, hung either on the keel or on a skeg, provide mitigation for the effects of seas on rudder effectiveness.

A final but essential point influencing the coursekeeping and steering capabilities of the yacht arises from the balance between the sail plan and the underwater hull, keel, and rudder. If the aerodynamic force on the sails (lift and drag resultant) does not act along the same line of action as the hydrodynamic force on the underwater profile (sideforce and resistance resultant), then a yawing moment arises. The rudder must be held at some angle to compensate for this moment and maintain course. If the yaw moment acts push the nose of the vessel into the wind, or 'luff up', it is termed weather helm; if it acts to push the vessel off the wind, or 'fall off', it is lee helm. The yawing moment magnitude and direction depend on the relative locations of the aerodynamic center of effort of the sail plan and the hydrodynamic center of lateral resistance of underwater profile. As the yacht heels, the aerodynamic center moves to leeward and the hydrodynamic center shifts forward due to the asymmetric shape under heel, though the latter may be insignificant in narrow yachts of high symmetry.[5] Because of these effects, it is impossible to achieve a balanced yacht at all heel angles, so the emphasis is typically placed on achieving good balance at smaller heel angles, and tolerating more weather helm at greater heel angles.[5] Good balance requires a small amount of weather helm, around 3 or 4 degrees, for reasons of both safety and performance. This gives the helmsman a feel on the vessel's response, and ensures that if the helm is let go, the yacht will naturally depower, turning into the wind. [27],[2] It also mitigates the effects of sudden gusts by the same mechanism.[5] By contrast, lee helm is dangerous in anything but light airs, as it may potentially result in a dangerous broaching situation. Beyond reasons of safety, a few degrees of weather helm may actually reduce the resistance of the yacht, by partially unloading the keel. Excessive weather helm, however, imposes a drag penalty; the optimum angle may be found for a given yacht through tank testing, but a simple rule of thumb approximation gives this as 5 degrees.[5]

The dominant parameter in establishing balance is the sail plan's lead, the horizontal distance between the sail plan CE and the underwater CLR (the sail plan CE is always positioned forward of the CLR).[27],[5] Semi-empirical rules of thumb, based on practical experience, exist to estimate the necessary lead for different sail plans and hulls to achieve reasonable balance.[5],[27],[30], which may nonetheless require adjustment after initial sailing tests.[5] In applying such rules, it is typical to use the geometric center of the sail plan, and either the geometric center of the underwater profile or a an approximation to the hydrodynamic center, such as the 25% chord line.

Cockpit The cockpit design warrants mention, as it can prove vulnerable to pooping in heavy seas. To mitigate this, the cockpit should be relatively small and quick-draining, and the companionway entry should be protected by a substantial bridgedeck or sill at the same height of the cockpit seating.[2] ISO 11812 provides requirements on cockpit bottom height above waterline, H_B, maximum acceptable drain time, t_d, and sill height, h_s for the different design categories, displayed in table 8.[31] The required drain time is a function of cockpit volume coefficient, k_c, which relates the volume of the cockpit, V_c to the yacht's reserve bouyancy as determined by her L_H, B_{OA}, and midships freeboard, F_M:

$$k_c = \frac{V_c}{B_{OA} * L_H * F_M} \tag{18}$$

Parameter	Design Category	
	A	B
H_B(m)	0.15	0.1
t_d(min)	$0.3/k_c$	$0.45/k_c$
h_s(m)	0.3	0.25

Table 8: ISO 11812 Cockpit Requirements

Summary The above discussion indicates design features that should be emphasized for a sea-worthy yacht, bearing in mind the inherent compromises they entail. These key features may be summarized:

1. Comparatively narrow beam, higher L/B and lower B/D ratios.
2. Higher relative displacement, higher Δ/L ratios or lower $L_{WL}/\nabla^{1/3}$ ratios.
3. Initial tenderness, GM adequate but not excessive.
4. A low center of gravity, located below the waterline.
5. A balanced hull, displacing roughly similar volumes fore and aft on heeling.
6. V shaped or wineglass sections of slack bilges.
7. Relatively deep hull, of large lateral wetted area.
8. Moderate flare forward.
9. Short or minimal overhangs.
10. A long, low aspect ratio keel.
11. A low aspect ratio, large and deeply immersed rudder that is mounted on the keel or a on a skeg.
12. Good balance between the sails and underwater profile.
13. Small, quick-draining cockpit with bridgedeck.

2.3 Hull and Deck Structures

The structural strength of the hull and deck must be adequate to withstand the loads imposed by heavy seas. One approach to ensure this is through the proper application of scantling rules, based on the structural dimensions of successful vessels of similar type to the design. This approach is relatively quick and simple to employ, which is advantageous particularly in the early stages of an iterative design approach. The resulting structures may then be checked for adequate strength by analysis at design loads specified by an accepted design standard. The other general approach is a detailed engineering analysis of the boat structures. While this is a more time consuming, and thus costly, approach, it allows designers to push the performance envelope of materials or adopt unusual hull forms or structures for which existing scantling rules do not apply.[24] The scantling rule approach was adopted for this design.

2.3.1 Scantling Rule

A formalized scantling rule establishes the required materials and dimensions for a vessel's structures based on a few principal dimensions and key design parameters. The rule is based on "engineering analysis cross-checked against a database of successful vessels."[24] Scantling rules have been one of the principal methods of boat construction for over a hundred years, and have been adopted by class societies such as American Bureau of Shipping (ABS) and Lloyds. However, because of their empirical basis, scantling rules are applicable only to the size and type of vessels intended by the rulemaker.[24] The scantling rule selected for use in this design applies to all monohull vessels, sail or power, between 3 and 37 m L_{OA} and of speeds up to 45 knots. The rule is detailed in reference [24].

The key reference point of the selected rule is the Scantling Number, Sn, which depends on L_{OA}, B_{OA}, and midships depth, D_M, measured from the hull bottom to the sheerline at midships. If the dimensions are expressed in meters:

$$Sn = \frac{L_{OA}B_{OA}D_M}{28.32} \tag{19}$$

If hull has significant overhangs or flare, the length or beam dimension is replaced with the average of the waterline and overall dimensions. If the hull has hollow garboards where the tuck of a keel is faired in, the hull bottom is found by extending the hull lines from the tuck base to centerline.

Given a construction material, for example Fiber Reinforced Plastic (FRP), the Sn is used to calculate all required dimensions for hull plating and internal structures. The rule for FRP assumes an alternating layup of Chopped-Strand Mat (CSM) and bidirectional Woven Roving (WR) E-glass layers in a polyester resin matrix, with a 35% glass content by weight in the resulting laminate. This is the most common layup procedure for FRP production boats.[24] The application of this rule to the present design is discussed in more detail in section 4.

2.3.2 ISO 12215

ISO 12215-5 defines design pressures on plating and stiffeners for use in checking the adequacy of scantlings to withstand local loading. The design pressures are reduced by 20% for each drop in design category, such that design category A vessels are subject to the full design pressure and have the highest load requirements.[32] These design pressures may be used in conjunction with material properties and one of several methods to determine adequacy of scantlings. The simplified method of analysis was selected as a means to cross check the results obtained from the scantling rule.

The simplified method of analysis is permissible for solid or cored FRP laminates with the same properties along both principal plate axes. This requirement is satisfied by using alternating layers of CSM and WR. The method uses the design pressures and ultimate stresses of the laminate to compute minimum thickness for solid plating, minimum section modulus for cored plating and stiffeners, and minimum web area for stiffeners. Under this method, the ultimate stresses of the laminate may be approximated by treating the plies that make up the laminate as one 'thick' layer, provided the flexural strengths of the individual plies do not vary by more than 25 to 30%.[32], which is achievable for CSM and WR. Formulae for computing an individual ply's mechanical properties are given, and values are pre-computed for different fiber types and fiber mass fractions. The "effective" elastic modulus, E, and shear modulus, G, of the laminate are calculated as averages of the respective moduli of the constituent plies, weighted by the ply thickness. The breaking strains, ϵ_u (or γ_u), in tension, compression, bending and shear of the laminate are set to the limiting values of the constituent plies, which are tabulated in [32] for different combinations of fiber and resin. The moduli and breaking strains are then used to determine the laminate's ultimate stresses in compression, tension, and bending, e.g.

$$\sigma_u = E\epsilon_u \tag{20}$$

$$\tau_u = G\gamma_u \tag{21}$$

These are reduced by a design factor and used with the design pressures to determine the minimum thicknesses and section moduli of the scantlings.

2.4 Yacht Resistance

While speed was not an overriding concern in the design, an attempt was made to incorporate some design factors that minimized resistance encountered by the yacht, in cases where such factors did not conflict with achieving a seaworthy design.

2.4.1 Sources and Design Factors

Beyond the drag induced on appendages, yacht resistance results from viscous effects and wave-making effects. The viscous effects may be further separated into frictional resistance and viscous pressure resistance.[5] The frictional coefficient of resistance may be estimated by the equivalent flat plate resistance by the International Towing Tank Conference (ITTC) 1957 correlation, and depends only on the Reynolds number of the flow.[26].

$$C_F = \frac{0.075}{(log(Re) - 2)^2} \tag{22}$$

$$C_F = \frac{R_F}{0.5\rho A_W V^2} \tag{23}$$

This coefficient is related to the resistance by the wetted area of the hull or appendage, so that minimizing wetted area is advantageous from a speed perspective, particularly given that frictional resistance is a dominant component at speeds of interest to a sailing yacht.[5] However, carrying this to the extreme results in shallow hulls with small, high AR fin keels that exhibit a decrease in seaworthiness, for the reasons previously discussed.

Viscous pressure resistance results from the fluid boundary layer modifying the pressure distribution around the hull, resulting in a higher pressure at the bow than the stern.[5] Flow separation at the stern greatly increases this pressure drop and resulting resistance. This effect depends on the shape of the hull, in particular the bluntness of the stern, with blunter sterns increasing the degree of flow separation. To prevent this, the diagonals of the stern should be kept to maximum slopes of less than about 22 degrees.[5] However, bluff sterns do help decrease wavemaking resistance that becomes dominant at higher speeds, so the optimization of the hull shape for resistance must be conducted for the speed of interest to the vessel.[5]

At higher speeds, the resistance of a ship is dominated by the waves created by the hull's motion. Because the associated wavelength depends on the ship's speed, a sharp rise occurs when the speed of the ship is such that the wavelength is equal to the ship L_{WL}. This occurs at a Froude number, Fn, of 0.4, the so called 'critical Froude number'[26]. Froude number is a dimensionless speed:

$$Fn = \frac{V}{\sqrt{gL}} \tag{24}$$

While optimized racing yachts may be capable of exceeding this barrier, most heavier displacement cruising vessels cannot, so that this forms a maximum 'hull speed' that a vessel can attain given her waterline length. This was used as one criteria in evaluating early design options. The parameter that predicts whether a yacht will be capable of reaching the 'high speed' regime is the length/displacement ratio ($L/\nabla^{1/3}$). Relatively lighter displacement vessels, of sufficiently high $L/\nabla^{1/3}$, will be capable of exceeding the barrier at Fn=0.40. Values of $L/\nabla^{1/3}$ around 5.7 are quoted as a threshold for this capability.[5] However, hulls that are light for their length make a tradeoff in performance and safety in heavy weather conditions.

An extensive series of tests at the Delft University of Technology have been conducted to determine empirical correlations for sailing yacht residuary resistance, which are summarized in [5]. Residuary resistance combines the effects of wave resistance and viscous pressure drop, both of which depend on the hull shape. Because frictional resistance can be easily computed from equation 22, these correlations enable prediction of a yacht's total resistance and form the basis for many Velocity Prediction Program (VPP) used to estimate yacht sailing performance under different wind conditions.[5] The studies reveal important design parameters that influence resistance. In addition to the impact of $L/\nabla^{1/3}$ discussed above, these factors are the prismatic coefficient, C_P, and the location of lateral center of buoyancy (LCB). For optimum performance at lower speeds, LCB should be about 3% of L_{WL} aft of midships. The optimum location moves aft to around 4% of L_{WL} for the highest upwind speeds, and moves aft even more for high speed regime yachts sailing off the wind.[5] The optimum value for C_P when beating upwind in a breeze is around 0.55.[5]

In addition to viscous and wavemaking resistance, sailing yachts experience added resistance from heeling, and added resistance from waves. The former results primarily from changes to the wetted area of the hull as it heels, changes to the hull wave generation, and changes to the keel wave generation, and the effect is captured in the Delft series correlations.[5] The latter refers to the increase in resistance a yacht experiences when she sails in other than calm water, i.e. in an appreciable seastate. The added resistance results from the coupled motions of pitch and heave induced on the yacht by the seas, which causes the generation of additional waves by the hull.[5] Added resistance in waves is sensitive to the pitch moment if inertia, with larger pitch gyradii, k_p corresponding to increased added resistance. An approximation for the gyradius of a sailing yacht is 25% of the L_{WL}[5] However, the gyradius is sensitive to the mass contribution of the rig, due to its distance from the pitch axis. Racing yachts then benefit considerably from a light mast, which reduces the pitch gyradius and added resistance in waves. However, as discussed in the preceding section, this results in less seakindly motions due to the higher accelerations experienced by the yacht of lower inertia.

Another method of reducing added resistance in waves is by the addition of winglets on the rudder or on a keel bulb. Adding high aspect ratio winglets to the rudder alters the damping characteristics of the yacht and results in reduced pitch motion amplitudes, and a corresponding reduction in added resistance in waves.[33]

3 Design Process and Tradespace Reduction

The design process followed an iterative, point-based design approach, with key early decisions established by a Pugh convergence methodology after the design tradespace was reduced by qualitative feasibility and performance considerations. Following these decisions, the principal dimensions and ratios of a starting design were established based on the material presented above. This design was developed in an iterative process, following a 'design spiral' similar to that presented in figure 4.

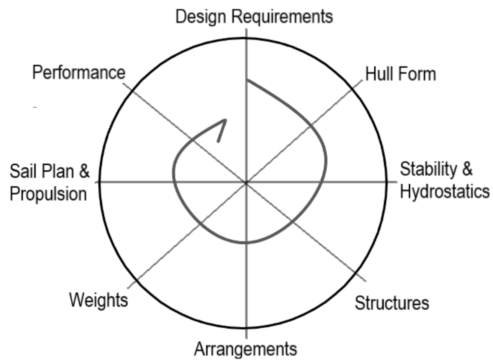


Figure 4: Design Spiral

3.1 Tradespace Reduction

Certain design options were eliminated from consideration at an early stage based on qualitative judgments regarding feasibility and performance.

Multihulls were excluded for two key reasons: characteristic large areas of inverted stability, and complexity in trailer operations. While trailerable multihulls exist, they necessitate a folding or articulating connection between sponsons. This represents an additional potential failure point, and the added complexity of such an arrangement was deemed inconsistent with the design philosophy. Furthermore, while catamarans and trimarans exhibit incredibly high initial stability due to their large waterplane moments of inertia, they are also characterized by a very large region of negative stability. This means that if inverted, they will almost certainly remain so without external assistance. There is always the risk that a breaking wave of sufficient size can capsize a vessel small enough to be put on a trailer, so it was judged prudent to account for this possibility in the design by minimizing the range in which the vessel is stable in the inverted position.

Multi-masted sailplans were not considered, based on the short length and small displacement of the vessel. Because the rigs are located high above the deck, their weight has a large impact on the position of vertical center of gravity, a critical parameter of ultimate stability. Additionally, a multi-masted rig would complicate and lengthen pre-launch rigging procedures for a trailerable yacht, and result in a less simple control schema for a singlehanded vessel.

Deck-stepped mast arrangements were ruled out, based on structural concerns. In these arrangements, the heel of the mast is supported on the deck, rather than a mast step affixed to a floor structure on the bottom of the hull interior. This confers an advantage to trailered boats by simplifying the stepping or unstepping of the mast, and may also result in more useable interior volume. The latter advantage, however, may be reduced by the necessity of locating a compression post under the mast heel to transmit the mast compression load to the internal yacht structures. While successful ocean cruiser designs with deck stepped masts have been demonstrated [4], they entail additional structural risk that was judged unsuitable for this design.

Ferrocement and steel construction materials were deemed infeasible based on weight concerns for a vessel of this displacement range. High modulus exotic laminates, such as carbon fiber, S-2, or Kevlar, were not considered for the hull construction, though they were considered for rig and mast construction. The higher cost of these laminates was deemed potentially justifiable for the mast because weight savings in the mast can have a large impact on stability characteristics. However, the cost of fabricating the hull out of these laminates was deemed prohibitive for a boat that was not designed as an extreme racer.

Centerboards and retractable or lifting keels were not considered feasible for the design. Both have an advantage for trailer launched boats, in that the draft at launch and recovery may be kept very small by raising the board or keel, and the performance advantages of a deeper keel can be gained by lowering it once sailing. However, their disadvantages are believed to outweigh this advantage for an open ocean yacht. Both require devoting a portion of interior volume to the board or keel trunk. A centerboard, being unweighted, is incapable of conferring the stability advantage of a ballast keel. A ballasted lifting or retractable keel adds considerable complexity and additional potential failure points and maintenance burden. In keeping with the design philosophy, these options were excluded from the tradespace.

3.2 Material Selection

Fiberglass, wood, and aluminum were all deemed potentially suitable materials for a vessel design of this scale and purpose. Because fiberglass is a very common material for production small yachts, it was selected as a datum against which to compare the other materials. The comparison criteria were shape control, hull weight, and maintenance load. The comparison is presented in table 9, where a "+" is assigned when a material exceeds datum performance for a given criteria, a "-" is assigned where a material under-performs the datum, and a "0" is assigned if the material is judged equivalent to the datum for that criteria. The results are combined for all criteria, and indicate that fiberglass is the preferred material for the design.

	Materials		
Criteria	Fiberglass	Wood	Aluminum
Shape Control	DATUM	-	-
Hull Weight		+	+
Maintenance Load		-	-
Σ+		1	1
Σ-		2	2
Total	0	-1	-1

Table 9: Material Selection

Shape control refers to the ability to develop curved surfaces in the material, including curvature along both principal axes of a plate. A higher degree of shape control is preferred, as it enables greater freedom in hull form design. The layup procedure of fiberglass laminates permits greater shape control than either wood or aluminum construction.

Lighter hull weights are preferred, since this enables a greater proportion of weight to be allocated to ballast for a yacht whose total weight is constrained by trailer requirements. Hull weight includes both the shell plating and internal structure. In this comparison, the fiberglass laminate was assumed to consist of a solid layup of alternating CSM and WR. The mechanical properties of wood show considerable variation across species, but in general the density of wood is on the order of half the density of such a laminate. The modulus of rupture of wood is roughly comparable to the ultimate flexural strength of the laminate and the Young's moduli are also comparable. Thus, wood may be selected that has a higher specific stiffness and specific strength than fiberglass; this could reasonably be expected to deliver a lighter structural weight. Aluminum is more difficult to compare, as marine grade (5000 series) aluminum alloys exhibit lower specific strengths than the nominal fiberglass laminate, but considerably higher specific stiffness. However, at least for small vessels, roughly similar boats tend to be heavier when constructed from fiberglass.[34]

Regarding maintenance load, hulls of all materials would require periodic bottom painting. For fiberglass boats that are not stored in the water, this requirement is expected to comprise the bulk of the total hull maintenance, indicating a very low maintenance material. Of the three materials, maintenance load is expected to be highest for wooden hulls and structures. This is due to the requirement for sealing the wood and the constant battle against water incursion and rot. Aluminum hulls come next, because while marine grade aluminum alloys exhibit excellent corrosion resistance in seawater, galvanic corrosion is a concern at any junction with a dissimilar metal. Such junctions may occur at through hulls, deck hardware, rudder bearings or shaft penetrations. Even if properly installed to prevent metal-metal contact, these points represent a risk of rapid corrosion that should be inspected periodically to ensure that, e.g. a protective paint layer insulating the contact point has not chipped. For these reasons, the advantage on maintenance load was given to fiberglass.

Relative to both wood and aluminum, the weakness of fiberglass is the higher resultant hull and structural weight. However, this shortcoming may be addressed in part by incorporating cored hull or deck sections into the design, which were not included in the assumptions of the above comparison. By separating two relatively thin fiberglass skins by a low density core material, the plating section modulus can be increased considerably. This can permit the use of less fiberglass, and also result in fewer internal structures due to the higher stiffness of this arrangement.[24]

3.3 Keel

Because of the trailer constraints, various shoal draft keel options were explored. These included a "traditional" full or modified full keel, integral to the hull underbody and extending from the stern to forward of amidships; a long, low aspect-ratio fin keel; twin keels, which consists of two foils offset from centerline; wing keels, which have wings extended laterally from the tip of a foil or from the foil bulb; and Scheel keels, a shoal draft keel whose thickness flares outward from the root of the keel to the tip. Examples of these keel types are shown in figure 5. Images of the fin and full keels are reproduced from [27], the twin keel from [35], the wing keel from [36] and the Scheel keel from [37].

The keel variants were compared across the criteria of sailing performance, damping ability, coursekeeping, ballast carrying ability, and resistance to damage/fouling. The low aspect ratio fin, a typical configuration on cruising vessels [27], was selected as the datum, and qualitative judgments were made on the basis of the material presented in the preceding section and the published model testing results comparing performance of a low AR fin keel, a Scheel keel, and a wing keel.[36] These results indicate that, among those shoal draft keel designs, a wing keel delivers a higher speed made good than a Scheel keel, which itself outperforms a low AR fin keel. Traditional full keels and twin keels were not included in that study, but due to their larger wetted area may be expected to deliver slower speeds in the speed regime of interest to a sailing yacht.

All variants were judged to have better damping characteristics than the datum: the full keel has greater lateral area and a longer keel tip, the twin keel will act similar to bilge keels used for roll damping on larger ships, the wing keel's wings provide additional damping and the Scheel keel has a flared, expanded keel tip. Similarly, ballast carrying capability of all variants were judged better than the datum, in which it was assumed that a long, low aspect ratio shoal fin keel was not fitted with a bulb.

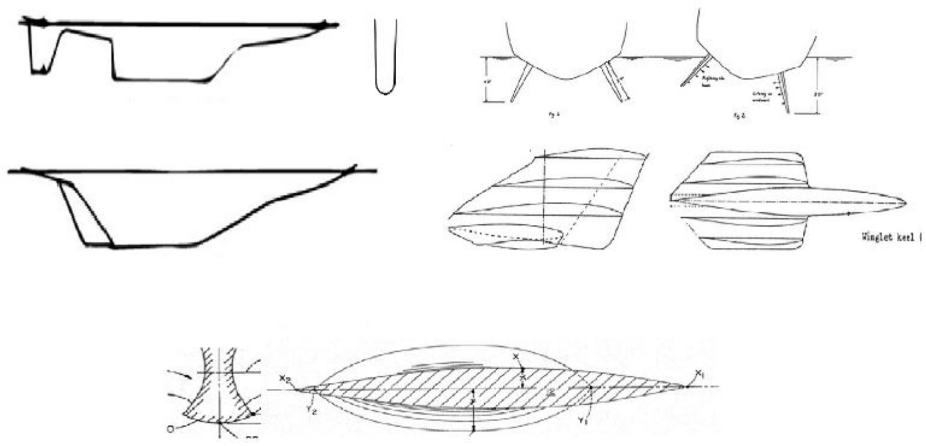


Figure 5: Keel Types

On left: Long Fin keel and modified full keel. On right: Twin Keel and wing keel. Bottom: Scheel keel

Following the discussion presented in the preceding section, coursekeeping ability was assumed to depend predominantly on keel length and aspect ratio, with longer keels and lower aspect ratios providing improved coursekeeping ability.

All variants except wing keels were judged equal in terms of their ability to resist fouling and damage. The nature of the wing keel design makes it more likely to be fouled by lines (crab pots, anchors lines,etc) or seaweed, and potentially at greater risk of damage in a grounding event.

Keel Types					
Criteria	Low AR Fin	Twin	Wing	Full	Scheel
Performance	DATUM	-	+	-	+
Damping		+	+	+	+
Course Keeping		-	-	+	0
Damage/Fouling Resistance		0	-	0	0
Ballasting Capability		+	+	+	+
Σ+		2	3	3	3
Σ-		2	2	1	0
Total	0	0	1	2	3

Table 10: Keel Type Selection

From this comparison, both full keels and Scheel keels emerge as likely candidates for the design. While the Scheel keel scored higher in the analysis, preference was initially given to the full keel. This was due to a judgement, in keeping with the design philosophy, that the improved course keeping ability of the full keel was more significant than the performance advantage of the Scheel keel. However, early in the modeling and hull design process, it became apparent that the full keel actually had a negative impact on stability characteristics of the design. Because it represented a significant submerged volume addition considerably deep below the waterline, it impacted the vertical center of buoyancy in a manner that reduced metacentric height (see equations 1 and 2). Thus initial stability and total righting energy were reduced below acceptable levels. This could, of course, be corrected by increasing the ballast weight to further lower the center of gravity. However, this resulted in the design quickly reaching the weight constraints imposed by the requirements. Because of this, the Scheel keel was selected for the design.

3.4 Sail Plan and Rig

Key design decisions for the rig and sail plan include general sail geometry or type and whether the mast is to be stayed or unstayed (so-called freestanding masts).

Freestanding masts take the sail load in bending, similar to a cantilever, while the more typical stayed mast arrangement relies on fore and aft tensioned stays and athwartships shrouds for structural integrity, behaving as a column. The former requires a mast of higher section modulus, a mast of greater diameter and wall thickness, in order to carry the bending moment from the sails, and the mast partner at the deck must be capable of carrying the reaction load. Stayed masts, however, deliver greater compression loads to the hull at the mast step, and higher loads on the deck where the chainplates connect to the shrouds and stays.

Freestanding masts exhibit several key advantages over stayed rigs, beyond putting less load on the hull. Not least among these, for a trailered vessel, lies in much easier rigging procedures, as there are no stays and no shrouds and attendant spreaders to connect and tune. They are also more reliable and lower maintenance by virtue of their simplicity: there are far fewer failure points, no highly loaded wires, shackles or pins, so reliability increases while maintenance requirements are significantly reduced.[38] Additionally, they can be designed to be more aerodynamically efficient, both by eliminating the drag from stays and shrouds, and by removing the constraints on sail shape and sail position that these elements impose.[39] It appears that the main reason for the prevalence of stayed rigs on small sailing yachts today is that such configurations became embedded in the ratings rules that govern yacht racing, and by extension yacht design trends.[38] For the present design, which is unconstrained by ratings rules, a freestanding rig was selected based on their improved reliability, simplicity, reduced maintenance, lighter hull loads, reduced drag, and simpler rigging process.

Sail types considered were the Bermuda, the junk sail, and the gaff sail. The Bermuda sail is characterized by a roughly triangular planform of characteristically high aspect ratio, and generally strong windward performance.[27] The gaff sail is more rectangular in planform, with a broad head connected to a yard, and typically has a lower aspect ratio and mast height for a given sail area, spreading the sail more fore-and-aft.[27] A junk sail is somewhat similar to a gaff, with the primary differences being that the luff extends slightly forward of the mast, and the sail is divided into panels which are fully battened. While this increases the total weight of the rig, the arrangement also makes reefing a very simple operation.[30] Additionally, the sail fabric is less stressed, meaning lighter fabric may be used, it is more easily repaired, and the sail tends to retain effectiveness even if damaged.[30] Examples of the sail types considered are depicted in figure 6. Bermuda and gaff images from [27], Junk image from [40].

A comparison of the sail types was performed. The Bermuda sail, being an ubiquitous choice for small yachts, was selected as the datum. The sail types were assessed using the criteria of ease of singlehanded operation (ease of reefing and sail handling for a solo sailor), mast height, windward performance, total rig weight, and rig "tune-ability".

Rig tune-ability refers to the ability to adjust the sailplan's center of effort after construction, to correct any helm balance issues that are present. This is a consideration related to design risk, as in the absence of extensive wind tunnel and tow tank testing, it is possible that the empirical relationship used to determine the rig lead may necessitate refinement of the CE position based on trials after construction. It is desirable that this adjustment can be accomplished with minimal effort. Any of the rigs considered could alter the lead by altering the rake of the mast or the longitudinal position of the mast, but either option would necessitate some structural modifications. However, because a portion of junk sail's luff extends forward of the mast, the junk rig center of effort can be adjusted by relocating the connection point between the yard and the masthead.[30] By moving the halyard off the center of the yard, CE can be brought forward or aft without structural modification. For this reason, the tune-ability advantage was given to junk rigs.

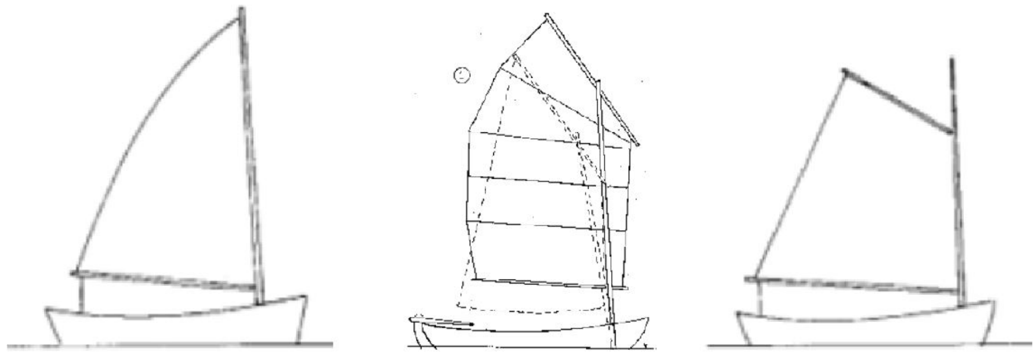


Figure 6: Sail Types

From left to right, Bermuda, Junk, and Gaff rigs

Mast height and total rig weight both impact the vertical center of gravity, with heavier rigs and taller masts negatively impacting stability characteristics. Mast height also affects the heeling moment, with lower heights corresponding to reduced heel angles for a given hull and wind conditions. This is particularly advantageous for yachts designed to be initially tender for seakeeping purposes, and can help prevent excessive heeling in moderate conditions without sacrificing sail area. The yards required for gaff and junk rigs make them heavier than a Bermuda rig of equivalent sail area. However, the bermuda rig's higher aspect ratio necessitates a taller mast for the same sail area.

Gaff and Bermuda rigs were judged equally suitable for singlehanded use. The advantage was given to junk rigs primarily due to the simplified reefing system conferred by the battened sail, which requires only easing the halyard to drop one panel, and avoids the crew having to tie in reef points on the sail.

The Bermuda rig is expected to perform better to windward than either the gaff or junk. This is due to the higher aspect ratio of the former, which delivers higher thrusts when sailing close to the wind.[41]

The results of the comparison are displayed in table 11, and indicate that the junk rig is the preferred concept for the design. This conclusion assumed that the heavier weight of this rig can be implemented without an unacceptable increase in total weight or decrease in stability characteristics, which was verified in weight and stability calculations presented in the following sections.

Sail Types			
Criteria	Bermuda	Junk	Gaff
Singlehanded Use	DATUM	+	0
Mast Height		+	+
Windward Performance		-	-
Rig Weight		-	-
Tune-ability		+	0
$\Sigma+$		3	1
$\Sigma-$		2	2
Total	0	1	-1

Table 11: Sail Plan Selection

4 Hull, Keel, and Topsides Design

Having established the key decisions for the point based design, a starting set of dimensions and design ratios was developed based on the trailer constraints, review of similar vessels, and considerations presented in the review of literature. This starting hull was heavily iterated upon to achieve satisfactory stability and performance characteristics while meeting design requirements. A keel was designed in a similarly iterative fashion, and faired into the hull. The hull, keel and rudder designs were modeled and analyzed using Rhino Computer Aided Design (CAD) software and the Orca3D naval architecture package.

4.1 Hull Form Development

Initial target values for several design ratios are displayed in table 12. It was desired to maintain length-to-beam at around 3 or larger, to balance initial stability and sail carrying capabilities against the greater capsize tendencies and diminished ultimate stability of too wide a beam. The ratio of overall length to waterline length is a measure of the yacht's overhangs, which are intentionally kept short for several reasons. As discussed above, long overhangs can increase added resistance in waves, and may also serve as a lever arm for wave action. Additionally, in a design whose overall length is constrained by trailer dimensions, overhangs reduce the available waterline, which negatively impacts speed and seaworthiness. Longer overhangs are effectively penalized by the STIX computation. Prismatic coefficient was targeted at 0.55 for resistance considerations. A substantial ballast ratio of 0.45 was selected based on the importance of vertical center of gravity to ultimate stability, and a sail area to displacement ratio of 15 was selected to provide adequate powering capability.

Parameter	**Target Value**
L_{OA}/B	3.0
L_{OA}/L_{WL}	1.06
C_P	0.55
Ballast Ratio	0.45
$SA/\nabla^{1/3}$	15.0

Table 12: Design Starting Ratios

The 'hull assistant' feature of Orca3D permitted the rapid evaluation of a number of different canoe hulls. This function creates hulls based on user inputs for principal dimensions and shape parameters, and was useful in the early design process to converge on appropriate principal dimensions. The hydrostatics module returns the relevant hydrostatics and stability results of a hull, including the full curve of stability with trimming effects, based on user inputs on weight centers or flotation plane conditions. It uses an iterative approach to solve for equilibrium at user specified heel angles based on the defined hull surface(s). While the hydrostatics module did not return total righting energy, it could be simply computed by numeric integration of the GZ curve using the trapezoidal rule. A simple MATLAB code was written to calculate the STIX based on inputs from the hydrostatics results. See appendix I. The hydrostatic module and STIX code were used to explore the extent of impact of design choices like flare/tumblehome, freeboard, displacement, draft and length/beam on early canoe hull models. For purposes of analysing and comparing hulls, a center of gravity was assumed at 12 cm below the waterline. Subsequent weight tabulations were later used to refine this. This enabled exploring a large number of hulls, since such effects could be observed without the detailed manual surface shaping necessitated in later stages of design development. This process resulted in important insights that shaped the outcome of the design. Examples of some of the canoe hulls examined are displayed in figure 7.

At the outset of the design, hulls of lengths near the high end of the trailer constraint, 8-9 meters, were explored. Longer waterlines confer greater hull speeds, and, all other things equal, can better

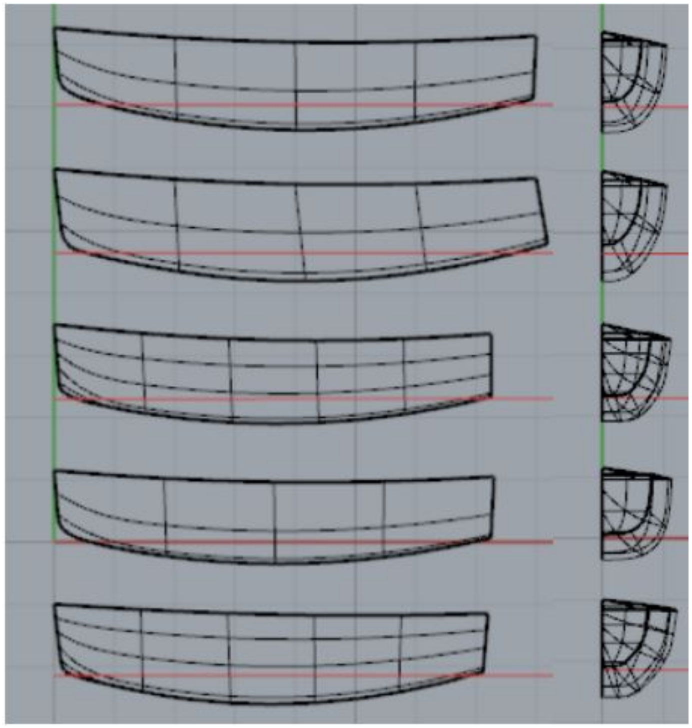


Figure 7: Canoe Hulls

A sample of of the canoe hulls explored. Divisions are in meters.

handle severe weather. However, early design work established that the length of the hull was effectively constrained by structural weight rather than trailer dimensions. Initially, the structural weight requirements were roughly estimated by similar vessels, by subtracting their ballast weight from their displacements, and dividing by their waterline lengths. This gave a 'weight per length' metric that was used to approximate the structural weight requirements of the design in early work, at an average of around 270 kg per meter waterline length. Later design work on the structures and the weight tabulation confirmed this as a reasonable approximation. In order to maintain a reasonable ballast ratio while keeping total displacement within the trailer constraints, overall length had to be reduced to between 7 and 8 meters. The short overhangs resulted in waterline lengths of around 6.5 to 7 meters. On the low end of lengths, waterline lengths below 6 m were not considered because their theoretical hull speeds drop below 6 knots, making it potentially difficult to meet speed requirements.

Initial design iterations also focused on displacements at the lower end of the requirement range, near the objective value of 2265 kg. This was due to a desire to enhance the trailer capability of the vessel. However, this was seen to have unacceptable impacts to stability and seaworthiness characteristics. In particular, it was impossible to meet the minimum righting energy for design category A at displacements less than around 2800 kg, so design iterations quickly moved to displacements closer to the upper end of the range, between 2800-3000 kg. Lower total displacements also resulted in less ballast and therefore a higher vertical center of gravity, which negatively impacted the stability curve. For similar reasons, the initial targeted draft of 1 m was eventually increased by 10% to 1.1 m, in order to keep the center of gravity as low as practical without exceeding design requirements on draft.

Beam between 2.1 and 2.6 m was explored. A parent canoe hull, shown in figure 8, was adjusted in beam only, and the stability results compared. Reducing the beam resulted in greater ranges of positive stability, but lower metacentric heights and peak righting arms. The latter effect was dominant in righting energy, such that lower beams exhibited slightly smaller righting energies, despite their larger angle of vanishing stability. This represents the inherent trade off between form and weight stability inherent, and is presented in figure 9 and table 13.

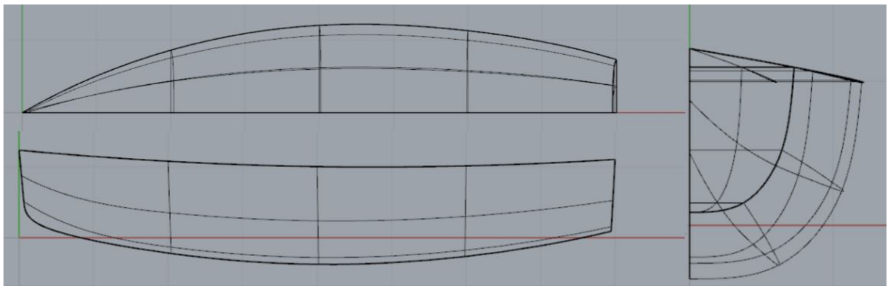


Figure 8: Parent Canoe Hull for Beam Comparison

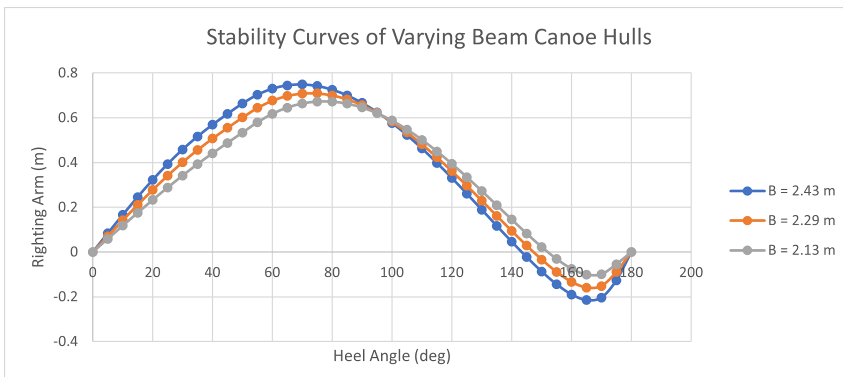


Figure 9: Stability Curves for Varying Beam Canoe Hulls

Beam (m)	GM (m)	ϕ_v (deg)	A_{GZ} (m-deg)
2.43	0.96	143	66.4
2.29	0.82	147	64.1
2.13	0.67	152	61.9

Table 13: Stability Characteristics with Varying Beam

Because both displacement (and thus ballast weight) and draft were constrained, the ability to increase weight stability by lowering the vertical center of gravity was limited; the lowest achievable center of gravity for the design was assumed to be 12 cm below the waterline. Subsequent weight calculations verified that this was a reasonable assumption. Then the minimum beam capable of delivering adequate righting energy and initial stability for sail carrying capability was selected. This was determined to be a beam of 2.43 m.

Freeboard was explored in a similar fashion, and also represented a trade-off. Higher freeboard conferred greater ranges of positive stability and greater righting energies, if the vertical center of gravity could be held fixed. However, for a fixed displacement, raising the freeboard will raise the center of gravity. Additionally, excessively high freeboard penalizes performance by increasing windage of the hull. Similar to beam, the lowest freeboard that resulted in adequate righting energy was selected. This was an average freeboard of 1.08 m.

Other considerations in developing the canoe hull form included LCB location, hull balance, and entry half-angle. For resistance purposes, LCB was maintained approximately 3% aft of midships, and the entry was kept relatively fine, with a half-angle of 25 degrees or less. Hull balance was gauged by comparing the trim angle variation throughout the range of heel angles assessed in the hydrostatics calculations. Relatively small changes in trim throughout this range indicate a well balanced hull, that should avoid developing large yawing moments while rolling.

The principal dimensions of a hull form were converged on for further development. This was an iterative process, as modeling of the cabin roof, cockpit, keel, and detailed hull shape all impacted the stability characteristics of a given baseline canoe hull. The selected baseline hull is depicted in figure 10. The dimensions and stability characteristics of the baseline hull form eventually selected for the design are presented in table 14 and figure 11.

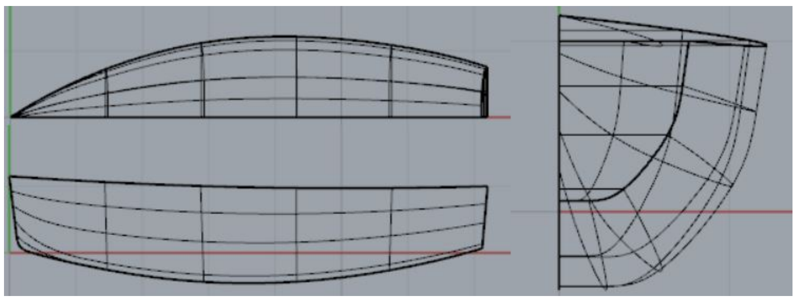


Figure 10: Baseline Hull Model

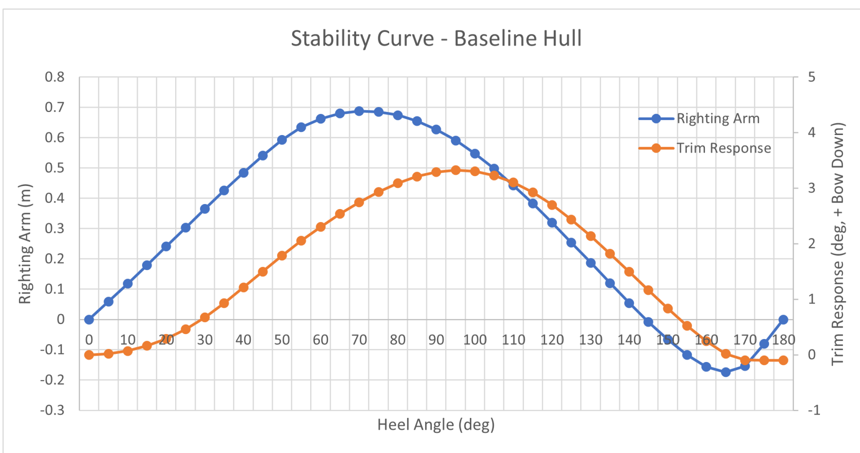


Figure 11: Design Baseline Hull Stability and Trim Response Curves

Parameter	Value
Δ (DWL)	2440 kg
L_{OA}	7.20 m
L_{WL}	6.56 m
B_{OA}	2.44 m
B_{WL}	1.84 m
T_c	0.46 m
FB_{FWD}	1.15 m
FB_{AFT}	1.0 m

Table 14: Design Baseline Hull Particulars

4.2 Keel Design

A Scheel keel was selected for the design, based on the dimensions and section geometries of the original patent.[37] A diagram of this keel type is displayed in figure 12, and the dimensions are described in table 15.

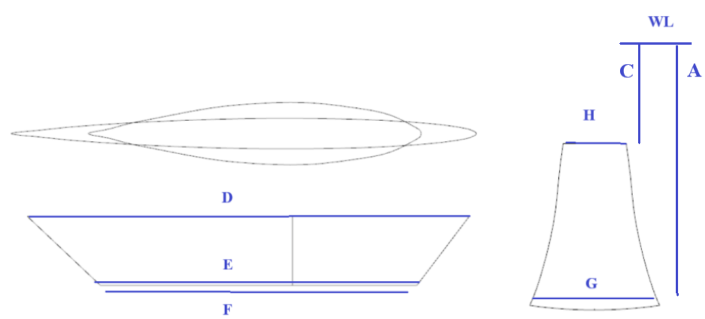


Figure 12: Scheel Keel Geometry

The Scheel keel design specifies a minimum, maximum, and optimum value for each dimension, as a function of yacht B_{WL} and L_{WL}. The thickness distributions of the foil cross sections at the minimum and maximum widths are specified by the patent. Beyond the optimum values specified by the Scheel design, several other considerations guided selection of the dimensions.

Ballast carrying capability was an important consideration in selecting values for the keel depth, chord lengths and maximum and minimum section thicknesses, as the internal volume of the keel had to be sufficient to carry the required amount of ballast. However, excessively large keel volume reduces the metacenter of the vessel, negatively impacting metacentric height and righting energy. Additionally, keel depth influenced the effectiveness of a fixed amount of ballast, but was also constrained by the trailer requirements on the vessel. Finally, as previously discussed, a long, low aspect ratio keel is beneficial from a seaworthiness perspective, for both roll damping and course keeping considerations. For this reason, keels with root chords of longer than 40% of the L_{WL} were considered.

In order to generate CAD models of the Scheel keel, a MATLAB script was written to generate a point cloud corresponding to the keel geometry, based on the foil section geometry specified in

Label	Description
A	Vertical distance from waterline to section of maximum thickness
B	Vertical distance from section of maximum thickness to keel bottom
C	Vertical distance from waterline to section of minimum thickness
D	Chord length at section of minimum thickness
E	Chord length at section of maximum thickness
F	Length of keel bottom
G	Maximum width of section of maximum thickness
H	Maximum Width of section of minimum thickness
R	Corner Radius where keel bottom connects to keel sides

Table 15: Scheel Keel Dimensions

[37] and a user provided input file specifying the dimensions described in table 15. The script is included in appendix J. After several iterations, the dimensions presented in table 16 were selected for the design. They are presented along with the maximum, minimum, and optimum values computed for a vessel of 6.8 m L_{WL} and 2.0 m B_{WL}.

Design Scheel Keel Dimensions, L_{WL}= 6.8 m, B_{WL}=2.0 m				
Dimension	**Minimum Value**	**Maximum Value**	**Optimum Value**	**Selected Value**
A	0.78	1.29	0.97	1.05
B	0.6 cm	1.6 cm	1.0 cm	1.1 cm
C	0.48	1.02	0.67	0.6
D	0.65	5.77	1.70	2.90
E	0.51	4.41	1.22	2.09
F	0.48	4.08	1.16	1.95
G	0.28	0.57	0.39	0.35
H	0.17	0.33	0.18	0.18
R	0.5 cm	4.5 cm	1.2 cm	2.5 cm

Table 16: Design Scheel Keel Dimensions

Using the above selected values, the Scheel keel code was used to generate a coordinates file for use in modeling the keel. The Scheel keel design cross section geometries result in a sharp trailing edge. Because the the keel is to be fabricated from fiberglass laminate, rather than cast in metal, the sharpness of the trailing edge was reduced to give 2 cm of thickness, judged adequate for the necessary laminate layup. This reduced the root chord by 7.6 cm, and the tip chord by 11.1 cm.

The final keel design is shown in figures 13 and 14.

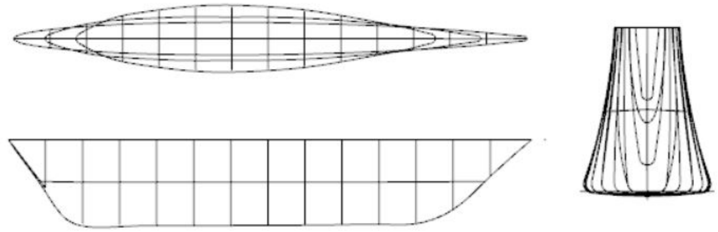


Figure 13: Design Scheel Keel Lines

Figure 14: Design Scheel Keel Renders

4.3 Hull Shaping

The baseline hull formed a starting point for developing the design. It was necessary to further modify the hull to achieve some of the seaworthiness-enhancing design attributes enumerated previously. This was accomplished through direct positioning of the control points that defined the Non-Uniform Rational B-spline Surface (NURBS). This permitted a much greater degree of control over the shape of the hull, and was employed to achieve the desired V-shaped sections forward, transitioning to wineglass sections of slack bilges where the keel was faired in. It also permitted designing in moderate flare forward, transitioning to moderate tumblehome aft. Finally, the keel, which was designed and modelled as a separate set of surfaces, was blended into the hull. The hull was then checked and adjusted for fairness using the built in CAD curvature analysis tools.

The longitudinal positioning of the keel was driven by two factors, the impact to Longitudinal Center of Gravity (LCG) and the impact to helm balance, by affecting the lead of the sail plan. Hull design occurred as part of an iterative sequence that also involved sail plan design and weights estimation. Because the keel contains weight comprising 40-45% of the total vessel displacement, its position has a significant impact on the trim condition of the vessel. The goal was for the LCG to be co-located with the Lateral Center of Flotation (LCF), so that the design condition exhibits zero trim. However, the keel's position is also crucial in determining helm balance. Following the design rules of reference [30], a lead of 9% of L_{WL} was targeted, where the lead was measured as the longitudinal distance between the geometric center of the sail plan and the geometric center of the yacht underwater profile, including rudder. Initial estimates regarding un-ballasted weight distribution, sail plan location, and rudder size and position were refined by iterating until convergence. The requirements on sail plan lead and LCG position were simultaneously satisfied by adjusting the position of the sail plan and, to a lesser extent, adjusting the weight distribution of the yacht.

In addition to developing the desired section shapes and depths and blending the keel into the hull at the correct location, the process of shaping the hull was guided by several other considerations. These included achieving a design displacement close to the calculated weight of the yacht, enforcing initial and ultimate stability requirements, maintaining LCB near the optimum point and a reasonable entry half-angle, and balancing LCF against predicted structural weight distribution. The results of this process are depicted in figures 16 and 15, and the shaped hull particulars are present in table 17.

Parameter	**Value**
Δ(DWL)	3012.5 kg
L_{OA}	7.20 m
L_{WL}	6.83 m
B_{OA}	2.44 m
B_{WL}	2.11 m
T	1.11 m
LCB_{FP}	3.77 m
GM_T	0.73 m

Table 17: Shaped Hull Particulars

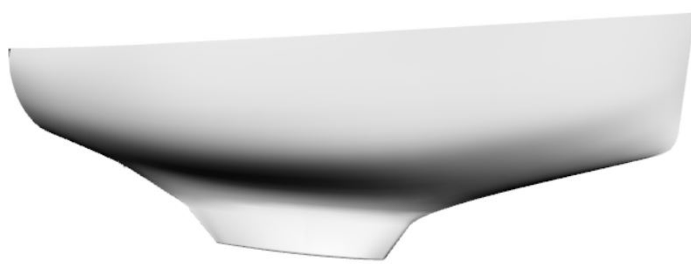


Figure 15: Shaped Hull Rendering

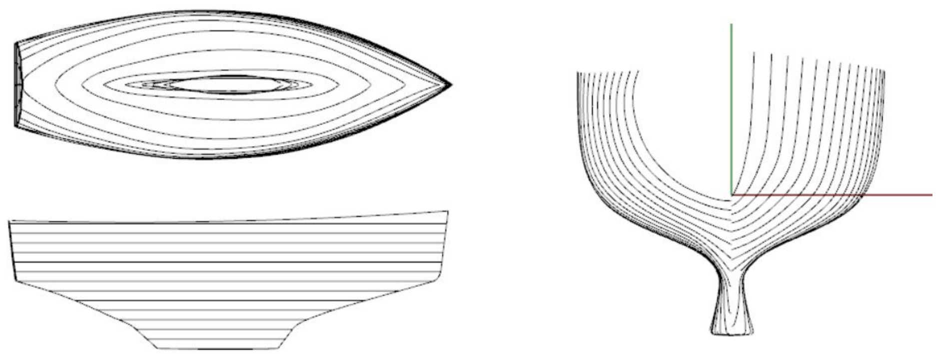


Figure 16: Shaped Hull Lines

Prior to joining the keel to the hull, the hull surface was checked and adjusted for fairness, using the Gaussian and mean curvature analyzer in the CAD suite. Figure 17 displays the resultant curvature heat maps for the hull surface alone. In order to blend the hull and the keel smoothly, the hull surface was trimmed slightly above the keel root section, and the keel surface was trimmed in a flat plane 11.1 cm down from the root. The hull station lines were adjusted to blend fairly into the keel section curves, and these curves were used to generate an intermediate surface to join the hull and keel surfaces, as depicted in figure 18.

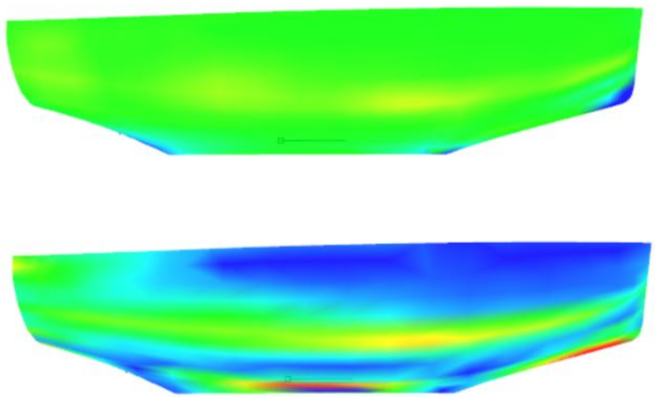


Figure 17: Curvature Analysis

Top image: Gaussian curvature, Red= 1.0, Blue= -1.0. Bottom image: Mean Curvature, Red = 1.5, Blue = 0. Model units in meters.

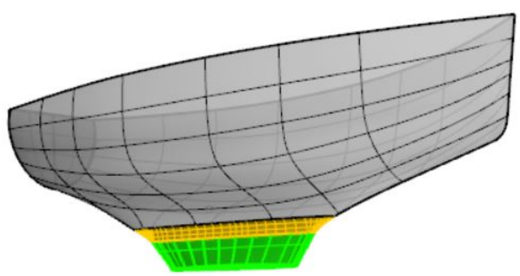


Figure 18: Joining Keel to Hull

This was not perhaps the best way to model the connection, as it resulted in some localized surface irregularities at the interface, as depicted in figure 19. However, this localized unfairness is consider an artifact of the manner in which the intermediate CAD surface was generated. It is believed that the degree of hull fairness is adequate for construction from the offsets or lines plan. Any changes to the modelled geometry at the keel tuck, if they should arise in the lofting process, are expected to be minor and cause no impact to hydrostatics, stability or performance predictions. This prediction was validated somewhat by the construction of 3D printed models from a surface mesh of the hull.

As a final, tactile check on the hull shape and fairness, an approximately 1/45 scale model of the hull was created using a 3D printer. The Rhino surface defining the hull was trimmed to a flat plane just below the sheerline, for ease of printing. The surface was exported as a stereolithography (.STL) file, meshed to a high degree of accuracy, scaled to fit on the print bed and printed on an

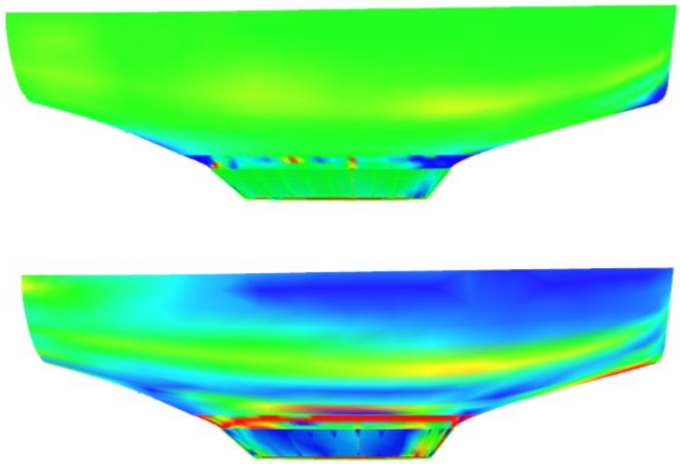


Figure 19: Curvature Analysis with Keel

Top image: Gaussian curvature, Red= 1.0, Blue= -1.0. Bottom image: Mean Curvature, Red = 1.5, Blue = 0. Model units in meters.

Ender-3 Pro Fused Deposition Modeling (FDM) machine using Polylactic Acid plastic (PLA). Pictures of the printed model are presented in figures 20, 21, and 22.

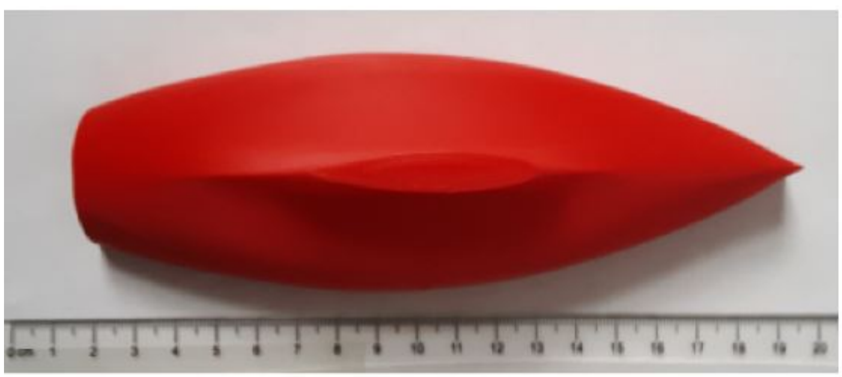


Figure 20: 3D Printed Scale Model of Hull: Top View

Figure 21: 3D Printed Scale Model of Hull: Side View

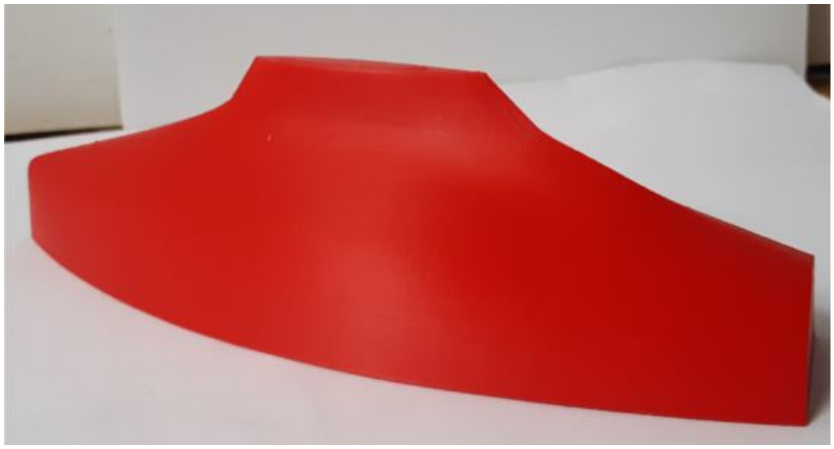


Figure 22: 3D Printed Scale Model of Hull: Quarter View

4.4 Topsides and Cockpit

The topsides of the yacht, including cockpit, cabin roof, sidedecks and foredeck were modeled from a single surface by direct positioning of the control points. Considerations for the cockpit design included ergonomic factors as well as minimizing risk of swamping or downflooding from taking a wave over the transom. The cabin roof volume impacts habitability belowdecks, but also influences the stability curve at extreme angles of heel in a manner similar to freeboard. And, as with freeboard, excessive cabin height raises the center of gravity and represents a penalty in air resistance. The side and foredecks were sized to provide adequate space for the crew to transit and handle lines and ground tackle. The foredeck is also sized to accommodate deck access to an anchor locker in the bow. The topsides arrangement and cabin geometry are displayed in figures 23 and 24.

Figure 23: Topsides Arrangement

The height of the cabin roof gives slightly more than 6 feet of standing headroom in the main cabin. Its shape also provides a significant improvement to the range of positive stability and the righting energy, as displayed in table 18 and figure 25, which compare the stability characteristics of the baseline canoe hull model, the shaped model with a flat deck, and the shaped model with cabin roof. The latter two models include the keel. For all cases, the estimated VCG is 12 cm below the DWL.

The cockpit benches were sized to comfortably seat two crew on either bench. They are 185 cm long, and narrow from 40 cm wide forward to 32 cm wide at the transom. The benches are

Figure 24: Yacht Profile with Cabin Roof

Model	ϕ_v **(deg)**	A_{GZ} **(m-deg)**
Baseline Canoe	144	59.81
Shaped, Flat Deck	143	53.26
Shaped, Cabin	167	62.71

Table 18: Stability Characteristics of Hull Design Progression

40 cm above the footwell, which is 42 cm wide at the narrowest point. Beyond ergonomics, the cockpit was designed with seaworthiness considerations as it represents a potential vulnerability for swamping or downflooding in heavy weather. In particular, the cockpit volume relative to yacht's reserve buoyancy, the height of the cockpit sole above waterline, and the height of the bridgedeck above the cockpit sole are important parameters. The bridgedeck protects the companionway from water ingress in the event of cockpit swamping. In the present design, the bridgedeck is at the height of the cockpit benches. ISO 12217-2 and 11812 provide requirements on bridgedeck height, H_S and cockpit bottom height above waterline, H_B for Category A designs. The latter also provides required cockpit drain size for Category A, based on the cockpit volume coefficient, k_c, which relates cockpit volume to reserve buoyancy:

$$k_c = \frac{V_c}{L * B * F_M} \tag{25}$$

where F_M is the midships freeboard. The cockpit drains are sized such that a full cockpit drains in less time than some maximum time, t_{max}, which depends on k_c and design category.

Table 19 presents ISO cockpit requirements for Category A classification alongside the design values in an assumed fully loaded condition of 3500 kg. In determining drain time and required drain diameter, a conservatively approximate method was employed using the tabulated reference times in [31] and assuming two drains with two elbows whose outlets were submerged below the waterline.

Parameter	**Category A Requirement**	**Design Value**
H_B (m)	≥ 0.15	0.187
H_S (m)	≥ 0.3	0.4
k_c	-	0.105
Drain Time (min)	≤ 2.86	2.6
Drain Diameter (mm)	≥ 65	65

Table 19: Cockpit Design Parameters and Requirements

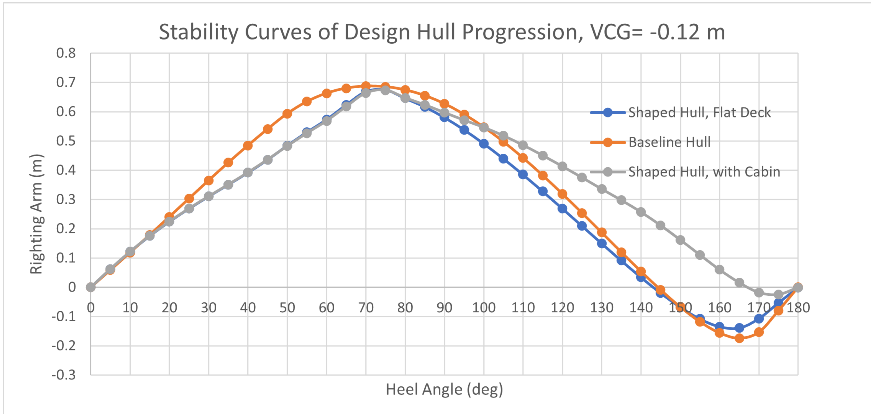


Figure 25: Stability Characteristics of Hull Design Progression

4.5 Hull and Deck Structures

4.5.1 Scantling Rule Application

The scantling rule detailed in reference [24] was used to determine the shell plating skin thicknesses and internal structural arrangements. A spreadsheet calculator was created to enable quick scantling determination and weight estimation upon design iterations. Cored fiberglass construction was selected for the deck and the portion of the hull above the waterline, while solid laminate was selected below the waterline. Cored construction enables stiffer hull structures with less laminate material, and eliminated the need for longitudinal stringers along the deck or hull topsides. The scantling rule applied is based on the scantling number in equation 19, which is used to compute the required thicknesses of laminate and/or core for different regions of the hull. The dimensions used to calculate Sn corresponded to an assumed fully loaded displacement condition as detailed in table 20. Laminate properties assumed by the rule are presented in the same table. The regions are displayed in figure 26, and the computed thicknesses in table 21.

Scantling Number	
Δ_{FL}	3500 kg
L_{OA}	7.20 m
L_{WL}	6.92 m
B_{OA}	2.44 m
B_{WL}	2.17 m
D_M	1.59 m
S_N	0.916

Scantling Rule Properties	
Materials	E-glass/Vinylester Resin
Layup	Alternating CSM/WR
Fiber Mass Fraction	0.35
Laminate Density	1,538 kg/m^3
Core Density	88 kg/m^3

Table 20: Scantling Number and Scantling Rule Properties

Skin Thickness (mm)				**Area Density (kg/m^2)**	
Region	**Inner Skin**	**Core**	**Outer Skin**	**Fiber Weight**	**Total Weight**
Internal Ballast	-	-	12.4	6.67	19.07
Keel	-	-	11.2	6.08	17.34
Hull Bottom	-	-	7.5	4.04	11.55
Hull Topsides	2.5	14.4	3.0	2.96	9.72
Deck	2.5	21.6	3.0	2.96	10.46

Table 21: Laminate and Core Thicknesses and Weights

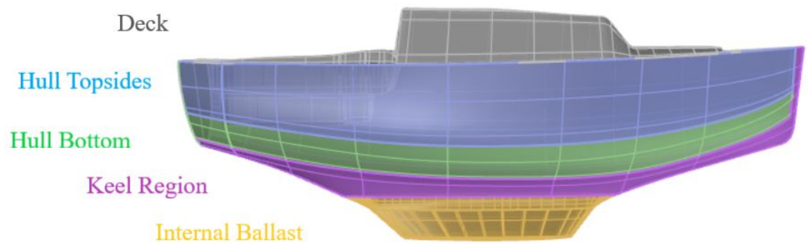


Figure 26: Structural Regions

All laminate and core thicknesses computed by the basic scantling rule were increased by 5% as a 'workboat' allowance for heavy duty usage. The cored sections were limited to 2.5 mm for inner skin thickness and 3 mm for outer skin; that is, the selected thicknesses are somewhat higher than those calculated by the basic scantling rule. This is because impact resistance becomes a practical limiting concern in cored sections on small yachts, and is a reason that weight savings from cored panels tend to be reduced for smaller yachts. However, the advantage of employing cored sections is that the increased stiffness in these regions can eliminate the need for stringers.[5],[24]

The internal ballast region exhibits additional laminate thickness, required by the scantling rule where ballast is carried encapsulated in the hull shell, rather than external ballast that is connected to the internal floor structures with keel bolts. Using bolted on external ballast has the advantage of lowering the ballast weight by the thickness of the shell plating, which can reduce the total ballast required to achieve the desired stability characteristics. However, the keel bolts themselves represent an additional potential failure point, corrosion risk, and maintenance burden. They also may shear in a grounding. Loss of a ballast keel from such an event would be catastrophic. In keeping with the design philosophy, encapsulated internal ballast was chosen for the design.

The weight per square meter of each region was calculated using the computed thicknesses of laminate and core in each region, and the laminate and core densities. These weights were used as inputs to compute total hull and deck shell structural weight and center of gravity. For the deck region, the baseline density was altered by assuming that 10% of the deck surface area was solid (uncored) laminate and an additional 5% of the deck area was reinforced with either plywood core or additional laminate for the purpose of mounting high load deck hardware, consistent with the scantling rule's requirements for such fittings.

The deck is joined to the hull by through-bolting it to an in-turned flange extending from the sheer of the hull around the entire perimeter. The entire joint is bedded in marine sealant. The scantling rule was used to determine the flange width, which is the same thickness as the calculated un-cored hull topsides thickness. The rule also determined the bolt diameter, and bolt spacing. The hull-deck joint scantlings are summarized in table 22.

The scantling rule also determined the arrangement and dimensions of the internal structural elements. These include engine bed stringers, hull stringers, floors, and bulkheads and ring frames. Figure 27 and tables 23 and 24 present the arrangement of the internal structures.

The scantling calculations resulted in a minimum requirement of 5 bulkheads and/or transverse ring frames. A sixth was added to account for usage in heavy weather. The minimum calculated

Deck Joint Scantlings	
Flange Width	25 mm
Flange Thickness	5.5 mm
Bolt Diameter	6.0 mm
Bolt Spacing	140 mm
Total Weight	5.66 kg

Table 22: Hull to Deck Joint Scantlings

Bulkhead Arrangements		
Bulkhead	**Location (m, from bow)**	**Type**
1	0.75	Full Blkhd
2	1.56	Web Frame
3	2.56	Full Blkhd
4	3.76	Partial Blkhd + Frame
5	4.94	Partial Blkhd + Frame
6	6.59	Full Blkd
Floor Arrangements		
Floor	**Location (m, from bow)**	**Core**
1	1.56	Wood
2	1.76	Wood
3	1.96	Wood
4	2.56	CC Foam
5	2.96	CC Foam
6	3.36	CC Foam
7	3.76	CC Foam
8	4.16	CC Foam
9	4.56	CC Foam
10	4.94	CC Foam

Table 23: Strucutral Arrangement: Floors and Bulkheads

bulkhead thickness for plywood construction was 12 mm. Where transverse frames are used in place of full bulkheads or to supplement partial bulkheads, they are top hat stiffeners with the basic laminate layup over a low density closed cell foam core. Their dimensions are given in table 25.

Three pairs of longitudinal stiffeners run between the aft most and forward most bulkheads, with roughly equal spacing athwarthships. All are top hat stiffeners of laminate over foam cores. The pair closest to the centerline are engine bed stringers of greater section modulus than the outboard stringers to withstand the loads imposed by the engine, which mounts to them via steel angles. At the location where the engine mount bolts to these stringers, their core is Douglas fir. Stringer dimensions are presented in table 25.

The scantlings calculation set a minimum of three reinforced floors at the location of the mast step. These floors have a core of Douglas fir sheathed in laminate. The remaining floors are spaced no greater than 40 cm apart along the full length of the keel tuck. Because the ballast is encapsulated internally, these floors may be foam cored to save weight. Dimensions of the floors are presented in table 26.

For weights calculations, the bulkheads and floors were modelled as surfaces and assigned an area density calculated from their dimensions and material properties. A 1 mm thick epoxy resin coat was included in bulkhead weight. The stiffeners were modelled as curves and assigned a linear

Longitudinal Stiffener Arrangement				
Stiffener	**Transverse Distance from CL (m)**			**Type**
	At BLKHD 1	At BLKHD 4	At BLKHD 6	
1,2	±0.187	±0.299	±0.200	Engine Bed
3,4	±0.315	±0.720	±0.426	Hull
5,6	±0.373	±1.091	±0.584	Hull

Table 24: Longitudinal Stiffener Arrangement

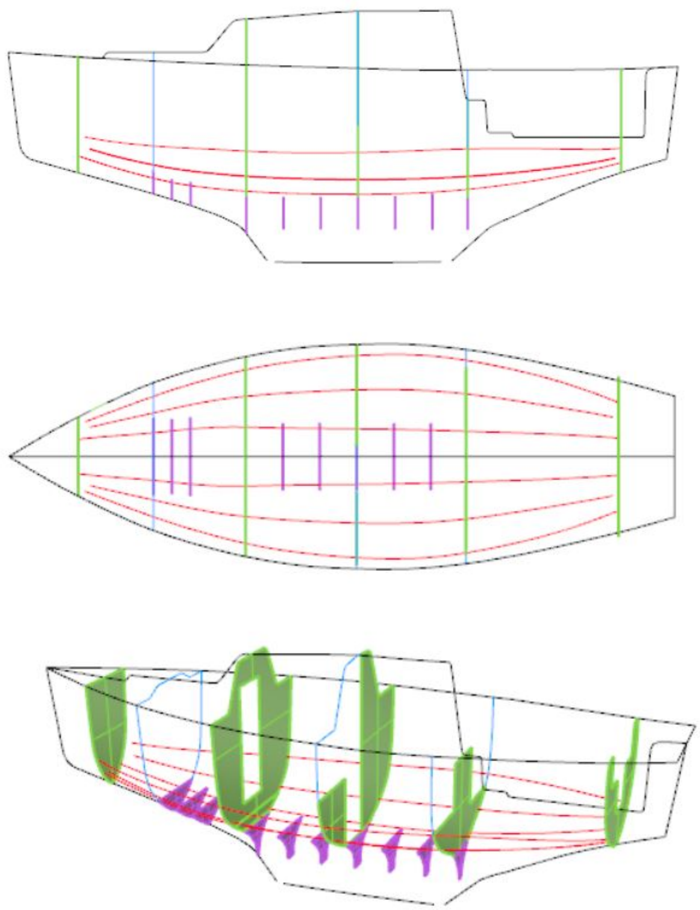


Figure 27: Internal Structural Arrangements
Bulkheads are displayed in green, transverse frames are in light blue, longitudinal stiffeners are in red, and floors are in purple.

density. The additional laminate tabbing required to bond the structural elements to the hull was calculated and included for all elements. The resulting structural weight and center of gravity is presented in table 27.

Stiffener	Core Height (mm)	Core Width (mm)	Laminate (mm)
Transverse Frame	80	80	4.7
Engine Bed	80	80	4.7
Engine Bed [1]	120	80	6.6
Hull Longitudinal	41	81	4.4

[1] Near the engine mount, the dimensions of the engine bed stringer are greater to accommodate the loads imposed by the engine. The upper third of the foam core is replaced with Douglas fir.

Table 25: Stiffener Dimensions

Floor Dimensions	
Core Thickness	80 mm
Laminate Thickness	4.7 mm
Minimum Height	24 cm
Minimum Span	73 cm

Table 26: Floor Dimensions

Element	Mass (kg)	LCG	TCG	VCG
Deck and Cabin Roof	202.60	4.309	0.000	0.992
Hull Shell	401.02	3.789	0.000	-0.040
Deck-Hull Flanged Joint	5.66	3.95	0.00	1.01
Floors	49.78	2.818	0.000	-0.416
Bulkheads	88.03	3.595	0.078	0.404
Transverse Frames	24.97	2.921	-0.179	0.762
Longitudinal Stiffeners	100.65	3.980	0.000	-0.135
Total	**872.71**	**3.833**	**0.003**	**0.242**

[1] LCG, TCG, and VCG are in meters from the bow, centerline, and the design waterline respectively.

Table 27: Structural Weights

4.5.2 Validation Cross Check

As a check on the results of the scantling rule calculation, structural elements at several locations were analyzed using the simplified method of ISO 12215-5. This method computes local design pressures used to determine minimum required section modulus of stiffeners and cored panels, minimum thickness of solid panels, and minimum bulkhead thickness. It also provides a means to check core compressive and shear strength, stiffener shear strength, and stiffener local buckling behavior. The results of these calculations indicate that the scantling rule delivered a generally conservative structure; all elements checked met the requirements for design category A, and many of them provided significantly greater strength and/or stiffness than required. While a full verification of all structures to ISO requirements was not conducted, this result provided some measure of validation to the choice of scantling rule.

The ISO calculations require as input material properties of the laminate in use, which are calculated based on the properties of the individual plies and their proportions in the bulk laminate. For the check, the laminate schedule was assumed to consist of repeating layers of 2 plies of 300 g/m^2 CSM and 1 ply of 800 Rovimat, which is a combined WR and CSM layer of total weight 800 g/m^2. This schedule results in an alternating CSM/WR layup with a fiber mass fraction very similar to that assumed by the scantling rule. The material properties for this laminate are pre-computed in the ISO and displayed in table 28.[32]

2x300 Mat + 800 Rovimat Properties		
Fiber Mass Fraction	ψ	0.363
Laminate Thickness/Fiber Weight	t/w	1.85
Young's Modulus	E	10396
Shear Modulus	G	2682
Ultimate Flexural Stress	$\sigma_{f,u}$	191
Ultimate Tensile Stress	$\sigma_{t,u}$	140
Ultimate Compressive Stress	$\sigma_{c,u}$	146
Ultimate Shear Stress	τ_u	46

[1] All stress and moduli units are in N/mm^2. t/w is the inverse of the dry fiber weight area density per millimeter of total laminate thickness, and is given in $\frac{mm}{kg/m^2}$.

Table 28: Material Properties of Assumed Laminate Schedule

Design stresses are computed as half the ultimate stresses, times a build quality factor, K_{BB}, and an analysis method factor, K_{AM}:

$$\sigma_{i,d} = 0.5 * K_{BB} * K_{AM} * \sigma_{i,u} \tag{26}$$

The build quality factor reflects departures from predicted material properties due to production methodology. For this analysis, a value of $K_{BB} = 1$ was assumed, which requires the laminate to be tested in the as built condition to verify mechanical properties. The analysis method factor accounts for simplifications used in the selected method to analyze the structures; for this analysis, $K_{AM} = 0.9$. Design stresses are displayed in table 29.

The ISO standard provides base pressures for the hull bottom and the deck, that are functions of the yacht mass in the fully loaded condition:

$$P_{BS}^{B} = (2 * M_{FL}^{0.33} + 18) * K_{SLS} \tag{27}$$

$$P_{DS}^{B} = 0.5 * M_{FL}^{0.33} + 12 \tag{28}$$

The slamming loads factor, $K_{SLS} = 1$ for the present design. Base design pressures are given in table 29.

Design Stresses and Pressures	
$\sigma_{f,d}$	86.0 N/mm^2
$\sigma_{t,d}$	63.2 N/mm^2
$\sigma_{c,d}$	65.7 N/mm^2
τ_d	20.7 N/mm^2
P_{BS}^{B}	47.6 kN/m^2
P_{DS}^{B}	19.4 kN/m^2

Table 29: Laminate Design Stresses and Structure Base Design Pressures

The base pressures are used together with panel or stiffener geometry and location to calculate the design pressures at each panel and stiffener. The base pressure is reduced by an area correction factor, K_{AR}, dependant on plate dimensions or stiffener spacing and length. It is also reduced by a longitudinal pressure factor, K_L, based on the element's location aft of the forward perpendicular. This accounts for the reduction in pressure loads experienced further from the bow. Finally, the design pressures are reduced by a design category factor, K_{DC}. For this analysis, the design category A factor of $K_{DC} = 1$ was used. For panels or stiffeners located below the waterline, the ISO design pressure is:

$$P_{BS} = P_{BS}^{B} * K_{AR} * K_{DC} * K_{L} \tag{29}$$

For panels or stiffeners located above the waterline, the pressure is reduced in relation to the mid-panel height above waterline, Z_Q, relative to the deck height, Z_{SDT}, to account for the greater pressure loads on the hull bottom.

$$P_{SS} = \left[P_{BS}^{B} - (P_{BS}^{B} - P_{DS}^{B}) * \left(\frac{Z_Q}{Z_{SDT}}\right)\right] * K_{AR} * K_{L} * K_{DC} \tag{30}$$

where $\frac{Z_Q}{Z_{SDT}}$ is never taken greater than 1.

In computing design pressures, the ISO standard also prescribes minimums for panels and stiffeners. If the design pressure as calculated from equations 30 and 29 is less than the minimum, the latter value is used instead. The appropriate design pressure, design stress, and element dimensions are used in calculating the minimum thickness of solid laminate panels, the minimum section modulus of stiffeners, and the design bending moment of cored panels, which is in turn used to compute the minimum section modulus of the cored panels. It is also used to calculate the minimum cored panel thickness and stiffener web area for adequate shear strength. These computations were conducted for a sampling of solid panels, cored panels, and longitudinal stiffeners at different locations. The key results are displayed in tables 30, 31, and 32.

Solid Panel ISO Laminate Requirements		
Panel	**Minimum Thickness (mm)**	**Fiber Area Density** (kg/m^2)
1B	3.14	1.70
1C	2.90	1.57
2B	4.41	2.38
2C	5.65	3.05
3B	4.97	2.69
3C	5.71	3.09
4B	4.74	2.57
5B	4.03	2.18

A panel is identified by the number of the forward most bulkhead and a letter corresponding to the top most longitudinal stiffener bounding them, with A representing the deck edge, B representing the most outboard stiffener, etc. I.e 2B is the panel bounded horizontally by bulkheads 2 and 3, and vertically by the two outboard hull stringers.

Table 30: Solid Panel ISO Laminate Requirements

The dry fiber mass resulting from the scantling rule calculations in the solid laminate panels varied from 4.04 to 6.08 kg/m^2 in the regions analyzed by the ISO calculations, more than 30% larger than the highest ISO required fiber weight in table 30.

Because the ISO keel structures computations pertain primarily to bolted keels, the laminate thickness where internal ballast is carried was checked with the Germanischer Lloyd classification standard for yachts.[42] This standard assumes a laminate composition of alternating CSM and WR with a total fiber weight fraction of 0.3, which is similar to the layup assumed by the scantling rule. The GL standard computes a keel and tuck fiber weight, G_{wk}, based on the average of the vessel hull length and waterline length:

$$G_{wk} = 1.7 * (350 + 5L)\sqrt{3.3L + 66.5} \tag{31}$$

which for the present design results in a dry fiber weight of 6.19 kg/m^2. This is below the weight of 6.67 kg/m^2 determined by the scantling rule for the internal ballast region, and comparable to the 6.08 kg/m^2 determined for the keel region.

Cored Panel ISO Requirement		
Panel	**Section Modulus** (cm^3/cm)	**Panel Thickness t_s** **(mm)**
1A	0.084	14.89
2A	0.102	15.99
3A	0.155	16.68
4A	0.092	14.17
5A	0.073	9.54

Section modulus is calculated per cm width of the panel.
Values given are required minimums.

Table 31: Cored Panel ISO Requirements

The cored panel thickness, t_s is the core thickness t_c plus the average of the inner and outer skin thicknesses:

$$t_s = t_c + \frac{t_i + t_o}{2} \tag{32}$$

The minimum required values for t_s arise from ensuring the cored panels exhibit adequate shear strength. In order to compute them, the ultimate core shear stress of a commercially available low density closed cell foam, was used in the analysis.[43] From the dimensions computed with the scantling rule, the core thickness $t_s = 17.15$ mm, greater than all ISO minimum thicknesses presented in table 31. Converting this to an equivalent thickness in the laminate used for the ISO analysis resulted in a negligible decrease of 0.03 mm, due to the fractionally higher fiber mass content of the ISO laminate relative to the scantling rule.

The scantling rule dimensions resulted in significantly greater section moduli of cored panels than those required by the ISO analysis. The section modulus of the inner skin is $SM_i = 0.381$ cm^3 per cm width, while the outer skin is $SM_o = 0.435$ cm^3/cm. These values are more than twice the limiting minimum requirement presented in table 31.

The stiffener web area, A_w, is a means of checking shear strength, while the stiffener section modulus checks bending strength. The hull stringers and engine bed stringers determined by the scantling rule have web areas of 3.52 and 7.52 cm^2, respectively, and section moduli of 24.48 and 63.40 cm^3 respectively. These values are greater than all calculated minimum requirements presented in table 32.

The standard places maximum limits on the ratios of stiffener height to web laminate thickness and stiffener width to flange laminate thickness, to prevent localized buckling of the stiffeners. These limits, as well as the computed values for the scantling rule stiffeners, are presented in table 33. All stiffeners pass the buckling verification.

Table 34 summarizes the results of the cross check by comparing the results from the scantling rule to the limiting value found in the ISO calculations. A full ISO structures validation was not performed, but the results of the elements checked indicate that the scantling rule approach delivered reasonably conservative results. In actuality, the results may be more conservative than they appear from table 34, because several conservative simplifications were made in the ISO calculations. In particular, the topmost longitudinal stiffener is the most heavily loaded, due to the greater spacing between the stiffener and the deck edge. This stiffener is the only one that doesn't exceed by wide margins the required minimum web area and section modulus. This stiffener is very close to the waterline, but the design pressure was conservatively calculated using only the

Longitudinal Stiffener ISO Requirements		
Location	**Section Modulus** (cm^3)	**Web Area** (cm^2)
1A	13.77	2.64
1B	5.03	0.84
1C	5.01	0.79
2A	17.20	2.93
2B	9.60	1.37
2C	9.15	1.30
3A	20.89	3.07
3B	13.067	1.67
3C	11.66	1.44
4A	17.00	2.54
4B	11.27	1.47
5A	22.90	2.41
5B	13.88	1.25

Values given are required minimums. Stiffener location is given by panel number, which references the bottom stiffener of the specified panel. I.e. location 1B refers to the middle hull stiffener between bulkheads 1 and 2.

Table 32: Longitudinal ISO Stiffener Requirements

	ISO Maximum	EB Stringer	Trans. Frame	Hull Stringer
h/t_w	32	17.02	17.02	9.09
b/t_f	21	17.02	17.02	18.41

Table 33: Stiffener Buckling Verification

bottom pressure, rather than using an average of bottom and side pressures, which would reduce the load experienced by the member. Additionally, the stiffener and panel spacing were taken at each location as the most conservative (longest) values, rather than by averaging. Finally, curvature of panels and longitudinal stiffeners was neglected for simplicity. Including the curvature effects would tend to decrease the panel design bending moment and stiffener required section moduli.

Element		**ISO Limiting Reqmt**	**Scantling Rule Result**
Bulkhead Thickness	mm	11.2	12
Long. Stiffener A_w	cm^2	3.07	3.52
Long. Stiffener SM	cm^3	22.90	24.48
Solid Panel Fiber Density	kg/m^2	3.09	4.04
Cored Panel t_s	mm	16.68	17.15
Cored Panel SM	cm^3/cm	0.115	0.382

Table 34: Summary of Cross Check Validation

5 Rudder Design

5.1 Rudder Type and Planform

The design process for the rudder involved first selecting between mounting configurations and positions. For seaworthiness considerations, a traditional, deeply immersed keel-hung rudder exhibits desirable characteristics, as discussed in section 2. However, the Scheel type keel selected for the design made such an arrangement awkward. Furthermore, in order to balance the sail plan and the underwater profile without moving the mast too far forward, it was helpful to mount the rudder further aft. Another consideration for locating the rudder was adequate clearance for the engine shaft and propeller, the location of which was significantly constrained by the available interior volume to mount the engine. Finally, because of the small size of the yacht, the simplicity and strength of an outboard mounted rudder with tiller steering was desirable. This arrangement eliminates the need for steering gear systems, and eliminates the through hull penetration, seal, and bearing of a common spade rudder design. The steering elements consist simply of a tiller and gudgeons and pintles connecting the stock to the transom. This arrangement is easy to inspect, maintain, and repair if needed. In addition to the transom connection points, the rudder was designed to mount to a partial skeg, faired into the hull at the aft perpendicular and extending down approximately half the span of the rudder. A skeg acts to delay the onset of flow separation that results in rudder stalls, and also provides additional structural strength.[3] A partial skeg configuration also provides a degree of balance to the helm, as a portion of the rudder extends forward of the axis of rotation.

Mounting the rudder further aft increases the yaw moment it can impart to the yacht. However, it also tends to reduce the depth of water the rudder operates in, making the rudder more susceptible to the unsteady flow effects of orbital velocities in waves. It also makes the rudder more susceptible to ventilation, although this is more a concern for high aspect spade rudders that develop a large suction peak near the leading edge.[3] Finally, in extreme cases, a bow down trimming moment when running before the wind, or yacht pitching induced by waves, can lift the rudder partially from the water and so markedly reduce steering power.[3] These factors were mitigated by selecting a relatively large planform area, a low aspect ratio, and reasonably large span. The rudder span was selected to be as deep as practical without protruding below the depth of the keel. The rudder area was selected to be close to 0.6 m^2. This is about 20% larger than the recommended 1.5% of sail area for a yacht of this waterline length.[5] Rudder planform, including skeg cut out, and configuration are displayed in figure 28, and planform dimensions are presented in table 35.

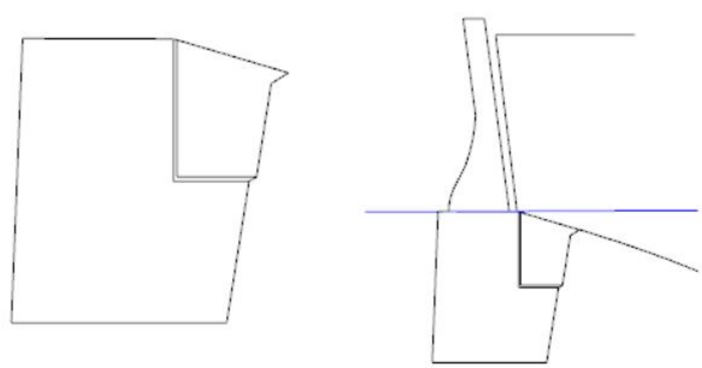


Figure 28: Rudder Planform and Configuration

Rudder Planform		
Area	A	0.584 m^2
Span	S	0.850 m
Root Chord	C_r	0.750 m
Tip Chord	C_t	0.640 m
Mean Chord	$\overline{C}$	0.695 m
Aspect Ratio	AR	1.22

Table 35: Rudder Planform Dimensions

5.2 Foil Section Selection

A suitable foil cross section was selected based on lift, drag, and stall angle considerations. Symmetric NACA 4 digit sections were considered, with the key variable being the maximum thickness ratio. The foil drag at zero and low angles of attack tends to increase with foil thickness.[5] This is significant, since this is the regime the foil will operate in during steady course transit, either under power or sailing with a few degrees of weather helm. However, thicker foils also exhibit higher stall angles, and more gradual stall characteristics.[5] This is a critical consideration for coursekeeping and steering control in heavy weather. A final consideration is the weight of the rudder, particularly if it is to be made from solid material such as wood or metal. In this case, given a rudder planform, the thickness ratio of the cross sections will have a considerable impact on the structural weight of the rudder. For this reason, thickness ratios greater than 14% were not considered.

Foils of thickness ratios ranging from 9 to 14% were examined at a Reynolds number of $R_n = 2x10^6$, which corresponds to the rudder mean chord traveling through the water at about 6 knots. Figures 29, 30, and 31 display the results of an analysis of four such foils using the open source program XFOIL.[44]

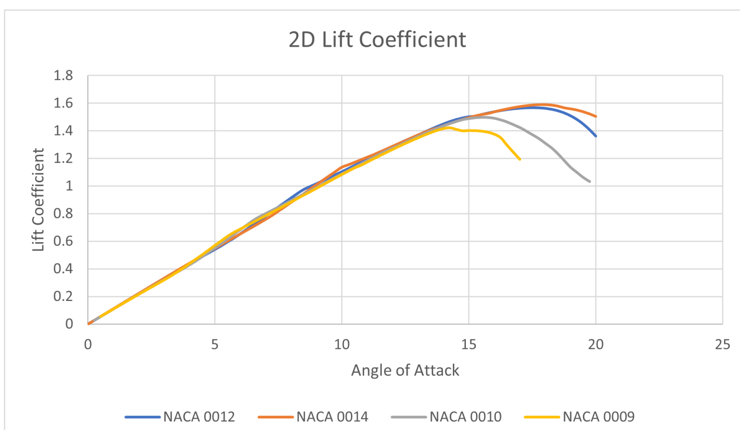


Figure 29: Rudder Foil Section Lift Coefficients

The lift coefficient and lift/drag behavior of the different foil sections, depicted in figures 29 and 30, demonstrates how stall angle increases with increasing thickness. Moreover, for these foils the incremental increase in stall angle appears to reduce at larger thicknesses: the stall angle increase between 10% and 12% thickness ratios is greater than that between 12% and 14%.

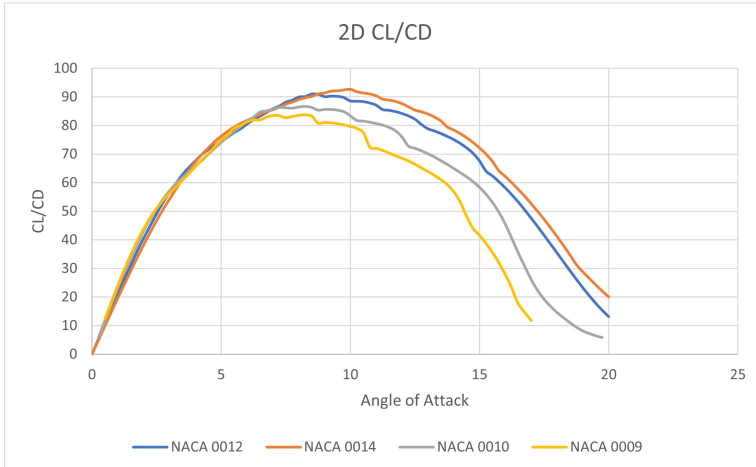


Figure 30: Rudder Foil Section Lift to Drag

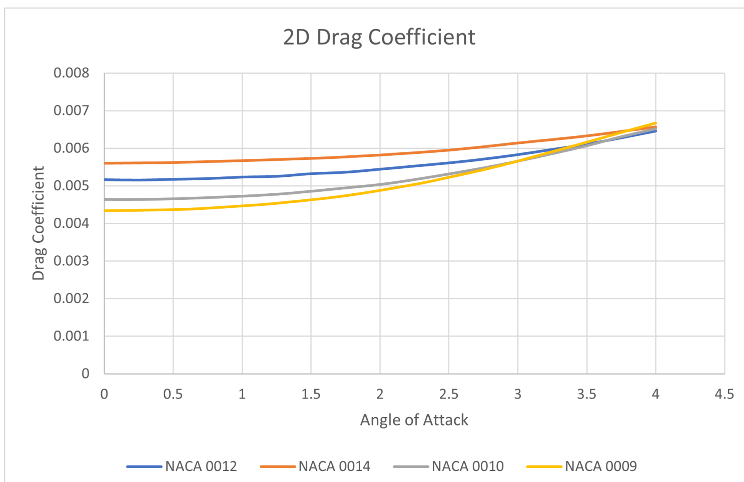


Figure 31: Low Angle of Attack Rudder Foil Section Drag Coefficients

The drag coefficient behavior of the foils at low angles of attack is depicted in figure 31, showing a drag reduction for decreasing thickness. The thickest foil considered has a zero angle drag coefficient about 30% larger than the thinnest foil.

The moderately thicc NACA 0012 section, whose XFOIL panelling is displayed in figure 32, was selected for the design as a good compromise between the higher stall angles of thicker sections and the lower drag at low angles of attack and reduced weight of thinner sections.

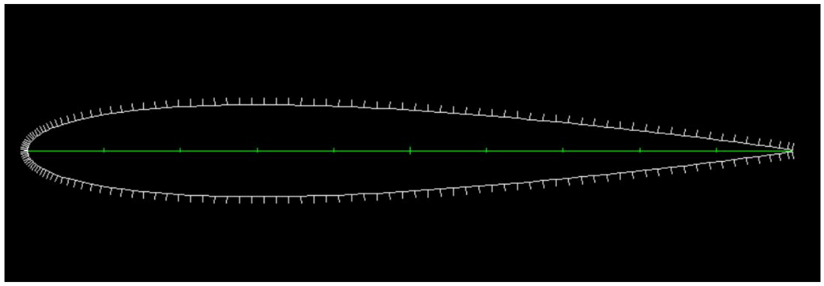


Figure 32: NACA 0012 Section

5.3 Rudder Scantlings

The rudder blade and integral stock are constructed together from Douglas fir sheathed in a 1 mm thick layer of fiberglass laminate for waterproofing. The rudder was analyzed using the structural criteria of ISO 12215-8.[45] The laminate was neglected in the structural calculations. The principal stresses of concern on the rudder and stock are the direct and shear stresses that arise from the bending moment and torsion that result from the rudder design force. The design force experienced by the rudder depends on rudder area, A_R, the vessel L_{WL}, and factors that account for design category, displacement-length ratio, 3D flow effects at the root, and vessel usage. It is given as:

$$F_d = 23 * L_{WL} * A_R * K_{SEA} * K_{LD}^2 * K_{GAP} * K_{USE} \tag{33}$$

The design force is used to calculate the resultant torsion, T, and bending moment, M, experienced by the rudder and stock. In the analysis, the skeg was considered to be flexible, so that the scantlings are determined by considering the rudder as a spade rudder mounted only to the two gudgeons on the transom. This represents a conservative simplification. The design maximum bending moment, M_d occurs at the lower gudgeon:

$$M_d = F_d * Z_b \tag{34}$$

Where Z_b is the vertical distance from the geometric centroid of the rudder area to the center of the lower gudgeon. The bending moment decreases linearly above the lower gudgeon until it vanishes at the upper gudgeon. The bending moment results in a direct stress in the rudder or stock, calculated using the section modulus in bending, SM_B

$$\sigma = \frac{M}{SM_B} \tag{35}$$

The design maximum torque, T_d is given in terms of the design force and the lever arm, r, which is calculated according to rudder geometry as detailed in the standard. Torsion is at its maximum at the lower gudgeon, and remains constant above this point to the tiller connection.

$$T_d = F_d * r \tag{36}$$

The torsion results in a shear stress, evaluated using the polar section modulus, SM_T:

$$\tau = \frac{T}{SM_T} \tag{37}$$

The section modulus in bending or torsion depends only on the geometry of the cross section, being the ratio of the applicable moment of inertia to the distance between the extreme fiber and the neutral axis.

The design criteria for wooden rudders under combined stresses is given as:

$$\left(\frac{\sigma}{\sigma_u}\right)^2 + \left(\frac{\tau}{\tau_u}\right)^2 \leq 0.25 \tag{38}$$

where σ_u is the ultimate flexural strength or modulus of rupture, and τ_u is the ultimate shear strength. Additionally, neither direct stress nor shear stress shall exceed half the applicable ultimate strength. The ultimate strengths for Douglas fir are given in table 36, assuming 12% moisture content Coastal grown Douglas fir.[46] The stock at the tiller connection is a rectangular section, 7 cm wide by 12 cm long. This is the cross section of smallest section modulus. Approximately halfway between the tiller and the rudder, the stock flares out to blend into the foil blade. Conservatively considering only this limiting cross section (neglecting contributions from the blade to moment of inertia), the results of the structural analysis are presented in table 36.

ISO Rudder Structural Validation		
F_d	(N)	3357
Z_b	(m)	0.617
r	(m)	0.057
M_d	(N-m)	2071
T_d	(N-m)	191
σ_{max}	(MPa)	21.1
τ_{max}	(MPa)	0.9
σ_u	(MPa)	85
τ_u	(MPa)	7.8
$\left(\frac{\sigma}{\sigma_u}\right)^2 + \left(\frac{\tau}{\tau_u}\right)^2$		0.077

Table 36: ISO Rudder Structural Validation

The design torque of 191 N-m, transmitted through a tiller arm of 1.5 m, results in a design steering force of 127.3 N, or about 29 pounds of force. This is within the capability of a single helmsman, and a reasonable amount of force for a limiting condition.

The skeg is designed with a Douglas fir core, sheathed in laminate of the same thickness as the keel region. The skeg may be constructed either integral to the hull shell in the molding process, or as an addition fitted through a key hole cut in the bottom, and connected to a supporting floor structure just forward of the transom. Weights for both the skeg and rudder were computed, and included in the total weights analysis presented in appendix F. The rudder, stock, and skeg are displayed in figure 33.

Figure 33: Hull with Rudder, Stock, and Skeg

6 Rig and Sailplan Design

6.1 Sail Plan

Having selected a junk rig, the key parameters of the sail plan design are sail area, shape, and aspect ratio. Sail area interfaces with both speed and stability; an under-canvassed yacht will not be able to reach her best speeds while an over-canvassed yacht will experience excessive heel angles. An additional consideration, given that the rig consists of a single sail, is the upper limit in sail size that a single crew member can reasonably handle. A sail area/displacement ratio at full load of 15 was selected to ensure adequate powering, resulting in a sail area of 33.25 m^2. This is considerably below the recommended single handed limit of 56 m^2 for junk type sails.[30]

For determining the sail shape, the form recommended by Hasler and McLeod was adopted.[30] This shape, depicted in figure 34, consists of 4 or 5 parallelogram panels below two triangular head panels. The selected sail shape is characterized by a luff distance, L, which is subdivided in lower panels of width P. The triangular head panels have edge length, N, and their battens are spaced by a distance U at the sail throat. All junk sails are divided into panels, which are fully battened. The boom and each batten, except perhaps the top batten, are sheeted. The head of the sail, or top edge of the topmost panel, is attached to the yard, while the foot of the sail is attached to the boom. In the selected design, the batten lengths and boom are all of equal length. The advantages of the selected sail shape are purely practical considerations related to reefing and batten length.[30] The reefing process on a junk rig involves easing the halyard to lower the yard, taking in the desired number of panels to reduce sail area, as depicted in figure 35, which shows a deeply reefed sail with only two panels hoisted. With the selected design, the battens stagger positively at the sail leech as panels are reefed, with each successive batten extending slightly aft of the previous batten. This prevents fouling the sheet spans when the sail is eventually rehoisted. It also keeps the furled battens securely within the topping lifts, discussed in more detail below. Finally, the sail design results in equal batten lengths at all panels, a practical consideration for carrying spare battens in the event of one breaking at sea.

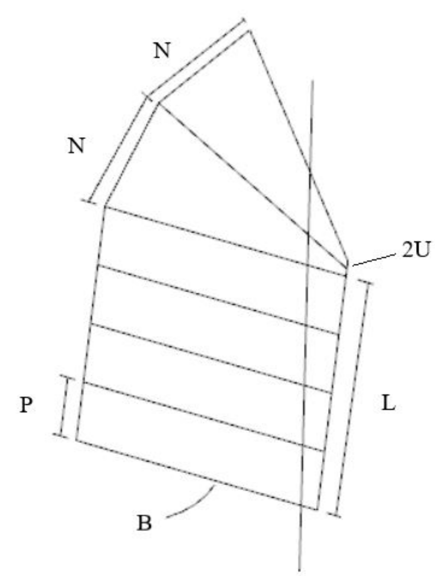


Figure 34: Selected Shape of Junk Sail with Mast Line

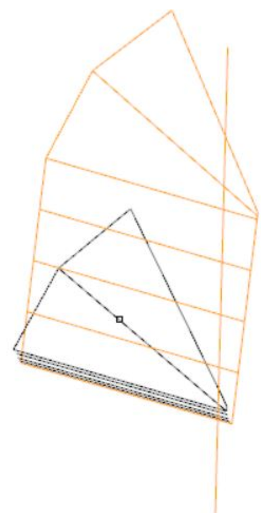


Figure 35: Deeply Reefed Junk Sail

Having determined the sail area and general shape, the aspect ratio was then selected. A higher aspect ratio sail may be expected to deliver better performance to windward, but also requires a taller, heavier mast. Lower aspect ratio sails deliver smaller heeling moments, which is advantageous to a tender yacht. However, they require heavier battens, boom, and yard; although if the aspect ratio is low enough, fewer panels may be required and one or two battens may be eliminated. The sail area is also spread more fore and aft for a low aspect ratio rig, and this may require moving the mast too far forward to achieve an appropriate lead. It may also negatively impact the sheeting positions. An important consideration for a trailer capable yacht was the length overall of the mast, which should not be much longer than 10 meters.

Four sails of aspect ratios ranging from 1.7 to 2.0 were drawn up according to the procedure detailed in reference [30]. The sail of aspect ratio 1.8 was selected, based on balancing the above considerations. This is the same sail depicted in figure 34, and its dimensions are given in table 37. Because the battened sail is less stressed, lighter fabric may be used relative to an unbattened sail. The fabric weight recommended for this size sail is 221 g/m^2.[30]

Sail Dimensions	
A	33.7 m^2
B	4.90 m
L	4.60 m
N	2.28 m
U	0.15 m
AR	1.8

Table 37: Selected Sail Design

With the sail plan determined, the mast was located to fix the sail plan center of effort about 9% of the L_{WL} forward of the center of lateral resistance for proper helm balance. The mast heel is located centerline, 1.81 m aft of the bow, or 1.57 m aft of the forward perpendicular. The mast position is also constrained by pitching consideration; it is recommended to place it at least 15% of the L_{WL} aft of the forward perpendicular.[30] The present location is about 0.5 m behind this limiting position.

6.2 Mast Design

Material options for mast construction include timber, both solid or hollow; light alloy, such as marine grade aluminum alloys; and carbon fiber. Because the rig weight has a large impact on vertical center of gravity, which is difficult to compensate for in a relatively shallow draft yacht of limited displacement, carbon fiber was selected for the mast to keep VCG as low as practicable.

The mast was designed according to the procedure detailed in reference [38], which partitions the mast into equal length intervals and determines the required thickness of unidirectional carbon fiber laminate for each interval. A design bending moment, equal to the yacht's maximum righting moment, is applied between the mast partner and the boom. From the mast partner to the heel, and from the boom to the masthead, this moment decreases linearly to zero. Based on the bending moment and the design stress of the unidirectional carbon fiber laminate, the required section modulus is calculated at each interval along the mast height.

$$SM_{min}(z) = \frac{M(z)}{\sigma_d} \tag{39}$$

where the design stress is the limiting ultimate strength of the laminate, divided by a safety factor, SF. Shear stress is not explicitly calculated in this method, relying on the observation that circular section composite tubes under a bending load fail on the compression side. Thus, the limiting strength is the laminate's ultimate compressive stress, and an appropriate safety factor is applied. To account for the shear load at the mast head and mast heel, where the bending moment vanishes but the shear force does not, the wall thickness at the mast head is set equal to the thickness at the mast height experiencing the highest load. [38] The method does not consider torsional loads, since the sail is attached to the mast with parrells that allow the sail to rotate without twisting the mast. Had a sail track been used instead, the resulting torsional loads would require assessment.

The design bending moment curve depends on the full load displacement, the maximum righting arm of the yacht, the mast overall length, and the distances between the mast heel and deck partner and between the deck partner and boom. The former distance is referred to as the mast bury, and must be sufficiently large: a minimum bury of 9% of overall mast length is recommended for freestanding junk masts.[30]. The distance between boom and deck partners is a practical matter of boom travel clearance over the deck house. These dimensions are presented in table 38, and the design bending moment curve for the mast is depicted in figure 36.

Bending Moment Factors		
Full Load Displacement	Δ_{FL}	3500 kg
Max Righting Arm	GZ_{MAX}	0.66 m
Design Moment	M_{MAX}	22.6 kN-m
Mast Length	L_M	9.75 m
Mast Bury	L_B	1.36 m
Boom Clearance	H_B	0.58 m

Table 38: Mast Bending Moment Factors

The mast inner diameter at the partners and the mast head is set, and the resulting linear taper is used to calculate mast inner diameter at each interval along the mast height. The mast diameter tapers from the boom to the masthead, and remains constant from the boom to the mast heel. From the required section modulus and the inner diameter, the necessary thickness of laminate is calculated. The thickness is calculated in terms of the required number of strips, N, of unidirectional carbon fiber tape required, with the strips of constant known thickness, t_{CF}, and width, w_{CF}.

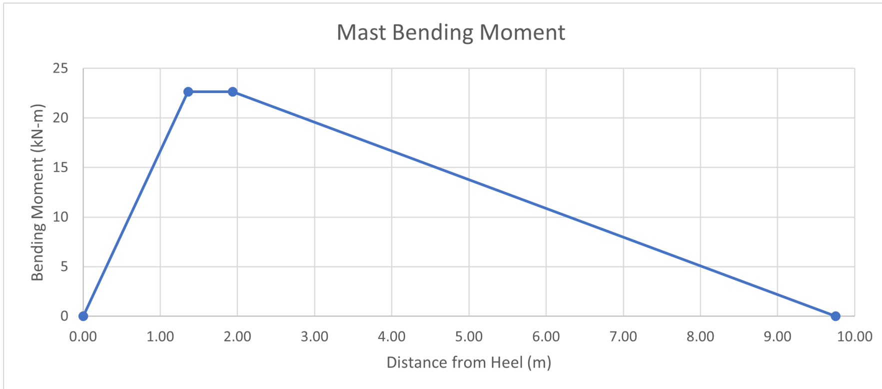


Figure 36: Design Bending Moment of Mast

The carbon fiber laminate is enclosed by inner and outer hoop layers of fiberglass. The fiberglass is unidirectional fabric aligned transverse to the mast axis, and of 814 g/m^2 weight and thickness of 1.02 mm. This assists in the fabrication process but also provides hoop strength and builds wall thickness. The fiberglass hoop layers have the added benefit of reducing the risk of galvanic corrosion that would otherwise occur between carbon fiber and any metal fittings in contact with the mast. The mast wall thickness is maintained $\geq 3\%$ of the mast diameter to prevent localized buckling from compressive stresses.[38]

To design the mast, a spreadsheet calculator was developed. The mast diameter at partners and mast taper ratio were varied to find the lightest mast that satisfied the buckling criteria. The mast length was shortened by 0.73 m relative to the height recommended by reference [30] for the selected sail plan. This was necessary for VCG considerations, and reduced the clearance between the yard sling point and the halyard crane at the masthead to 0.75 m, which is predicted to be acceptable for the required halyard block size and shackle spacing. Laminate material properties presented in reference [38] are used for the design, and considered achievable in practice. They were given in English units but have been converted to metric for display in table 39. A safety factor of 6 was applied to the ultimate stress to calculate the design stress. The particulars of the mast are presented in table 40. The laminate schedule is depicted in figures 37 and 38, which shows the number and sequence of carbon fiber strip layup. Details of the calculations are contained in appendix G.

Mast Materials		
CF Strip Width	w_{CF}	30.5 cm
CF Strip Thickness	t_{CF}	0.15 mm
CF Strip Linear Density	ρ_c	0.074 kg/m
Fiber Mass Fraction	ψ	0.6
Laminate Ultimate Stress	$\sigma_{u,c}$	1172 MPa
Safety Factor	FS	6
Design Stress	σ_d	195.3 MPa

Table 39: Mast Material Properties

The resulting mast structure is predicted to be rather conservative, as carbon mast sections with adequate section modulus for the region of highest loading are quoted in the literature at lower weights; a typical carbon mast section with a minimum section modulus of 124 cm^3 has a linear density of 5.4 kg/m.[5] If the entire mast was built from this section, it would weigh about 7 kg less than the designed mast. Such a mast would also be 'overbuilt' above the boom, since the

Mast Results		
Inner Diameter at Partner	D_P	19.5 cm
Inner Diameter at Masthead	D_H	7.0 cm
Mast Taper Ratio	T_M	1.64 cm/m
Mast Total Mass	M_M	59.6 kg
Mast Vertical Center	VCG_M	3.8 m

Table 40: Mast Particulars

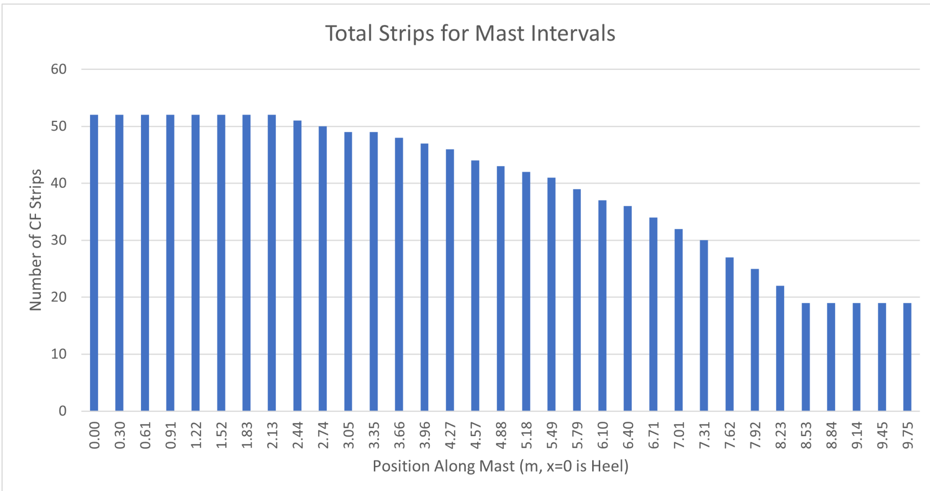


Figure 37: Total Carbon Fiber Strips Required at Each Mast Interval

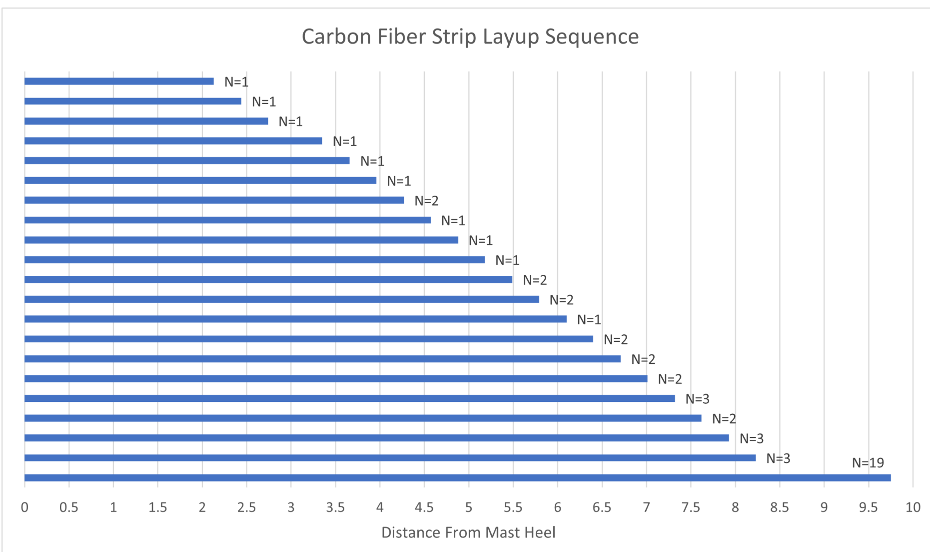


Figure 38: Mast Carbon Fiber Layup Sequence
Bars indicate the length along the mast a strip extends. Bar labels indicate the number of strips laid down for that length. I.e. 19 strips of 9.75 m lengths are laid down first, then 3 strips of length 8.23 m, etc.

maximum required section modulus is 115 cm^3 between the partners and the boom, and decreases above this point. This may indicate a lower factor of safety is warranted. The chosen factor of 6 was based on design practice recommended in references [38] and [47]. It represents a safety factor of 2 applied to the laminate ultimate stress to account for non-ideal material properties resulting from construction process quality, and a further factor of 3 applied to account for transient moments in excess of the maximum righting moment, and to ensure that the laminate is not stressed to more than about half its ultimate value, which results in permanent degradation of the laminate strength.[38]

The mast step is constructed from Douglas fir timbers that form a box notched over and lag bolted to the three reinforced floors forward, with a depth of 0.75 times the diameter of the mast heel.[30] The mast partner is a 5 mm aluminum flanged cylinder, through bolted to a plywood backing and incorporated into the fiberglass structure of the foredeck, similar to the design presented for a similarly rigged and sized vessel in reference [48].

The mast head is a 3 mm thick stainless steel cylindrical endcap, with length equal to its outer diameter. It is glassed into the mast tip and through bolted. This cap provides the connection points for the rigging, in the form of four welded tangs.

Lighting protection is provided in accordance with DNV GL classification rules.[49] This requires all non-metal masts to be fitted with a copper cable of 70 mm^2 cross section, terminating in a 12 mm copper rod rising 300 mm from the mast head. The cable runs inside the mast, and terminates in a copper plate bonded to the hull and in contact with seawater. A fitting near the mast heel enables breaking the connection for unstepping the mast.

6.3 Battens, Yard, and Boom

Scantling rules for the boom, yard, and battens for junk rigs are given in reference [30] for spars constructed from European ash. These scantlings are calculated from sail plan dimensions, and empirical in nature. In order to reduce weight, equivalent scantlings were derived in both carbon fiber and high strength aluminum, by enforcing both equivalent stiffness and equivalent strength in the carbon fiber or aluminum spars of initially unknown dimensions, and the wooden spars of known dimensions. For the carbon fiber spars, material properties were based on the advertised minimum properties of commercially available unidirectional carbon fiber/epoxy pultruded tubes.[50] For the aluminum spars, material properties were based on the high strength 7075 T6 alloy. These are shown with the material properties of European ash[51] in table 41.

Property		**Material**		
		Euro Ash	Carbon	7075 T6 Al
Density, ρ	g/cm^3	0.68	1.5	2.8
Flexural Strength, $\sigma_{u,f}$	MPa	104	1370	-
Yield Strength, σ_y	MPa	-	-	483
Elastic Modulus, E	GPa	12	127	72

Table 41: Spar Material Properties

To determine the required dimensions for the composite spars, first the dimensions of the wooden spars were calculated according to reference [30]. These were used to compute each wooden spar's moment of inertia, I_w, and section modulus, SM_w, displayed in table 42. When constructed to the wooden scantling rule, the boom is rectangular, of constant cross section. The battens' cross sections are also rectangular. Along most of the batten length the cross section is of constant depth, except for a short length of thicker stub at the luff end; the moment and section modulus of the long, thinner section was used in the analysis, as this is where the maximum bending moment and stress is predicted to occur. The yard is analyzed at both the center and the tips, as it is a doubly tapered rectangular beam, whose depth diminishes linearly from the center to both tips.

	Boom	Batten	Yard (Mid)	Yard (Tip)
I (cm^4)	48.0	33.6	502.0	96.1
SM (cm^3)	19.6	13.7	102.4	30.0

Table 42: Wooden Spar Moments and Section Moduli

To enforce equivalent stiffness between the wooden and composite spars, the products of the moment of inertia and elastic modulus were set equal in wood and in composite:

$$(EI)_w = (EI)_c \tag{40}$$

This resulted in a required minimum moment of inertia for the composite spar, I_c:

$$I_c \geq \frac{E_w}{E_c} * I_w \tag{41}$$

The section modulus of the wooden spar and the ratio of the wood and composite ultimate strengths were used to enforce equivalent strength without assuming a load. The stress resulting from a bending moment, M, applied to a beam is related to the section modulus:

$$\sigma_f = \frac{M}{SM} \tag{42}$$

where the section modulus may be computed from the spar geometry using the moment of inertia, I, about the neutral axis and distance, $\overline{y}$ from the axis to the extreme fiber.

$$SM = \frac{I}{\overline{y}} \tag{43}$$

If the composite spar is to withstand a bending load of equal or greater magnitude than the failure load of the wooden spar, then the required section modulus in composite is the product of the wood spar section modulus and the ratio of the materials' ultimate flexural strengths:

$$SM_c \geq \frac{\sigma^w_{u,f}}{\sigma^C_{u,f}} * SM_w \tag{44}$$

The required section moduli and moments of inertia for the composite spars are displayed in table 43. In calculating the required section moduli, an additional factor of safety of 1.5 was applied to the ultimate strength of the carbon fiber laminate. The same process was used to compute the moduli and moments required for aluminum, although no additional safety factor was employed. The wooden scantlings for the battens and the yard are rectangular in cross section, such that their moments and section moduli are greater for bending in the vertical plane than for bending in the plane of the sail. For both spars, the larger of the two moments and moduli were used to determine the required equivalent strength and stiffness for composite and aluminum spars. This is predicted to result in rather conservative scantlings, since the largest bending moments in limiting wind conditions are expected to occur in the plane of the sail, rather than the vertical plane.

Because moment of inertia and section modulus are functions of spar cross sectional geometry, the may be easily tabulated for a range of geometries. This was done for battens, the boom, and yard to select appropriate dimensions for the composite or aluminum spars that would result in equal or greater bending strength and stiffness than the wooden spars, at significant weight savings. All spars are of hollow tube cross section, with the outer diameter and wall thickness set to achieve the required stiffness and strength at the minimal weight. The same buckling criteria is applied as was used for the mast, such that the wall thickness is at least 3% of the inner diameter. All composite spars are formed around a low density structural foam core, and finished with an outer hoop layer of 200 g/m^2 fiberglass. The composite and aluminum spar scantlings are displayed in

Composite Spars				
	Boom	**Batten**	**Yard (Mid)**	**Yard (Tip)**
I (cm^4)	4.54	3.18	47.43	9.08
SM (cm^3)	2.23	1.56	11.67	3.42
Aluminum Spars				
	Boom	**Batten**	**Yard (Mid)**	**Yard (Tip)**
I (cm^4)	8.04	5.63	84.01	16.08
SM (cm^3)	4.22	2.96	22.06	6.46

Table 43: Composite and Aluminum Spar Required Moments and Section Moduli

table 44. All spars exhibit greater strength and stiffness than demonstrated by the wooden spars, with the exception of the composite battens, which are fractionally less stiff.

Composite Spars				
	Boom	**Batten**	**Yard (Mid)**	**Yard (Tip)**
D_o (cm)	4.25	4.00	8.00	4.75
t_w (cm)	0.21	0.14	0.28	0.28
I (cm^4)	5.45	3.17	50.66	9.86
SM (cm^3)	2.57	1.58	12.66	4.15
Mass (kg)	2.86	1.66	6.51	
Aluminum Spars				
	Boom	**Batten**	**Yard (Mid)**	**Yard (Tip)**
D_o (cm)	5.50	4.75	10.25	5.75
t_w (cm)	0.20	0.20	0.30	0.30
I (cm^4)	11.71	7.41	116.2	19.13
SM (cm^3)	4.26	3.12	22.66	6.65
Mass (kg)	4.84	3.96	10.48	

Table 44: Composite Spar Scantlings

The boom and battens are 4.95 m in length, while the yard is 5 m in length. The spar weights are included in table 44. The core and hoop layer are included in the weight calculations for the composite spars. Due to the height of the spars, the weight difference between aluminum and composites had a significant impact on overall yacht VCG, so the composite spars were selected for the design.

6.4 Rigging

The rigging plan was developed following the recommendations provided in reference [30]. Figures 39 and 40 shows the required lines, divided into running and standing rigging for clarity. The full rigging warrant, including required lines, shackles, blocks and cleats, is contained in appendix H.

The running rigging is used to control the position and the set of the sail, and consists of the halyard, the sheets, the yard hauling parrel, the luff hauling parrel and the topping lift. All running rigging except the sheets is led aft to cleats located on the side decks accessible from the cockpit, to enable all normal sail handling to be conducted by the helmsman.

The halyard raises and lowers the sail through a 4 part tackle between the mast head and the sling point located at the center of the yard. The 4 part tackle consists of a double block with becket shackled to the mast head, and another double block shackled to the sling plate. The

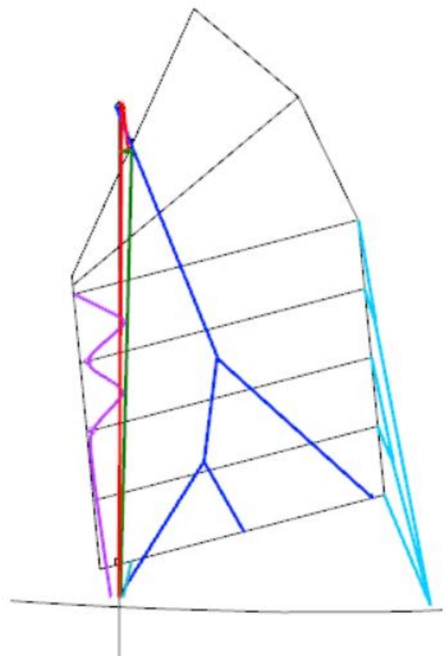


Figure 39: Running Rigging
Key: Light Blue = Sheets; Dark Blue = Topping Lift; Green = Yard Hauling Parrel; Red = Halyard; Purple = Luff Hauling Parrel

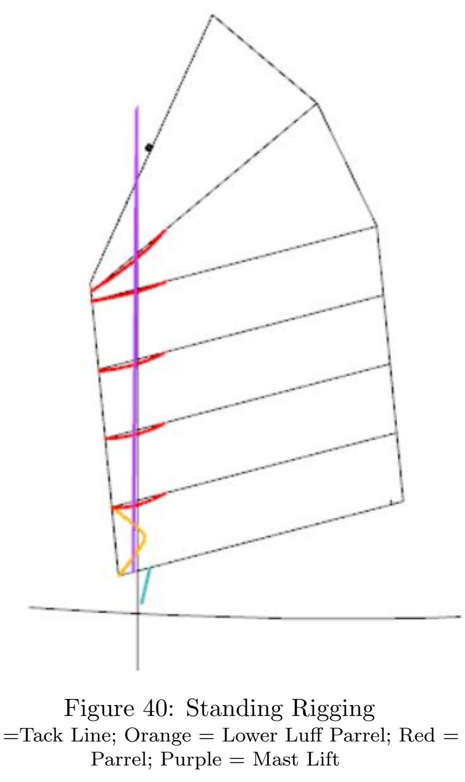


Figure 40: Standing Rigging
Key: Blue =Tack Line; Orange = Lower Luff Parrel; Red = Batten Parrel; Purple = Mast Lift

mechanical advantage conferred by a 4 part purchase is suitable for hoisting sails up to about 37 square meter in area.[30] The halyard runs from the becket on the mast head double block, through the sling plate and masthead blocks, down the side of the mast, and through a fairlead to its securing cleat.

The yard hauling parrel pulls the yard sling point close to the mast, maintaining the sail in its designed position and preventing fore and aft motion of the head of the sail in a seaway. It runs from the sling point, wraps around the mast, and back to the sling plate, where it passes through a block and runs down the mast to its cleat. The standing lower luff parrel provides a similar function at the tack of the sail, while the adjacent standing tack line keeps the lower panel pulled taut.

The topping lifts run from the mast head down both sides of the sail, splitting twice at thimbles. They support the boom when the halyard is eased, and gather together the battens and sail panels as the sail is reefed. They can be made either standing or running; running lifts enable the crew to easily lower the boom and sail to the deck at sea, which is an important practical consideration for a cruising vessel. For this reason, they were made running in the present design. Due to the size of the sail, 2 part purchases are used with the topping lift running lines.

The luff hauling parrel is made fast at the forward end of the second-highest batten, wraps around the mast, passes through a block at the next lower batten, then wraps around the mast again before being lead through a block at the next lower batten to the deck. Its purpose is to control the fore and aft set of the battens, providing increased control over the set of the sail.

The sheets enable the helmsman to control the angle and twist of the sail. Many different sheeting arrangements exist for junk rigs, although important considerations are sheeting power and sheeting block location.[30] In the present design, a 5 part, 5 point sheeting system is used. The arrangement is shown diagrammatically in figure 41, which illustrates how the sheet is rove through 5 blocks to provide mechanical advantage while delivering equal tension to the boom and each of the four lowest battens. The end of the sheet is connected to the boom, and connections at the four battens are made by way of two single blocks that are free to travel along a fixed-length span of line connecting each pair of adjacent battens. The three fixed blocks are combined in a single housing mounted on the aft deck, on a traveller of sufficient height and width to permit full swing of the tiller that passes underneath.

The sheeting arrangement must be positioned such that the span blocks remain clear of the lower fixed blocks when the sail is reefed, and the lower block must be located to give a reasonable sheeting angle. The recommended sheet block positioning is outlined in figure 42, based on providing minimum clearance between the clew and lower sheet block, and providing an adequate sheeting angle.[30]

The standing rigging consists of the lower luff parrel and tack line, discussed above, and the mast lift and batten parrels. The batten parrels are simply lengths of line attached forward and aft of the mast line to hold the battens to the mast without limiting fore and aft sail motion or causing excessive friction when raising or lowering sail.[30] The mast lift runs from the masthead, down the outside of the sail, and ends in a loop around the mast base, below the boom. Its function is to gather in and support the forward end of the sail bundle when the sail is reefed.[30]

The required length and diameter of each type of line was determined, as were the required numbers and sizes of all shackles and blocks. This was used to create the rigging warrant presented in appendix H, and used to conduct a detailed weights and centers calculation for the rigging.

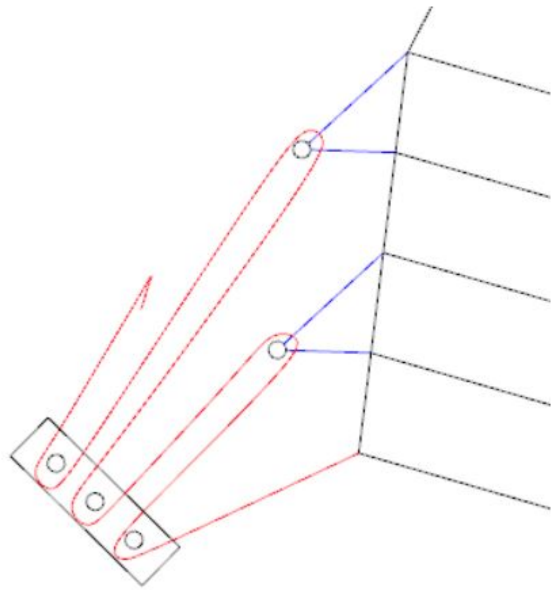


Figure 41: Sheeting Arrangement Diagram
The red curve is the running component of the sheeting system, rove through a 5 part purchase. Three of the blocks are fixed on deck, while two are free to travel along the blue spans connecting adjacent battens.

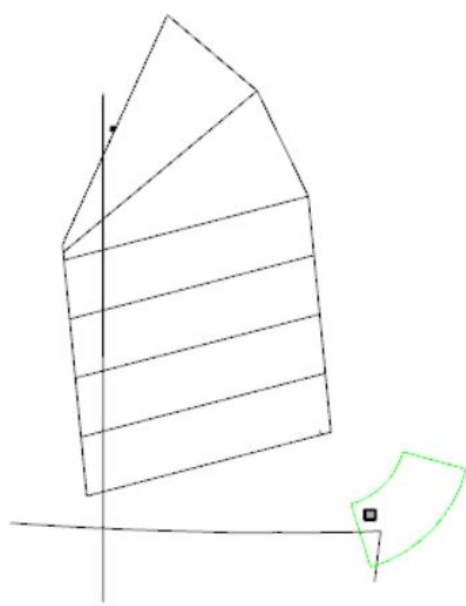


Figure 42: Lower Sheet Block Positioning
Acceptable positions for the lower sheet block outlined in green. The gray box is the design location for this block.

6.5 Mast Stepping

In order to fully realize the benefits of a trailer capable yacht, the mast should be capable of being stepped and unstepped with the crew's own resources, at least on the trailer during launch and recovery operations. A mast of close to 10 m and 60 kg is too heavy and awkward to manage manually, even with two crew members. Instead, a mast derrick may be used to accomplish this with a single crew member. Such a design, modeled after those presented in reference [30], is displayed in figure 43. The derrick consists of an aluminum column, shown in green, pinned on deck by a fitting located on the coachroof and supported at the head by three stays, shown in black, which are secured to cleats on deck. The derrick has a double block tackle connected to the top, shown in red. The tackle fall is run back to the foredeck where it can be taken to the anchor winch. The moving blocks of the tackle are hooked to a sling hitched to the mast near the mast center of gravity. The derrick is of sufficient length to lift the mast heel clear of the deck partner and coachroof, without two-blocking the tackle. The fitting at the column heel is located above the forward cabin bulkhead to transmit the point load without requiring excessive reinforcement of the cabin roof.

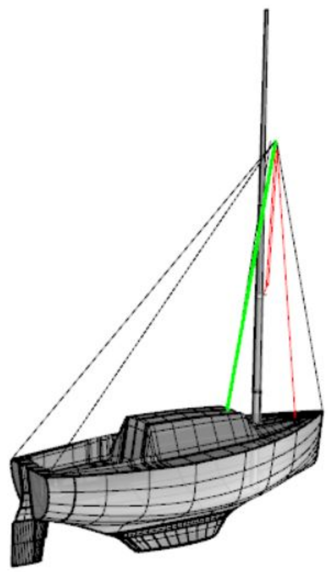


Figure 43: Derrick for Stepping and Unstepping Mast

The derrick is a 5.5 m length of aluminum tubing, with a 6 cm outer diameter and a wall thickness of 3.2 mm. It may be fabricated in sections that sleeve together for ease of storage and transport. Because it is primarily loaded in compression, the derrick is treated as a slender column with pin-pin connections. The critical buckling load is:

$$P_{Cr} = \frac{\pi^2 * EI}{L^2} = 5276N \tag{45}$$

which results in a safety factor greater than 4 for the anticipated load. The stays and the tackle line are the same type and diameter as the halyard: 8mm polyester 3 strand, with a breaking load well above the maximum load anticipated on the system. The mechanical advantage of the double block results in a hauling force of 15 kgf to lift the 60 kg mast. Combined with a winch, a single crew member will be capable of stepping or unstepping the mast, though a second person to help guide the mast would certainly simplify the process. The weight of the derrick is 8.8 kg; if it is carried at sea as part of the allotment for emergency gear set aside in the weight list, it could serve well as a spare yard or as an emergency mast.

7 Auxiliary Propulsion and Propeller Design

7.1 Engine Selection

A diesel engine was incorporated into the design for auxiliary propulsion. The yacht is not intended to be a motorsailer, so is not fitted with large tankage capacity needed for powered transits of considerable length. The purpose of the engine is to assist in navigating crowded harbors, in launch and recovery operations, and as a last resort in heavy weather near navigational hazards. The alternator also serves as a backup power supply to the solar panel to recharge the battery of the simple electrical system. A diesel, vice gasoline engine, was selected based on improved reliability and lower maintenance load.[2] Considerations for engine selection included power needs, weight, and compactness. Initially, power needs were estimated using displacement, by applying a rule of thumb of a minimum of 2 HP per long ton, or 1.47 kW per metric ton.[2] A light weight and compact commercially available marine diesel engine of adequate power was then selected: the Yanmar 1GM10. This is a single cylinder, seawater-cooled 6.6 kW diesel engine with a dry weight of 71 kg.[52] A simplified CAD model of the engine and reduction gear was created based on the manufacturer's drawings to ensure proper fit in the engine compartment. The model captures the required dimensions and clearances for placement while neglecting the engine details, although the removable manual start crank and location of exhaust elbow were modeled. It is displayed in figure 44.

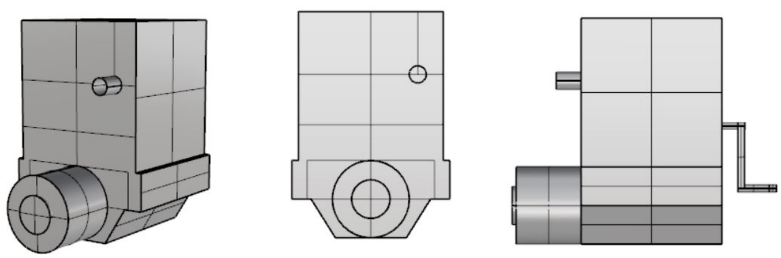


Figure 44: Simplified Model of Selected Engine for Fitting Purposes

7.2 Resistance Estimation

The calm water, upright resistance of the yacht was estimated using the correlations of the Delft series of yacht hulls, for speeds up to 8 knots. The computations were conducted by MAXSURF Resistance based on inputs of hull geometry. The Series I/II correlations were used, as they are based on hull forms more similar to the design than the Series III.[5] 'Canoe hull' geometry inputs that are based on the hull without keel, such as prismatic coefficient and draft, were determined from the yacht model by drawing tangents from the bilge curve to the centerline at several midships stations to generate an equivalent 'canoe hull' form without the keel. An appendage form factor of 2.5 was assumed. Results are displayed in figure 45.

In order to analyze the engine and propeller off-design performance, the added resistance in rough weather was estimated by calculating the wind resistance and the added resistance in waves. A constant apparent wind speed of 30 knots and waves of 3 m significant height ($H_{1/3}$), 5 second period, and 35° heading relative to the bow were assumed for the rough weather condition. The wind resistance of the hull and mast were calculated:

$$R_w = \frac{1}{2}\rho_{air} * V_{air}^2 * A_F * C_D \tag{46}$$

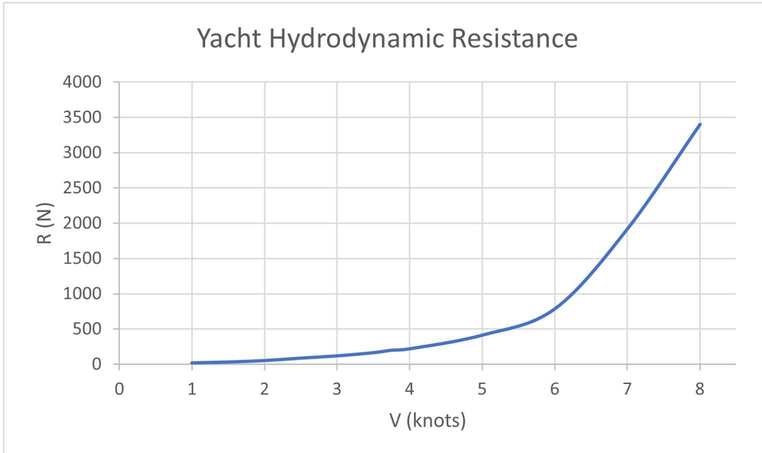


Figure 45: Estimated Yacht Upright Resistance in Calm Water
Resistance estimated with Delft Series I/II correlations for upright, calm water conditions.

where the frontal area of the hull is equal to the product of the midships freeboard and the maximum beam, and the frontal area of the mast is the product of the mast height and average diameter. The drag coefficient, C_D, was taken as 1.13.[5] For 30 knot apparent wind speed, the added resistance was 624 N.

The added resistance in waves, R_{AW}, was calculated from a statistical model based on Delft series hulls, summarized in reference [5].

$$\frac{100 * R_{AW}}{\rho * g * L_{wl} * H_{1/3}^2} = a * \left[100 * \left(\frac{\nabla_c^{1/3}}{L_{wl}}\right) * \left(\frac{k_{yy}}{L_{wl}}\right)\right]^b \tag{47}$$

where the coefficients a and b depend on wave period, wave heading, and vessel Froude number. The pitch gyradius k_{yy} was estimated as 25% of the hull length.[5]

The added resistance from waves was computed at Froude numbers of 0.15, 0.25, and 0.35. These resistances were combined with the added resistance from 30 knot winds and the calm water resistance to determine the total rough weather resistance curve for the assumed conditions in the speed regime of interest. Figure 46 displays the results. Within this speed regime, the combined added resistance of waves and wind is well approximated by a linear fit, which was used to interpolate added resistance in rough weather at 3.0, 3.25, and 3.5 knot vessel speeds. These speeds were selected to bracket the expected maximum attainable speed under the rough weather conditions, for analysis of propeller and engine operation. A different, higher set of speeds was selected for evaluation near calm water maximum speed. The two resistance curves are displayed in figure 47.

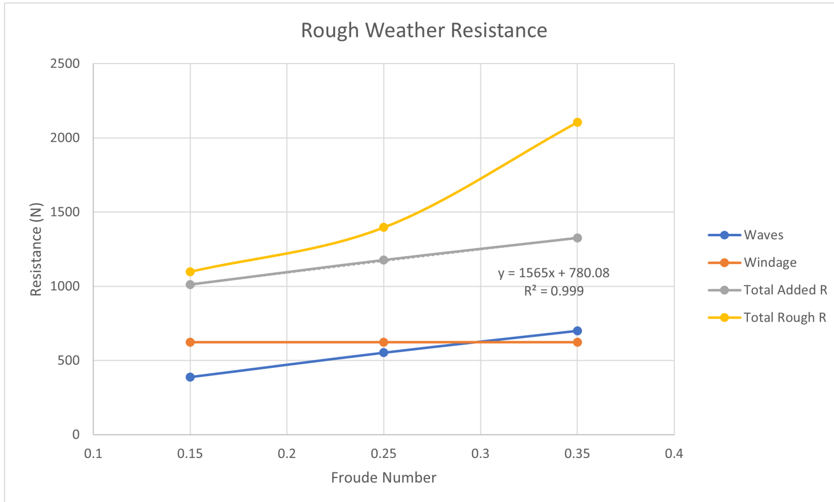


Figure 46: Rough Weather Resistance
Added resistance calculated for 30 knot apparent wind speed, 3 m significant wave hieght of 5 second period.

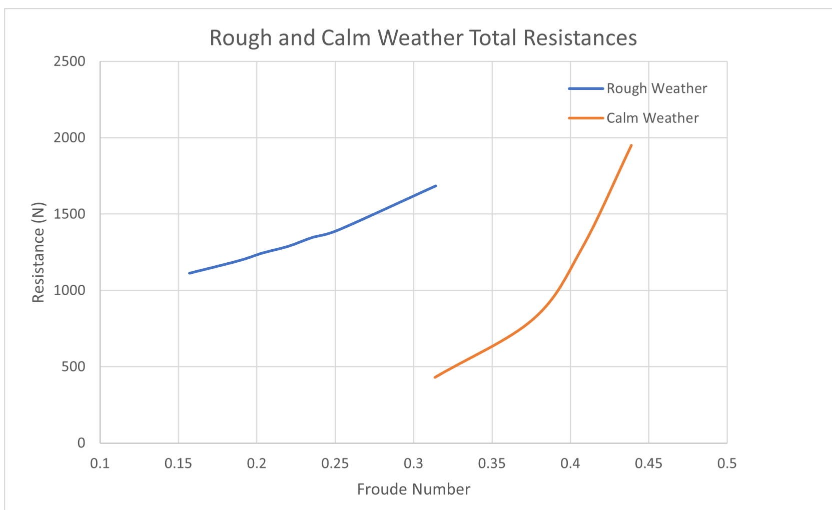
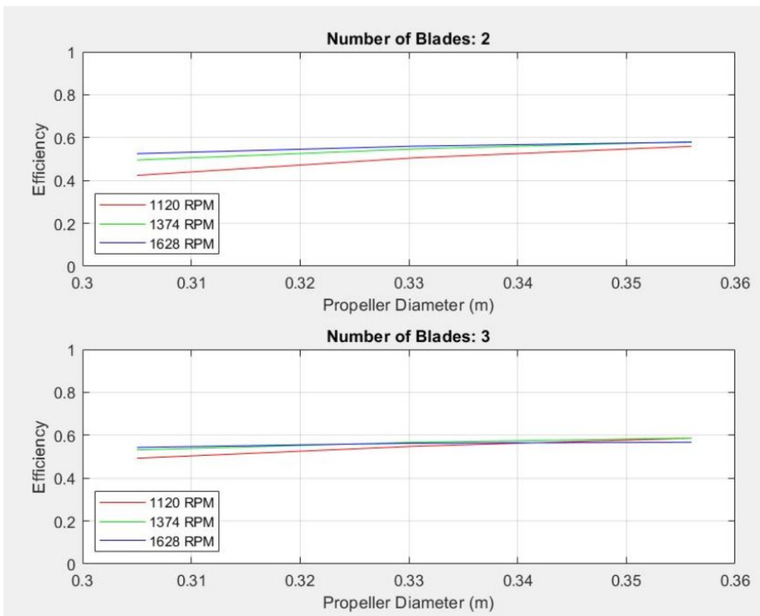


Figure 47: Rough and Calm Weather Resistance in Speed Regimes of Interest

7.3 Propeller Design

The estimated resistance of the yacht was used to design and analyze a propeller. The open-source software OpenPROP[53] was used to perform a parametric sweep across propeller speed, propeller diameter, and blade number, then to develop and analyze the design of the optimal variant within the sweep. The parametric study and propeller design were conducted for a ship speed of 6.5 knots and the corresponding resistance of 1281 N. This speed approximately corresponds to the critical Froude number of 0.4, at which the wave generated by the vessel motion is of the same length as the vessel. For the analysis, both the thrust deduction factor and wake factor were neglected: the propeller thrust was assumed equal to the yacht resistance, and the advance velocity of the propeller equal to the yacht velocity. These effects are expected to be small for a sailing yacht with propeller mounted below the hull and forward of the stern.[5]. The design was developed using a NACA a=0.8 meanline, NACA 65A010 thickness distribution, and the chord distribution displayed in figure 48. A 2.5 cm hub diameter was assumed, based on the required shaft diameter.

The selected engine has a rated speed of 3600 Revolutions per Minute (RPM) at full rated power, and the option of three different reduction gear ratios. The three resulting shaft speeds were used as inputs in the parametric study. Propeller diameter was constrained by clearance from the shaft to the hull underbody. The minimum clearance from the propeller tip to the hull was set to 20% of the propeller diameter, as a typical minimum value used in merchant vessel design.[54] The propeller hub location was constrained by the engine mount angle and the location of the rudder skeg. Given the hub location and the required clearance, the maximum size propeller that could be fit was 14 inches, or 0.356 m, in diameter. Propeller diameters of 12 and 13 inches were also included in the parametric sweep. Both 2-bladed and 3 bladed propellers were considered. The results of the parametric sweep are displayed in figure 48.



r/R	c/D
0.2	0.16
0.3	0.1812
0.4	0.2024
0.5	0.2196
0.6	0.2305
0.7	0.2311
0.8	0.2173
0.9	0.1807
0.95	0.1388
1	0.001

Figure 48: Propeller Parametric Sweep and Chord Distribution

The highest diameter propeller is most efficient, and the efficiencies of the low and medium speed 3 bladed propellers are fractionally higher than the 2 bladed propellers. More significantly, for equal chord distributions, the 3 bladed propeller has a greater expanded area and thus may be expected to have better performance in regards to the onset of cavitation. A 0.356 m diameter, 3 bladed, 1374 rpm propeller was selected for further design. The resulting propeller geometry is displayed in figure 49, and the performance curves in figure 50.

At the design condition, the propeller operates at an advance ratio, $J_s = 0.41$, and has an efficiency of $\eta = 58.7\%$. A blade cross section near the r/R=0.7 position was evaluated in XFOIL to ensure no cavitation. In order to prevent cavitation, the negative of the pressure coefficient, $-C_P$, must remain below the local cavitation number, σ.

$$C_P = \frac{P - P_\infty}{0.5\rho V^2} \tag{48}$$

$$\sigma = \frac{P_\infty - Pv}{0.5\rho V^2} \tag{49}$$

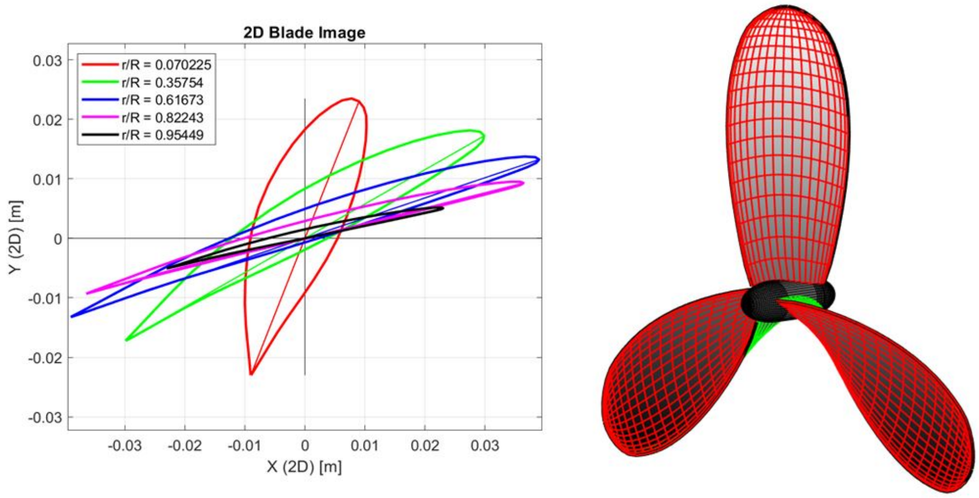


Figure 49: Propeller Geometry

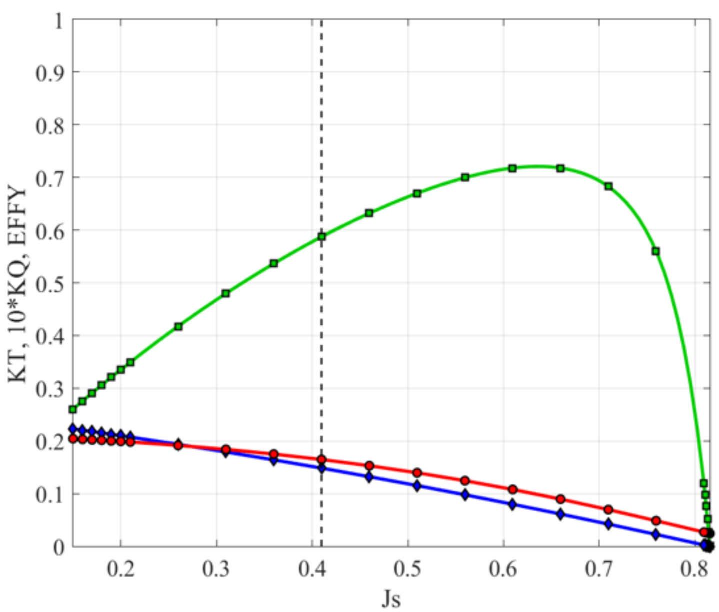


Figure 50: Propeller Performance Curves

where P_∞ is the ambient pressure in the absence of foil motion, and depends only the hydrostatic head at the depth of the foil, and P is the pressure experienced by the foil in fluid flowing with velocity V. P_v is the vapor pressure of seawater and ρ is the density of seawater.

In order to analyze the foil cross section in XFOIL and calculate the cavitation number, cross section geometry at r/R=0.7277 was taken directly from OpenProp output at this radial position. The angle of attack was calculated by subtracting the local hydrodynamic pitch angle, β_i, from the local geometric pitch angle θ_p. The former is the angle made by the axial and tangential velocities at the foil, with induced velocities in both directions included, while the latter is the angle made by the blade. The output for β_i and V was given by OpenProp at slightly different radial positions than the foil cross section geometry, so these values were calculated at the correct radial position by fitting curves to the output data. Foil data used for the analysis is summarized in table 45.

Foil Cavitation Analysis			
Radial Position	r/R	0.7277	
Depth below WL	d	0.26	m
Vapor Pressure	P_v	2300	N/m^2
Apparent Velocity	V	18.86	m/s
Cavitation Number	σ	0.5571	
Chord Length	c	8.160	cm
Reynold's Number	Re	1.45e6	
Angle of Attack	α	0.3453°	

Table 45: Propeller Section Cavitation Analysis

The cavitation number at the depth corresponding to the foil section of interest was calculated to be 0.5571. The pressure coefficient plot of the foil, displayed in figure 51, demonstrates that the foil in the design condition has substantial margin to cavitation.

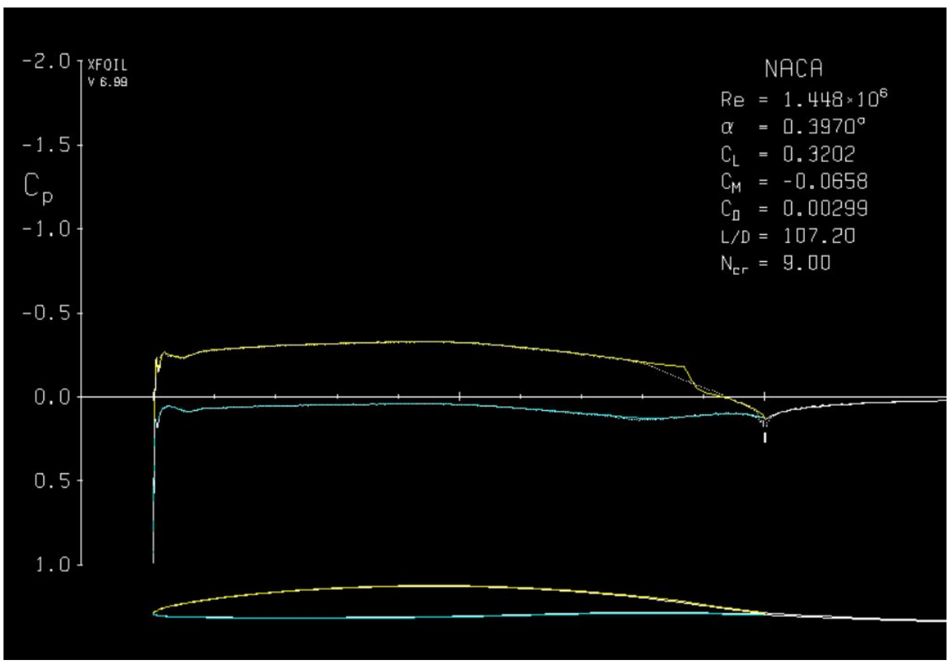


Figure 51: Pressure Coefficient Plot of Propeller 2D Section

7.4 Off-Design Propulsion Analysis

The resistance estimates in calm and rough weather were used, together with the propeller performance curves and manufacturer-supplied data on the engine, to analyze the propulsion system in off-design conditions.

The operating points at off-design conditions were determined by drawing ship thrust coefficient curves and finding the intersection points with the propeller thrust coefficient curve. This intersection point determines the advance ratio, torque coefficient, and efficiency at the off-design speed and thrust, which can then be used to determine the propeller delivered power. The ship thrust coefficient, $K_{T,ship}$, may be expressed for a known combination of thrust and advance velocity:

$$K_{T,ship} = \frac{T}{\rho * D^2 * V_A^2} * J^2 = c_i * J^2 \tag{50}$$

were D is the propeller diameter, and c is a constant for a specified off-design condition. The advance ratio J is the advance velocity made non-dimensional with the propeller rotation speed and propeller diameter:

$$J = \frac{V_A}{n * D} \tag{51}$$

As before, wake and thrust deduction effects are neglected, so that thrust, T, is assumed equal to the total resistance at the given ship velocity, which is assumed to equal the advance velocity, V_A, of the propeller.

The off design points selected for analysis are presented in table 46.

Weather Condition	**Speed (knts)**	**Resistance (N)**
Rough	3.25	1246
Rough	3.50	1289
Rough	3.75	1348
Calm	5.00	430
Calm	6.00	812

[1] Rough weather is defined as previously: 3m significant waves of 5 sec period, 30 knot apparent winds.

Table 46: Off Design Propulsion Points

The resulting $K_{t,ship}$ curves for these points are displayed in figure 52, with the design operating point shown as a vertical line at J=0.41.

From the intersection of these curves with the propeller thrust coefficient curve, the off-design efficiencies, advance ratios, and torque coefficients may be determined. The propeller delivered power, P_D, and propeller speed may then be calculated. The selected gear reduction ratio of 2.62:1 can be used to convert the propeller speed to engine speed.

$$P_D = \eta * T * V_A = \eta * R * V \tag{52}$$

$$n_p = \frac{V}{J * D} \tag{53}$$

The delivered power and engine speed at the design and off design conditions are displayed in table 47.

Condition	P_D **(kW)**	**Engine Speed (rpm)**
3.25 R	5.3	3076
3.50 R	5.7	3154
3.75 R	6.2	3228
5.00 C	1.7	2300
6.00 C	4.0	3008
6.50 C	7.4	3604

Table 47: Delivered Power and Engine Speed at Design and Off Design Conditions

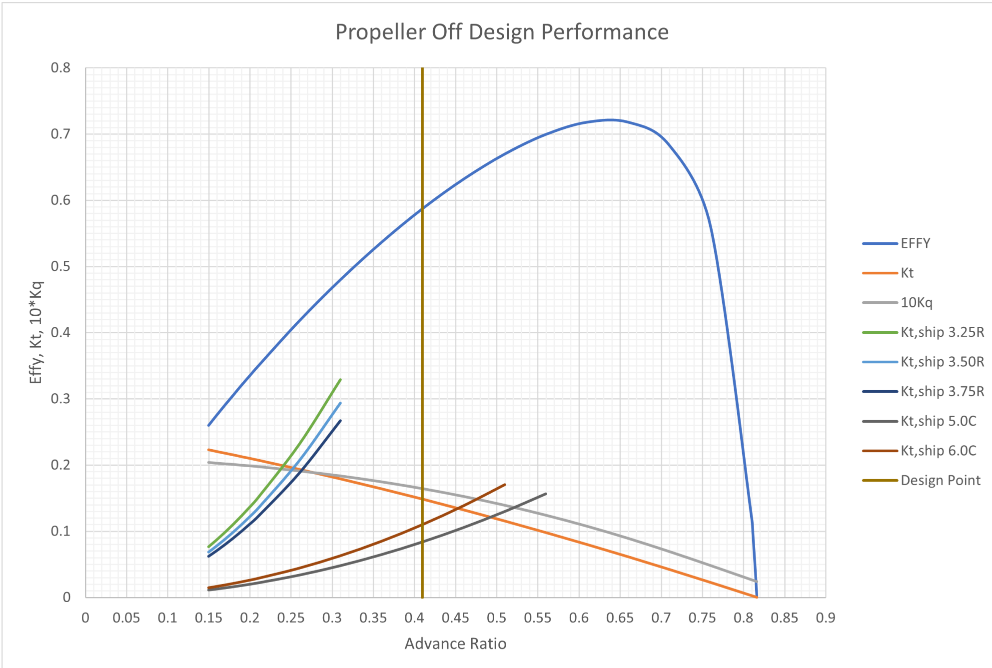


Figure 52: Propeller Curves at Off-Design Operating Points

The power curves in rough and calm water are plotted superimposed over the engine operating envelope in figure 53. The operating envelope was derived from manufacturer data, and includes the expected power loss through the gear and shaft.[52]

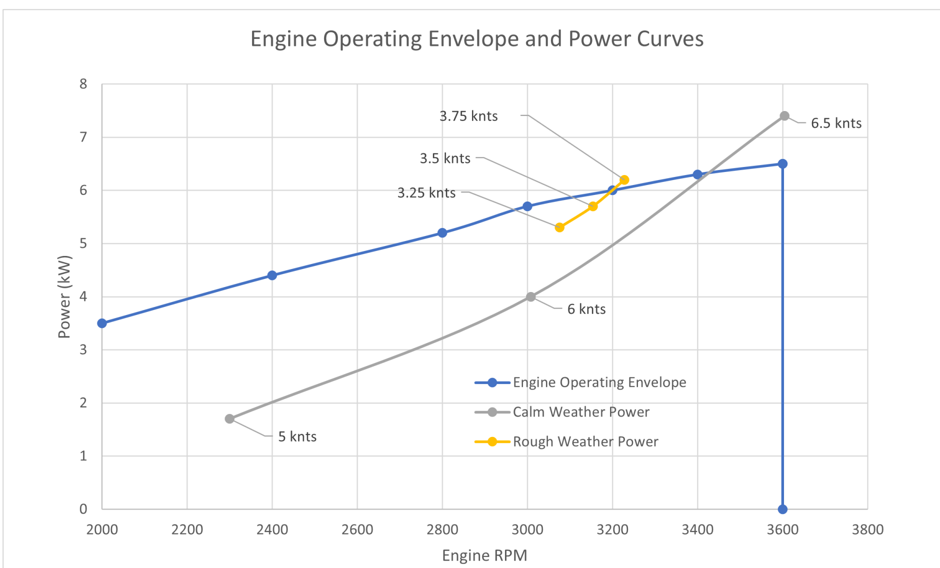


Figure 53: Engine Operating Envelope and Power Curves for Rough and Calm Weather

As evident from the figure, the engine and propeller combination are capable of pushing the vessel along at at over 6 knots in calm weather, and around 3.5 knots in the assumed rough weather conditions analyzed. However, they are not quite capable of driving the yacht at the original design speed of 6.5 knots, which was selected as the top speed based on an initial estimate of efficiencies. Of more concern is that the power curves for both calm and rough weather fall to the left of the maximum power corner, which means that the engine is not capable of reaching its maximum rated power. Any increase in resistance, from hull fouling for instance, or more severe weather, will shift the power curves further to the right and further limit the power output. A more efficient propeller would help with this, which might result from selecting a different design condition and iterating. However, with the diameter already at the maximum, and propeller speed constrained by available reduction gear ratios, it is unlikely enough gains in efficiency could be made through further attempts at optimization. Ideally, a more powerful engine would be fitted, and consideration was given to this. However, the additional space and weight requirements made this option unattractive. Furthermore, the above analysis did not consider the option of 'motorsailing.' In heavy weather near navigational hazards, if the captain felt the need to utilise the engine, it could be unloaded somewhat by the thrust from a reefed sail employed at the same time. Ultimately, given the intended purpose of the design, this was the preferred option and the original engine selection was unchanged.

A powered range estimation was conducted using the engine power and speed results for calm weather and fuel consumption data provided by the manufacturer.[52] Results for assumed tankages of 8 and 10 gallons are displayed in table 48, and indicate that even with a relatively small tankage of 8 gallons, a range in excess of 100 nautical miles at 6 knots speed may be achieved with the fuel efficiency of the selected engine. The 6 knot figures are more reasonable to apply, since it is generally hard on a diesel engine to run it at an excessively light load for long periods of time. Additionally, allowances must be made for residual fuel that can't be sucked from the tank, and for any differences between the manufacturer supplied fuel consumption data and real world conditions. Nonetheless, for a design not intended for long powered passages, a range under power of around 100 nautical miles is satisfactory.

Speed (kts)	P_D **(kW)**	**RPM**	**GPH Fuel**	**Hours**		**Range (nm)**	
				8GAL	10GAL	8GAL	10GAL
5	1.7	2300	0.2	40	50	200	250
6	4	3008	0.36	22.2	27.8	133.3	166.7

Table 48: Fuels Endurance and Range

The cavitation performance of the propeller in the off-design conditions was analyzed using the Burril method, which computes minimum developed area ratio of the propeller to prevent cavitation of a propeller operating with cavitation number σ and delivering a given thrust at a specified advance velocity. [5] The computation depends on the propeller pitch to diameter, which OpenProp computes for each radial control position. In the present design, it varies slightly along the propeller blade radius, but for most of the blade is close to 0.66, so this value was used in the calculations. Additionally, it was assumed that the developed area ratio was approximately equal to the expanded area ratio. The expanded area ratio of the present design is 0.3395. The minimum required area ratios according to the Burril method are displayed in table 49 for each condition considered.

Condition	**Minimum Area Ratio**
3.25 R	0.3347
3.50 R	0.3255
3.75 R	0.3168
5.00 C	0.1505
6.00 C	0.1760

Table 49: Burril Method Cavitation Check of Off-Design Conditions

The minimum area ratios for all conditions analyzed are below the propeller's expanded area ratio of 0.3395. There is no concern of cavitation in the calm weather scenarios. However, the required minimum area ratios in rough weather are quite close to the design ratio. Given the uncertainties in the assumptions and analysis method, this may be cause for concern. Again, however, the combination of the engine and a reefed sail may assist, by unloading some of the thrust required from the propeller. Alternatively, the propeller chord distribution may be altered to provide wider blades that better spread the load, increasing margin to cavitation by distributing the lift over a wider area and reducing the suction peaks on the blades.

Further optimization of the designed propeller was not conducted because, realistically, for a vessel of this design and purpose it is considerably more cost effective to purchase a commercially available propeller. This is particularly true given that high propulsive efficiency is not a vital design parameter for vessel primarily intended to sail. The preceding design and analysis was conducted primarily to assess the engine adequacy in off design conditions, to estimate a powered range for assumed tankage, and to generate recommendations for commercial propeller acquisition, which was necessitated due to lack of access to propeller geometry data or performance curves of commercially available sailing yacht propellers. Such a propeller should be 14" in diameter, have a P/D of between 0.6 and 0.7, and an expanded area ratio of above 0.34. Although not modeled above, the propeller should also be the folding type. Folding propellers create far less drag on the yacht while under sail relative to fixed propellers: a fixed propeller, even when free to spin, can add more than twice the drag of the equivalent folding propeller.[5] This is a significant consideration when considering lengthy passages of days or weeks.

8 Arrangements and Systems

Arrangements were driven by the need for adequate space for tankage and stores, ergonomic considerations for crew seating and berthing, and habitability considerations for a minimum 3 week passage with two crew. Accommodations are provided only for two crew, due to space and weight considerations.

8.1 Living Facilities

A plan view of the yacht arrangements is displayed in figure 54. The companionway ladder descends to the main saloon. Port and starboard quarterberths are provided, each of 1.88 m or greater length and a minimum of 60 cm in width from the waist up. The berth heads extend into the saloon and double as seating. The starboard berth doubles as the settee, and the head of the port berth forms a seat adjacent to a fold down navigation table. Forward of the saloon, separated by a partial bulkhead, the galley is to port and the head to starboard. A hanging locker is provided opposite the head. The coachroof provides just over 6 feet of standing headroom in the saloon and galley.

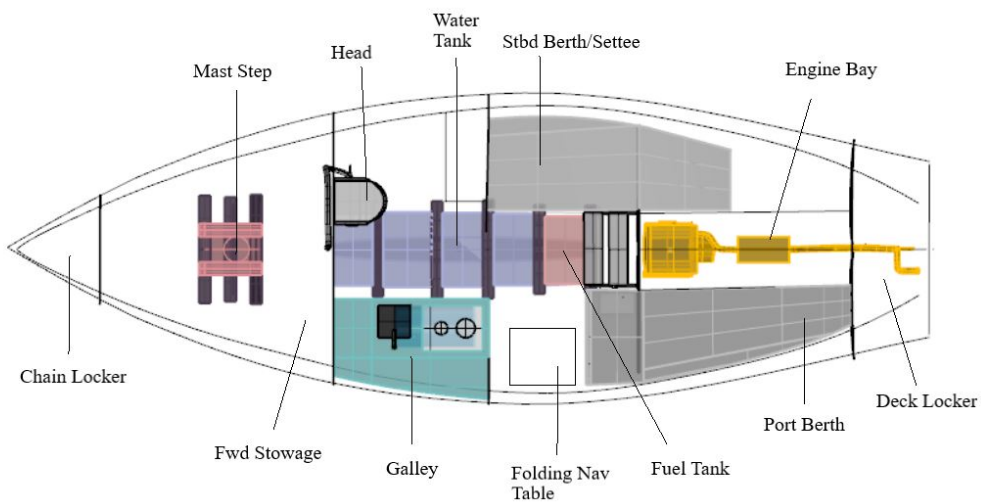


Figure 54: Yacht Arrangements

The galley is a simple affair, consisting of a 2 burner propane stove, and a 10" square sink with faucet, mounted on a counter. Stove dimensions and weight were taken from vendor data of a commercially available model. A locker underneath the stove provides storage for a standard sized 33 lb propane tank, which is estimated to provided 18 weeks of cooking service for 2 crew.[2] Stowage is also provided for cookware and utensils in the galley cabinet. The sink receives water from a foot-powered manual pump connected to the in-service water tank, and drains overboard through a seacock. The galley is located close to the lateral center of flotation through which the pitch axis passes, to minimize the effect of pitching motions on food preparation.

The head consists of a composting marine toilet. This eliminates the necessity for a holding tank, reducing maintenance burden and eliminating the need for port calls to pump out facilities. It also eliminates an additional thru-hull and the accompanying flooding risk, while saving both weight and space. The dimensions and weight of the head are based on those of an available commercial model. The composting chamber is vented through a hose and 12v fan to weather. There is adequate clearance for a 6' tall person to sit upright on the head comfortably. The head is screened by a nonstructural fore and aft divider. The head is oriented fore-and-aft and positioned close to centerline to minimize the effects of roll motion.

8.2 Tankage and Stowage

The forward compartment is dedicated to stowage. It is typical for yachts of this size to place a V-berth here. However, such an arrangement makes a poor sea berth and reduces valuable storage space on a cruising vessel, while also adding weight. There is also dedicated storage underneath both berths and aft of the starboard berth in the engine compartment. Deck gear such as life jackets, fenders, or mooring lines may be stored in the transom void, accessed by a watertight hatch from the cockpit. The chain locker in the bow stores the anchor and rode.

All tankage is located centerline, in the void spacing between keel floors, and near the lateral center of flotation. This arrangement keeps weight low for stability considerations, prevents any list or trim effects from unfilled tanks, and minimizes free surface effects. A fuel tank located immediately forward of the engine bay provides 8 gallons of diesel, giving an approximate range of 130 nm in calm weather. Four water tanks are located underneath the cabin sole. The tanks provide segregated storage of 30 gallons of fresh water, adequate for two crew to cruise 30 days. Segregating the fresh water ensures that contamination of a single tank does not put the crew into extremis. Each tank is fitted with a connection for the manual pump, so that the in service tank may be swapped after a tank is emptied.

8.3 Engine Bay

The engine is mounted below the cockpit on the reinforced engine bed stringers, by means of 6 mm steel angles through-bolted to the stringers with 10 mm bolts.[24],[2] The engine was mounted at a 7° angle to the horizontal, in order to give adequate clearance to the hull bottom, cockpit bottom, and forward engine bay bulkhead. The mount angle was also influenced by the need to achieve adequate propeller depth. Higher mount angles provide a deeper propeller, which is beneficial for cavitation considerations, but also reduce the forward driving thrust the engine is capable of supplying. The selected mount angle is well below the manufacturer specified limit of 15°. The engine position and mount angle necessitated locally increasing the height of the engine bed stringers from the required scantling minimum to 19 cm.

The selected engine is an electric start with manual handcrank backup. The handcrank is removable and mounts to the forward face of the engine. In order to access the engine for manual starts, a hatch is provided in the engine bay bulkhead. Access to the engine for maintenance and repairs is via either berth, through access hatches in the longitudinal engine bay bulkheads that separate the berths from the engine bay.

The engine is fitted with a waterlift muffler, that uses exhaust gas pressure to discharge cooling seawater through a shared port. Figure 55 displays the exhaust system, as well as the engine mounting position. Minimum clearance is at 2 cm at the lower forward engine edge, to the forward engine bay bulkhead. Because the seawater discharge is close to the waterline, a vented loop extending into the void formed by the bridgedeck is added on the seawater discharge line prior to its connection to the mixing elbow. This prevents drawing a siphon from the seawater system that would flood the engine. The combined exhaust/seawater discharge line is 45 mm in diameter, and has a lift of 26 cm relative to the water box, to prevent water backflow into the engine at any pitch or heel condition. The waterlift exhaust box is mounted on top of the stuffing box that provides watertight shaft support as the shaft exits the hull. A strut bearing, through bolted to hull with a backing plate, provides additional support near the propeller. The seawater/exhaust discharge loops to a height of 540 mm above the waterline before discharging at an outlet port 150 mm above the waterline. This ensures that wave action at the stern does not draw a siphon that would flood the waterlift box and engine. The 540 mm is greater than the minimum recommended 300 mm.[2]. If the height from the waterlift box to the loop is too great, the engine encounters

Figure 55: Engine Mount and Exhaust Configuration

excessive backpressure. The 766 mm height here is below the recommended maximum of 914 mm.[2] Seawater cooling intake is via a seacock mounted below the reduction gear. A 50 mm diameter vent is mounted on the cabin roof and directed to the engine bay to deliver adequate air intake.

8.4 Electrical

Electrical requirements for the vessel were estimated to determine the 12VDC electrical system needs. Loads are limited to engine starting, navigation lights, cabin lights, a handheld VHF radio and GPS unit, and fans for the cabin and head vent. Load requirements are estimated in table 50 for an underway passage condition. Current draw estimates are based on vendor quoted values for commercially available equipment.

Load	Current (A)	Run Time/Day (h)	Daily Capacity (A-h)
Cabin Lights (3)	0.3	6	1.80
Running Lights	0.2	12	2.40
Stern Light	0.1	12	1.20
Cabin Fan	0.5	8	4.00
Head Fan	0.06	24	1.44
VHF/GPS	-	-	1.80
Total	-	-	**12.64**

[1] VHF/GPS capacity is based on recharging 1.8 Ah batteries once per day.

Table 50: Electrical Loads

Because the electrical requirements are minimal, a single battery was selected to serve dual purpose for engine starting and for running electrical loads with the engine off. The manual start backup for the engine mitigates the risk of inadvertently discharging the battery too deeply to start the engine, and using a single battery saves both weight and space. A commercially available, 78 Amp-hour (A-h) capacity dual purpose marine 12V battery was selected. Given the electrical load requirements, it would take slightly more than 3 days to discharge the battery to 50%.

The battery charge is maintained primarily through a 100 W 12V solar panel and regulator. The panel rated optimal current output is 5.3A. Assuming 4 hours of sun exposure per day and a charging efficiency of 70%, the panel is capable of delivering 14.8 A-h per day, 17% more than the daily draw on the battery from the loads. The 3 day time to discharge 50% provides some buffer for days of poor sun exposure, and the engine alternator provides a back up means of recharging the battery. Assuming the same 70% charging efficiency, the 40 A alternator could replace one day's electric use by running the engine for slightly less than 30 minutes, which would consume about 0.25 gal of fuel. By carrying spare batteries for the radio and GPS, and backup, battery powered navigation lights as part of the emergency equipment, the risk of running out of power for essential equipment is virtually eliminated. Solar panel and alternator outputs are summarized in table 51.

Solar Panel Output	
I (A)	5.3
η_{charge}	0.7
Sun Hours	4
A-H/Day	14.8

Alternator Output	
I (A)	40
η_{charge}	0.7
12.6 Ah Recharge (min)	27
Recharge Fuel Burn (gal)	0.25

Table 51: Output of Solar Panel and Alternator

9 Model Testing

A 1:7 scale model of the vessel was constructed to measure the upright resistance of the yacht in a tow tank. A secondary objective was to explore the feasibility of home-scale 3D printing as a rapid and inexpensive means of constructing experimental models.

9.1 Model Construction

Larger models typically result in more accurate measurements, in part since they can be manufactured with greater accuracy and also result in larger forces to measure. However, the model size was limited to less than the tank depth or half the tank width, to prevent introducing interference from the tank walls or bottom.[55] The towing tank used for measurements is 100 feet long, 8 feet wide, and 4 feet deep, putting an upper limit of 4 ft or 1.22 m on the model length. The model also had to be large enough to support the weight of the force sensor and attachment hardware at the DWL. A final consideration for scale selection was based on manufacturing criteria: the print volume of the 3D printer used was a 22 cm by 22 cm square print bed and a maximum print height of 25 cm. To build a reasonably sized model required printing half-sections of the hull and joining them together. The model scale was chosen such that the width of these half sections did not exceed the print bed dimensions, to minimize the number of joints in the final model. A 1:7 scale of the model was selected as producing the largest size model that fell below the limits imposed by the towing tank dimensions, and could be constructed without more than one longitudinal joint. The model did not include the rudder, but did include the keel, since it is not a 'typical' high aspect ratio foil and since it is designed molded into the hull.

The CAD model of the vessel was adapted for printing. First, the model was trimmed below the sheerline to reduce the required print material and time, since the topsides were not needed for resistance testing. Adequate freeboard was left to enable model testing at heel angles up to 25° without submerging the gunwale on the heeled side. Then, the model was scaled to the correct size, and thickness was added to the hull surface. The hull surface was given 15 mm thickness, to ensure adequate strength of the finished model, to provide a suitable surface area for gluing watertight joints, and to prevent warping issues from attempting to print overly thin sections. A level floor was modeled at the LCF for attachment of the free pitch plate to which the force sensor connects. Finally, the model was split into sections, and connection pieces, consisting of pegs and holes, were modeled where sections join, to help guide assembly and glue up. Each section was exported as a stereolithography (STL) file to an open source slicer program, CURA, which converted the STL files into machine code that can be interpreted by the printer. The slicer program also allows user control over printer settings. Figure 56 depicts an exploded view of the sections for the port half of the model. Note that the bow and keel sections did not require longitudinal splitting.

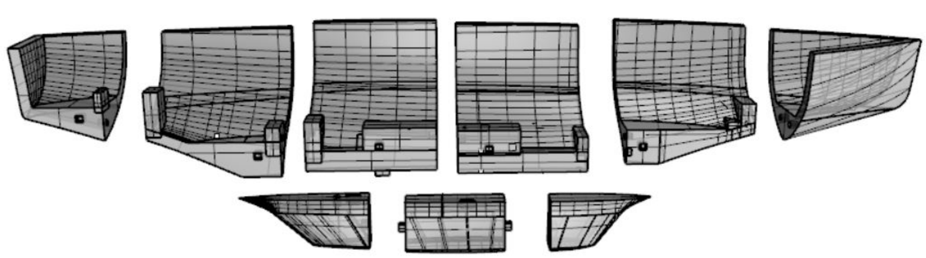


Figure 56: Model Print Sections - Port Side

The model sections were printed using a Creality Ender 3 Pro FDM home printer. This is a relatively inexpensive, popular hobbyist printer that retails for less than $250 USD. PLA was used as the print medium, which retails for less than $25 USD per kilogram. The hull model was printed in 11 sections, with an additional 3 sections required for the keel. Figure 57 displays the printing of a keel section and the bow section. The printed parts are built up in layers by deposition of heated PLA, and consist of an outer shell supported by an internal cubic framework of walls. The key printer settings selected pertain to infill percentage, outer wall thickness, nozzle size, layer height, print speed, print nozzle and bed temperatures, and adhesion support.

Infill percentage refers to the fraction of interior part volume filled with support material; higher infill percentage results in stronger, more durable parts that are also heavier and require longer print times and more material. Exterior wall thickness refers to the thickness of the shell outer walls; higher thickness increases weight, print time, and material usage, but increases part strength and resistance to puncture. In order to provide adequate strength at reasonable print times and for a reasonable model weight, 10% infill was set with a wall thickness of 1.2 mm. The total weight of all parts was checked before printing to ensure the finished model had sufficient margin to carry the weight of the force sensor and attachment hardware while floating at the DWL. Nozzle size and layer height both represent trade-offs between print speed and model quality. Larger nozzles and layer heights significantly reduce the time required to print a part, but result in lower quality surface finish and less resolution on model details. Due to the scale of the model, most of the printed sections were large pieces of relatively gradual curvature that did not require high detail resolution, so a large nozzle size of 0.8 mm and layer height of 0.32 mm were selected to reduce print times. However, because of the reduced resolution, this required designing the connection tabs between sections with relatively loose tolerances. Rather than tightly 'snapping' together adjoining sections, they acted more as guides during the glue up process. Printing temperatures are material dependant; for PLA on this machine, a 200°C nozzle temperature and 60°C bed plate temperature worked well. Print speed should be set as high as possible without causing print failure; 50 mm/sec was selected for all parts. To ensure adhesion of the part base to the printer bed and prevent warping of the part, all parts were printed on a 'raft', a thick filament structure laid down on the print bed first and on top of which the base layer of the part is printed. While this increases print time and material wastage, it was necessary to consistently prevent print failures from base layer separation or base warping. Printer settings are summarized in table 52. Figure 58 displays the printed sections prior to assembly.

Infill	10 %
Wall Thickness	1.2 mm
Nozzle Size	0.8 mm
Layer Height	0.32 mm
Nozzle Temp	200°C
Bed Temp	60°C
Print Speed	50 mm/sec
Bed Adhesion	Raft

Table 52: Printer Settings

The printed parts were glued together in stages with waterproof adhesives to assemble the model. Where surfaces fit together perfectly flush, a rapidly setting two-part cyanoacrylic plastics adhesive was used. However, some joints were found to have slight gaps due to small misalignments during glue up of a previous joint or an imperfectly flat surface. These joints were glued using a thicker layer of two part epoxy resin, and exterior surface imperfections at the joint were covered in epoxy putty and sanded down flush. The hull was then coated in a layer of two part epoxy resin for waterproofing and to fill in the surface ridges that result from FDM printing. Finally, two coats of high gloss oil paint were applied. Figure 59 displays the model assembled and painted. The surface was sanded with 800 grit sandpaper between coat applications.

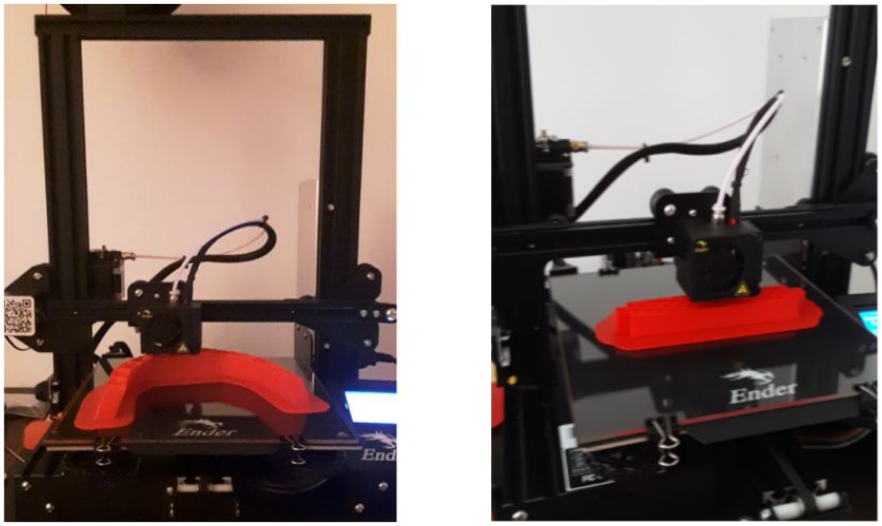


Figure 57: Printing Model Sections

Figure 58: Printed Model Sections

In addition to the model, adapters were designed and printed for the connecting hardware. The model testing was conducted with the model unconstrained in pitch or heave, by means of a free pitch plate and free heave bars. An adapter plate was printed for connecting the free pitch plate to the model. The plate was glued to the level floor of the hull model, and used four heat set threaded 3 mm inserts to enable bolting the free pitch connection plate to the model. Two adaptors were printed to make the connection between the free pitch plate, the single axis force sensor, and free heave bars. The adaptors are depicted in figure 60. Figure 61 displays the connection of the model to the sensor.

On the free pitch adaptor plate, four additional heat set inserts were provided to enable varying the heel angle of the model by bolting in printed angled shim blocks between the model and the free pitch plate, such as that depicted in figure 62. A full set of experimental runs on a sailing yacht would entail both upright towing tests, and towing tests at varying heel and yaw angles corresponding to realistic sailing conditions. The drag, side force, and yawing moment could then

Figure 59: Model Assembled and Painted

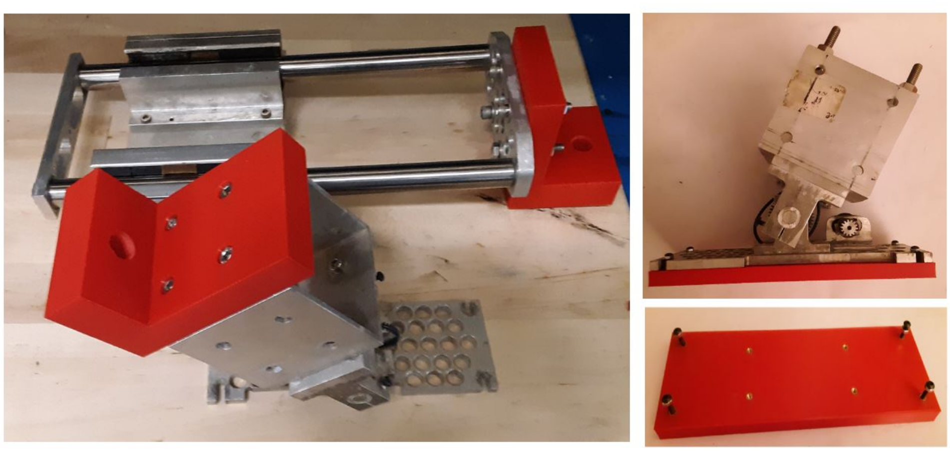


Figure 60: Adaptor Plates for Free Pitch, Free Heave, And Force Sensor Connection

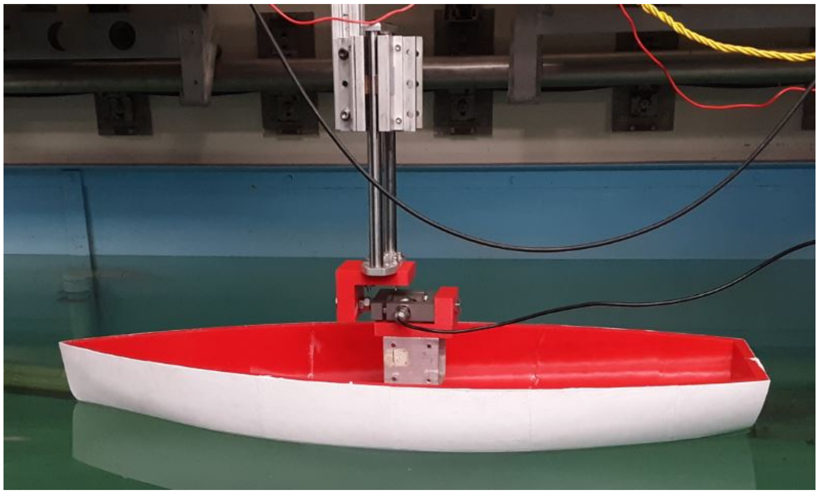


Figure 61: Model Connection to Force Sensor with Free Pitch, Free Heave

be used in a VPP to give predicted speed made good for different wind conditions. Unfortunately, the 6 axis force sensor with yaw position control around which the model and experiments were planned was broken before the experiments were conducted. With only one single-axis force sensor then available, only upright resistance was measured.

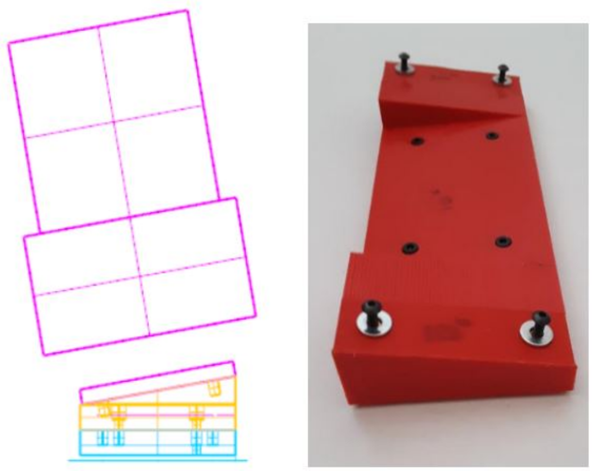


Figure 62: Shim Block for Heel Control

9.2 Experiment

The model upright resistance was measured at different tow speeds. The results were used to determine the upright resistance of the full size yacht, by first expressing the total measured model resistance, $R_{T,m}$ as a nondimensional total resistance coefficient using the fluid density, model speed, and wetted area, A_W:

$$C_T = \frac{R_T}{0.5 * \rho * A_w * V^2} \tag{54}$$

For both model and ship, the total resistance coefficient is considered the sum of a viscous component that depends only on Reynolds number, and a residuary component that depends only on

Froude number:

$$C_T = C_V(Fr) + C_R(Re) \tag{55}$$

The viscous coefficients for the model and the ship are calculated using the ITTC 1957 correlation line for frictional resistance coefficient, C_F, and a form factor, k, determined from the method of Prohaska during the experiment[55]:

$$C_V = (1+k) * \frac{0.075}{(log_{10}(Re) - 2)^2} = (1+k) * C_F \tag{56}$$

Froude scaling ensures that the residuary resistance coefficient of the model is equal to that of the full size yacht:

$$C_{R,m}(Fr) = C_{R,s}(Fr) \tag{57}$$

so that the ship total resistance coefficient may be determined based on model measurements. First the model total resistance coefficient is calculated from the measured resistance using eq (54). The model viscous coefficient determined from eq (56) is subtracted from this value to find the model residuary coefficient, which is equal to the ship residuary coefficient. To this, the full size yacht viscous coefficient is added to find the total resistance coefficient of the full sized yacht.

The model and full size yacht parameters used for the experiments are displayed in table 53, including the different densities and viscosities of the tank freshwater and seawater. In order to achieve the correct scaled displacement, the necessary amount of ballast was added in the form of bagged metal shot. These were taped into a position that ensured the model trim remained zero.

Parameter		**Yacht**	**Model**
L_{WL}	m	6.831	0.976
A_W	m^2	14.18	0.289
Δ	kg	3012.5	8.58
ρ	kg/m^3	1025.9	1002.4
μ	Ns/m^2	0.001090	0.001002

Table 53: Yacht and Model Parameters

The Froude numbers corresponding to full size yacht speeds of interest were computed, and used to compute the corresponding required model speeds. The model was then towed by the carriage at the appropriate speeds, with pauses after each run to ensure that waves fully dissipated before commencing the next run. The model resistance was measured using a single axis S-beam load cell of 20 kg capacity. The logged data for each run was processed in MATLAB using a butterworth filter to reject the noise in the measurements. The tested model speeds and measured model resistances are displayed in table 54, along with the corresponding full scale yacht speeds and resistances, computed following the procedure discussed above. For speeds where a repeat run was performed, the average of the runs is reported. All of the runs were well below the critical speed for tank depth h=1.22 m, $V = 0.7 * \sqrt{gh}$, at or above which shallow water effects skew the resistance measurements. [55]. Figures 63 shows two different speed runs of the model, with visible differences in wavemaking.

The form factor was determined by plotting the model ratio of total to frictional resistance coefficient, $\frac{C_{T,m}}{C_{F,m}}$ against the ratio of the Froude number raised to fourth power to the frictional resistance coefficent, $\frac{Fr^4}{C_{F,m}}$. A linear fit was taken, excluding the three lowest Froude number data points which were considered outliers. The y-intercept of this fit is equal to 1+k. The plot is displayed in figure 64. The form factor was determined as $k = 0.1431$.

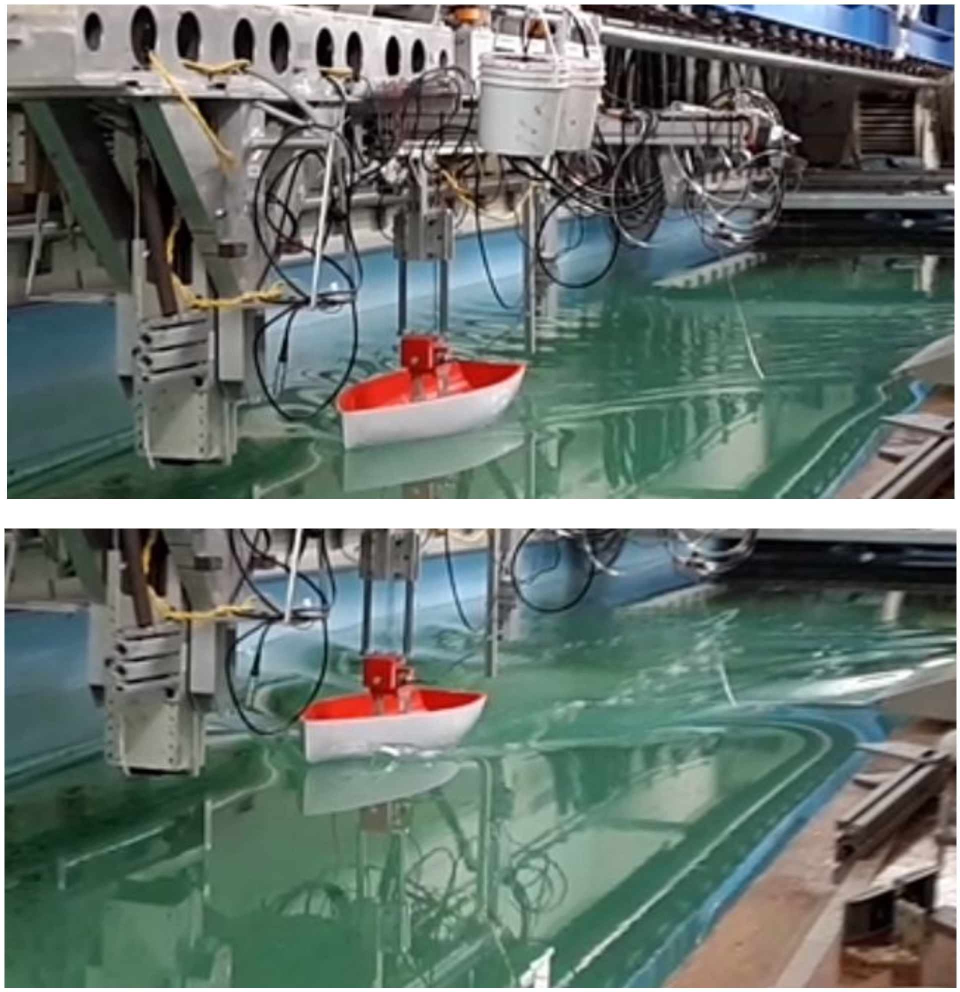


Figure 63: Model Runs

The upright full size yacht resistance is displayed in figure 65, along with the predicted resistance from the Delft series correlations. Figure 66 displays the resistance coefficients of the yacht, along with the Delft series predicted total resistance coefficient. Figure 67 display the Delft and experimental effective towing power curves, where effective towing power is related to the speed and resistance of the yacht by:

$$P_E = R_T * V_s \tag{58}$$

Froude Nr	V_m **(m/s)**	$R_{T,m}$ **(N)**	V_s **(knts)**	$R_{T,s}$ **(N)**
0.063	0.194	0.082	1.00	22.2
0.094	0.292	0.153	1.50	40.6
0.126	0.389	0.228	2.00	59.0
0.157	0.486	0.254	2.50	58.3
0.189	0.583	0.309	3.00	66.3
0.220	0.680	0.448	3.50	103.0
0.236	0.729	0.470	3.75	103.9
0.251	0.778	0.573	4.00	132.7
0.314	0.972	1.025	5.00	260.0
0.377	1.167	2.081	6.00	595.2
0.440	1.361	4.376	7.00	1358.4
0.503	1.555	7.621	8.00	2451.7

Table 54: Model Resistance Measurements

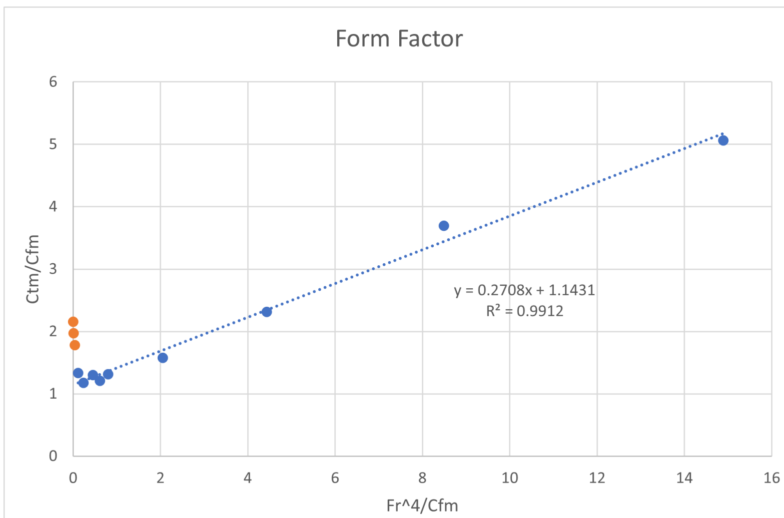


Figure 64: Form Factor Plot

9.3 Discussion

The measured values demonstrate a resistance below that predicted for similar hulls on the basis of the Delft series, indicating a relatively low resistance hull and keel. As evident from the figure, the experimental values fall considerably below those predicted. However, further testing is warranted to confirm these values. It is likely that laminar flow effects are present in the model measurements. Turbulence stimulators were not used in the experiment, because it was assumed that the roughness on the model surface from the fabrication method would be sufficient to trip turbulent flow. Given the results relative to predictions, this may not have been the case. It is likely that part of the discrepancy arises from an overly conservative assumption on the appendage form factor adopted when generating the Delft series resistance estimate. Finally, since the present design has a somewhat lower midships section coefficient than the range of hulls on which the Delft correlations are based, it may be that the correlations overestimate the yacht's resistance. In any case, re-performing the model runs with turbulence stimulators would develop a more accurate assessment of yacht resistance, and help ascertain to what degree the observed discrepancy is an artifact of laminar flow effects. This was not accomplished in the present work due to time

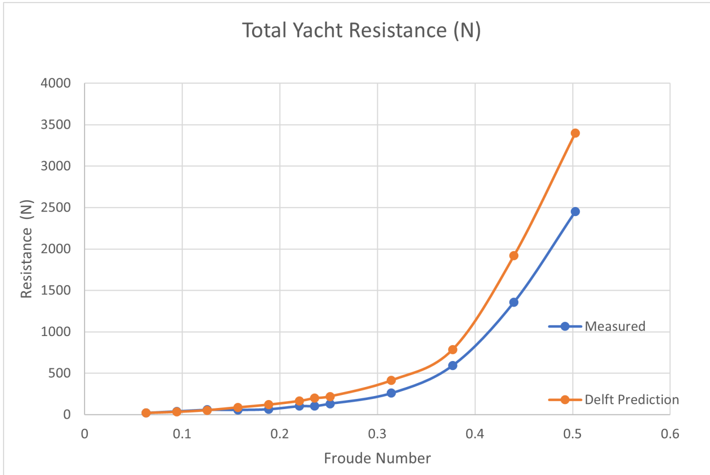


Figure 65: Measured and Predicted Yacht Resistance

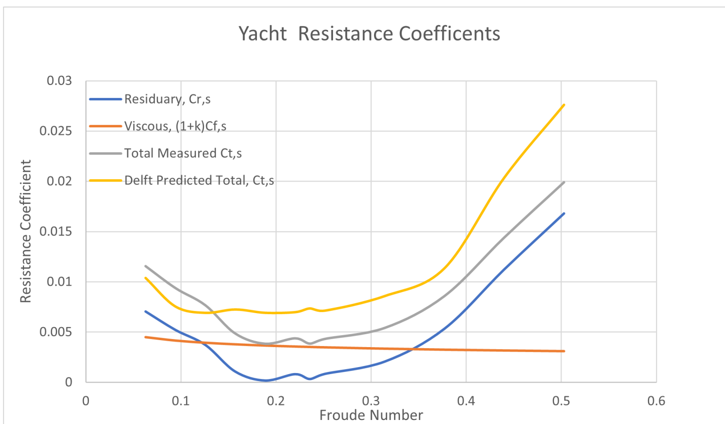


Figure 66: Measured and Predicted Yacht Resistance Coefficients

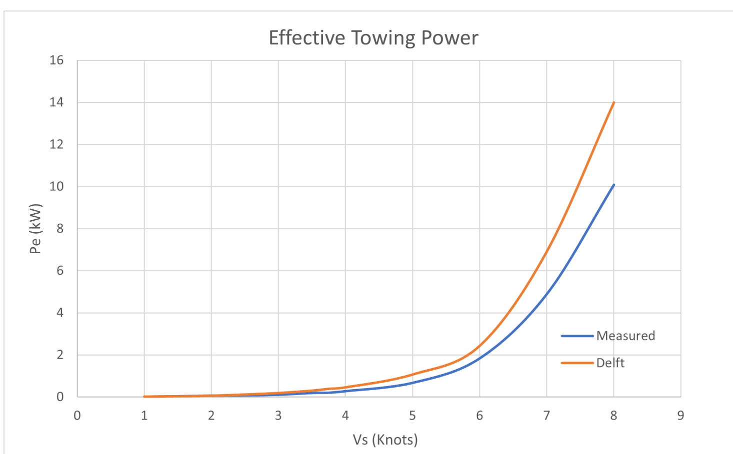


Figure 67: Effective Towing Power

constraints. However, if such testing confirms that resistance values are indeed lower than initially predicted, the propulsion system performance under limiting conditions would be better than predicted in the preceding analysis. At present, the more conservative predicted values are used in the propeller and engine analysis.

It was initially planned to sand the model surfaces to a high degree of smoothness, then add turbulence stimulators to the bow and the keel leading edge. However, while sanding, the author became concerned that this might excessively reduce the relatively thin hull wall, leading to inadvertent puncture of the skin. Instead, the ridges formed by the print layer height were filled by coats of epoxy and paint, which were sanded after application. Nonetheless, some surface roughness remained, so the stimulators were not included.

Overall, the use of 3D printing is deemed a feasible and attractive option for fabricating hull or appendage models for use in testing, though significant improvements are possible over the model here developed. Such a fabrication process requires no specialized machining, casting, or woodworking skills, and is well suited to complex hull or appendage forms. With the print settings employed, the model required less than 4 kg of PLA and a total print time of less than 140 hours, using a single, hobby-scale machine. There were no issues with water intrusion, and the resulting structure was strong enough and light enough to carry the test equipment for free pitch and free heave testing with several kilograms of margin to spare. Including the cost of the PLA filament and the adhesives and paint, the model total cost was under $200 USD. Because of the selected settings, the model did require a fair amount of post-processing, and had some imperfections at joints and surface roughness issues that had to be addressed. This was due partially to alignment issues during glue up, and partially to the 'ridges' formed on the sides from using a large nozzle and layer height. Both of these could be addressed in part by using higher quality print settings to ensure the parts 'snap' together tightly with flush surfaces. This would also result in a smoother surface that requires less sanding. A slightly higher wall thickness would enable sanding to any desired finish without concern for puncturing the model walls. The cost of these improvements are increased print time. However, using multiple printers in parallel provides one option to offset this cost, particularly with the rise of 3D printing facilities at university labs and shops.

10 Design Synthesis and Analysis

Although the preceding chapters have described the design progression in a somewhat linear fashion, the actual process was highly iterative. The final result for each major element of the design was established after multiple passes around the design spiral, until the overall design converged and satisfied requirements. The result is displayed in figure 68. Design particulars are summarized in table 55 for the minimum operating condition.

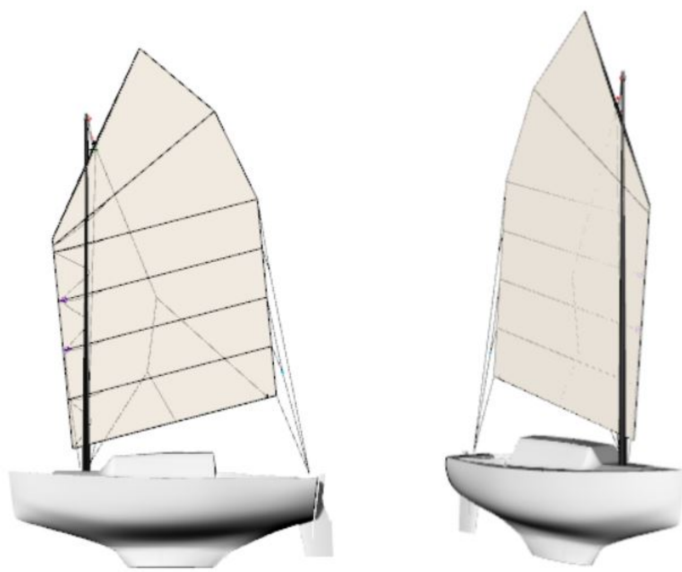


Figure 68: Converged Design

Yacht Particulars		
Δ_{MO}	2935	kgf
L_{OA}	7.56	m
L_H	7.20	m
L_{WL}	6.78	m
B_{OA}	2.44	m
B_{WL}	2.09	m
T	1.10	m
FB_{fwd}	1.15	m
FB_{aft}	1.0	m
SA	33.7	m^2
$C_{P,c}$	0.555	
$C_{M,c}$	0.575	
C_{WP}	0.633	

Table 55: Yacht Particulars, Minimum Operating Condition

10.1 Weights and Load Capacity

Weights and centers were critical to design requirements pertaining both to trailer capability and seaworthiness, so considerable effort was spent developing a thorough and accurate accounting of weights, presented in appendix F. The minimum operating condition is summarized in table 56.

Min Op Condition Weights	
Structural	1020.4
Topsides	197.3
Rig and Sail	115.5
Interior	174.2
Systems	205.9
Ballast	1221.7
Total Weight	2935 kgf
VCG	-0.10 m

Table 56: Minimum Operating Condition Weights

The minimum operating condition is limiting for stability and seaworthiness considerations, because the limiting design parameter is the ISO minimum righting energy for design category A. The additional load for the design cruising condition is 345 kg, resulting in a full load displacement of 3280 kg. The components are listed in table 57. Based on the provided storage, fuel and water tank locations, the net effect of the weight additions is to shift the vertical center of gravity 1 cm upward.

Full Load Displacement		
Crew Size	1	2
M_{MO}	2935	2935
Additional Crew	0	75
Crew Effects	20	40
Fresh Water	149.8	95
Fuel	25.4	25.4
Stores	149.8	109.6
Endurance	74 Days	25 Days
Total M_{FL}	3280 kg	
VCG	-0.09 m	

Table 57: Full Load Displacement Weight Allocations

The following assumptions are used in assessing weight allocations in the full load condition: crew weight of 75 kg/person, crew effects of 20 kg/person, fresh water needs of 0.5 gal/person per day, food needs of 2 kg/person per day, 8 gallon diesel fuel tank. Under these assumptions, a 345 kg weight capacity results in an endurance range of 25 days for a crew of 2, and 74 days for a crew of one.

The hull and mast structures were designed to a full load displacement of 3500 kg. The difference between this displacement and the design full load displacement represents a weight margin of 220 kgf, or 7% of design minimum operating weight, to account for differences in the as-built versus design weights and vertical center. This could enable adding some additional ballast if, for instance, the actual vertical center of gravity came out slightly higher than predicted. The margin could also ensure the ability to carry the designed load capacity of 345 kg if the as-built yacht turns out somewhat heavier than predicted. The amount of margin was set by trailer requirement constraints, such that even if all margin is used, the design will meet threshold requirement for trailer weight. The trailer weight condition is defined as the minimum operating condition, less the weight of the minimum crew and the weight of removable tools and spares. Weight conditions are summarized in table 58.

Weights, Capacity, and Margin	
DWL Condition	3045 kgf
Min Op Condition	2935 kgf
Trailer Condition	2830 kgf
Design Full Load	3280 kgf
Load Capacity	345 kgf
Maximum Load	3500 kgf
Design Margin	220 kgf

Table 58: Weight Conditions

10.2 Stability and Hydrostatics

The stability curve of the yacht in the minimum operating condition is displayed in figure 69. The station area curve is displayed in figure 70, and key stability parameters are summarized in table 59. A full printout of the hydrostatics data is contained in appendix E.

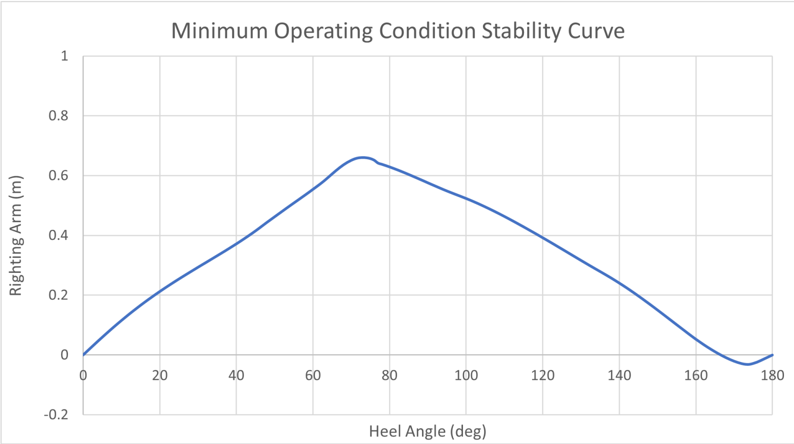


Figure 69: Min Op Condition Righting Arm Curve

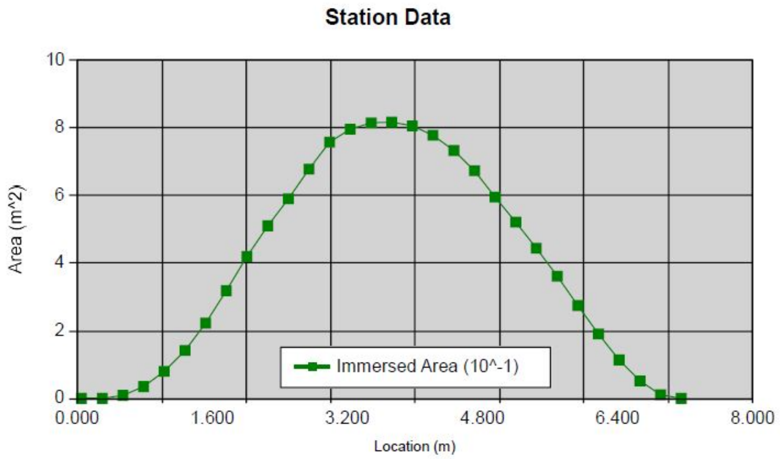


Figure 70: Station Immersed Area Curve

Minimum Operating Stability Parameters		
Parameter	Design Value	ISO 'A' Minimum
GM_T (m)	0.69 m	-
GZ_{MAX} (m)	0.66 m	-
ϕ_{GZMAX}	73°	-
ϕ_{DF}	100°	40°
ϕ_V	166°	124.1°
STIX	37.6	32
RE (kg-m-°)	176,592	172,000

Table 59: Stability Characteristics at Minimum Operating Condition

The ISO specifies a minimum Stability Index of 32, and a minimum righting energy of 172,000 $kg - m - deg$ for classification as design category A. Because the stability characteristics depend strongly on VCG, which is predicted here from a weights calculation, a sensitivity study was performed. Table 60 displays how STIX and RE change with different positions of VCG. All VCG positions are given relative to the position of the DWL. All calculations are performed at the minimum operating condition. The impact to stability curves is shown in figure 71.

VCG (m)	GM_T (m)	STIX	RE (kg-m-deg)
-0.1	0.69	37.6	176,592
-0.09	0.68	37.2	173,137
-0.08	0.67	36.5	169,838
-0.05	0.64	35.1	159,458
0.00	0.59	33.2	143,286

Table 60: STIX and RE Sensitivity to VCG Position

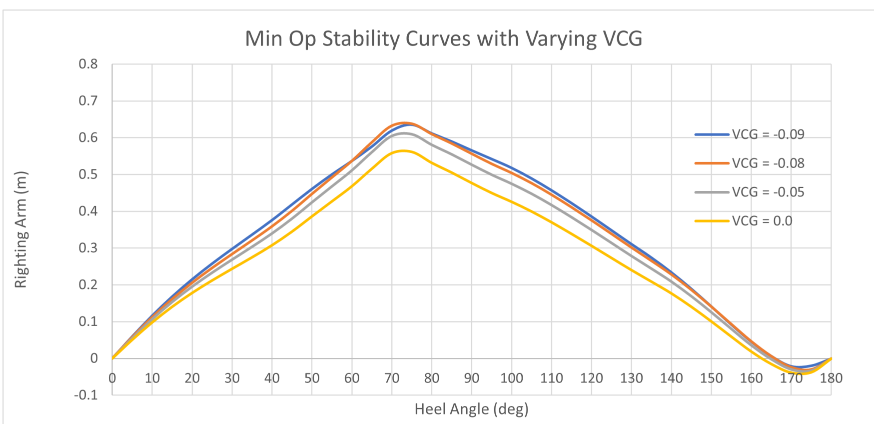


Figure 71: Min Op Righting Arm Curves with Varying VCG

The STIX remains above the design category A threshold for all VCG positions considered, up to a VCG at the DWL, 10 cm above the predicted position. The righting energy requirement for design category A is met for VCG positions of 9 cm below the DWL or lower. Righting energy requirements for design category B are met for VCG positions up to the DWL. A VCG any higher than the DWL would not be acceptable on the grounds of initial stability, since a metacentric height below about 0.6 m would result in an overly tender yacht for the given sail area.

If the fresh water tanks are considered to serve dual purpose as ballast tanks, the category A righting energy requirement may be satisfied with a VCG as high as 8 cm below the DWL (2 cm higher than calculated). This would require refilling each of the tanks with seawater after the freshwater had been exhausted. In place of four 7.5 gallon tanks, eight containers of 3.75 gallon capacity each could be stored in the tank locations. These containers would be small enough for a single person to refill and replace. With eight containers holding a total of 114 kg of water, refilling them in this fashion corresponds to an effective increase in ballast of at least 100 kg (since one tank at a time may be considered empty just before refilling). While this is not an ideal solution, it does provide some additional margin to ensure meeting objective design requirements. By designing the water tanks to be removable and portable, cleaning them thoroughly before refilling with fresh water at the next port should be a simple affair.

As evident from the stability curves, the angle of vanishing stability decreased only 4 degrees as the VCG was varied from 10 cm below the waterline to the waterline. This indicates that the geometry of the topsides and cabin roof plays an important role in the yacht's ultimate stability, similar to some self-righting life boat designs. The righting energy and metacentric height are more heavily impacted by VCG within the range considered, with metacentric height decreasing approximately 1 cm for each cm rise in VCG, as expected.

10.3 Velocity Prediction

The sailing performance of the yacht was predicted using gvpp, an open source velocity prediction program.[56] The results, displayed in figure 72, predict the yacht speed when sailing at different angles to true wind speeds ranging from 5 to 20 knots. The program utilizes the Hazen model for sail aerodynamics, and the Delft series correlations for yacht resistance. It accepts stability and geometry data for the hull and sailplan, and solves a constrained optimization problem to determine best yacht speed for true wind speed and heading, by enforcing equilibrium of heeling moment and righting moment and of sail driving force and total resistance force.[56]

The results are viewed only as a rough approximation of performance, subject to several sources of error. The Hazen model lift and drag coefficients pertain to the more typical Bermuda sail plans; the design sail plan may reasonably be expected to yield worse performance when sailing close-hauled, but better performance on a reach or off the wind. To roughly account for the difference in rig type in a conservative manner, the sail area input to the program was reduced by 10% relative to the design sail area, following a rule of thumb that conversion of a Bermuda rig to a junk should increase the latter's working sail area by about 10% to achieve good light air performance.[30] Another potential source of error lies in the fact that the design midship section coefficient, $C_M = 0.575$, falls slightly outside the the range of hulls contained in the Delft series, for which $C_M = 0.646$ is the lower bound.[5] This parameter is one input to calculating added resistance in heel. Finally, residuary resistance of the keel was neglected, as the calculation was returning nonphysical negative values, likely due to the program not handling very low aspect ratio keels. Neglecting keel residuary resistance should not affect the results too significantly, as this is a small contribution to total yacht resistance.[5]

Hull form factor was set to 1.14, based on the results of the model testing described above. Keel and rudder form factors were set to 1. Heel angle was constrained to 35 degrees maximum.

The results indicate that the yacht should be easily capable of exceeding the threshold passage making speed requirement of 4 knots under favourable conditions. In true winds of 10 knots speed, true wind angles between 47-133° result in yacht speeds of 5 knots or greater. If true winds are 15 knots, wind headings between 54-153° deliver 6 knots or greater speed. Even if the predictions are off by as much as 15%, cruising speeds in excess of 4.25 and 5.1 knots can be expected in 10 and 15 knot winds, respectively, for wind headings of interest to cruising yachts on established passage routes.

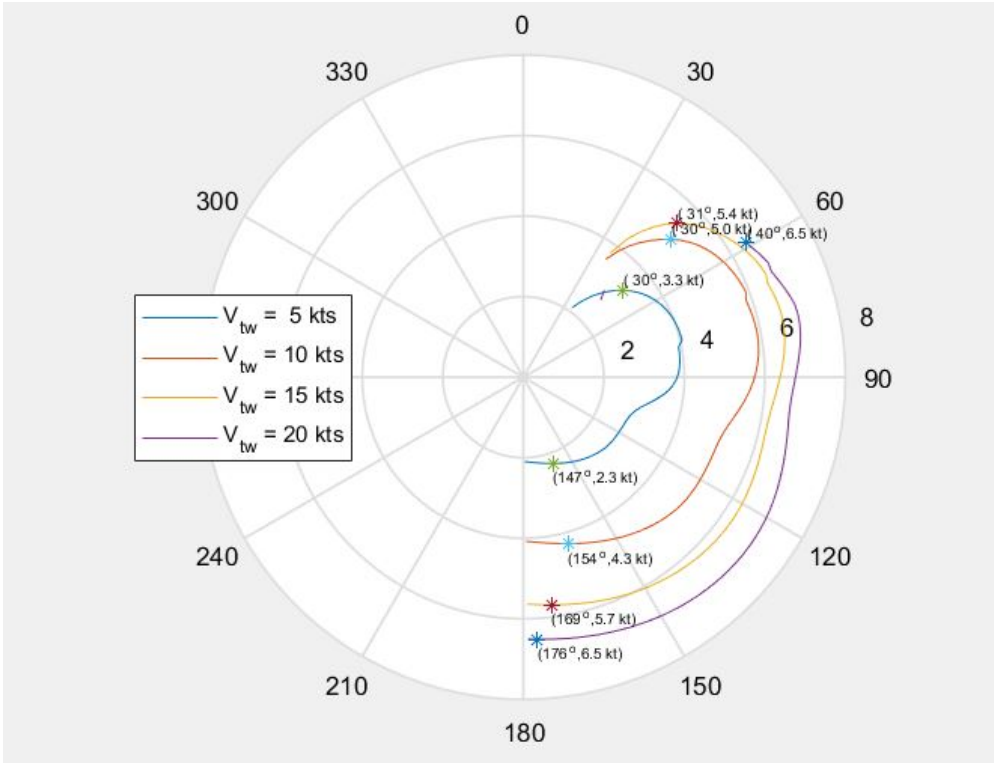


Figure 72: Sailing Performance Velocity Prediction Polar Plot

10.4 Design Risks and Cost Considerations

The yacht weight and vertical center of gravity represent significant sources of design risk. Deviation from design values in the as-built condition could result in a yacht that does not meet either trailer constraints or stability requirements. In order to mitigate this risk, a thorough evaluation of weights was conducted, and margins were included. Where possible, such as for structural components, the weights were calculated directly from the scantlings and material property. For commercially supplied equipment, such as the engine, stove, deck fittings, etc, manufacturer or vendor data was used. Where estimates were used, they are believed to be conservative. 5% build margins were included for each weight group, in addition to the 7% design weight margin discussed above.

A secondary risk is the sailplan's lead, the distance from sail center of effort to the hull center of lateral resistance. The lead is a key parameter in determining helm balance, and is set to provide a few degrees of weather helm while beating upwind. An appropriate lead could be accurately determined with extensive wind tunnel and towing tank testing. However, in practice experience-based design rules are often used. Such a rule was applied in the present design to set the lead to about 9% of the L_{WL}. If the yacht demonstrates excessive weather helm or lee helm, a means of adjusting the lead is necessary. The least intrusive way to accomplish this is by adjusting the sail plan center of effort, moving it forward to address excessive weather helm or aft to correct lee helm. Several design choices were made with consideration to reducing this design risk by permitting such adjustments with minimal intrusion. Selection of a junk rig was one such choice, since the set of the sail may be adjusted somewhat by either moving the yard sling point off center, adjusting the cant of the sail on the mast, or both. The mast may also be raked forward or aft, which would require re-positioning the mast partner and so is more intrusive than adjusting the sail set. The floor structure on which the mast step is secured was designed wide enough to permit this raking. Finally, the mast longitudinal position may be changed. There is space on the mast floors and the foredeck to move the mast, step, and partner forward somewhat without significant structural change, and without positioning the mast too far forward for pitching concerns. Moving the mast aft would require more significant structural changes, since the floors would need to be repositioned and the partner would need to be by incorporated into the cabin roof, which would require reinforcement.

Cost was not an explicit design requirement for the yacht, however it was qualitatively considered throughout the design process. The yacht hull and deck are made from standard production boat materials. The freestanding mast choice eliminates the cost of stays, spreaders, shrouds, chainplates and associated hardware, while the junk sail may be very inexpensively fabricated: it requires light material and is cut flat, so can be made by an amateur without need of sailmaking expertise. The largest cost driver in the design is undoubtedly the use of carbon fiber for the mast and spars, which was selected due to the considerable impact the rig weight has on stability. Scantlings were determined for aluminum battens, yard, and boom, using high strength 7075 T6 aluminum, and a mast was designed in marine grade 6065 T6 aluminum. The changes resulted in a net addition of 37 kg. This amount was lower than initially expected; however, the use of an aluminum mast removes the need for the mast head and lighting conductor required in the carbon mast. Additionally, the aluminum mast was designed with a safety factor of 2 applied to the material's yield stress, vice the SF of 6 applied to the carbon laminate's ultimate compressive stress. The lower safety factor is justified by the fact that the aluminum material properties are not sensitive to the mast construction process as laminates are; in a carbon fiber mast, deviations in cloth alignment or fiber mass fraction can cause reduction in strength. Additionally, aluminum may be stressed up to its yield strength without permanent strength degradation, while micro-cracking in a laminate can permanently reduce the material strength if it is loaded more than about 50% of the ultimate strength.[38] The net weight addition raised the minimum operating condition vertical center of gravity by 7 cm, to a position 3 cm below the DWL. Such a design would meet ISO design category A requirements for STIX, but only design category B requirements for righting energy.

11 Conclusion

A design was developed for a seaworthy, ocean capable sailing yacht that can be transported, launched and recovered from a trailer hauled by a standard sized pick up trick or sport utility vehicle. Design requirements were developed based on the yacht's intended purpose. The design tradespace was constrained by qualitative feasibility considerations, and a multi-criteria decision making process was used to further down-select to preferred options in the major functional areas of build material, rig type, and keel type. A review of similar vessels and design factors affecting seaworthiness of yachts were used to establish initial parameters for the point-based design approach that followed. In an iterative process, the yacht dimensions, hull form, keel and rudder designs were developed; structural scantlings were determined; the sail plan and rig was developed; and engine and propeller were selected and analysed. Weights and centers were analysed by direct calculation, and yacht stability was assessed using the hydrostatics solver in the selected CAD suite. Upright resistance in rough and calm weather were estimated with Delft series correlations and sailing performance was predicted with an open source VPP. A 1:7 scale model of the hull was 3D printed and used to measure upright resistance in towing tank experiments.

The design meets or exceeds all threshold requirements. Design requirements and predicted values are summarized in table 61.

Parameter	**Threshold**	**Objective**	**Design**
Stability Index	23	32	37.5
Righting Energy	57000 kg-m-deg	172000 kg-m-deg	176592 kg-m-deg
Minimum Crew	1	-	1
Passage Speed	4 knots	7 knots	5 knots
Accommodations	2 adults	3 adults	2 adults
Stores Endurance	21 days	30 days	25 days
Weight, yacht + trailer	3855 kg	2720 kg	3396 kg
Length, on trailer	10.67 m	9.14 m	9.1 m
Width, on trailer	2.59 m	2.44 m	2.44 m
Height, on trailer	4.11 m	3.96 m	3.3 m
Launch/Recover	2 crew, No crane	1 crew, No crane	1 crew, No crane
Structural Strength	Open Ocean	-	Open Ocean

Table 61: Design Requirements and Results

As for any design, compromises were made to balance conflicting design goals. Broadly speaking, the largest conflict was between seaworthiness considerations and trailer capabilities. The need to maintain a low vertical center of gravity and a comparatively narrow beam was complicated by the desire to keep both the draft and total weight as low as practicable. In converging upon a design that satisfied stability requirements, all of these variables were in play. Both the final design draft of 1.1 m, and the trailer condition yacht weight of 2830 kg were somewhat higher than initially hoped, though still within requirements. This was necessary to achieve adequate stability, in particular to achieve sufficient righting energy. Similarly, rather more reliance was placed on form stability than initially anticipated; the beam needed to be increased about 10 cm relative to early iterations, in order to provide enough initial stability to carry sail. This resulted from the inability to achieve a vertical center of gravity as low as initially desired, due to the draft constraining ballast depth somewhat. However, the final center is still comfortably below the waterline, and the length/beam ratio reasonably high for a vessel of this size. Finally, the weight and size constraints, as well as the impact of crew size on stores endurance, necessitated accepting the threshold value for crew accommodations. It was deemed infeasible to design accommodations for more than three people, though the yacht could certainly accommodate up to four people for day sails.

There exist opportunities for further analysis and optimization of the design. Several design elements were incorporated to enhance seaworthiness of the vessel, based on information in the literature; a comparatively large lateral area, long keel, and relatively deep hull of V-shaped and wineglass sections were designed to take advantage of viscous effects to promote damping, and added mass effects to increase inertia. The impacts of these design choices were not quantified in the present work, but future analysis of the yacht in seakeeping would provide an indication of their effectiveness. Additionally, the prediction for yacht sailing performance could be improved by conducting a full series of semi-captive model towing tests, in which heel angle, yaw angle, and model speed are all varied. Combining this with data on lift and drag characteristics of junk rigged sails would enable a VPP to more accurately predict yacht speed than reliance on the statistical Delft series alone. However, for a cruising vessel, such accuracy is not likely warranted. Finally, opportunities exist to optimize the structural design of the vessel, which was shown to be rather conservative. Such optimization could result in some additional weight savings.

The yacht design presented meets all established design requirements, and is believed to be a good solution for a trailer capable, but inherently seaworthy, yacht. The predicted performance of the yacht indicates that she is capable of accomplishing the original intent of the design: to safely carry at least two sailors across an ocean, then be pulled out on her trailer and carted home with minimal fuss. She is by necessity a small yacht; there is inherent risk in sailing any such vessel over a horizon, regardless of the engineering standards she meets. Presented here was an attempt to mitigate that risk through design.

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A Yacht Renderings

Figure 73: Profile View, Rigged

Figure 74: Aft Quarter, Rigged

Figure 75: Forward Quarter, Rigged

Figure 76: Starboard Forward Quarter, Rigged

Figure 77: Top View, Rigged

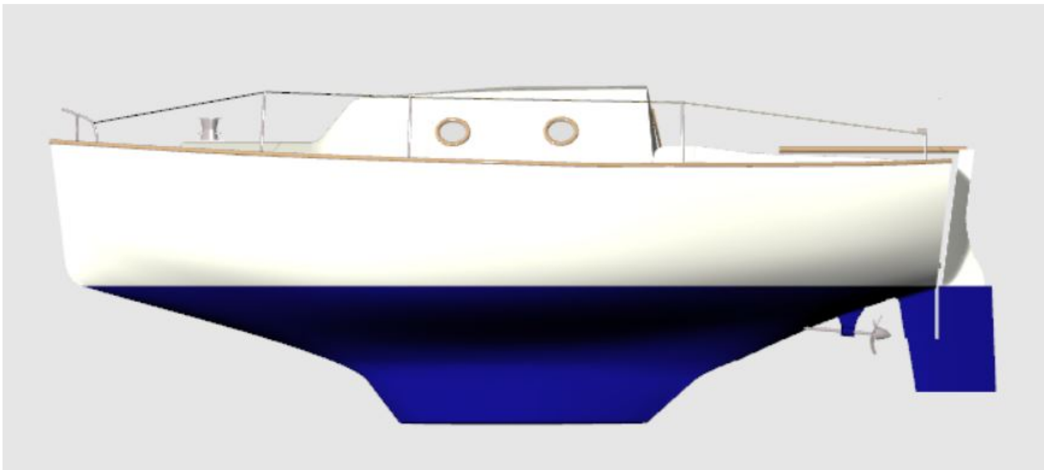


Figure 78: Profile View

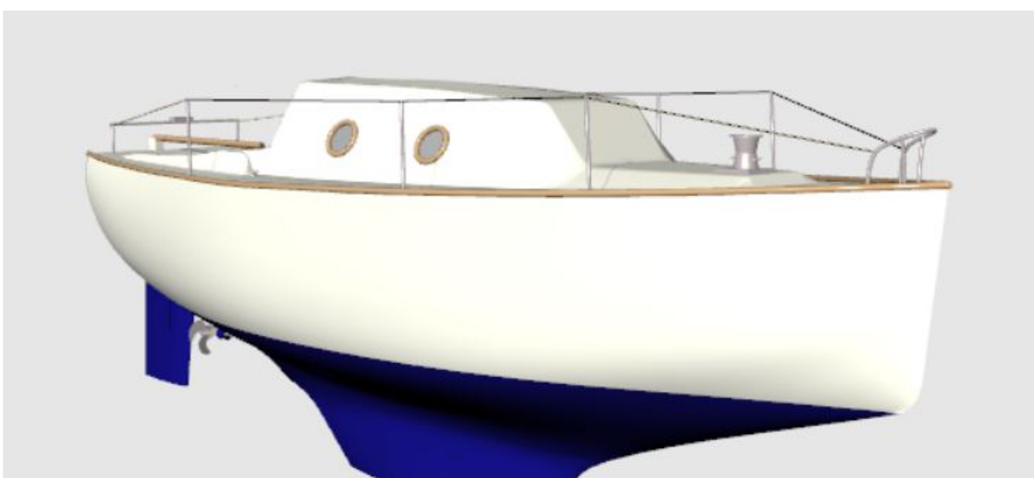


Figure 79: Fore Quarter

Figure 80: Aft Quarter

B Lines Drawings

Lines drawings for the hull are displayed in figures 81 and 82. Increments are in meters. Drawing for the cabin roof, cockpit, and decks are displayed in figures 83 through 85. Increments are 10 cm.

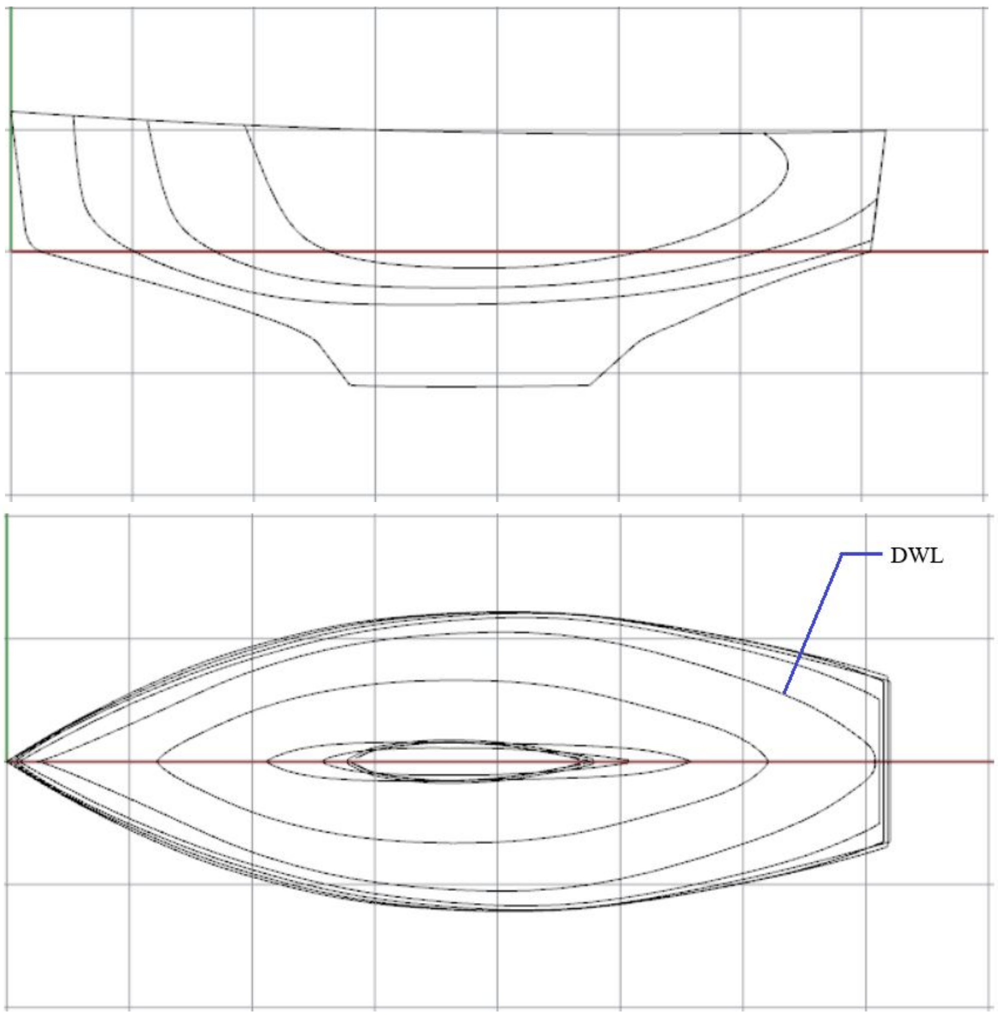


Figure 81: Hull Profile and Waterlines

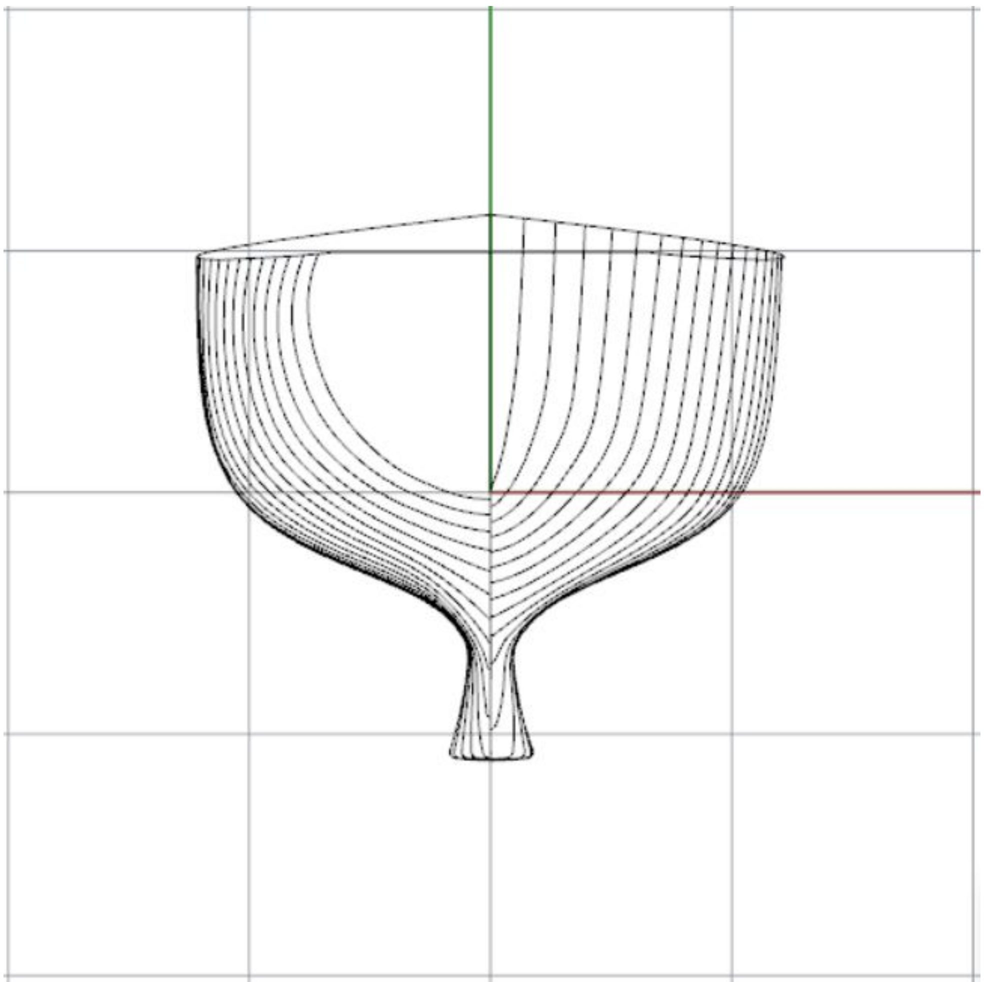


Figure 82: Hull Body Plan

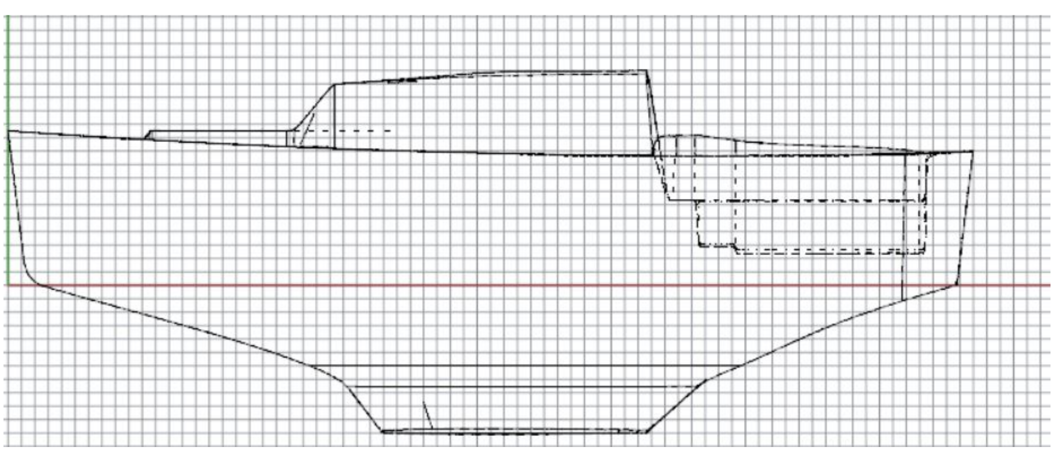


Figure 83: Deck, Cockpit Cabin Profile View

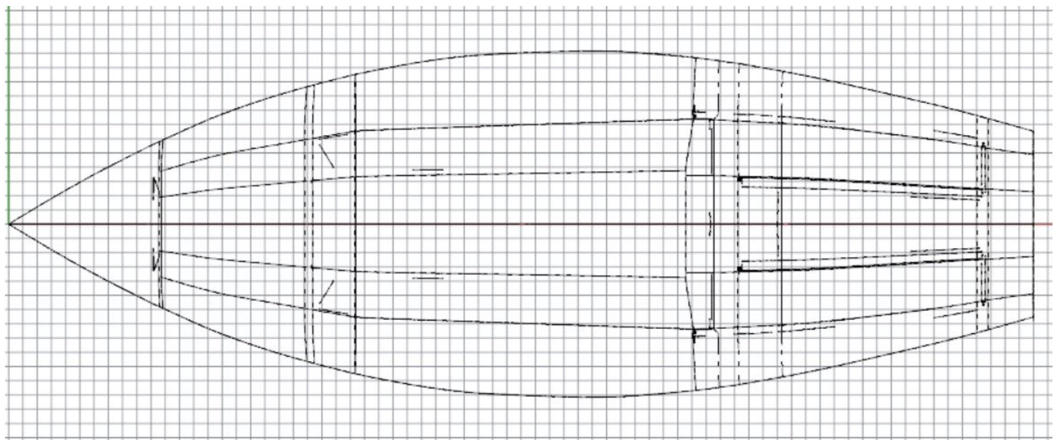


Figure 84: Deck, Cockpit, Cabin Overhead View

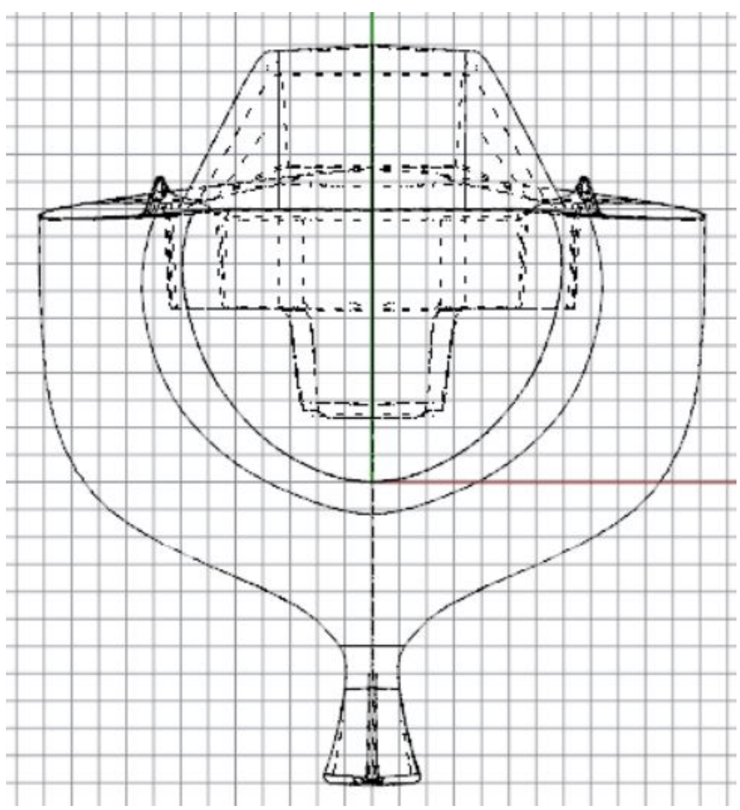


Figure 85: Deck, Cockpit, Cabin Body View

C Table of Offsets

Offsets for the hull are presented in figures 86 and 87.

TABLE OF OFFSETS											
File Name:		Hull Offsets									
Units:		Meters									
Absolute Tolerance:		0.005									
WATERLINE HALF-BREADTHS											
Station	Waterline -1.080	Waterline -0.825	Waterline -0.550	Waterline -0.275	Waterline 0.000	Waterline 0.275	Waterline 0.550	Waterline 0.825	Waterline 1.100	Sheer Line	Station
0.225						0.072	0.110	0.126	0.138	0.140	0.225
0.450					0.067	0.193	0.242	0.258	0.269	0.270	0.450
0.675					0.159	0.311	0.358	0.377	0.391	0.391	0.675
0.900					0.257	0.419	0.464	0.487		0.505	0.900
1.125					0.356	0.518	0.562	0.589		0.611	1.125
1.350				0.108	0.459	0.610	0.654	0.684		0.709	1.350
1.575				0.228	0.563	0.696	0.740	0.772		0.799	1.575
1.800				0.331	0.661	0.775	0.818	0.851		0.877	1.800
2.025				0.420	0.745	0.846	0.888	0.920		0.945	2.025
2.250			0.065	0.491	0.817	0.910	0.952	0.984		1.006	2.250
2.475			0.118	0.547	0.880	0.970	1.011	1.042		1.061	2.475
2.700		0.051	0.143	0.591	0.931	1.021	1.061	1.090		1.107	2.700
2.925	0.091	0.090	0.154	0.623	0.971	1.063	1.102	1.129		1.144	2.925
3.150	0.142	0.104	0.157	0.645	1.001	1.097	1.135	1.160		1.171	3.150
3.375	0.167	0.111	0.157	0.658	1.024	1.124	1.161	1.183		1.191	3.375
3.600	0.176	0.110	0.154	0.664	1.041	1.146	1.182	1.200		1.206	3.600
3.825	0.170	0.110	0.150	0.662	1.051	1.162	1.197	1.212		1.215	3.825
4.050	0.153	0.105	0.146	0.653	1.054	1.172	1.206	1.218		1.219	4.050
4.275	0.120	0.094	0.142	0.635	1.049	1.174	1.209	1.218		1.217	4.275
4.500	0.081	0.076	0.137	0.607	1.033	1.167	1.203	1.209		1.206	4.500
4.950		0.028	0.116	0.515	0.966	1.121	1.165	1.168		1.160	4.950
5.175			0.095	0.455	0.916	1.085	1.134	1.137		1.125	5.175
5.400			0.056	0.393	0.858	1.043	1.099	1.101		1.085	5.400
5.625				0.328	0.793	0.997	1.060	1.062		1.041	5.625
5.850				0.250	0.721	0.947	1.017	1.020		0.993	5.850
6.075				0.134	0.645	0.892	0.971	0.974		0.943	6.075
6.300					0.561	0.831	0.923	0.927		0.890	6.300
6.525					0.462	0.761	0.866	0.875		0.834	6.525
6.750					0.335	0.676	0.800	0.818		0.776	6.750
6.975					0.175	0.577	0.722	0.756		0.715	6.975

Figure 86: Table of Offsets: Waterline Half-Breadths

BUTTOCK HEIGHTS						
Station	Buttock 0.000	Buttock 0.305	Buttock 0.610	Buttock 0.915	Sheer Line	Station
0.225	0.009				1.135	**0.225**
0.450	-0.060				1.120	**0.450**
0.675	-0.123	0.259			1.106	**0.675**
0.900	-0.185	0.063			1.092	**0.900**
1.125	-0.248	-0.055	1.068		1.079	**1.125**
1.350	-0.311	-0.146	0.276		1.066	**1.350**
1.575	-0.376	-0.224	0.060		1.055	**1.575**
1.800	-0.445	-0.291	-0.057		1.044	**1.800**
2.025	-0.519	-0.345	-0.139	0.780	1.034	**2.025**
2.250	-0.599	-0.384	-0.198	0.301	1.025	**2.250**
2.475	-0.710	-0.409	-0.238	0.071	1.016	**2.475**
2.700	-0.981	-0.424	-0.265	-0.025	1.008	**2.700**
2.925	-1.103	-0.432	-0.281	-0.076	1.001	**2.925**
3.150	-1.108	-0.436	-0.291	-0.106	0.995	**3.150**
3.375	-1.110	-0.437	-0.297	-0.123	0.989	**3.375**
3.600	-1.111	-0.436	-0.299	-0.133	0.984	**3.600**
3.825	-1.110	-0.433	-0.298	-0.136	0.979	**3.825**
4.050	-1.109	-0.427	-0.294	-0.134	0.975	**4.050**
4.275	-1.106	-0.419	-0.286	-0.125	0.972	**4.275**
4.500	-1.103	-0.408	-0.274	-0.109	0.968	**4.500**
4.950	-0.930	-0.375	-0.233	-0.048	0.964	**4.950**
5.175	-0.733	-0.351	-0.204	-0.001	0.964	**5.175**
5.400	-0.626	-0.322	-0.169	0.055	0.965	**5.400**
5.625	-0.529	-0.288	-0.127	0.124	0.967	**5.625**
5.850	-0.426	-0.245	-0.079	0.211	0.969	**5.850**
6.075	-0.328	-0.196	-0.025	0.329	0.972	**6.075**
6.300	-0.241	-0.139	0.035	0.510	0.974	**6.300**
				0.887		
6.525	-0.163	-0.077	0.106		0.977	**6.525**
6.750	-0.093	-0.013	0.196		0.982	**6.750**
6.975	-0.028	0.054	0.318		0.987	**6.975**

Figure 87: Table of Offsets: Buttock Heights

D Design Data on Similar Vessels

Particulars for similar vessels, consulted in early design work for to aid in establishing initial parameters for the point-based design, are displayed in figure 88.

Yacht	LOA (m)	LWL (m)	Disp. (kg)	Draft (m)	Beam (m)	Ballast (kg)	SA (m^2)	Aux HP	Rig	Keel	Note
JESTER	7.62	6.10	1960	1.22	2.13	1050	22.30	N/A	Junk	mod full	1
North Atlantic 29	8.90	7.01	4218	1.68	2.90	1905	39.86	13	Junk	mod full	2
Dana 24	7.36	6.53	3356	1.17	2.62	1406	33.26	18	Cutter	mod full	3,4
Cal 20	6.10	5.49	884	1.01	2.13	408	18.12	N/A	Frac sloop	Bulb fin	3,4
Contessa 32	9.75	7.31	4308	1.68	2.90	2041	40.23	24	Mast sloop	fin w/ skeg	3,4
Albin Vega 27	8.25	7.01	2299	1.12	2.46	915	27.50	13	Mast sloop	mod full	3,4
Falmouth Cutter 22	6.71	6.35	3356	1.07	2.44	1134	30.75	7	Cutter	full	3,4
Pearson Triton 28	8.63	6.55	3143	1.19	2.51	1369	34.47	30	Frac sloop	mod full	3,4
Allied Seawind II	9.63	7.77	6757	1.37	3.18	2449	51.56	27	Ketch	mod full	3,4
Cape Dory 25D	7.62	5.79	2322	1.07	2.44	930	28.24	7.5	mast sloop	mod full	3,4
Catalina 27 (shoal)	8.18	6.63	3107	0.91	2.69	1224	31.77	N/A	mast sloop	wing	3
Catalina 27	8.18	6.63	3107	1.22	2.69	1224	31.77	N/A	mast sloop	fin	3,4
Tartan 30	9.12	7.39	3968	1.50	3.05	1633	41.71	30	mast sloop	fin w/ skeg	3,4
Contessa 26	7.77	6.40	2449	1.22	2.29	1043	26.01	7	mast sloop	mod full	3,4
Flicka 20	7.31	5.54	2721	0.99	2.44	794	23.23	N/A	cutter	full	3,4
Nor'Sea 27	8.23	7.62	3673	1.17	2.44	1406	34.93	9	mast sloop	long keel	3
Westerly 26	7.92	6.50	3039	0.91	2.57	1270	30.10	13	mast sloop	bilge keel	3
Pacific Seacraft 25	7.47	6.40	2154	0.99	2.44	794	23.23	8	mast sloop	mod full	4
Folkboat	7.64	6.00	1960	1.19	2.19	1050	23.97	6.5	fract sloop	full keel	4
Frances 26	7.92	6.48	3084	1.17	2.49	1587	30.38	10	sloop	mod full	4

Figure 88: Particulars of Similar Yachts
The 'Note' column indicate the source of the data. Note 1 is from reference [30]; note 2 is from reference [57]; note 3 is from reference [58] and note 4 is from reference [4].

E Hydrostatics

A hydrostatics and stability report is provided in figure 89. The calculation used the model in the minimum operating weight condition, and included the skeg and transom rudder. Inclusion of the rudder skewed some of the calculations because it inflated the waterline length. Corrected values for affected parameters, using the waterline length of the hull, are summarized in table 62.

Values for Correct L_{WL}					
L_{WL}	DLR	L_{WL}/B_{WL}	B_{WL}/T	AF/L_{WL}	FF/L_{WL}
6.78	263	3.23	1.90	0.468	0.532

Table 62: Hydrostatic Parameters Corrected by Exclusion of Rudder Length from L_{WL}

Default Project
Hydrostatics & Stability Analysis
Default Company
Report Time: Tuesday, March 2, 2021, 6:22:19 PM
Model Name: C:\Users\nmaxw\OneDrive\Documents\MIT\Thesis\Hull Models\Final Models\Hull K_Lines.3dm

Orca3D

Condition Summary

Load Condition Parameters

Condition	Weight / Sinkage	LCG / Trim	TCG / Heel	VCG (m)
Condition 1	2935.000 kgf	0.000 deg	0.000 deg	-0.1

Resulting Model Attitude and Hydrostatic Properties

Condition	Sinkage (m)	Trim(deg)	Heel(deg)	Ax(m^2)
Condition 1	-0.011	0.000	0.000	0.81

Condition	Displacement Weight (kgf)	LCB(m)	TCB(m)	VCB(m)	Wet Area (m^2)
Condition 1	2935.001	3.802	0.000	-0.248	15.298

Condition	Awp(m^2)	LCF(m)	TCF(m)	VCF(m)
Condition 1	9.551	3.878	0.000	-0.011

Condition	BMt(m)	BMl(m)	GMt(m)	GMl(m)
Condition 1	0.839	7.807	0.691	7.659

Condition	Cb	Cp	Cwp	Cx	Cws	Cvp
Condition 1	0.172	0.487	0.631	0.353	3.361	0.272

General Info

Analysis Type	FreeFloatEquilibrium	Up Direction = Positive_Z	
		Fwd Direction = Negative_X	

Surface Meshing Parameters

Density	1	Minimum edge length	0.0001 m
Maximum angle	0	Maximum edge length	0 m
Maximum aspect ratio	0	Max distance, edge to surf.	0 m
Minimum initial grid quads	0	Jagged seams	False
Refine mesh	True	Simple planes	True

Load Condition Parameters

Weight	2935.000 kgf
Model Trim	0.000 deg
Model Heel	0.000 deg
VCG	-0.1 m
Fluid Type	Seawater
Fluid Density	1025.900 kg/m^3
Mirror Geometry	False

Resultant Model Attitude

Heel Angle	0.000 deg	Sinkage	-0.011 m
Trim Angle	0.000 deg		

Overall Dimensions

Length Overall, LOA	7.555 m	Loa / Boa	3.098
Beam Overall, Boa	2.439 m	Boa / D	0.900
Depth Overall, D	2.711 m		

Waterline Dimensions

Waterline Length, Lwl	7.242 m	Lwl / Bwl	3.465
Waterline Beam, Bwl	2.090 m	Bwl / T	1.901
Navigational Draft, T	1.099 m	D / T	2.466

Volumetric Values

Displacement Weight	2935.001 kgf	Displ-Length Ratio			215.360
Volume	2.861 m^3				
LCB	3.802 m	FB/Lwl	0.486	AB/Lwl	0.514
TCB	0.000 m	TCB / Bwl			0.000
VCB	-0.248 m				
Wetted Surface Area	15.298 m^2				
Moment To Trim	31.039 kgf-m/cm				

Waterplane Values

Waterplane Area, Awp	9.551 m^2				
LCF	3.878 m	FF/Lwl	0.497	AF/Lwl	0.503
TCF	0.000 m	TCF / Lwl			0.000
Weight To Immerse	97.981 kgf/cm				

Sectional Parameters

Ax	0.811 m^2		
Ax Location	3.637 m	Ax Location / Lwl	0.464

Hull Form Coefficients

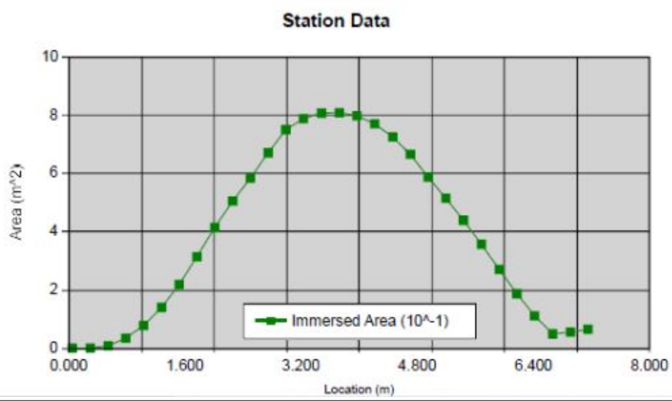
Cb	0.172	Cx	0.353
Cp	0.487	Cwp	0.631
Cvp	0.272	Cws	3.361

Static Stability Parameters

I(transverse)	2.401 m^4	I(longitudinal)	22.334 m^4
BMt	0.839 m	BMl	7.807 m
GMt	0.691 m	GMl	7.659 m
Mt	0.603 m	Ml	7.570 m

Points Of Interest

Name	Long'l (m)	Transv (m)	Vert (m)	Dist Abv WL (m)
Port DF	4.778	-0.383	1.541	1.552
Stbd DF	4.778	0.383	1.541	1.552



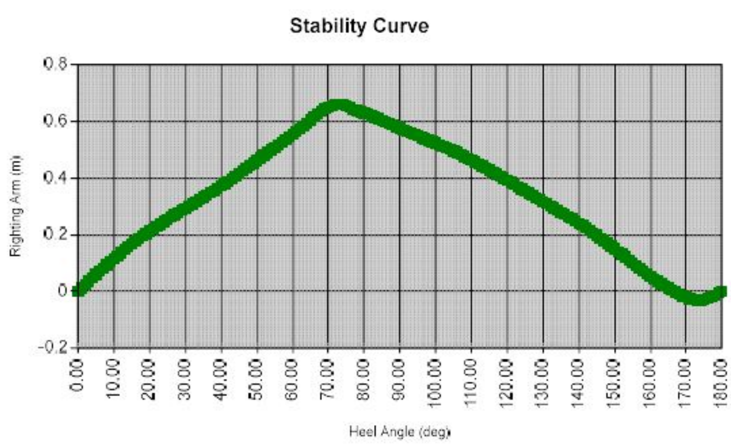


Figure 89: Hydrostatics Report, Min Op Condition

F Weights and Centers

Weights and centers are tabulated for the minimum operating condition in figure 90. The results are summarized in table 63.

Weights Summary				
Condition	Δ (kg)	VCG (m)	LCG (m)	TCG (m)
Min Op	2935	-0.10	3.88	0.00
Full Load	3280	-0.09	3.88	0.00

Table 63: Weights Summary

Item	m (kg)	VCG	LCG	TCG
Structural Weights				
Hull, Deck, Internal Structures	867.05	0.237	3.802	0.003
5% Build Margin	43.35	0.237	3.802	0.000
Mast Partner	4.40	1.200	1.800	0.000
Mast Step	18.00	-0.260	1.800	0.000
Skeg	9.95	-0.270	6.970	0.000
Skeg Floor	3.00	0.120	6.970	0.000
Deck Flanged Connection	5.66	1.010	3.950	0.000
Gelcoat/paint	43.00	0.306	3.907	0.000
Cabin Sole	21.20	-0.260	3.790	-0.020
Head Sole	4.80	-0.190	3.020	0.400

Deck and Cockpit				
Anchor	10.00	0.250	0.500	0.000
Anchor chain&rode	7.50	0.250	0.500	0.000
Bow pulpit	4.00	1.300	0.200	0.000
Stanchions	4.90	1.300	3.990	0.000
Stanchion Supports	3.28	1.010	3.990	0.000
Lifelines	0.80	1.540	4.020	0.000
Toe rail	8.00	1.030	4.020	0.000
Cleats	5.00	1.000	4.500	0.000
Ventilation box	3.00	1.620	3.000	0.000
Winch	14.00	1.250	1.000	0.000
Windows	4.00	1.250	3.700	0.000
Mooring Cleats (x4)	2.00	1.050	4.300	0.000
Tiller	3.00	1.100	6.900	0.000
Rudder blade	13.54	-0.473	7.190	0.000
Rudder stock	7.10	0.454	7.270	0.000
Gudgeons and pintles	5.00	0.100	7.200	0.000
Mooring Line	2.80	0.000	7.000	0.000
Helmsman	75.00	0.400	6.650	0.000
Wind vane autohelm	15.00	0.400	7.200	0.000
Misc/Margin	9.40	1.000	3.880	0.000

Item	m (kg)	VCG	LCG	TCG
Rig and Sails				
Mast	59.60	3.800	1.810	0.000
Boom	3.10	2.290	3.790	0.000
Sail	7.75	5.840	3.380	0.000
Rigging	12.05	3.400	6.200	0.000
Battens	8.32	5.700	3.460	0.000
Yard	6.76	8.724	1.800	0.000
Masthead fittings	2.50	9.550	1.530	0.000
Lighting Cable	6.12	4.760	1.800	0.000
MH lightning rod	0.32	9.850	1.800	0.000
Ground Plate	4.50	-0.300	1.900	0.000
Mast Margin	2.98	3.800	1.810	0.000
Boom Margin	0.16	2.290	3.790	0.000
Batten Margin	0.42	5.700	3.460	0.000
Yard Margin	0.34	8.724	1.800	0.000
Rigging Margin	0.60	3.400	6.200	0.000

Head				
WC	12.75	0.070	2.750	0.370
Lockers	6.00	0.500	3.430	0.293
Margin	0.94	0.100	3.000	0.300

Galley				
Sink	4.54	0.260	3.030	-0.560
Faucet	0.50	0.260	3.030	-0.560
Drain/seacock	1.00	-0.200	3.030	-0.640
FW manifold and hoses	0.50	-0.300	3.000	0.000
Pump	0.30	0.000	3.000	-0.420
Fuel	17.00	-0.030	3.350	-0.600
Stove	6.00	0.370	3.490	-0.610
Countertop	5.00	0.370	3.130	-0.790
Counter, Drawers	7.00	0.010	3.180	-0.380
Cookware	5.00	0.200	3.200	-0.500
Shelving	6.00	0.400	3.200	-0.500
Margin	2.64	0.400	3.200	-0.500

Item	m (kg)	VCG	LCG	TCG
Systems				
Engine	71.00	-0.050	5.180	0.000
exhaust	5.00	0.200	5.900	0.000
water intake and pipe	2.00	-0.200	5.200	0.200
fuel filter	2.00	-0.200	5.200	-0.200
water filter	2.00	-0.200	5.200	0.200
Reduction gear	10.00	-0.280	5.420	0.000
Bearing	3.00	-0.328	6.373	0.000
Shaft seal	3.00	-0.329	5.950	0.000
Shafting	6.10	-0.350	6.140	0.000
Propeller	3.00	-0.390	6.630	0.000
Fuel tank	3.05	-0.420	4.360	0.000
Water tank	12.18	-0.420	3.340	0.000
Electrical panel	5.00	0.050	3.770	-0.600
Wiring	5.00	0.800	4.000	0.000
Battery	23.80	-0.143	4.780	-0.420
Radio	2.00	1.080	4.830	0.475
Solar panel	2.90	1.600	3.200	0.000
Nav Lights	1.00	1.200	0.000	0.000
House lights	1.00	1.500	4.000	0.000
Chartplotter/GPS	2.50	1.080	4.830	0.475
Bilge pumps and piping	2.00	-0.500	6.000	0.000
Essential Safety Gear	30.00	0.200	6.000	0.500
Margin	8.38	0.000	5.000	0.000
Quarterberths				
Port Berth	17.98	0.180	5.430	-0.610
Stbd Berth	18.26	0.060	4.640	0.640
Engine Bay Blkhds	16.00	0.000	5.780	0.000
Margin	2.61	0.000	5.500	0.000
Saloon				
Nav Table	3.00	0.500	4.160	-1.000
Tools & Spares	30.00	-0.200	2.300	0.600
Ladder	4.05	0.200	4.720	0.000
Shelving	5.00	0.500	4.200	0.500
Margin	2.10	0.000	4.500	0.000
Ballast				
Ballast	1221.7	-0.940	3.680	0.000

Figure 90: Weight List, Minimum Operating Condition

VCG is from the plane of the DWL, positive being above this plane. LCG is given aft of the forward-most part of the bow. TCG is given from centerline, positive is to starboard.

G Mast Design Calculations

Figure 91 displays the inputs for the design of the carbon fiber and aluminum masts. The mast design parameters that were controlled in the composite case were the diameter at partners and at masthead, and the starting point of this taper. The necessary number of carbon fiber strips to achieve required section modulus at each interval was then computed, and wall thickness checked against buckling criteria. For the aluminum mast, the masthead diameter and taper ratios for both diameter and wall thickness were controlled to give adequate section modulus, rather than calculating the necessary thickness based on required section modulus. This was done in order to give a feasible linear taper in both diameter and thickness that would be possible to form in an extrusion process, since metal does not lend itself to the freedom of form of composites.

Figures 92 and 93 display the mast design calculations in 1 foot increments. Mast weights were calculated from material properties, and vertical centers were computed by summing over the section weights in these increments, since each section is located a fixed height above the mast heel.

Carbon Mast Inputs:			
Position of mast heel, from DWL	zo	-0.82	ft
Length of mast	L	32	ft
Bury	B	4.25	ft
Length abv partners	LAP	27.75	ft
Maximum RM	Mr	16700	ft-lb
Safety factor	FOS	6	
Ideal comp. yield stress	si	170000	psi
Design max stress	sd	28333.3	psi
Inner D @ partners	Dp	7.66	in
Inner D @ masthead	Dm	2.75	in
Taper start	Ts	7	ft
Boom H abv heel	Hb	6.4	ft
Width of CF strips	w*	12	in
Thickness of CF strips	t*	0.006	in
Weight/lin ft CF	p	0.05	lb/ft
Fiber weight ratio	f	0.6	
Taper slope	m	0.1964	in/ft
E-glass hoop layer thickness	te	0.040	in
E-glass weight	pe	1.5	lb/sq yd

Thin wall approximations used for moment of inertia, area

Aluminum Mast Inputs			
Position of mast heel, from DWL	zo	-0.82	ft
Length of mast	L	32	ft
Bury	B	4.25	ft
Length abv partners	LAP	27.75	ft
Maximum RM	Mr	16700	ft-lb
Safety factor	FOS	2	
6065 T6 AL Yield Stress	si	40000	psi
Design max stress	sd	20000	psi
Wall thickness at base	tb	0.3	in
Wall thickness at masthead	tm	0.2	in
Masthead Diameter	Dm	3.5	in
Diameter Taper	md	0.124	in/ft
Thickness Taper	mt	0.00417	in/ft
Taper Start Point	Ts	8	ft

Thin wall approximations used for moment of inertia, area

Figure 91: Mast Design Inputs

Carbon Mast Design							
Position along mast (ft), z	Inner Diameter (in), Di	Design Moment (ft-lb), BM	Reqd SM (in3)	Number of CF strips	CF Wall thickness (in), tc	Total Wall thickness, (in), t	Thickness/ Diameter
(z=0 is heel)	Depends on Dp,m, and z	BM = Mr, z<Ts BM=(Mr/(L-Ts))*(L-z), z>Ts	SM=BM/sd	N= SM/(w*t*Di/4)	tc = Nwt*/(Diπ)	t = tc+2te	r = t/Di
0	7.66	16700	7.073	52	0.1556	0.2356	0.031
1	7.66	16700	7.073	52	0.1556	0.2356	0.031
2	7.66	16700	7.073	52	0.1556	0.2356	0.031
3	7.66	16700	7.073	52	0.1556	0.2356	0.031
4	7.66	16700	7.073	52	0.1556	0.2356	0.031
5	7.66	16700	7.073	52	0.1556	0.2356	0.031
6	7.66	16700	7.073	52	0.1556	0.2356	0.031
7	7.66	16700	7.073	52	0.1556	0.2356	0.031
8	7.46	16032	6.790	51	0.1566	0.2366	0.032
9	7.27	15364	6.507	50	0.1577	0.2377	0.033
10	7.07	14696	6.224	49	0.1588	0.2388	0.034
11	6.87	14028	5.941	49	0.1634	0.2434	0.035
12	6.68	13360	5.658	48	0.1647	0.2447	0.037
13	6.48	12692	5.375	47	0.1662	0.2462	0.038
14	6.29	12024	5.093	46	0.1677	0.2477	0.039
15	6.09	11356	4.810	44	0.1656	0.2456	0.040
16	5.89	10688	4.527	43	0.1672	0.2472	0.042
17	5.70	10020	4.244	42	0.1690	0.2490	0.044
18	5.50	9352	3.961	41	0.1709	0.2509	0.046
19	5.30	8684	3.678	39	0.1685	0.2485	0.047
20	5.11	8016	3.395	37	0.1660	0.2460	0.048
21	4.91	7348	3.112	36	0.1680	0.2480	0.051
22	4.71	6680	2.829	34	0.1653	0.2453	0.052
23	4.52	6012	2.546	32	0.1623	0.2423	0.054
24	4.32	5344	2.263	30	0.1591	0.2391	0.055
25	4.12	4676	1.980	27	0.1500	0.2300	0.056
26	3.93	4008	1.698	25	0.1459	0.2259	0.057
27	3.73	3340	1.415	22	0.1351	0.2151	0.058
28	3.54	2672	1.132	19	0.1232	0.2032	0.057
29	3.34	2004	0.849	19	0.1304	0.2104	0.063
30	3.14	1336	0.566	19	0.1386	0.2186	0.070
31	2.95	668	0.283	19	0.1478	0.2278	0.077
32	2.75	0	0.000	19	0.1583	0.2383	0.087

Figure 92: Carbon Mast Design

Aluminum Mast Design						
Position along mast (ft), z	Design Moment, (ft-lbf), BM	Reqd SM (in3)	Reqd Outer Diameter (in), Dor	Actual Outer Diameter (in), Doa	Wall Thickness (in), t	Thickness/ Diameter
(z=0 is heel)	BM = Mr, z<Ts BM=(Mr/(L-Ts))*(L-z), z>Ts	SM=BM/sd	Dor =2*(SM/tπ)^0.5	Doa = Dor+0.1, z<Ts Then tapers to Dm	t =tb, z<Ts Then tapers to tm	r = t/Di
0	16700	10.02	6.5	6.6	0.300	0.045
1	16700	10.02	6.5	6.6	0.300	0.045
2	16700	10.02	6.5	6.6	0.300	0.045
3	16700	10.02	6.5	6.6	0.300	0.045
4	16700	10.02	6.5	6.6	0.300	0.045
5	16700	10.02	6.5	6.6	0.300	0.045
6	16700	10.02	6.5	6.6	0.300	0.045
7	16700	10.02	6.5	6.6	0.300	0.045
8	16032	9.62	6.4	6.5	0.300	0.046
9	15364	9.22	6.3	6.4	0.296	0.046
10	14696	8.82	6.2	6.3	0.292	0.047
11	14028	8.42	6.1	6.1	0.288	0.047
12	13360	8.02	6.0	6.0	0.283	0.047
13	12692	7.62	5.9	5.9	0.279	0.047
14	12024	7.21	5.8	5.8	0.275	0.048
15	11356	6.81	5.7	5.6	0.271	0.048
16	10688	6.41	5.5	5.5	0.267	0.048
17	10020	6.01	5.4	5.4	0.263	0.049
18	9352	5.61	5.3	5.3	0.258	0.049
19	8684	5.21	5.1	5.1	0.254	0.049
20	8016	4.81	4.9	5.0	0.250	0.050
21	7348	4.41	4.8	4.9	0.246	0.050
22	6680	4.01	4.6	4.8	0.242	0.051
23	6012	3.61	4.4	4.6	0.238	0.051
24	5344	3.21	4.2	4.5	0.233	0.052
25	4676	2.81	3.9	4.4	0.229	0.052
26	4008	2.40	3.7	4.3	0.225	0.053
27	3340	2.00	3.4	4.1	0.221	0.053
28	2672	1.60	3.1	4.0	0.217	0.054
29	2004	1.20	2.7	3.9	0.213	0.055
30	1336	0.80	2.2	3.8	0.208	0.055
31	668	0.40	1.6	3.6	0.204	0.056
32	0	0.00	0.0	3.5	0.200	0.057

Figure 93: Aluminum Mast Design

H Rigging Warrant

The types, sizes, and lengths of lines are depicted in the rigging warrant of figure 94. Also displayed are the required blocks, cleats, and other fittings for each line. All line is 3 strand polyester.

Line	Diameter (mm)	Length (m)	Fittings
Running Rigging			
Halyard	8	45.5	x1 4-part double block w/ beckett (50mm sheave) x1 Deck Cleat x1 Single block (50 mm sheave)
Upper topping lift	10	11	x2 Thimbles
Lower topping lift	8	21.5	x2 Thimbles x4 Single blocks (50 mm sheave) x2 Deck cleats
Sheets	8	38.7	x5 single blocks (50 mm sheave) x1 cleat
Yard Hauling Parrel	8	12.9	x2 Shackles (6mm SS) x2 Single blocks (50 mm sheave) x1 Deck cleat
Luff Hauling Parrel	6	11.2	x3 Single blocks (40 mm sheave) x1 Deck cleat
Burgee Halyard	6	21	
Standing Rigging			
Batten Parrels	6	8.5	x5 Shackles (6mm SS)
Mast Lift	8	8.7	
Tackline	6	1	x1 Shackle (10mm SS)
Standing Lower Luff Parrel	6	2	

Figure 94: Rigging Warrant

I STIX Calculator

Code for computing stability index is presented in figure 95.

```
%STIX Calculator%
%Read a file with yacht parameters, output stix calculations%


%Read in text file with yacht parameters from user input.

Parameters=readtable(input('Enter yacht filename: ','s'));

%Pick out values, convert to array, and assign to variables.
%Text file is formatted with one line break at top, then 12 rows in
%the order given below. Each row is 'parameter units value'. Metric units.
%I.e first row is "LH m 7.198"

Values=table2array(Parameters(:,3));

Lh=Values(1,1);  %Hull length
Lwl=Values(2,1); %Waterline length
Bh=Values(3,1);  %Max beam, excl bolted appendages
Bwl=Values(4,1);  %Beam at the waterline
Mmo=Values(5,1);  %Min op condition mass. Repeat calculations for other conds
Asp=Values(6,1);  %Sail area
hce=Values(7,1);  % Distance WL to center of sail area, upright
hlp=Values(8,1); %Distance WL to center of lateral area below WL, upright
GZr=Values(9,1);   %Righting arm at 90 degrees heel
PhiV=Values(10,1); %Angle of vanishing stability
PhiD=Values(11,1); %Downflood angle
GZd=Values(12,1); %Righting arm at PhiD
Agz=Values(13,1); %Area under GZ-curve from upright to PhiV

%Calculate Base length
Lbs = (Lh+2*Lwl)/3;

%Calculate Displacement length factor

Fl=(Lbs/11)^(0.2);
FDLc=(0.6+(15*Mmo*Fl)/((333-8*Lbs)*Lbs^3))^0.5;

%Enforce allowed limits
if FDLc<0.75
    FDL=0.75;
elseif FDLc>1.25
    FDL=1.25;
else
    FDL=FDLc;
end
```

```
%Calculate Beam Displacement factor

Fb = 3.3*Bh/((0.03*Mmo)^(1/3));

if Fb>2
    FBDc = ((13.31*Bwl)/(Bh*(Fb^3)))^(0.5);
elseif Fb<1.45
    FBDc = ((Bwl*Fb^2)/(1.682*Bh))^0.5;
else
    FBDc = 1.118*(Bwl/Bh)^0.5;
end

%Enforce allowed limits
if FBDc<0.75
    FBD=0.75;
elseif FBDc>1.25
    FBD=1.25;
else
    FBD=FBDc;
end

%Knockdown Recovery factor

Fr=GZr*Mmo/(2*Asp*hce);

if PhiV<90
    FKRc=0.5;
elseif Fr>=1.5 && PhiV>=90
    FKRc=0.875+(Fr*0.0883);
elseif Fr<1.5 && PhiV>=90
    FKRc=0.5+0.333*Fr;
end

%Enforce limits
if FKRc<0.5
    FKR=0.5;
elseif FKRc>1.5
    FKR=1.5;
else
    FKR=FKRc;
end
```

```
%Inversion Recovery Factor

if Mmo<40000
    FIRc=PhiV/(125-Mmo/1600);
else
    FIRc=PhiV/100;
end

if FIRc<0.4
    FIR=0.4;
elseif FIRc>1.5
    FIR=1.5;
else
    FIR=FIRc;
end
```

```
%Dynamic Stability Factor

FDSc=(Agz/(15.81*(Lh)^0.5));

if FDSc<0.5
    FDS=0.5;
elseif FDSc>1.5
    FDS=1.5;
else
    FDS=FDSc;
end

%Wind Moment Factor
%Calculate apparent wind speed
Vaw = (13*Mmo*GZd/(Asp*(hce+hlp)*(abs(cos(PhiD))^1.3)))^0.5;

if PhiD>=90
    FWMc=1;
else
    FWMc=Vaw/17;
end

if FWMc>1
    FWM=1;
elseif FWMc<0.5
    FWM=0.5;
else
FWM=FWMc;
end

%Downflood Factor
%Replace PhiD with PhiV if PhiV<PhiD

FDFc=PhiD/90;

if FDFc>1.25
    FDF=1.25;
elseif FDFc<0.5
    FDF=0.5
else
    FDF=FDFc;
end

%Stability Index Calculation

SI=(7+2.25*Lbs)*(FDL*FBD*FKR*FIR*FDS*FWM*FDF)^0.5;
```

```
if SI>=32
    cat='A';
elseif 23<=SI<32
    cat='B';
elseif 14<=SI<23
    cat='C';
else
    cat='D';
end


parameter = {'FDL '; 'FBD '; 'FKR '; 'FIR '; 'FDS '; 'FWM '; 'FDF '; 'Stability Index '};
value =[FDL; FBD; FKR; FIR; FDS; FWM; FDF; SI];
value = round(value,3);
output = table(value,'RowNames',parameter)

writetable(output, input('Specify output file: ','s'),'WriteRowNames',true,
'Delimiter',' ','QuoteStrings',false,'WriteVariableNames',false)
```

Figure 95: Stability Index Code

J Scheel Keel Code

The point cloud generated by the Scheel keel code for specified inputs is displayed in figure 96. The code is reproduced in figure 97.

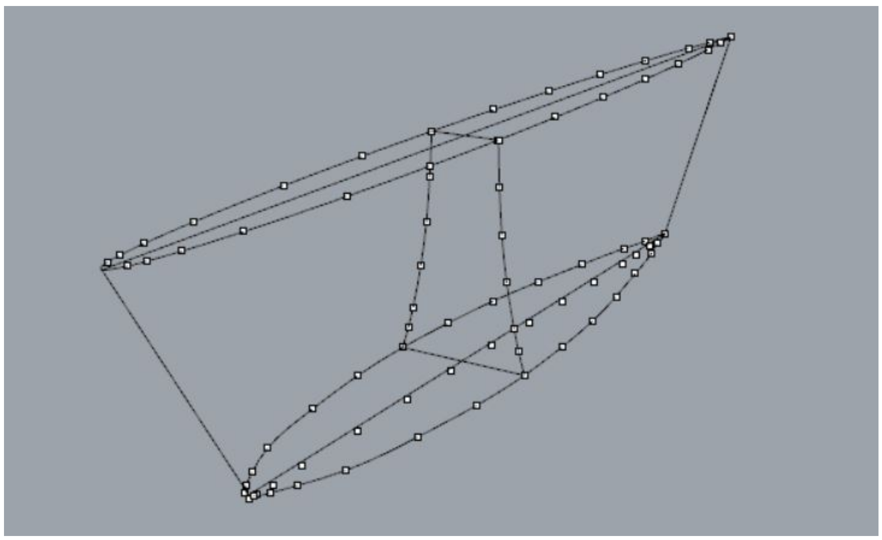


Figure 96: Scheel Code Point Cloud

```
%Scheel Keel Generator - takes a text file input specifying key dimensions,
%of the keel and outputs an xyz coordinate text file for upload into a CAD
%program.

%Read in keel file in which parameters are specified.

Parameters= readtable(input('Enter input filename:','s'));

%Pick out values, convert to array, and assign to variables.
%Text file is formatted with one line break at top, then 12 rows in
%the order given below. Each row is 'parameter units value' in Metric
%units. IE first row is "A m 1.05".

Values=table2array(Parameters(:,3));

A=Values(1,1); %Distance from WL to thickest section
B=Values(2,1); %Distance from thickest section to keel bottom
C=Values(3,1); %Distance from WL to thinnest section
D=Values(4,1); %Length of chord @ thinnest section
E=Values(5,1); %Length of chord @ thickest section
F=Values(6,1); %Length of CL along bottom of keel
G=Values(7,1); %Max width
H=Values(8,1); %Min width

%col 1, 2, and 4 create foil cross-sections

% x/chord postition vector for both min and max thickness cross sections
col1 = [0.0125,0.025,0.05,0.1,0.2,0.3,0.4,0.5,0.6,0.7,0.8,0.9,0.95,1];

%t/tmax for min thickness cross section (=thickness(x)/(H/2))
col2 = [0.25,0.35,0.48,0.66,0.87,0.97,1,0.98,0.90,0.78,0.6,0.35,0.19,0];

%t/tmax for max thickness cross section (=thickness(x)/(G/2))
col4 = [0.064,0.18,0.33,0.58,0.81,0.95,1.0,0.96,0.87,0.7,0.49,0.26,0.12,0];

%col 5 and 6 create curves defining the lateral surface of the keel between
%heights A and C (max and min sections)

%relative height above plane of max thickness (G); z/(|A-C|)
col5 = [0,0.1,0.2,0.4,0.6,0.8,1.0];

%offset (rel. width from vert center of keel @max width position y/(G/2)
col6 = [1.0,0.9,0.82,0.69,0.6,0.55,0.52];

x=[];
y=[];
z=[];
points=[];
d = 0.4*(D-E);
```

```
%offset in x by which max width section (lower curve) must be shifted to align
% the maximum thickness of the lower and upper curves

fileID = fopen('Scheel_output.txt','w');

%create the cross section of minimum thickness
for i=1:length(col1)
    x(i)=D*col1(i);
    y(i)=(H/2)*col2(i);
    z(i)=-1*C;
    points=[x(i) y(i) z(i)];
    fprintf(fileID,'%.4f %.4f %.4f\r\n', points);
end

%create the cross section of maximum thickness
for i=1:length(col1)
    x(i)=E*col1(i)+d;
    y(i)=(G/2)*col4(i);
    z(i)=-1*A;
    points=[x(i), y(i), z(i)];
    fprintf(fileID,'%.4f %.4f %.4f\r\n', points);
end

%create the bottom CL points

for i=1:length(col1)
    x(i)=F*col1(i)+d;
    y(i)=0;
    z(i)=-1*(A+B);
    points=[x(i), y(i), z(i)];
    fprintf(fileID,'%.4f %.4f %.4f\r\n', points);
end

%create the lateral surface curves

for i=1:length(col5)
    x(i)=0.4*D;
    y(i)=(G/2)*col6(i);
    z(i)=(A-C)*col5(i)+(-1*A);
    points=[x(i) y(i) z(i)];
    fprintf(fileID,'%.4f %.4f %.4f\r\n', points);
end

fclose(fileID);
```

Figure 97: Scheel Keel Code