The Type 600 Project

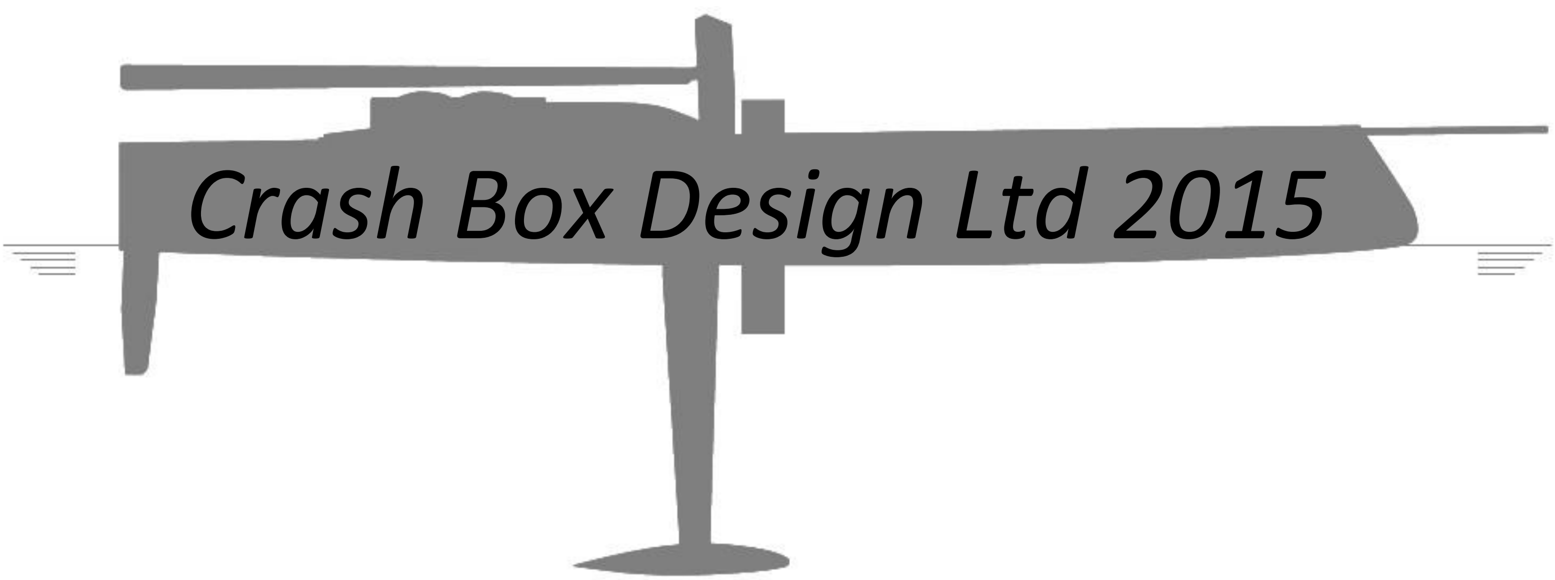
IMOCA 60

Adam Day

Liam Johnston

Tristan Edwards

Carolyn O'Rourke



Acknowledgements

We would like to thank the following people for their assistance and support throughout the design of The Type 600 Vessel, Crazy Horse.

Nathan Smith

For his time spent review a rather rough first draft

Ryan Williamson

For his steady hand, camera skills and willingness to take photos of the tow campaign during midterm break.

Morgen Watson

As a valuable reference regarding the many difficulties of offshore sailing

Bruce Colbourne

For his understanding, patience and willingness to accept this "crazy yacht thing" as a capstone design project

Dan Walker

For his support in allowing this project get off the ground in term seven

Technical Services

For all the hard work done on our model

Trevor Clark

For his endless hours riding the tow carriage with the team

To all the Professors, past and present, who gave us the tools to make this possible.

Executive Summary

The Vendée Globe 2020 will be one year shy of the centennial of the launch of the Bluenose. This famed Canadian schooner inspired people from across the nation, both directly through its success and indirectly through the many songs and homages to its achievements. While the days of wooden ships and iron men have passed, the need for inspiring Canadian figures remains. The Vendée Globe represents a race where such historic personalities are made.

The Vendée Globe is a round-the-world single handed sailing race which takes place every four years. The vessels must be designed to International Monohull Open Classes Association 60 class rules. While there have been two past Canadian entrants into the race, a Canadian finish, much less a Canadian win has never been achieved. Currently, *Canadian Offshore Racing* is looking to enter skipper Eric Holden in the 2016 Vendée Globe. While an entry would be a major step forward in developing a Canadian presence in offshore racing, Mr. Holden will need a world class vessel if he is to chase the 2020 Vendée Globe title. In the following report, Crash Box Design Ltd. presents the work done in The Type 600 Project for the concept design of *Crazy Horse*, a world class IMOCA 60.

By evaluating race statistics it was found that roughly half of the vessels entered into the Vendée Globe retire. Due to these statistics, Type 600 Vessel design was approached with the primary mission being to safely complete the race, while as much as possible optimizing performance. Performance levels were enhanced both through design elements as well as improvements in habitability.

Performance elements such as Dynamic Stability System foils, weight optimization, both structurally and elsewhere, sail plan and hull form were all evaluated to maximize performance returns while operating as an effective, cohesive unit. As part of the hull form evaluation, Crash Box Design Ltd. conducted a model test campaign to evaluate the effect of heel on resistance and the seakeeping improvements of a reverse-stem bow.

Increasing skipper comfort and ease of vessel operations was achieved through careful arrangement of the cockpit and main hold area. A 3D arrangement of the cockpit was created ensuring proper geometric proportions and ergonomics were attained. The most notable outcome of these habitability efforts was a retractable hard dodger with viewport bubble under which the skipper can helm while protected from the elements.

As this vessel was designed to inspire and keep pace with modern developments, a complete renewable energy generation package was installed on the vessel. While other IMOCA 60s rely partially on renewable energy, *Crazy Horse* will be the second in its class with near 100% green energy rating.

While the design of the vessel is what makes it a viable competitor, to maintain such a campaign requires the financial support of a corporate sponsor, the general public, or both. To increase pubic engagement and the benefits to corporate sponsors, a system of video cameras have been placed throughout the vessel, providing personal daily updates of lift as sea. It was determined through analyses of past Canadian campaigns that the financial structure of the campaign should be split such that construction costs are supported by large corporate donations while the operational costs are derived from private donations. A brief list of potential sponsors is also presented.

Overall *Crazy Horse* represents a design that can compete at an international level while being attractive to both the skipper and potential sponsors. To summarize, this will be a vessel designed by Canadians, for a Canadian skipper, and supported by the people of Canada.

A supporting Drawing Package is provided to supplement this report and associated appendices.

Table of Contents

Table of Figures

Table of Tables

Acronyms

The following are a list of acronyms used in this report:

AWA	Apparent Wind Angle
AWS	Apparent Wind Speed
C_D	Coefficient of Drag
CE	Center of effort
CFD	Computational Fluid Dynamics
CL	Centerline
C_L	Coefficient of Lift
CLR	Venter of Lateral Resistance
DSS	Dynamic Stability System
ELA	Electric Load Analysis
GA	General Arrangement
GHS	General HydroStatics software
HM	Heeling Moment
HWG	Heavy Weight Gennaker
IMOCA	International Monohull Open Classes Association
ISO	International Standards Organization
ITTC	International Towing Tank Conference
IWO	In Way Of

I_{xx}	Second Moment of Area
J1	Jib 1
J2	Jib 2
J3	Jib 3
LCG	Longitudinal Center of Gravity
MOD	Multi One Design
NOAA	National Oceanic and Atmospheric Administration
RA	Righting Arm
R&D	Research and Development
RM	Righting Moment
RMS	Root Mean Square
S.A.L.T.S.	Sail And Life Training Society
SAN	Styrene Acrylic Nitrate
S_M	Section Modulus
SW	Salt Water
TCG	Transverse Center of Gravity
TYP	Typical
VCG	Vertical Center of Gravity
VG	Vendée Globe
VLM	Vortex Lattice Method
wrt	With Respect To

1. Design Concept

The goal of the Type 600 project is to develop a vessel designed to International Monohull Open Classes Association (IMOCA) 60 Class rules with the intent being to compete in the 2020 Vendée Globe (VG) non-stop, round the world, sailing race. The VG demands a vessel capable of circumnavigating the globe singlehandedly, unassisted in a reliable and competitive fashion.

Vendée winners have been getting faster finishing times every event with the winner of 2012 VG finishing the course in just over 78 days. This is a significant improvement over the 109 days winning time of the first VG in 1989 (VG, 2013). Ultimately, the goal is to complete the race in the fastest time. Finishing the race at all is considered a success and is a huge challenge in itself. As a non-stop race of up to 100 days long, it is an endurance challenge on both the vessel and the skipper. The vessel must be self-sustained for that time with very robust rigging and equipment, and enough stores to last the duration of the race.

Past IMOCA 60 design failures were analyzed by the team. On average over the past VG races, 43 percent of the boats do not finish the race for one reason or another. Thirty percent of past failures are due to the steering system, half of these are from losing control and half from rudder failure. Another 27 percent of failures are due to mast or rigging failures, while 21 percent of failures are due to keel damage or loss of stability. Although keel failure is a lesser occurrence, the severity of the consequences makes it a serious concern. Most of these design failures occur while the skipper is asleep (Vendee Globe).

While focusing on optimizing the design of the steering system, mast and rigging system, and keel system, The Type 600 Project design team also aims to develop a habitable arrangement to minimize preventable damage leading to severe failure. By combining these four key aspects into the design, The Type 600 Project focuses on optimizing the safety, speed, and comfort of the vessel.

In 2005 IMOCA 60 class rules were implemented into the VG Rules. All competing vessels must fit within the ‘box rule’ of the class. However; there is still some room for innovation in aspects of the design such as the rudders, foils, hull shape, arrangement and equipment selection. As far as monohull circumnavigation sailing goes, the VG is the premiere international event. It has consistently pushed for advances in composite hull construction, composite spar construction and introduced innovative design features such as the canting keel and active ballast tank systems. With The Type 600 wave piercing bow, Dynamic Stability System (DSS) foils, powering by renewable energy, a focus on creating a comfortable design, and with skipper Eric Holden at the helm, The Type 600 Vessel will be a strong competitor in the upcoming 2020 VG race.

2. Statement of Requirements

The primary mission of The Type 600 Project design is to finish the race safely with the secondary mission of finishing first. The main criteria for this design were established as follows:

1. The Type 600 Vessel shall meet or exceed IMOCA 60 class rules.
2. Maximum static righting moment (RM) shall be 32 ton-m.
3. The maximum lightship displacement shall be 7 tons.
4. The Type 600 Vessel shall have a maximum design speed of 28 knots.
5. The Type 600 Vessel shall be operational in category 0 weather conditions as defined by Offshore Sail and Racing.
6. The Type 600 Vessel shall have 100 days endurance while single handed.
7. Auto Pilot settings shall provide options for either steering to a course or steering to the wind.
8. There shall be a clear path from the navigation station to the cockpit.
9. All electronics shall be powered by renewable energy.
10. There shall be two (2) bunks.
11. The navigation station shall be accessible from one (1) of the bunks.
12. The keel shall cant 38 degrees to each side.
13. The canting keel system shall be hydraulic with manual lockout.
14. The DSS foil lifting system shall be mechanical with a manual option.
15. There shall be two (2) rudders, one port and one starboard.
16. The cockpit shall provide shelter from harsh weather to the skipper.
17. The skipper shall be able to change the sails single headedly.

3. The Vendée Globe Route

The VG route takes the sailors from France, south into the Southern Atlantic Ocean, then east through the Antarctic circumpolar current until reaching Cape Horn where they turn north and race back towards the Northern Atlantic Ocean and cross the finish line in Les Sables d'Olonne, France. A summary of the route in Figure 1 and some of the conditions and challenges the skippers will face is provided by the Vendée Globe website (The Route, 2012).

Figure 1: The Vendée Globe Route (The Route, 2012)

At the start of the race skippers will be faced with the challenge of travelling through the Bay of Biscay which often contains violent winds. With any luck, a south blowing wind will provide the skippers with a quick ride south where they will need to pick up the trade winds to be in position to travel through the Doldrums.

Once entering the Doldrums, the skippers will be faced with another significant challenge. The weather in this area can include erratic winds, violent thunderstorms and sometimes torrential rain. The skippers will have spent a lot of time studying wind charts and preparing for this leg of the race.

Once approaching the Indian Ocean and the roaring 40's, the skippers will be in a completely new environment of low light, dangerous seas, cold weather and violent winds, while completely alone. This area is where the tradeoff of safety and speed comes into play. Traveling further south into the raging 50's would be a shorter path, but would increase the risk of collision with ice.

Once crossing the International Date Line, the ocean conditions will be much more calm, but dangers of icebergs drifting north will still be a concern. The vessels radar will detect large icebergs but is incapable of detecting smaller ice pieces such as growlers. The skipper must be much more alert to avoid collision in this region.

Once rounding Cape Horn, the sailors will sail upwind through sometimes violent winds and back through the Doldrums. Ideally the skipper will catch the westerly winds to sail directly back to the finish line.

4. Operating Conditions - Environmental Data

The vessel must be designed and built to withstand the environmental conditions expected for the duration of the race. Holden is trained in navigation and will be prepared for the environmental conditions he will encounter. CBDL based The Type 600 Vessel design on environmental data extracted from a web application created by NOAA (National Oceanic and Atmospheric Administration) which allows users to generate global wind diagrams given a date and location. The 2020 VG will begin in November and last no more than 100 days, therefore the wind speeds and direction for the months of November, December, and January were analyzed. Figure 3 shows average wind speeds and directions.

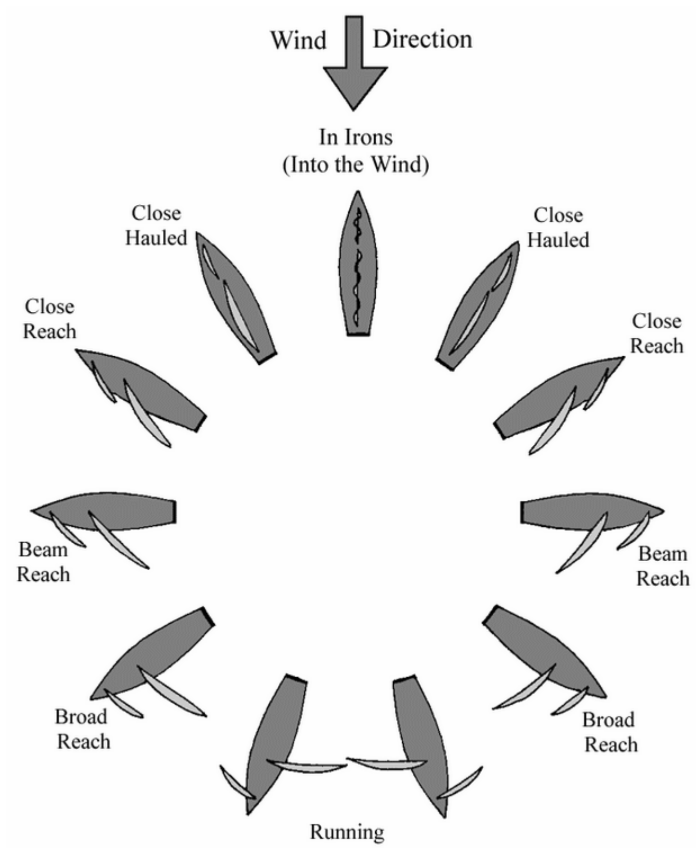


Figure 2: Sailing Conditions

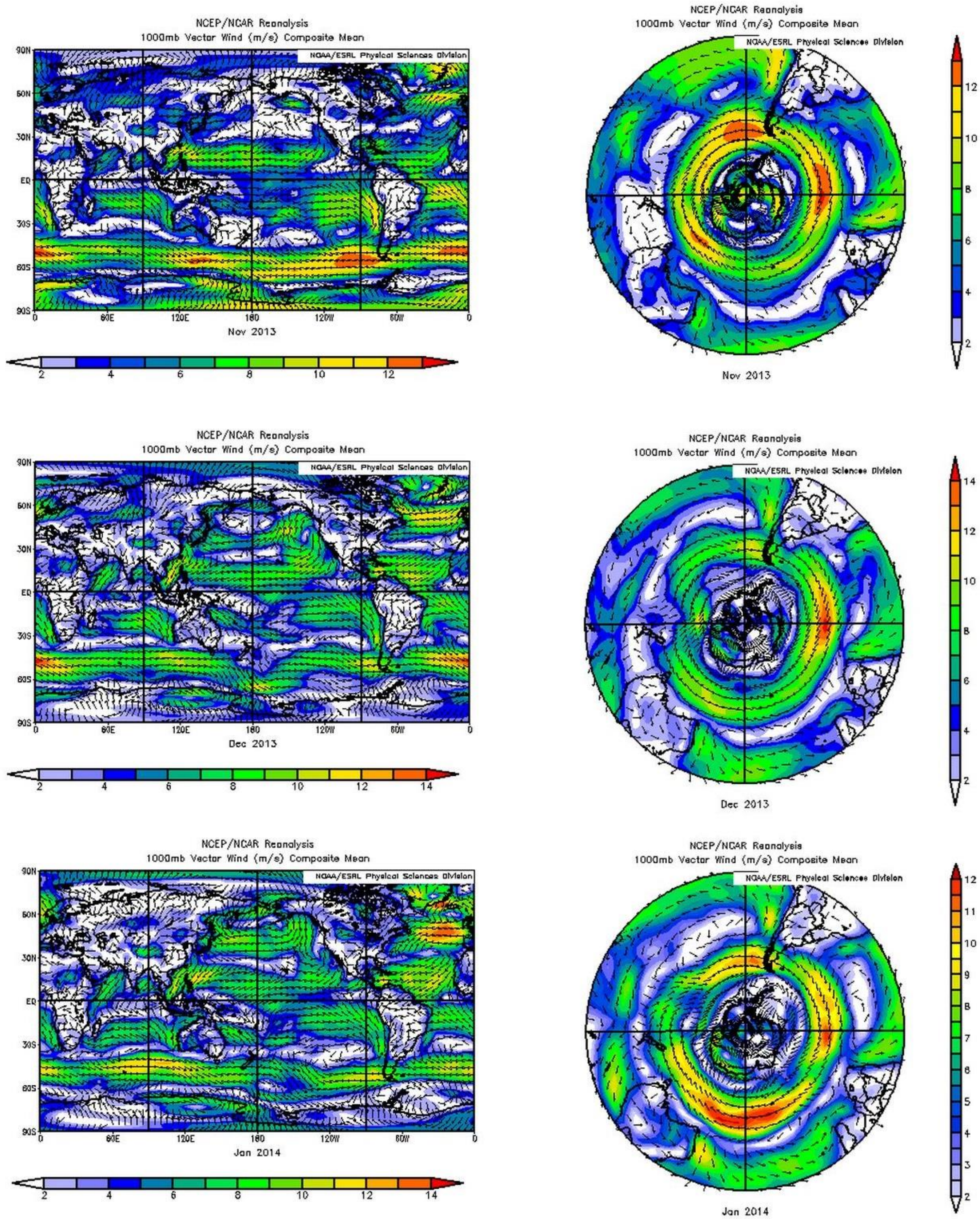


Figure 3: Wind Charts (Monthly/Seasonal Climate Composites)

Based on these charts, environmental data for specific points along the route as labeled in Figure 1 were determined. In Table 1, these wind speeds were related back to the Beaufort scale to determine a wave height approximation. The wind speeds presented are an average for each month which does not necessarily reflect the severity of the conditions. For example, when navigating the Antarctic Circumpolar Current, the vessels can encounter vicious winds of up to 35 knots but the averages show 14 to 19 knot winds.

Table 1: Environmental Data

Location (Figure 1)	Wind Speed	Wind Speed	Sailing Condition	Wave Height	Beaufort Scale
[-]	[m/s]	[kts]	[-]	[m]	[-]
1	7	14	Running	1.75	4
2	8	16	Broad Reach	2	5
3	10	19	Close Hauled	2.75	5
4	11	21	Close Hauled	3	5
5	6	12	Close Hauled	1.5	4
6	9	18	Running	2.7	5
7	9	18	Running	2.7	5
8	10	19	Running	2.75	5
9	8	16	Running	2	5
10	7	14	Broad Reach	1.75	4
11	9	18	Running	2.7	5
12	4	8	Running	0.7	3
13	5	10	Close Hauled	0.8	3
14	10	19	Close Reach	2.75	5
15	14	27	Running	4	7

The Sailing conditions in Table 1 are based on the wind directions as displayed in Figure 2.

5. Lines Plan Drawing

The lines development in Figure 4 of the Type 600 Project began in the Summer of 2014 with a parametric study of hull lines from previous VG competitors. The lines of Macif, the VG 12-13' winner, were used as a starting point. This hull exhibited many of the qualities which we considered important of the VG within the Open 60 Rules. A fairly beamy and flat aft section would provide great hullform stability and righting moment, which also acts as a good planing surface for higher speeds. The narrow sharp stem provides some wave piercing qualities for a cleaner entry in rough seas. These lines were modified throughout the Fall of 2014. A more dramatic wave piercing bow, with reverse rake was used in place of the vertical stem, as an attempt to further reduce energy loss when hitting waves in rough seas. The lower portion of the bow was increased in volume to provide greater buoyancy to prevent submarining. The chine was also further accented as it moves aft to encourage planing mode at higher speeds.

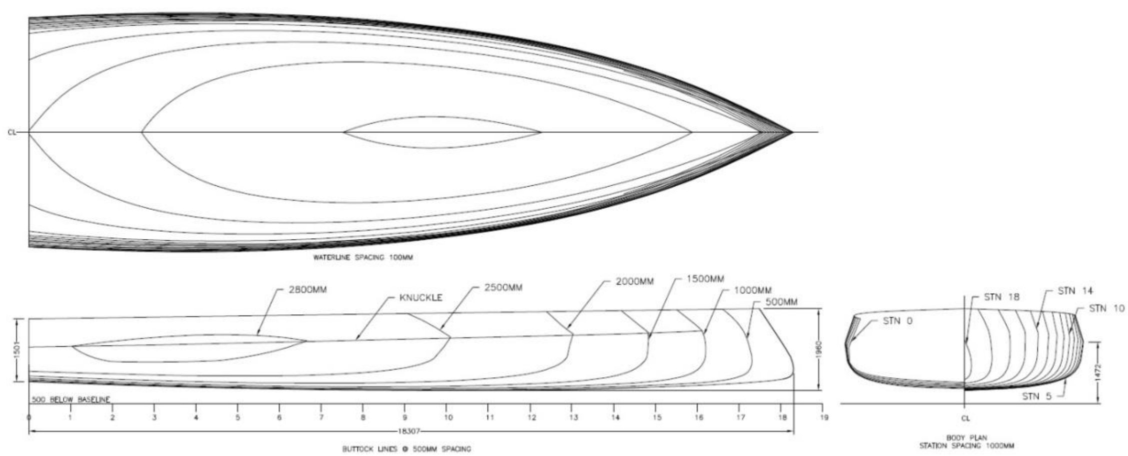


Figure 4: The Type 600 Lines Plan

6. General Arrangement

The general arrangement (GA) for the Type 600 Project has evolved along with the hull form and weight requirements, with focus on ergonomics and usability of the vessel. It is important to note that the usability and habitability of a VG sailing vessel is much more extreme than the habitability of day to day life on land or on any other type of vessel. Since the main tasks of the skipper are to steer, navigate and adjust the sails accordingly, the main area of focus of the GA is the habitability of the navigation station, cockpit and sleeping areas.

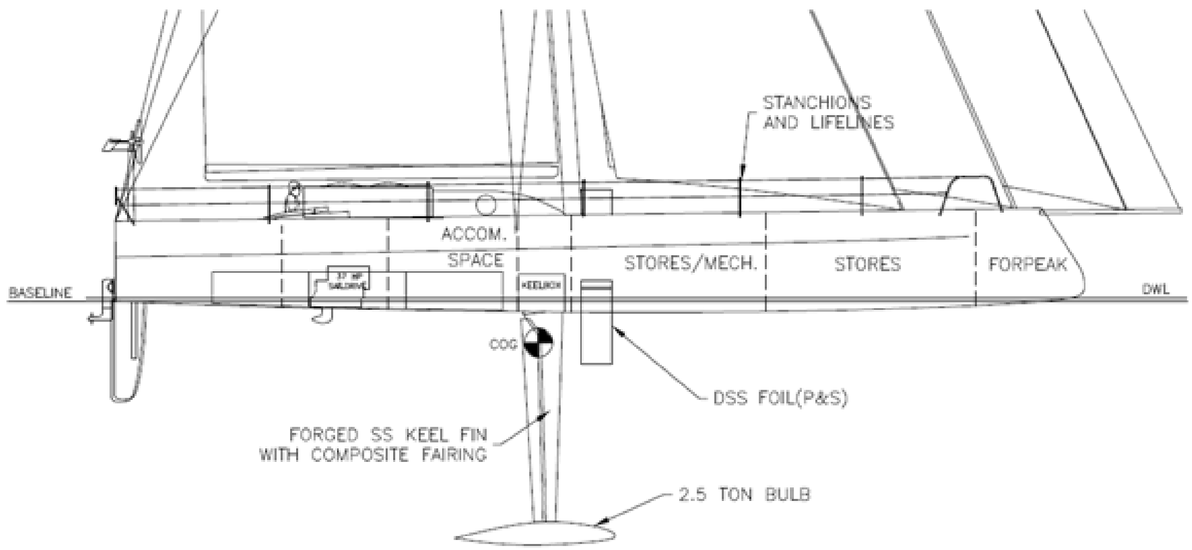


Figure 5: Profile view of The Type 600 Project

Having a small crew of one (or two in other events), space is not a big limitation in the arrangement of the cockpit and accommodation area. Instead, practicality and ergonomics are the major drivers for the arrangement of much of the spaces. Some of these arrangement decisions are skipper dependent and can be customized as it is outfitted. In many IMOCA 60 builds, a full scale mock-up of the cockpit and accommodation area will be made in order to get a better feel for final arrangements. As The Type 600 Vessel is being designed for Eric Holden, the arrangement in Figure 5 is designed to suite his preferences. The full general arrangement drawings are in the attached Drawing Package.

6.1. Living Spaces

Down below in the accommodation/navigation station space, cooking/eating, sleeping, maintenance, communication and navigation are the primary activities to be performed. Given the extreme nature of ocean sailing, IMOCA Rules require that all items must be fastened securely below, whether permanently or temporarily for storage. The layout of controls and living spaces must be optimized to make operations as comfortable and as simple as possible for the skipper.

6.1.1. Navigation Station

As the navigation is a key component to the VG and because the skipper, Eric Holden, has a background in navigation; a critical part of The Type 600 Vessel design is to make the navigation station comfortable and modern. Shown in Figure 6, the navigation station will be located on the forward cabin bulkhead, with a gimbaled berth/navigation seat aft of it. This transverse bunk will allow for access to communications and navigation gear while lying down, and be useable on either tack (Vendee Globe TV, 2015).

6.1.2. Sleeping

IMOCA 60 class rules require that the vessel be designed to sleep two. One berth as mentioned above is at the navigation station. The alternative berth will run longitudinally along the starboard side shell near the companionway. This berth allows for resting with quick access to the cockpit for sail or steering adjustments and to the navigation station to check the course and radar.

6.1.3. Galley

All galley gear will be lightweight and portable, but with straps and dedicated surface space for strapping them down for use. A propane burner, pot, kettle, bowl, fork, spoon and knife will be onboard. There is a surface in the galley area to allow for easy and quick food preparation of all meals. The pantry area in the galley on the port side of the cabin space will provide space for the freezer dried meals, powdered food and energy bars to be stored in tubs labeled with their content. The skipper will eat while they man the navigation station or steer the vessel and therefore a settee is not necessary (Vendee Globe TV, 2015).

6.1.4. Head

Due to weight saving measures, it is typical of IMOCA 60 vessels to use a bucket instead of a head and dump the waste overboard (Vendee Globe TV, 2015). A wide plastic circle has been designed to sit on the rim of the bucket to replicate a toilet seat and increase the comfort for the skipper.

6.1.5. Sails up and down

A large deck hatch is located in the sail compartment two meters forward of the mast to allow the direct transport of sails from the sail store to the forward deck and back again. This makes switching sails easier, faster and much safer that dragging them aft out the main companionway and then forward onto the deck.

6.1.6. General Comfort

The duration of the 2012 VG winner was 78 days. It is expected that the 2020 VG winners will be faster but going into the race the skippers are prepared to be sailing for up to 100 days. Even though there are communication systems and contact with shore crew and family, small comforts from home will make the time in isolation more bearable. Most skippers will have a small stuffed toy for comfort. One of the main comments from past VG competitors is the comfort they get from music (Vendee Globe TV, 2015). There is a stereo system to allow music to be played throughout the vessel. The vessel color scheme is red and white to represent Canada. The color red also has many positive and inspirational psychological effects. It is seen as a warm, positive and energetic color. Red is associated with motivation, ambition and determination for the will to survive. Although these impacts seem small, they contribute to helping the skipper maintain their drive to complete the race.

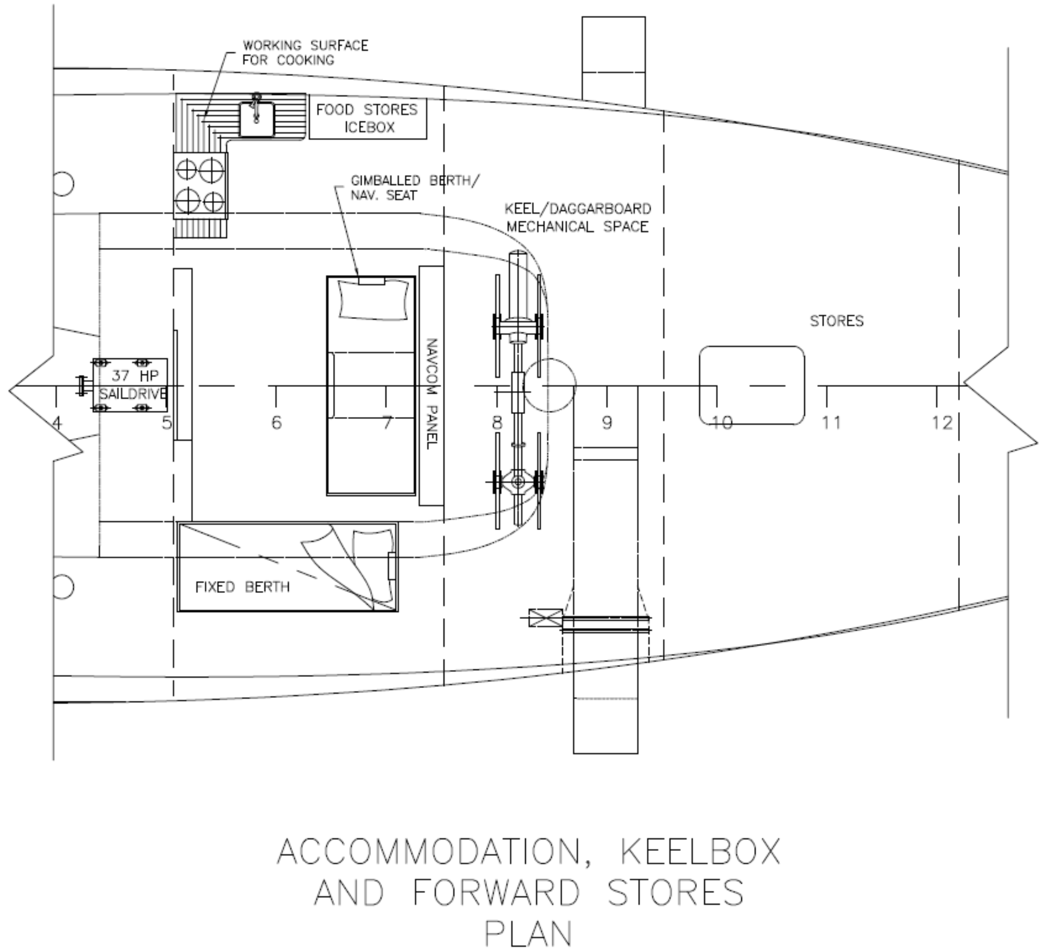


Figure 6: Plan of accommodation space and hold

6.1. Cockpit

In single handed endurance sailing the skipper will miss out on sleep to allow for a few more hours at the helm, especially in extreme circumstances of bad weather, ice infested waters or equipment failure. Maximizing the comfort of the cockpit allows the skipper to maintain a close watch on the course and the sails with minimal distractions.

6.1.1. Steering System

A mechanical steering system has been selected over a hydraulic system because it is easier to perform maintenance and repair. A wheel steering system uses a lot of valuable cockpit space compared to a tiller system. A tiller also provides better control when constant adjustments are required. A mechanical tiller is the preferred set up for many skippers and racing vessels. The rudder is properly balanced with about 20 percent of the total rudder side area forward of the rudder stock and the rudder stock forward of the center of pressure. This provides good control and requires a comfortable force from the skipper to turn the tiller and steer the vessel. There is a reliable autopilot system which can be set to maintain a course in relation to the wind or maintain a compass course and the skipper can adjust the sails as needed.

6.1.2. Protection from Elements

It is important that the skipper be as comfortable as possible while they are steering the vessel so they are able to stay at the helm longer. One of the main comments from sailors is how important protection from the elements is to their comfort. There is a hard dodger on The Type 600 Vessel design which can be pushed forward and out of the way or extended aft for shelter by a sliding motion to protect the skipper from weather. There is a bubble window on the port and starboard top side of the hard top dodger so that the skippers' line of site does not vanish in bad weather when the dodger is extended aft (Vendee Globe TV, 2015).

6.1.3. Navigation

There is a small navigation screen located in the cockpit near the entrance to the cabin with a bench for the skipper to sit and sleep on. This allows for the skipper to take short naps while still being within reach of the rigging and steering controls for minor adjustments without having to get up or move too far. The cockpit is located near midship where the motions of the vessel are minimized for increased comfort.

6.1.4. Rigging Control

From Figure 7, all of the lines are fed down the deck of the boat and into the cockpit on the port and starboard sides. Each sail has a designated color for its lines which can be arranged in designated cleats within the cockpit, based on the skipper's preference. This provides a simple system to adjust the sails which allows for quicker reactions and less chance of mistakes which is essential in rough seas especially as the skipper will most likely be sleep deprived. There is a large grinder in the center of the cockpit which can assist in tightening each of the lines. The grinder is aligned longitudinally which provides better footing and support when traveling in wind and waves than a transverse arrangement (Vendee Globe TV, 2015).

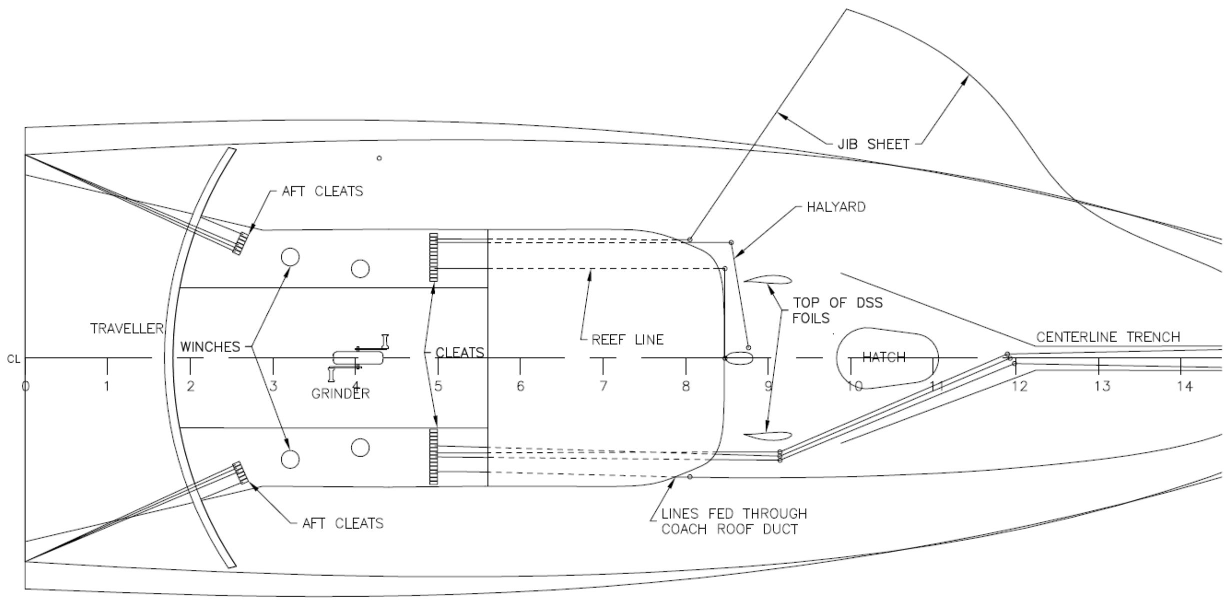


Figure 7: Lines General Layout

7. Weight and Stability

Weight control has been critical throughout the development of the Type 600 Project. The initial approach to a weight estimate was to obtain an understanding of the lightship displacement and lightship longitudinal center of gravity (LCG) of existing IMOCA 60s, preferably from the lighter vessels from the most recent VG 2012-13races. Lightship displacement was readily available as published data, and displacement values were found for each event winning vessel dating back to the first race in 1989. Lightship displacement has been steadily dropping in this class over the last 25 years, from 13 tons in 1989, to just 7.7 tons in the 2012-13 event. (VG, 2013) This trend is shown in Figure 8.

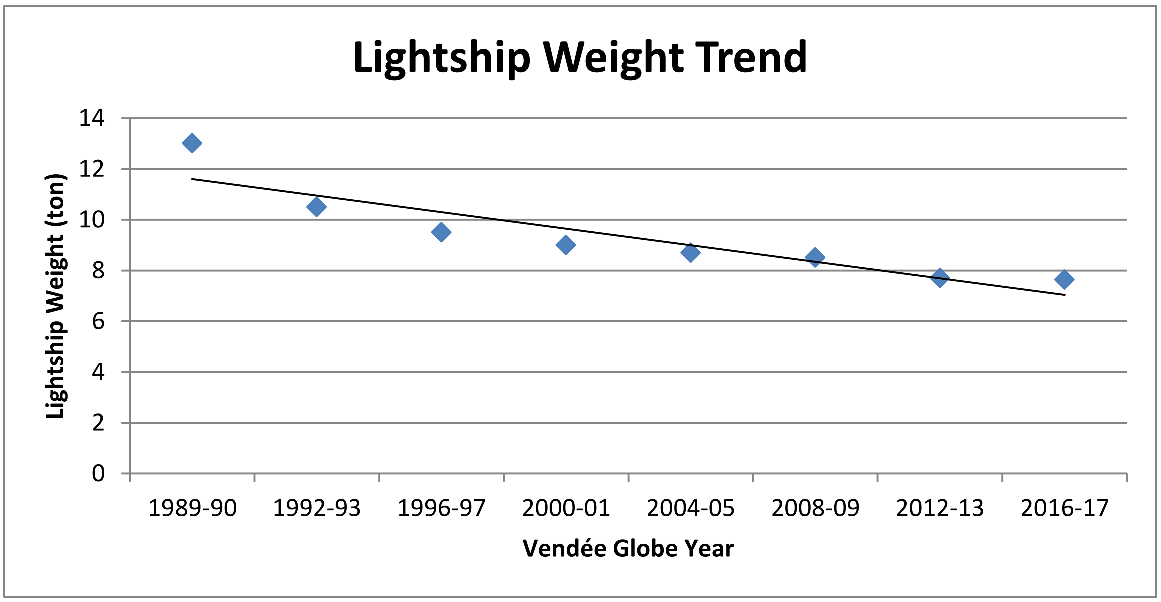


Figure 8: Lightship weight reduction throughout the history of the VG

This dramatic weight reduction is a good representation of the innovation which this class has seen so far. Some of these weight saving innovations are; composite construction advances, structural optimization, and new design approaches made possible by the canting keel and variable ballast tanks. Any competitors in future events with new boats will continue to reduce weight as technology advances further.

7.1. Weight Reduction Method

To be competitive, The Type 600 Vessel will follow this weight reduction trend, and make use of a canting keel, ballast tanks, and lightweight structure. There is a compromise between a heavier boat for a more powerful hull and larger righting moment, versus a lighter boat which will plane earlier and have greater acceleration and speed in low to moderate wind speeds. It was decided to minimize the weight to 7 tons, without compromising strength or reliability.

7.1.1. Canting Keel and Bulb

Limiting the keel bulb weight would be one simple example of an approach to this. IMOCA 60s in the past have used keel bulbs as heavy as 3.5 to 4.5 tons, and this provides excellent righting moment resulting in a very powerful hull/ballast arrangement. Keeping the Type 600 keel bulb within the 2.5-3 tons range would allow for significant weight savings, although at the cost of some righting moment. Even with this reduced keel bulb weight, the benefits of the canting keel can clearly be seen in figures below, which compare the righting arm (RA) curve of the vessel with the keel canted at just 5 degrees, and canted at its max of 38 degrees. It can be seen that the righting energy, and angle of heel at max righting moment shifts further left, providing this energy earlier on in the heeling process. The static righting arm also is seen to increase as the keel cant angle is increased.

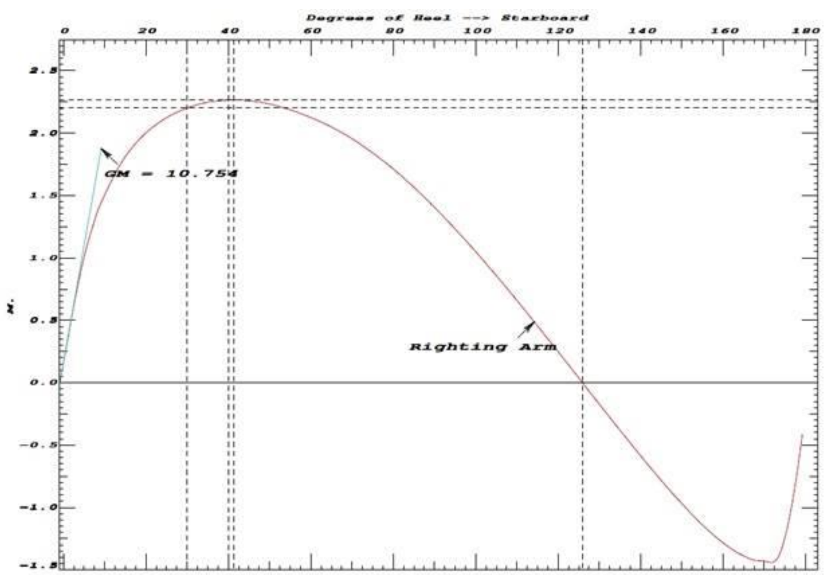


Figure 9: Righting Arm curve for 5 degrees cant

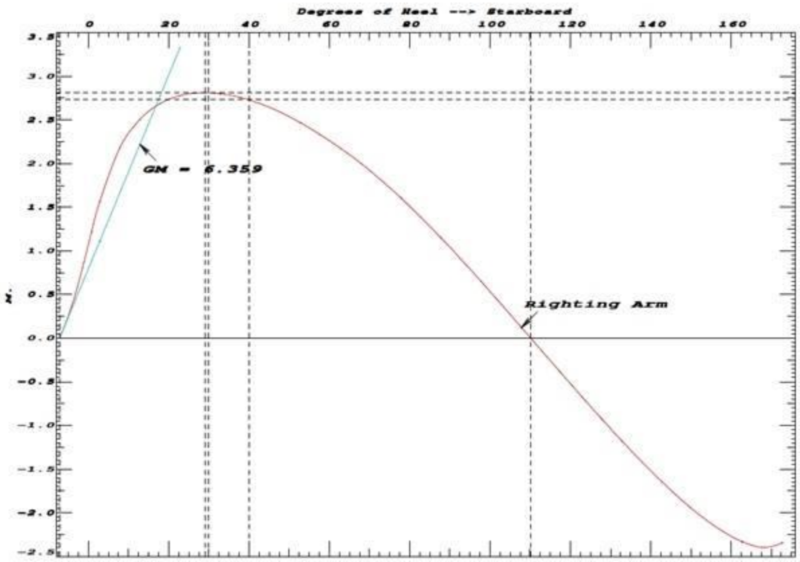


Figure 10: Righting Arm curve for 38 degrees cant

The full stability analysis from General HydroStatics software (GHS) can be found in Appendix 1 (General Hydrostatics, 2012).

7.1.2. Ballast Tanks

With weight reduction being a key objective, focus was also put on other methods of gaining back some righting moment and hull power. An early solution to this was to rely more on saltwater ballast tanks located port and starboard to aid in righting moment, this would mean the vessel could be very lightweight in light and moderate winds, and ballast-up to gain power in strong winds.

Three sets of port and starboard wing tanks with capacities between .7 and .9 m^3 were selected based on The 10 Degree Rule. The 10 Degree Rule requires that in the most extreme ballasting condition (ballast tanks filled to one side, keel canted 38 degrees to same side), the vessel does not heel more than 10 degrees. This rule is one of the fundamental stability requirements of the IMOCA class, and is meant to be physically tested once the vessel is launched and ballasted, prior to beginning a race. During the design phase, this was checked through GHS. The ballast tanks were sized to achieve an angle of heel as close to 10 degrees as possible without going over. The ballast tank arrangement from the GHS model can be seen in Figure 11.

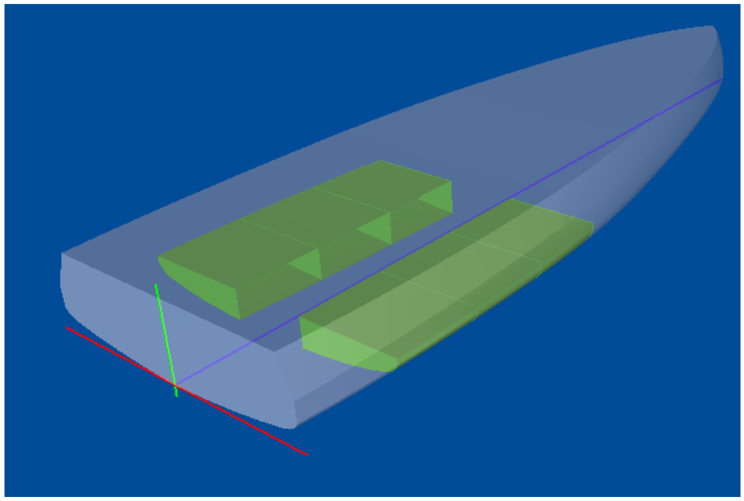


Figure 11: Ballast Tank Arrangement in Stability Model

7.1.3. Dynamic Righting Moment

Another way of creating greater righting moment is with the use of DSS foils, a proven appendage technology with maxi-yachts, which had yet to make it to IMOCA class vessels. In Fall of 2014, the IMOCA committee announced that it would be allowing the use of DSS style foils (although with some limitations), and the decision was made for the Type 600 Project to utilize this technology. An example of existing DSS foil technology is shown in Figure 12.

Figure 12: Existing DSS foil technology as seen on an Infiniti Yacht

The dynamic righting moment produced from such foils would further aid in keeping the weight down, without performance compromise. As the VG route is about 80% downwind/across wind, DSS foils seem perfectly suited as they function best in reaching conditions. The design of these appendages is discussed in section 12. Figure 13 shows the static and dynamic righting arm of the vessel at 10 degrees heel with the keel canted 38 degrees in lightship condition. This clearly shows the potential increase of righting arm at speeds over 10 knots, meaning that if the conditions are right, dynamic righting moment can be a significant advantage.

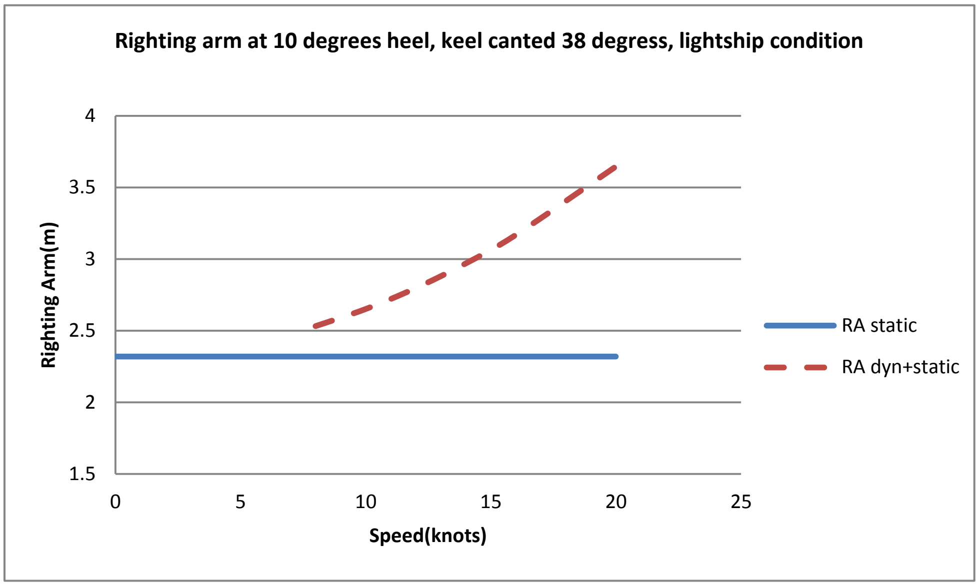


Figure 13: Static and Dynamic Righting Arm as speed increases

7.1.4. LCG and Trim Control

Initial longitudinal center of gravity (LCG) estimates were based on stillwater photos and scanned lines plan sketches of existing IMOCA 60 vessels of a known displacement. Trim was roughly measured and an LCG was calculated using GHS.

LCG position is very important for a planing craft of this type, and has been monitored and maintained to achieve a small amount of aft trim in lightship. In operation, the three (3) sets of ballast tanks will be used for LCG and trim control. This LCG control would be fine-tuned for various conditions during sea trials and IMOCA events leading up to the 2020 VG. However, the basic trimming methodology would be to fill the forward most tanks when sailing upwind, and gradually shift ballast aft for points of sail off the wind. In varying conditions, ballasting would have to be decided while sailing based on point of sail, sea state and wind conditions.

7.2. Weight Estimate

Although the lightship value is over the initial goal of 7 tons, it is still within an accepted range. One area where this weight could be reduced is in structure optimization, which is discussed further in section 9. Another strategy may be to further reduce the weight of the keel bulb, putting more reliance on DSS foils for dynamic stability. The weight summary table below shows that appendage weight and hull structural weight combine to make up almost 80% of the total lightship weight. The full weight estimate is shown in Appendix 2.

Table 2: Summary of Weight Estimate

Sub Category	Weight	LCG	VCG	TCG
[-]	[kg]	[m]	[m]	[m]
Structure	2320	11	0.785	0
Outfit and Miscellaneous	200	7	1.5	0
Appendages(including bulb)	3793.3	8.04	-3.29	0
Rigging	773.3	9.1	7.8	0
Systems	215	6.12	0.242	0
Electrical	335.6	4.27	0.237	-0.115
Totals	7637	8.0	-0.6	-0.005

7.3. Future DSS Development

In future VG events, it may be possible to drastically reduce lightship values of these vessels if technology such as DSS proves to be practical for solo circumnavigation. If the DSS foils are a success in the 2016 VG, and if IMOCA adapts its appendage rules accordingly to allow for more flexibility, then it is very likely that the next generation of IMOCA 60s will be made much narrower and hence lighter. This will result in a less powerful hull from a static perspective, but ideally the dynamic righting moment provided by proper DSS foils would make up for this.

7.4. IMOCA Stability Requirements

The IMOCA class rules with regards to stability are mainly focused on a “powering limitation”, setting maximums for righting moment, as opposed to conventional commercial stability criteria which usually sets required minimums for these values.

These powering limitations are in place partly for safety, limiting the loads on rigging and attainable sailing speeds, as well as serving to keep the vessels a true(or close to) conventional monohull form.

Table 3: Stability Criteria Summary Table

Rule	Description	Approach	Pass/Fail
D1	Self-righting, normally done as an experiment with the completed outfitted vessel	RA curve compared for various keel cant angles, negative RA observed at 180 degree	Pass at 5 degree to 38 degree cant
D2	10 degree initial heel angle	Keel fully canted, tanks full on same side	Pass
D3	AVS > 127.5	Read From 0 cant RA curve	Pass
D4	AVS worst case > 108 degree	Read from 38 degree RA curve lightship	Pass
D5	Stab curve ratio, > 5:1	Read From 0 cant RA curve	Pass
D6	Max righting moment requirement < 32 ton-m	Very light vessel, achieved ~30tonm with ballast tanks full	Pass

7.5. Final Remarks

The ballasting arrangements have been designed to make full use of the righting moment limitation, and all other criteria have been met. If advancements in dynamic righting moment technology are further developed and class rules evolve to allow that, then it is likely the next set of IMOCA rules will have quite different stability limitations which could drastically change ballasting arrangements of the IMOCA class.

8. Tow Tank Testing

VG competitors average 12 knots over the duration of the race. This also represents the transition point of the hull from the displacement zone to the planning zone. There is very little knowledge about how to analyze the full speed spectrum of a planning hull using empirical software making it challenging to evaluate the performance of The Type 600 Vessel. There is also limited empirical information on analyzing hulls at angles of heel. As a sailing craft, the Type 600 Vessel will spend most of its time heeled over. The vessel also has an innovative reverse rake bow. The raked back wave piercing bow is not generally seen in VG competitors, however it is not uncommon on other high speed sailing vessels such as Volvo Ocean Racing and Americas Cup competitors. CBDL built a model and completed three days of tow tank testing to evaluate the bare hull design performance of the planning hull with a wave piercing bow.

8.1. The Model and Set Up

A 1:10 scale model was built by The Department of Technical Services in ENGR 1023 at Memorial University. The model was built out of layers of foam with renshape reinforcement at the tow post attachment and a Hydraulic Crush Point (HCP) 60 backbone for strength on the bottom along center line. The layers were glued together with epoxy. An epoxy coating was applied to the outer surface of the hull to provide additional strength. The model was finished with two coats of Duratek and fairing putty. The final weight of the model was 7.2 kg.

Traditionally the tow post weight would then have to be considered as part of the ballast weight of the model. However; with a full scale weight goal of just 7 tons and 1:10 model scale there was no weight allowance for the addition of the 7 kg tow post. CBDL designed an innovative counter balance for the tow post. A minimal stretch line was attached to the tow post and run through a pulley system to a 7 kg kettle bell as displayed in Figure 14. An additional line was tied to the kettle bell to minimize the impact of the kettle bell hitting the tow post particularly in the accelerating stages of the tests.

The Type 600 Project will operate in many heeled positions during the race and CBDL wanted to replicate these realistic conditions by performing tow tank tests at different angles of heel. There was no current mechanism to hold the model at a constant heel so the team designed and built renshape wedges to attach the model to the tow post at 10 degrees heel and 20 degrees heel. These wedges allowed the same tow post to be used for all the tests which maintained freedom of motion in pitch and heave. The wedge built for 10 degrees of heel is shown in Figure 15.

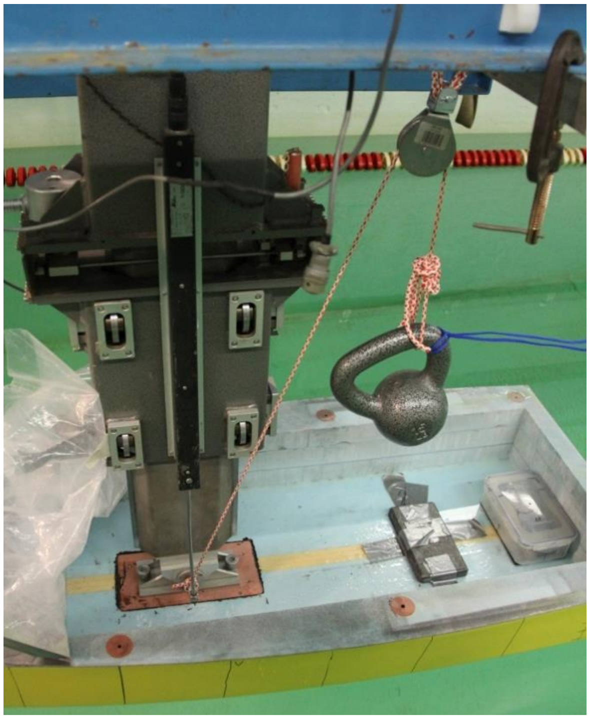


Figure 14: Counter Balance Set Up

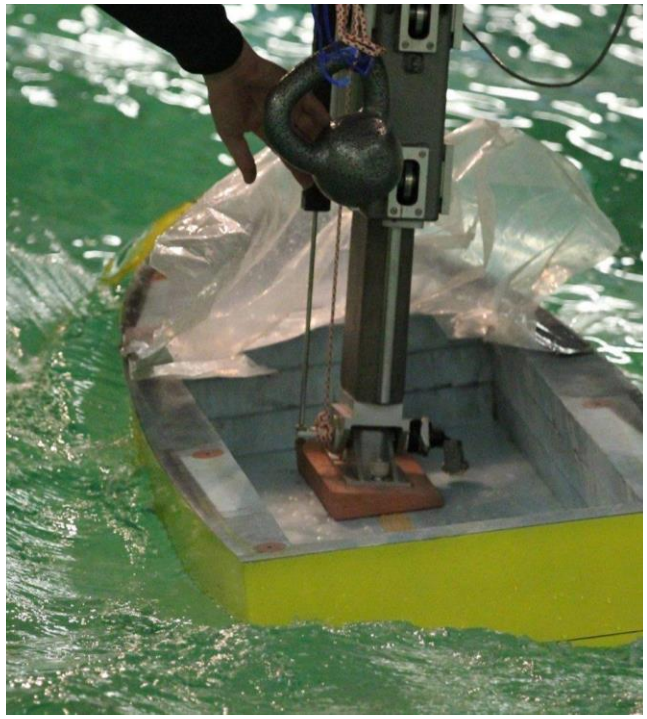


Figure 15: 10 Degree Heel Block Set Up

8.2. Experimental Objectives

Based on the limited time available for tank testing, the team outlined the following experiment objectives:

1. To estimate the bare hull resistance to compare with NavCad.
2. To evaluate the planing trend of the hull as the speed increases.
3. To compare vessel performance at three (3) angles of heave(0, 10, and 20 degrees) and two ballasting conditions.
4. To observe the performance of the wave piercing bow in waves.

8.3. Bare Hull Calm Water Tests

The model was tested in calm water at Froude numbers ranging from 0.23 to 1.08 model speeds which equates to full scale speeds from 6 to 28 knots at 2 knot intervals. The model was run at three (3) different angles of heel; 0 degrees, 10 degrees and 20 degrees. Each test was performed in two conditions; unballasted and ballasted with 3.422 kg mass in the stern giving a trim of 0.9 degrees aft. The ship and scaled model parameters are displayed in Table 4.

Table 4: Parameters

Parameter		Model	Full Scale	
Scale	λ	1	10	[-]
Length	L_m, L_s	1.8288	18.288	[m]
Beam	B_m, B_s	0.4153	4.153	[m]
Draft	T_m, T_s	0.0184	0.184	[m]
Mass	M_m, M_s	7.2	7200	[kg]
Ballast Mass	m_m, m_s	3.422	3422	[kg]
Density of water	ρ_m, ρ_s	999.2	1027.8	[kg/m^3]
Kinematic Viscosity	ν_m, ν_s	1.16×10^{-6}	1.63×10^{-6}	[m^2/s]

Experimental resistance and heave, which will be referred to as ‘Dynamic Heave’, values were extracted from the tank test results.

To scale the measured resistance to full scale, the wetted surface area for each test was required. These values were obtained by estimating the waterline and pitch angle from video footage as well as estimating planing displacement. Using these estimates, the model orientation was recreated within Rhinocerous 3D, which allows for a calculation of wetted surface area. The full scale wetted surfaces were then scaled accordingly.

Based on the conventional International Towing Tank Conference (ITTC) 1957 correction line methodology from *Engineering 4011 Ship Resistance and Propulsion Course Notes*, the model resistance values were scaled to full scale (Colbourne, 2012). Model and full scale non-dimensional coefficients of total resistance, frictional resistance and residual resistance were found based on the experimental resistance and estimated wetted surface areas at each speed, heel and ballast condition. The correlation allowance for a standard planning hull used in this analysis is 0.00025. Once the total ship resistance coefficient was found, it was converted into a ship resistance value based on the water density, velocity of the full scale vessel and corresponding wetted surface area. Refer to Appendix 3 for full ITTC scaling methodology.

8.4. Bare Hull Sea Keeping Tests

A major design aspect of the hull is the reverse rake wave piercing bow. The model was tested at speeds equivalent to full scale 8, 14, 18 and 22 knots with the same range of heel angles, and ballast conditions as noted in section 8.3, in combination with two (2) sinusoidal waves; 6hz and 8hz as displayed in Table 5. These tests were performed for qualitative purposes only due to the expected differences in the models inertia from the full scale vessel.

Table 5: Wave Information

Wave Data	Wave One	Wave Two
Model Wave Frequency	6 hz	8 hz
Model Wave Period	1.66s	1.25s
Model Wave Height Approximation	0.20m	0.11m
Full Scale Wave Period	5.27s	3.95s
Full Scale Wave Height	2.0m	1.1m
Full Scale Wind Speed	19 knots	13 knots
Beaufort Wind Scale	5	4

8.5. Results and Discussion

8.5.1. Bare Hull Resistance

Scaled ship resistance vs Froude number comparison between ballasted and unballasted conditions are plotted at 0, 10 and 20 degrees of heel in Figure 16, Figure 17, and Figure 18, respectively. Scaled ship resistance verses Froude number comparison between 0, 10 and 20 degree heel are plotted for ballasted and unballasted condition in Figure 19 and Figure 20 , respectively.

It can be seen from the trends in Figure 16 and Figure 17 that the planing hump is more prominent in unballasted condition than in ballasted condition. This is due to the placement of the ballast weight which created 0.9 degrees trim aft, making it difficult for the vessel to get up onto a plane causing the vessel to perform as a semi-planing hull at the speed range tested. Comparing the trends of ballasted verses unballasted conditions from Figure 19 and Figure 20, this trend is more prominent as there is no planing hump in the ballasted condition. Consideration will need to be taken with how the vessel is ballasted. This is a very complex and skipper specific task. Further testing and integration with the skipper’s preferences would be considered in the next phase of design to create a detailed ballasting arrangement. Further testing will need to be done to develop a full ballasting plan for the vessel.

From Figure 18, at 20 degree heel there is no planing hump in either ballast condition. This is due to the shape of the submerged hull while heeled having a finer form and a deeper draft. This can be seen again from the unballasted condition results in Figure 19. The 20 degree heel curve follows a very different trend compared to the 0 and 10 degree curves. When operating from 13 to 23 knots (between Froude numbers of 0.5 and 0.9) the resistance at 20 degree heel is much greater than at 0 or 10 degree heels. As most VG competitors spend most of the time in this speed range, the vessel ballast arrangement will need to take this into consideration so that the vessel can limit the heel angle while traveling between 13 and 23 knots.

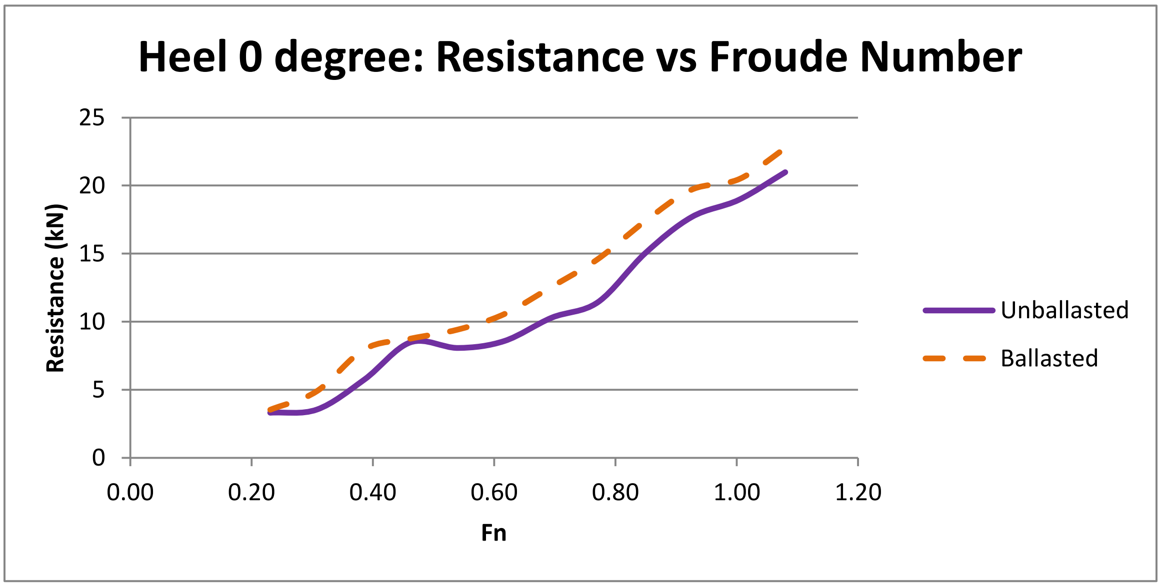


Figure 16: Resistance vs Froude Number at 0 degrees heel

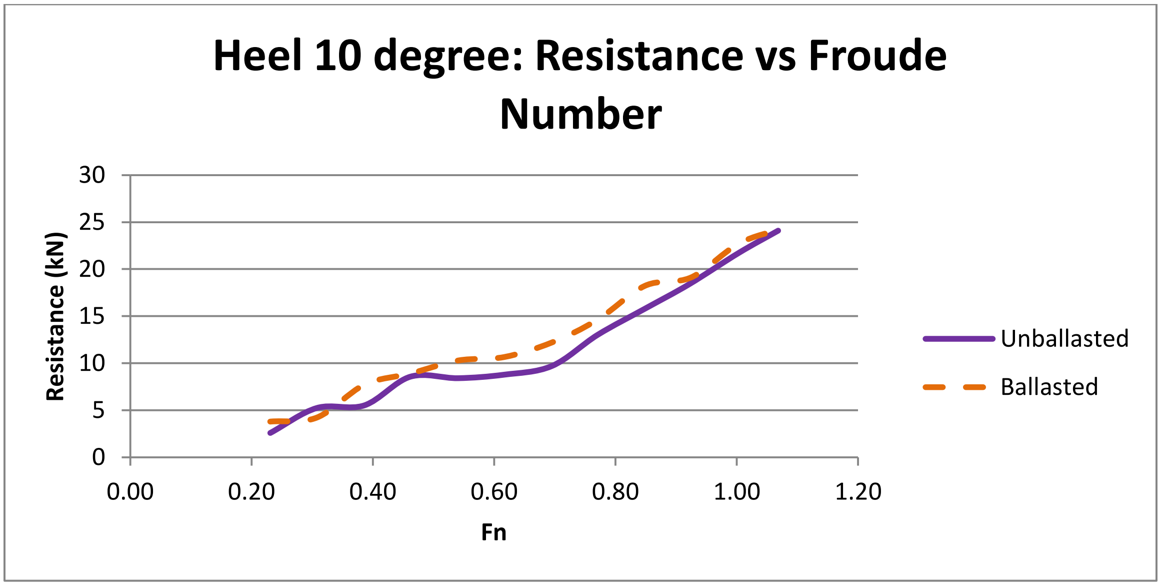


Figure 17: Resistance vs Froude Number at 10 degrees heel

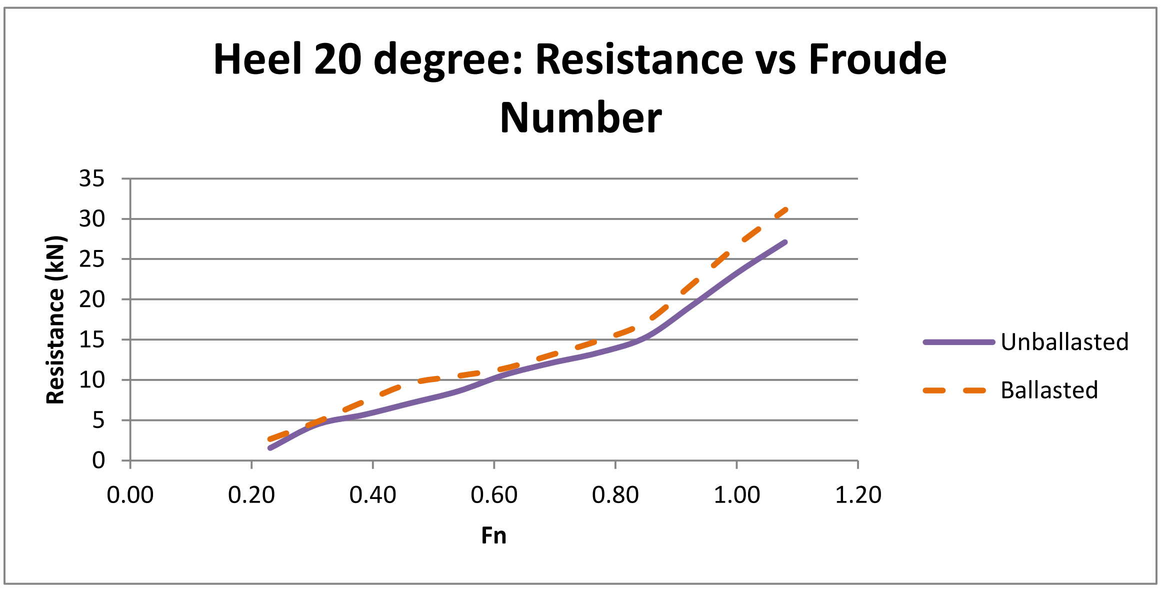


Figure 18: Resistance vs Froude Number at 20 degrees heel

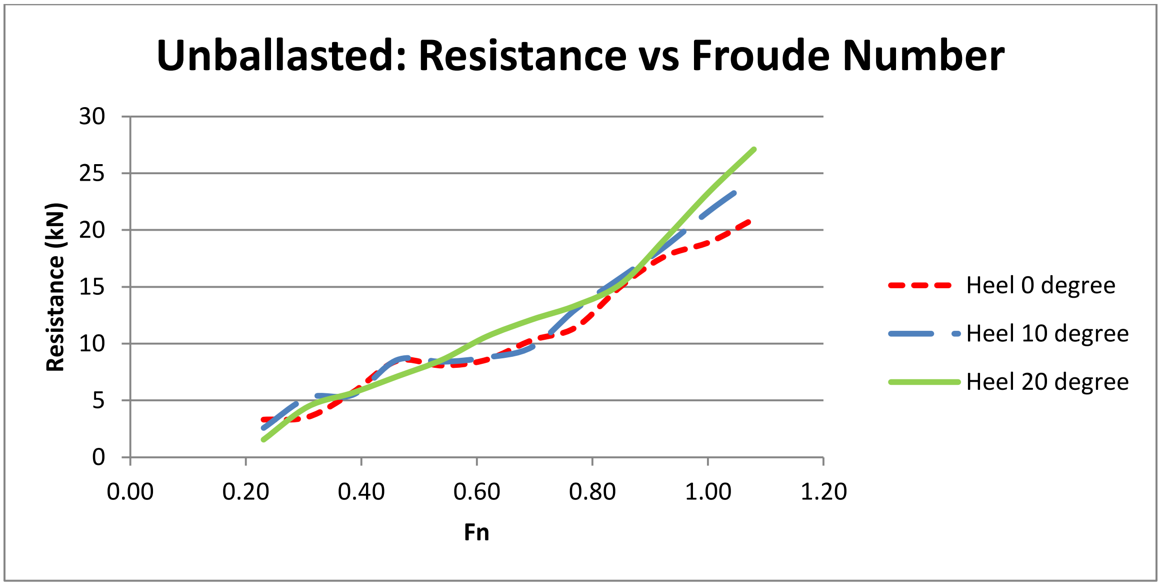


Figure 19: Resistance vs Froude Number in Unballasted condition

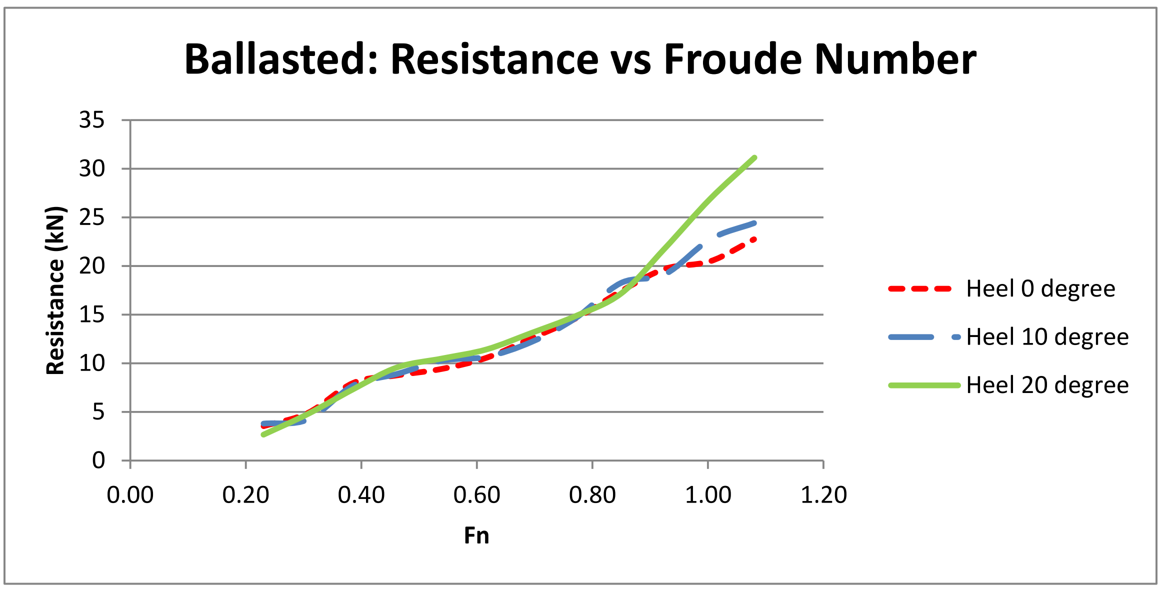


Figure 20: Resistance vs Froude Number in Ballasted condition

8.5.2. Bare Hull Dynamic Heave

Model dynamic heave verses Froude number is plotted for each ballast condition and angle of heel in Figure 21. This figure shows the dynamic heave trend of the vessel as it transitions from displacement mode up into planning mode. The ballast conditions have higher dynamic heave values than the unballasted condition, however, when considering Figure 19 and Figure 20, which shows that the balasted condition does not reach the planning hump, it was thought that the reason for the large heave value is due to the model trimming aft in semi-displacement mode and the location of the heave sensor. This was confirmed by watching the video recordings of the tests.

Sinkage begins to appear during around a Froude number of 0.5, equivalent to 13 knots full scale, which is the speed where the vessel begins to transitioning up onto a plane. This is due to the location of the heave sensor and the trim of the vessel as it transitions. Once up on a plane, the trim levels out and the dynamic heave results are in the positive.

At 0 degree heel there is higher dynamic heave results than to 20 degree heel. This is because of the trim on the model as discussed in Section 8.5.1, the model was not planing with 20 degrees of heel.

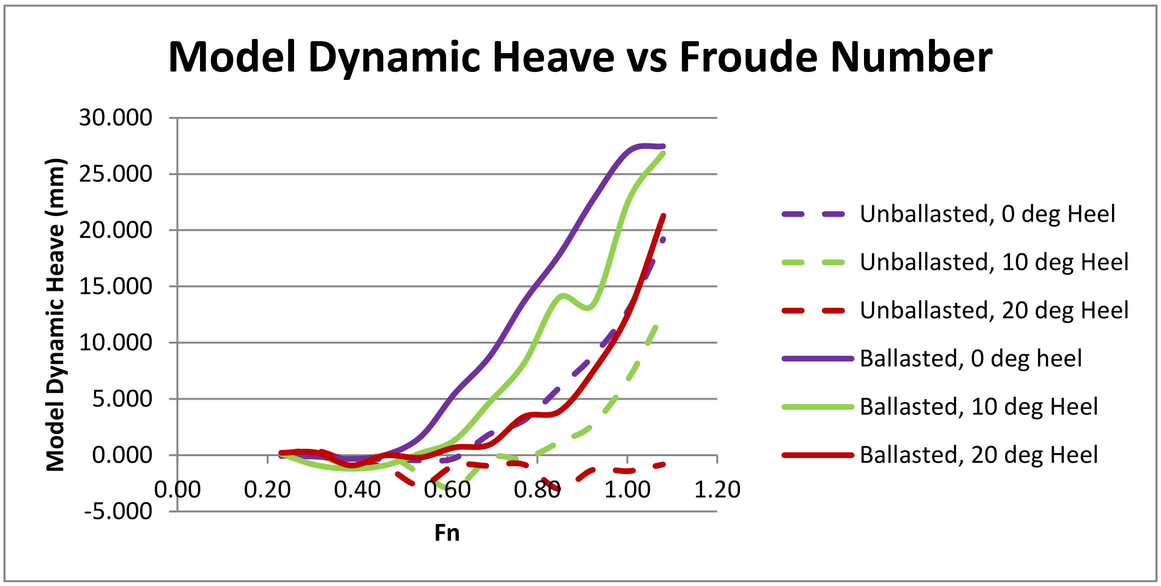


Figure 21: Dynamic Heave vs Froude Number

8.5.3. Sea Keeping

The images were reviewed and analyzed to evaluate the seakeeping characteristics of the hull.

Figure 22 shows the model in unballasted condition moving through 0.6 hz waves with 0 degree heel at Froude numbers of 0.31, 0.54, 0.69, and 0.85 which are equivalent to full scale speeds of 8, 14, 18 and 22 knots. It was noted that at low speeds, 8 and 14 knots, the wave piercing bow cut smoothly through the waves. The pictures in Figure 23 provide a sample of the sea keeping of the model hull in unballasted condition at 14 knots full scale speed and 0 degree of heel in 0.6 Hz wave as the bow pitched in and out of the water. As the speed increased, the bow plunged deeper into the water. Due to the beamy shape of the hull, these dives would create a significant amount of additional buoyancy forcing the bow to rise out of the water. With the bow out of the water, it did not interact with the waves and showed no performance benefit at 18 and 22 knots. During the high speed tests, much more water was taken into the model cockpit area than during low speed tests.

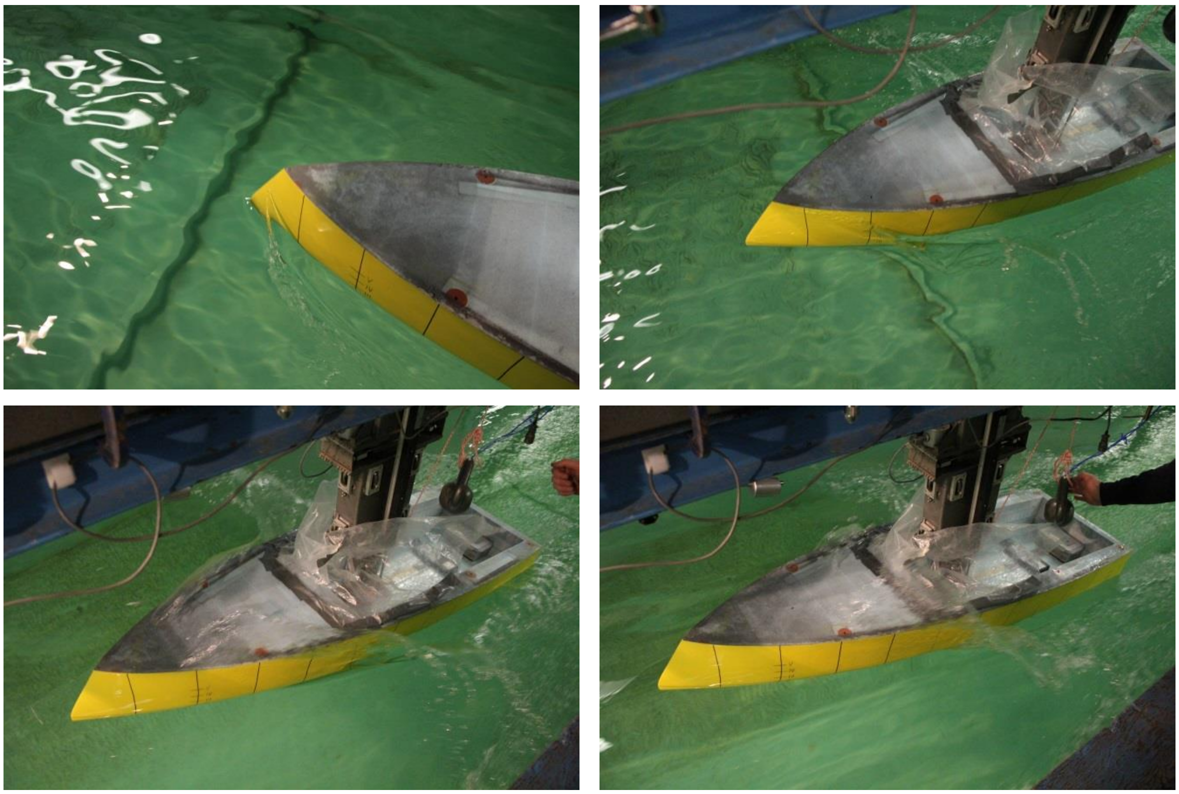


Figure 22: Ballasted condition with 0 degree heel in 0.6 Hz wave at: 8 kts (top left), 14 knots (top right), 18 knots (bottom left), 22 knots (bottom right). (Photo Credit to Ryan Williamson)

Figure 23: Unballasted condition at 18 knots with 0 degree heel and 0.6 Hz wave. (Photo Credit to Ryan Williamson)

The unballasted model tests had more violent heaving and pitching motions than the ballasted model tests. This may have been due to the location of the ballast creating a trim aft, as well as the added inertia. Additional experimentation with ballasting options to keep the bow interacting with the waves in each sea state would be considered in the next phase of design.

Due to the extreme motions of the unballasted model and time constraints in the tow tank, unballasted testing at angles of heel were not preformed. At 0 degree heel, the waves would splash up and in towards the middle of the model. By comparing Figure 22 with Figure 24, at angles of heel, the waves were projected out and away from the model cockpit and took on less water.

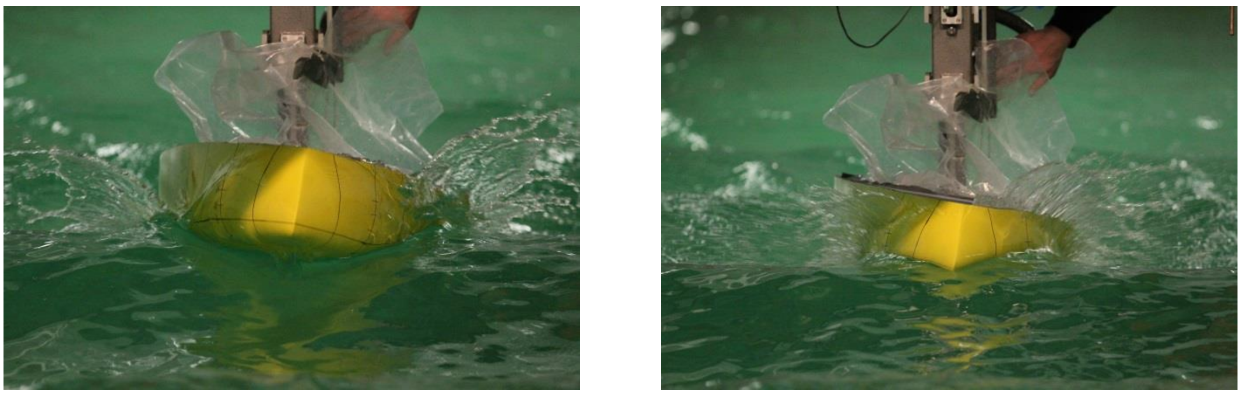


Figure 24: Ballasted condition with 10 degree heel and 0.8 Hz wave at 14 knots (left) and 18 knots (right). (Photo Credit to Ryan Williamson)

8.6. Validation

The bare hull resistance tank tests were compared with Hydro Comp NavCad 2007 Results. A NavCad resistance prediction analysis was performed for the unballasted condition at 0 degree heel. In Figure 25, the full scale resistance results from the tow tank results are plotted along with the NavCad Savitsky and Delft 2/3 Methods. The Delft 2/3 method improves upon the parent Delft prediction method by re-analyzing high speed results as well as specializing on light displacement vessels with high Beam over Draft ratio (B/T) values and deep keel sailing yachts. The Savitsky prediction method is meant for a prismatic planning hull with a constant deadrise. This method works from theoretically derived and empirically corrected formula of lift and drag. Over the development of this prediction method, practical corrections have been made to the input data making this method more universally applicable and compatible with modern hull forms. Savitsky is the ‘general case’ resistance prediction for planning hulls in NavCad (NavCad, 2014).

It is evident from Figure 25 that near a Froude number of 0.5, which corresponds to approximately 13 knots full scale, the transition is made from displacement mode of the Delft prediction method up onto the 'planing hump', into semi-planing/planing mode of the Savitsky prediction method. The results from the tank test are within 10% of the NavCad results; however without performing the tank tests, it would have been impossible to determine where the transition was made from displacement to planing mode. As this transition occurs around the average expected speed of the vessel during the race, the tank test data is very valuable to the design.

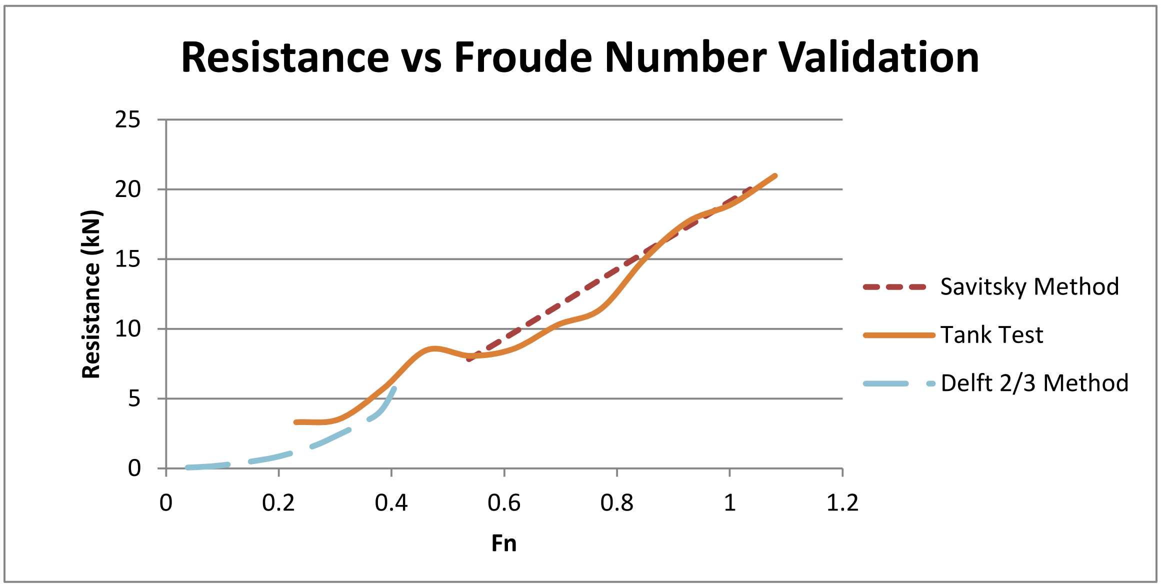


Figure 25: Compare Tow Tank Resistance with NavCAD Methods (NavCad, 2014)

8.7. Final Remarks

Although the model was tested with regular periodic waves, in operating condition the vessel would experience irregular sea states rather than sinusoidal waves as tested. The 0.6hz and 0.8hz waves used in the tank test are representative of 1.1 m and 2 m full scale wave height which correspond to a Beaufort Wind scale of 4 and 5. From section 0 The Type 600 Vessel is likely to encounter these conditions during the race.

There may have been additional sources of error due to the counter balance of the tow post. The counterbalance system added in some frictional heave damping, as well as a change in inertia due to the extra mass in the system. The arrangement of the counterbalance had a lifting line angle of 30 degrees, meaning there was some horizontal force component pulling on the post. This was fine for the calm water tests, but in seakeeping tests this dynamic horizontal component had an effect on the resistance values which could not be compensated for. However, due to time constraints in the tow tank, a validation of the counterbalance method would be completed in the next phase of design.

As the Type 600 tow tank model did not have any appendages or rigging, the mass moment of inertia and radius of gyration are very different from the fully outfitted ship, in both pitch and roll. The keel bulb alone makes up 35 percent of the total lightship displacement of the fully outfitted boat, and is located 4.4m below the hull baseline. It is this large mass concentration and distance from the hull that has the largest effect on this difference in mass properties.

To confirm this, the radius of gyration and mass moment of inertia for the complete vessel was calculated using mass properties from the 3-D model for the structural deck and shell, and point masses for all outfit and equipment from the weight estimate. An estimate was completed in this way for both a fully rigged vessel with keel/bulb, and an appendageless “barehull” version with no mast. These values were then scaled and compared with swingframe results for radius of gyration of the unballasted model.

As shown in the Radius of Gyration Summary Table below, the swingframe radius of gyration in pitch and roll compare well with the weight estimate calculated values for the “barehull”, but are significantly less for the values for the fully outfitted ship. For this reason, as well as the tow post counterbalancing effects, seakeeping results from the towtank are only used qualitatively. The full swing frame radius of Gyration calculations are in Appendix 4.

Table 6: Radius of Gyration Summary Table

Motion	Radius of Gyration			
	Swingframe	Ship scaled with Mast/keel	Barehull Ship scaled	**Rule of Thumb Estimate
[-]	[m]	[m]	[m]	[m]
Roll	0.10	0.31	0.09	0.20
Pitch	0.27	0.68	0.35	0.46
$** \; Radius\ of\ Gyration, Roll = 0.35 * Beam, Radius\ of\ Gyration, Pitch = 0.25 * Length$				

9. Structure and Structural Integrity

Due to the combined requirements of both high strength and low weight for the structure of the vessel, there is no modern alternative to use for hull structure beyond composite construction. Though costly, composite construction allows the material properties as well as the structure to be tailored to the applied load resulting in extremely strong and weight efficient design. The Type 600 vessel employs a unidirectional carbon fiber skin over and aramid fiber honeycomb core for the vast majority of its structure. Through variation of skin and core thicknesses, vessel panel properties were modified to suit the regional loading conditions. Within the scope of this report, a first approximation of panel weight has been produced based on application of *International Standards of Organization (ISO) 12215*. There are many areas of local loading that will require further analyses and optimization, thus at this stage only an estimate of the added weight due to these local refinements is presented.

9.1. General Considerations

The following briefly describes the primary loadings that the structural members of the vessel are exposed to.

1) Dynamic Loads: slamming and green water on deck, keel and mast acceleration
    - Exposure to these loads located forward of midships on the hull (slamming) and fore-deck (green water) as well as twisting and bending of hull throughout vessel.
2) Static Loads within the hull: hull forces from hydrostatic forces, bending moments caused by keel and standing rigging tension, and torsion forces caused by an imbalances of loads.
    - Stresses range from compression to shear and are distributed throughout the monocoque of the hull as well as into the bulkheads.
3) Local Loading: imposed by keel and DDS foil as attachment regions, mast compression and standing rigging tension and point loads from attachment points.

The use of unidirectional carbon allows for directionally-specific application of material properties in response to the above described loads. The aramid fiber honeycomb core greatly increases the panel stiffness and strength with minimal impact on weight. This sandwich construction is employed for the monocoque hull, deck and bulkheads. The longitudinal stiffeners are constructed with unidirectional carbon laid over a core material of Styrene Acrylic Nitrate(SAN). The reason for the variation in core material is twofold: first, SANS-A can be shaped to fit the hull far easier than honeycomb and secondly, it is more suitable for fatigue and slamming applications. These properties also make it an attractive choice for specific hull-core locations as noted later.

9.2. Design Methodology –Code and assumptions

A first estimate of structure was done through implementation of the *ISO 12215-1, Small Craft – Hull construction and Scantlings* (hereto *referred to as 'ISO Standard'*). The standard covers pleasure vessels under 24m in length, providing requirements for both sailing and powered craft, constructed of composites, aluminum or glued-wood (plywood). It is based largely on two ABS standards: The Ocean Racing Yacht Guide and The Motor Yacht Guide (Lars Larsson, 2014). The ISO Standard was applied through instruction presented in *Principles of Naval Yacht Design* (Lars Larsson, 2014).

9.2.1. ISO Standard Application

The primary concern of the structural rules is to derive structural members that are capable of the expected environment while at the same time having the flexibility to be applicable over a range of conditions. To scale the intensity of seas states that a vessel may encounter, the ISO Standard implements a 4 category system, 1 being an offshore vessel and 4 being applied to vessel of coastal nature.

Structural elements are typically of two types, panels (hull, deck and bulkheads) and stiffeners (longitudinals and local reinforcing structure). Each panel was given a design pressure based on longitudinal location, panel geometry and region type: hull, side shell, deck or bulkhead. For certain areas such as the hull and side shell, the design pressure takes into account dynamic actions such as slamming. For these regions it results in the design pressure at the bow being almost twice that at the stern. Design pressure is also applied to stiffener dimensioning, thus overall, the forward portion of the vessel will be comprised of more robust structure.

Once the design pressure was established for each panel, its geometry was evaluated and minimum section modulus (S_M) and second moment of area (I_{xx}) were established. As all panels evaluated where of cored construction, shear stress and minimum effective thickness were also evaluated.

In addition to these structural calculations, a minimum skin thickness for the inner and outer skins was calculated based on the length and lightship displacement.

Over the course of the evaluation it was found that the minimum skin thickness and minimum effective thickness dictated the minimum panel geometry and thus drove how much panel weight could be optimized.

9.3. Structural Arrangement

Presented below are the panels' regional properties. The full analysis can be found in Appendix 5 and the panel locations and supporting sections found in the Drawing Package.

Table 7: Panel Dimensions

Panel Geometry		Hull	Side Shell	Deck	Bulkhead
Core	[mm]	25	25	13	25
Inner	[mm]	1.5	1.3	1.3	1.9
Outer	[mm]	2.1	2	1.9	1.9
Regional Density	[kg/m^3]	7.9	6.7	5.5	7.1
Regional Area	[m^2]	74	55	71	43
Regional Weight	[kg]	582	371	394	303
Total Weight	**[kg]**	**1650**			

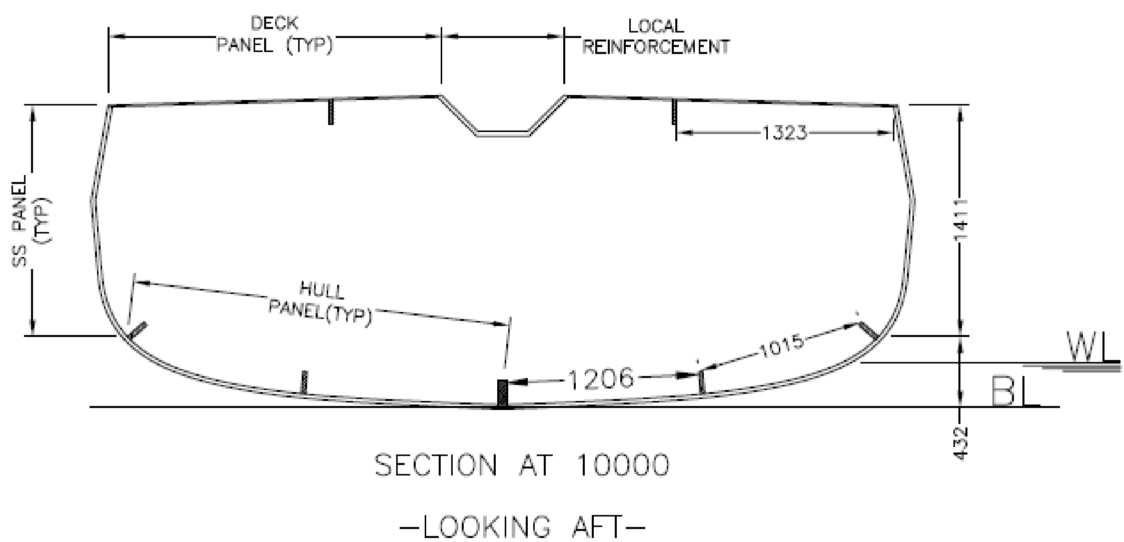


Figure 26: Typical Section

As can be seen from the table above, the hull requires the strongest material followed by the sideshell and bulkheads. It should also be noted that all panels (with the exception of the bulkheads) employ an asymmetric skin thickness with the outer skin being the thicker of the two. This is a direct result of the ISO Standard's minimum skin thickness values which assume a great change and magnitude of exterior impact. Reductions in panel weight were achieved by reducing the core density and therefore shear strength as far as the panel requirements would allow. The variation in design pressure of the hull from the bow to the stern meant that two types of core had to be employed to minimize weight while meeting panel requirements. Aramid fiber honeycomb was the primary core and SAN-A core was used in the forward section of the hull. See section 9.4 for further description of local core modifications.

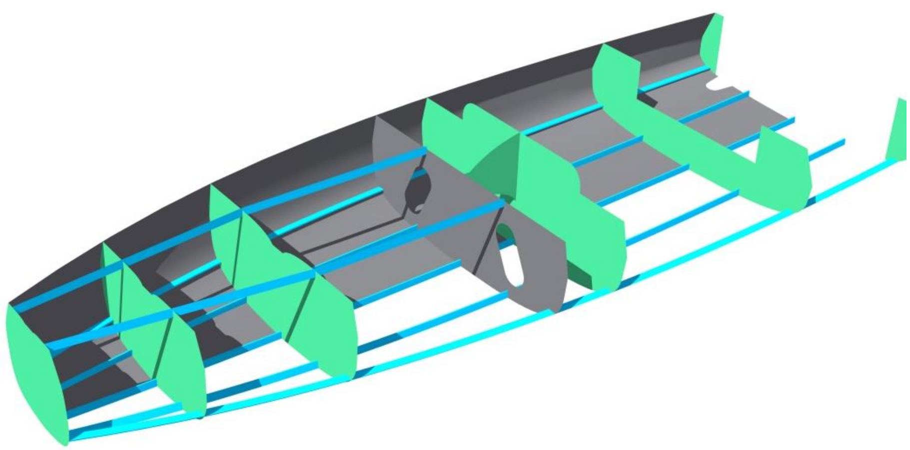


Figure 27: Hull Structure

9.3.1. Transverse Structure

The transverse structure saw five (5) watertight bulkheads (green-Figure 27), as per IMOCA regulation, as well as a structural non watertight bulkhead directly below the mast (grey-Figure 27). Increasing the number of bulkheads in this region also benefited the arrangement of the DSS foil trunk and keel tie-in structure.

9.3.2. Longitudinal Structure

Longitudinal structure is comprised one centerline (CL) stringer with two typical (TYP) outboard stringers (Figure 28). The outermost stringer sits just above the waterline, with the inner stringer placed between the outer stringer and the center line stringer. As the stringers carry aft, their separation increases in proportion to the hull width. This greater spacing works in concert with the decreasing design pressure. Stiffener dimensions and weights are displayed in below.

Table 8: Stiffener Dimensions

Stiffener Geometry		TYP Stiffener	CL Stiffener
Height	[mm]	135	150
Width	[mm]	25	50
Skin Thickness	[mm]	3.5	3.5
Weight	[kg/m]	2.1	2.75
Length -Total	[m]	80.6	16.3
Structural Weight	[kg]	169	44
Total Weight	**[kg]**	**214**	

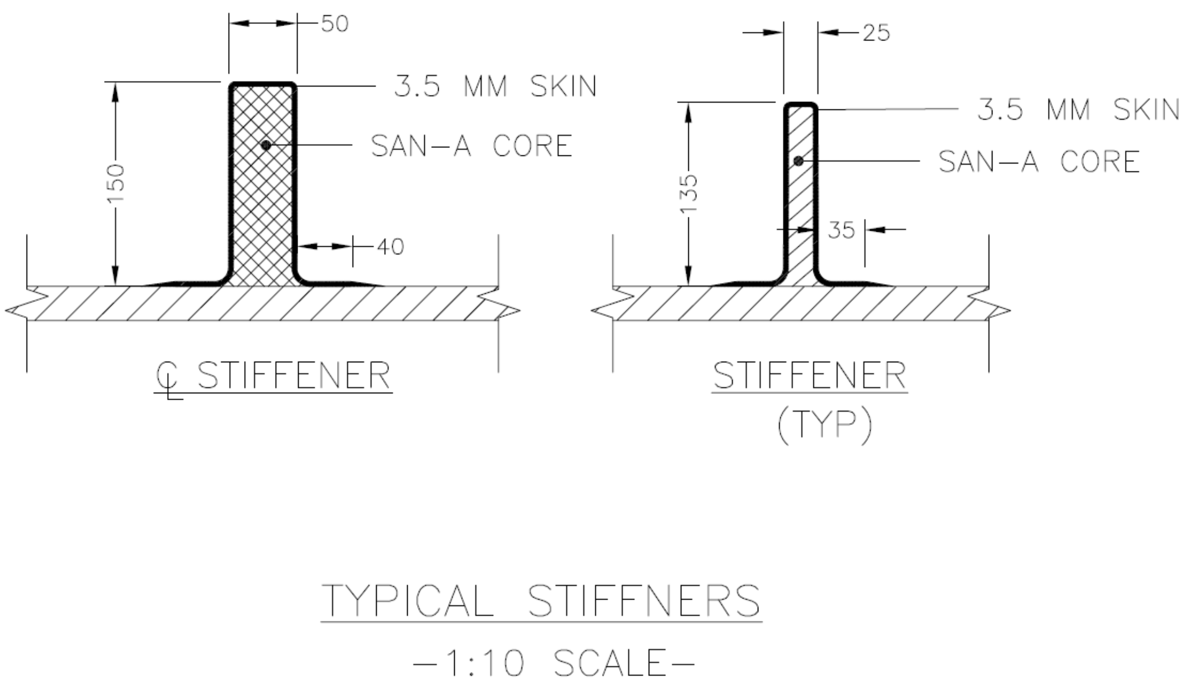


Figure 28: Stiffener Section

9.4. Areas of Local Reinforcement

Although no calculations were carried out for structural strength, an estimate of the increase in weight due to the added local structure or panel skin thickening. Weight increase was estimated in various manners depending on load and required reinforcement (core exchange, local structure or skin thickening – Table 9.

As noted earlier, the forward areas of the hull are exposed to significant slamming and wave loads. To combat this typical aramid fiber honeycomb core was substituted for a high density SANS-A foam core. Although it has a significantly higher weight penalty, it also has much greater shear strength and better overall impact resiliency.

Beyond the exchange of hull core material, local reinforcement takes the form of either additional structure or laminate thickness increase for chain plates to better distribute the point loads (i.e. stay and shroud chain plates). In some cases skin laminate thickening was required to the extent that it becomes a solid laminate – as is the case IWO keel pins.

Figure 29 identifies the areas of local reinforcement and relates them to Table 9 which describes the type of loading, required reinforcement and presents and estimate of weight increase.

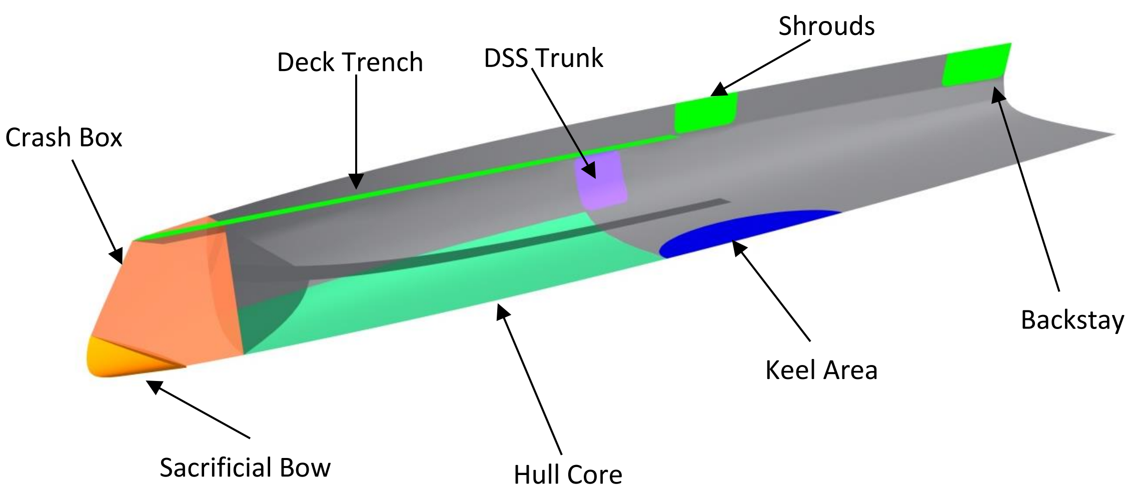


Figure 29: Area of Local Reinforcement

Table 9: Areas of Local Reinforcement

Location	Load Description	Estimated Load		Reinforcement/structural modifications	Weight Increase [kg]
Crash Box	Direct and catastrophic impact, either with submerged or floating object.	-NA-	[-]	Lower section of bow composed of sacrificial foam. Fwd-most compartment (crash box) filled with closed cell foam for impact energy absorption.	30
Hull ($x>.6L_{wl}$)	Slamming	90	[kPa]	Standard Aramid Fiber honeycomb core replaced with SAN-A closed cell foam (higher shear strength)	118
Deck Trench	Point load of attachment and larger loads on deck due combined stay tension	180	[kN]	Increased panel thickness IWO deck trench. Skin thickened IWO stay attachment points	10
Mast Step	Compression force of mast step	180	[kN]	Bulkhead below mast step reinforced in proportion to increased compression	40
DSS Trunk	Dynamic load of foils while sailing, impact load transmitted from foil upon impact with object	1.28	[MN]	Local structure added IWO DSS trunk. Hull and deck skin thickened IWO penetration	27
Keel Box	Static Moment	135	[kN]	Local structure added IWO keel pins and rams	70
Keel Box	Impact	-NA-	[-]	Local structure (mentioned above) further augmented.	35
Shrouds	Point load of attachment and regional increase in stress from shroud tension	175	[kN]	Stiffeners added to distribute load to hull. Skin thickened IWO shroud attachment point	10
Back Stay	Point load of attachment regional increase in stress from backstay tension	105	[kN]	Stiffeners added to distribute load to hull. Skin thickened IWO back stay attachment point	10
Total Weight			**[kg]**	**350**	

9.5. Future Developments

While the ISO Standard is the forerunner for small offshore sailing vessel design, it is not specific to the IMOCA rules, nor is its primary focus on developing extremely lightweight structures. Further research, analysis and review of the structure would identify areas where structure could be reduced beyond what the ISO Standard would allow.

Refinements visible at this stage would be further tailoring the core weight, which currently represents approximately 30 percent of the total panel weight. Panel geometry would also be refined to develop a more favorable geometry from a membrane-theory standpoint (lower aspect ratio). This would most likely result on more transvers structure in conjunction with less longitudinal stiffeners.

Future design would also see a large investigation into the magnitude of the local loading and the most efficient methods to resolves these high stress areas. The investigation would also have knock-on effects into the overall structural concept.

Assumptions of a purely unidirectional lay-up were assumed to simplify concept development, however in response to the torsional loads (among others) a more robust lay-up schedule would have to be produce and analyzed.

Most of these further refinements would require a resolution level that FEA analyses would be the only viable option. Although time consuming, there are considerable weight savings to be achieved. The current generation of IMOCA 60s are ~ 6.5 tons lightship (Almeida, 2015) and as the class has both a standardized mast and keel, there are few other areas beyond the structure for significant weight savings.

10. Steering System

Steering control is an important part of ocean sailing, and especially important for this design because of the high number of IMOCA 60 steering system failures. Three key factors which contribute to the design of the rudder are the foil section, the side area and the rudder stock strength. A balanced rudder is vital as it contributes to both the safety and the comfort of the skipper. The arrangement of the steering system also contributes to the comfort of the skipper.

10.1. Rudder Section

Foil sections are a well-developed area, without extensive research and testing it is best to use an existing proven design. Two common foil sections for rudders are 6-series NACA foil and 4-series NACA. The 6-series NACA foil section has a lower drag at small angles of attack than the standard 4-series NACA shapes. Within the 6-series, thinner shapes have less drag at very small angles of attack. From the drag coefficient (C_D) verses angle of attack (alpha) graph shown in Figure 30, the bucket shape is narrower for thinner foil sections. Thinner sections tend to stall abruptly with a large loss of lift at small angles of attack. Thinner sections are also designed for laminar flow. As the rudder will be operating in the transition of turbulent flow zone with Reynolds numbers ranging from 9.5×10^5 at 6 knots to 4.4×10^6 at 28 knots, a thicker 6-series section is beneficial. The 63 NACA section is designed for lightweight, fast hulls. As the vessel will be sailing offshore, small angles of the rudder will be required to change course, but a substantial thickness is required for strength and reliability and therefore the 63 A015 NACA shape was selected. This is a symmetric shape with a max thickness of 15% chord length located at 35% chord length from the leading edge. Figure 30 below shows coefficients of drag versus angle of attack at 6 knots. From the graphs, the stall angle is approximately 18 degrees. Additional graphs for the 63 A015 NACA foil section are shown in Appendix 6.

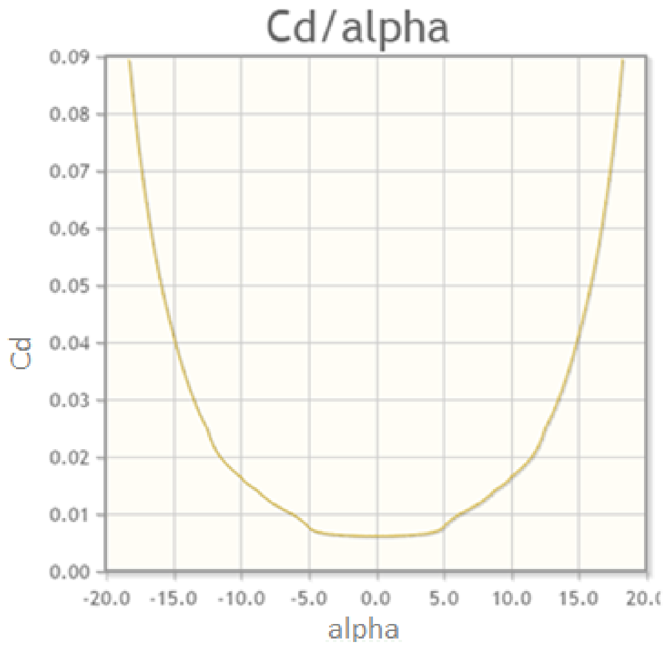
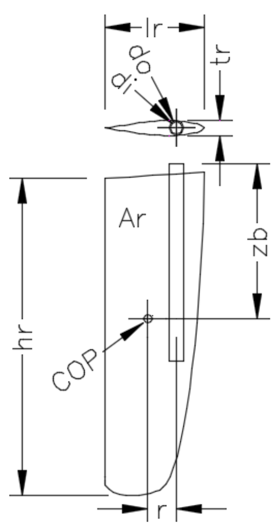


Figure 30: Selected foil section: NACA 63 A015 foil section with Reynolds number of 1x10^6 (NACA 63-015A AIRFOIL, 2015)

It is important that the rudder is properly balanced to increase comfort for the skipper while steering as it can be tiresome to steer the vessel if the rudder is unbalanced. The rudder stock is positioned so that 20% of the total rudder area is forward of the rudder stock. This places the center of the stock 0.1 mm forward of the center of pressure. The rudder dimensions are labeled in Figure 31.



Rudder Dimensions			
Rudder Foil Length	l_r	0.600	[m]
Rudder Foil Thickness	t_r	0.090	[m]
Outer Diameter of Rudder Stock	d_o	0.088	[m]
Inner Diameter of Rudder Stock	d_i	75	[m]
Distance from COP to Rudder Bearing	z_b	0.9	[m]
Rudder Height	h_r	1.9	[m]
Distance from COP to Rudder Stock	r	0.1	[m]
Side Area of Rudder	A_r	1.014	[m^2]

Figure 31: Rudder Dimensions

10.2. Rudder Shaft

The rudder shaft must be strong enough to withstand the bending moment caused by the force on the rudder. The size of the rudder shaft was designed to IMO standards from *Principles of Yacht Design* (Lars Larsson, 2014).

Table 10: Rudder and Vessel Parameters for Sizing of Rudder Stock

Vessel Parameters			
Length of hull	L_{WL}	17.8	[m]
Mass of hull	m_{LDC}	7000	[kg]
Density of salt water	ρ_{SW}	1028	[kg/m^3]
Speed of ship	V_{max}	28	[kts]
Speed of ship	V_{max}	14.39	[m/s]
Rudder Parameters			
Height of rudder	h_r	1.9	[m]
Area of rudder	A_r	0.7	[m^2]
Distance from center of pressure to center of bearing	z_b	0.9	[m]
Distance from center of pressure to rudder stock (0.1*c)	r	0.1	[m]

Using the rudder parameter values from Table 10 above, the rudder side force was found to be 26 kN. Based on this force, 23.3 kNm of rudder bending moment and 2.6 kNm of rudder torque equating to 23.4 kNm equivalent bending moment is felt by the rudder stock.

Now that an equivalent bending moment was found, the rudder shaft was sized to meet the strength requirements outlined in the IMO standards. A minimum Fiber Reinforced Plastic (FRP) skin thickness of 2.27 mm is required. Due to the rudder thickness and to minimize weight, a carbon epoxy composite tubular shaft with an inner diameter of 75 mm and an outer diameter of 88 mm with a foam core will be used. A detailed calculation of the rudder stock can be found in Appendix 7.

10.3. Steering System Arrangement

The steering system shown in Figure 32 is a fully mechanical system. It has been designed to require a maximum force of 47 kg on the tiller to hold the rudder position at the stall angle (18°) at 28 knots vessel speed. This amount of force will only be required in extreme situations. Regular steering activities will require much less force and will provide a comfortable steering experience for the skipper. For example at the average operating condition of 12 knots only 18 kg of force would be required to hold the rudder position at stall angle. The force and moment calculations for the force required at 28 knots and stall angle are in Table 11.

Table 11: Steering System Forces at 28 knots and Stall Angle

Steering System		
Torque on ruder post	2593	[Nm]
Arm to pivot	0.409	[m]
Torque at Pivot	6340	[N]
Pivot to rudder	0.3	[m]
Force rudder	1902	[Nm]
Distance to rudder post	3.6	[m]
Torque on rudder post	528	[N]
Required force on ruder post	**54**	**[kg]**

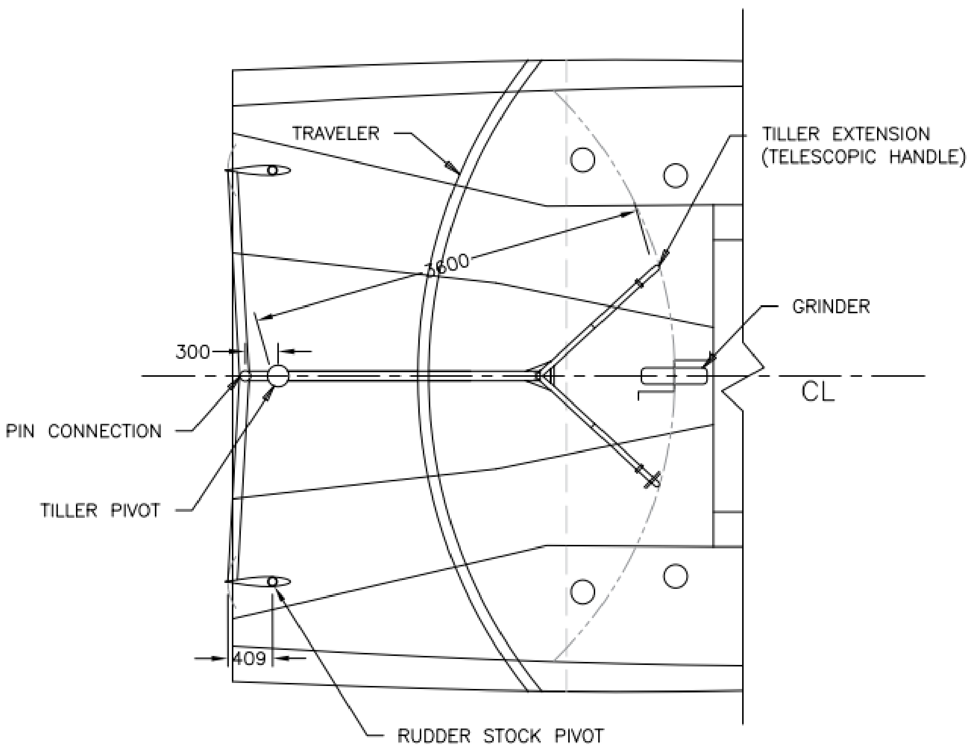


Figure 32: Steering System (Dimensions in mm)

10.3.1. Safety and Contingency Plan

It is not uncommon for vessel appendages to hit semi-submerged objects such as containers during the race. To increase fault tolerance of such events, and to increase ease of repair, the rudders have been designed to kick up. This allows the skipper accessibility to repair damages during the race. It also allows for easier access to inter-change the port and starboard rudders in the event of complete rudder failure. This kick up action may also sacrifice reliability and performance of the rudder and steering system. For example it may kick up unexpectedly causing the vessel to lose control and veer off track. However, given the nature of the race, the kick up design feature is considered an asset. In the case of collision, the tips of the rudders are made out of foam. This sacrificial tip allows energy to be absorbed by the rudder without catastrophic failure. Additional material will also be carried on board in the case of rudder damage so the skipper will be able to flip the rudder up and perform maintenance if necessary.

11. Sail Plan and Rigging

11.1. Mast

Within the IMOCA 60 class rules there are two (2) options for a mast. A rotating wing mast with outriggers, or a fixed mast with spreaders. The rotating mast allows for better performance due to the reduction of mast disturbance and drag caused by rigging, while the fixed mast is structurally safer. Because mast failure is a major concern, and the primary goal of this vessel is to finish the race safely rather than finish first, the fixed mast with spreaders was selected.

IMOCA 60 class rules state dimensional constraints for the mast cross section. The maximum chord is 330 mm, the max thickness is 160 mm, and the width of the back face must be 35 mm. There are also restrictions on the types of carbon fibers to be used for the mast. The mast must be constructed of fibers with an elasticity modulus less than or equal to M46J and HS40 fibers. The mast for this design will be of M46J with the following assumed properties for the composite matrix:

- Tensile strength = 2210 MPa
- Compressive strength = 1080 MPa
- Flexural strength = 1420 MPa
- Young's Modulus = 220 GPa
- Density = 1600 kg/m^3

The structural calculations for the mast can be seen in appendix 9. The structural requirements for the mast has been broken down into transverse and longitudinal strength requirements and calculated as per the methods outlined in section 7 of *Principles of Yacht Design* (Lars Larsson, 2014). The transverse stiffness is calculated by observing each of the 4 panels which are separated by the spreaders and then applying formulas which are functions of the design righting moment, rig geometry, material properties, and appropriate safety factors. The longitudinal stiffness was calculated using a similar set of equations.

Based on the calculated minimum transverse and longitudinal moments of inertia in Table 12, and dimensional restrictions for the mast section, the following cross section was chosen for the mast.

Table 12: Mast Stiffness

Moment of Inertia			
Minimum Moment of Inertia in x-direction	I_{x_min}	1.31E+07	[mm^4]
Minimum Moment of Inertia in y-direction	I_{y_min}	5.74E+07	[mm^4]
Moment of Inertia in x-direction	I_x	1.35E+07	[mm^4]
Moment of Inertia in y-direction	I_y	6.01E+07	[mm^4]

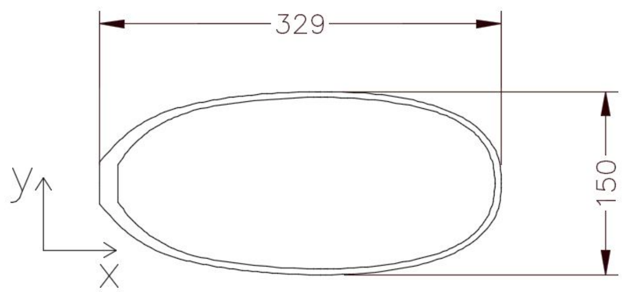


Figure 33: Mast Cross Section (Dimensions in mm)

Based on the dimensions in Figure 33, the weight of the mast is approximately 212 kg.

11.1.1. Mast – Sail Interference

A major drawback of choosing the fixed mast rather than a rotating wing mast is the additional flow separation on the mast which not only produces a drag force, but also disturbs the flow around the sails and can severely reduce the driving force of the sails. This disturbance can be reduced or even eliminated with a proper mast section and roughness. It is predicted that a mast roughness with a grain size of one percent of the mast diameter could decrease the drag on the mast by up to 25 percent for an apparent wind speed (AWS) of 5-10 m/s (Lars Larsson, 2014).

11.2. Rigging

11.2.1. Stays

The IMOCA 60 vessels are equipped with five (5) forestays and three (3) sets of runners, the highest of which is considered the backstay. Each of the five (5) forestays are designed to carry a specific sail as shown in the figure below.

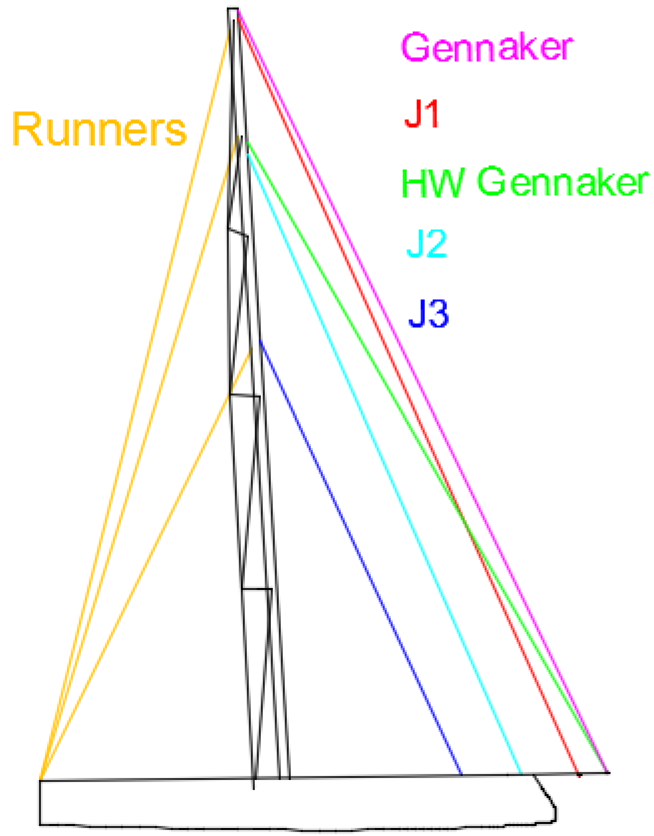


Figure 34: IMOCA 60 Stays

There are three (3) jib stays, J1, J2, and J3 for Jibs 1, 2, and 3, and two (2) gennaker stays. The J2 stay also serves as a permanent structural forestay. The design forces for these stays are calculated as per the methods outlined in section 11 of *Principles of Yacht Design* which are functions of the righting moment, altitude of each stay, and angle of each stay from the mast. The maximum static righting moment of 32 ton-m as set by the IMOCA class rules was used for these calculations (Lars Larsson, 2014).

Table 13: Stay Forces

Stay	Height Above Mast Step	Force
[-]	[m]	[kN]
Gennaker (foremost fore stay)	27	174
J3 (aftmost fore stay)	15.5	244
Back Stay	27.5	106

The forces on the other stays will be within the range for the two forestays shown above. The design force for each forestay will be equal to the force on J3.

The cables for the stays will be carbon T-700 fibers. This is a type of cable that can be purchased for the purposes of yacht rigging and many other applications. Based on properties provided by Fiber Max supplier, the stay cables have the following properties:

- Specific density = 1.76
- Tensile Strength = 4900 MPa
- Young's modulus= 230 GPa

With a known maximum tension and tensile strength, the minimum diameter for the cables can be calculated by a ratio of tension and yield stress. The minimum diameter for the cables is calculated to be 8.0 mm resulting in a total weight of 11.1 kg for all of the forestays.

11.2.2. Shrouds

The IMOCA 60 standard fixed mast has 3 sets of spreaders. The dimensions and locations of spreaders and shrouds for the vessel are taken from a drawing provided in an appendix to the IMOCA 60 class rules.

Section 11 of *Principles of Yacht Design* was used to calculate the forces on the rig. Different loading conditions for the rig were analyzed to determine the maximum horizontal forces experienced by each of the 3 sets of spreaders and mast head. The worst case scenario for each condition was taken to be the design condition for structural calculations. There were 6 cases considered for these calculations where the maximum 32 ton-m heling moment was achieved:

- Case 0A –the highest foresail loaded only
- Case 0B –the middle height foresail loaded only
- Case 0C – the lowest foresail loaded only
- Case 1 – mainsail reefed to first reef point (23.5 m up the mast)
- Case 2 – mainsail reefed to second reef point (19.4 m up the mast)
- Case 3 – mainsail reefed to third reef point (14.7 m up the mast)

For case 0A, 0B, and 0C, the full heeling moment (HM) is supported by only the one forestay, and thus the force on the mast at that stay is the HM divided by the stay height. For cases 1, 2, and 3, the HM is supported by the mainsail reefed at different heights on the mast.

The forces through the rigging are calculated using equations in Appendix 8. The highest tension was taken to be the design condition for each of the cables (374 kN). Using the same carbon fiber cables that were selected for the stays, a cable of diameter of 9.85 mm has been selected for the shrouds which weigh a total of 14.1 kg.

11.2.3. Drag

The drag resistance of the rigging was calculated based on section 7 of *Principles of Yacht Design*. If the diameter of the stays and shrouds are approximated to 13 mm, with a total length of 385 m, the following conservative values of drag were calculated.

Table 14: Rig Induced Drag

AWS	Drag/meter	Drag
[m/s]	[N/m]	[N]
5	0.5	193
10	1	385
15	2.5	963
20	5.5	2118

11.3. Spreaders

The structural requirements for the spreaders are calculated based on section 11 of *Principles of Yacht Design*. Three requirements were set for the structure of the spreaders: a minimum cross sectional moment of inertia at the middle section of the spreader, a minimum section modulus near the mast, and a minimum moment that the spreader should be able to withstand at the attachment point on the mast. These requirements are calculated in Appendix 8 using functions of spreader length, angle from the mast, material properties, and shroud forces. The spreaders will be constructed of the same composite material as the mast.

The requirements for each of the spreaders were calculated and the worst case results were used as the design requirement for all of the spreaders. The spreader midsection and spreader mast Inertia results and dimensions are shown in the tables and figures below.

Table 15: Spreader Mid Section Inertia

Spreader Middle Section			
Minimum Moment of Inertia	I_{min}	$1.2x\ 10^6$	$[mm^4]$
Moment of Inertia in the x-direction	I_x	$1.6x10^6$	$[mm^4]$
Moment of Inertia in the z-direction	I_z	$7.7x10^6$	$[mm^4]$
Area	A	$1.5x10^3$	$[mm^2]$

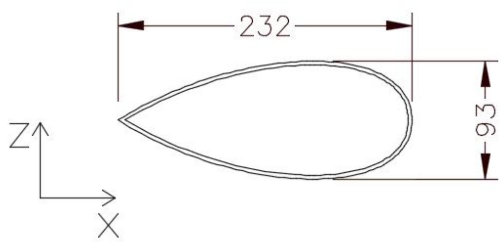


Figure 35: Spreader Mid Section (Dimensions in mm)

Table 16: Spreader Mast Section Inertia

Spreader Mast Section			
Minimum Section Modulus	SM_{min}	$1.6x10^5$	$[mm^3]$
Minimum Moment	M_{min}	$1.7x10^5$	[Nm]
Maximum Distance from Neutral Axis to Edge in x-direction	y_x	152	[mm]
Maximum Distance from Neutral Axis to Edge in z-direction	y_z	56.5	[mm]
Moment of Inertia in the x-direction	I_x	$1.1x10^7$	$[mm^4]$
Moment of Inertia in the z-direction	I_z	$5.8x10^7$	$[mm^4]$
Section Modulus in the x-direction	SM_x	$2.0x10^5$	$[mm^3]$
Section Modulus in the z-direction	SM_z	$3.8x10^5$	$[mm^3]$
Flexural Strength	σ	1420	[MPa]
Moment	M	538.6	[kNm]

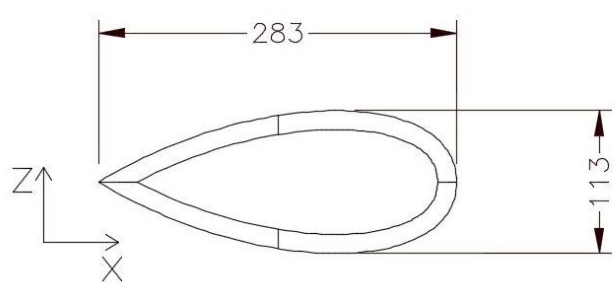


Figure 36: Spreader Mast Section (Dimensions in mm)

Based on these dimensions, the weight for each spreader is approximately 7 kg, for a total of 42 kg for all six (6) spreaders.

11.4. Boom

To determine the structural requirements of the boom, beam bending equations were used with the assumption that the forces on the sails would be distributed along the boom as a gradient. Using a safety factor of 3, the minimum section modulus of the boom is 45583 mm^3 which is achieved with the cross section geometry shown below.

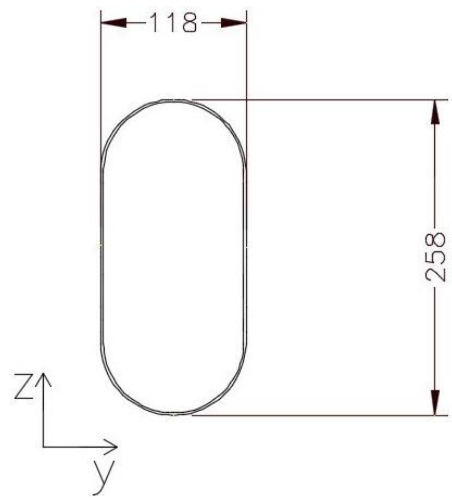


Figure 37: Boom Section

11.5. Sails

The IMOCA 60 class rules allow for nine (9) sails to be carried on board with the addition of a storm sail which must have a sail area of no more than 20 m^2. The sails to be carried onboard are as follows:

- 2 x main sail
- 1 x J3 – small size jib
- 2 x J2 – medium size jib
- 2 x J1 – large size jib
- 1 x gennaker
- 1 x heavy weight gennaker (HWG)
- 1 x storm sail

11.5.1. Power

The performance of the sails was analyzed using an open source simulation software program which uses vortex lattice method (VLM) to simulate flow around the sails, Sail7 (Sail7, 2012). This software was used to determine the forces and moments applied to the sails for any given wind speed, angle of attack, and heel angle. Since Sail7 relies on VLM, some restrictions have been set for the use of this software. VLM uses the assumption of potential flow and small angles of attack. Sail7 should therefore not be used for sailing conditions where sail drag is the predominant force on the sails, such as running condition or using a gennaker. This software has been used for the condition of close hauled and reaching only, with the use of the mainsail and appropriate jibs.

Examples of the results from this program are shown below. Throughout the VG route, contestants are close hauled for approximately 20 percent of the race, and during this time the true wind speeds are on average 30 – 50 km/h. The example shown in Figure 38 is for close hauled condition with wind speeds of 43 km/h using the J1 and the mainsail. At this speed with an apparent wind angle (AWA) of 30 degrees the sails are producing 12.4 kN of driving force and approximately 30 ton-m of HM.

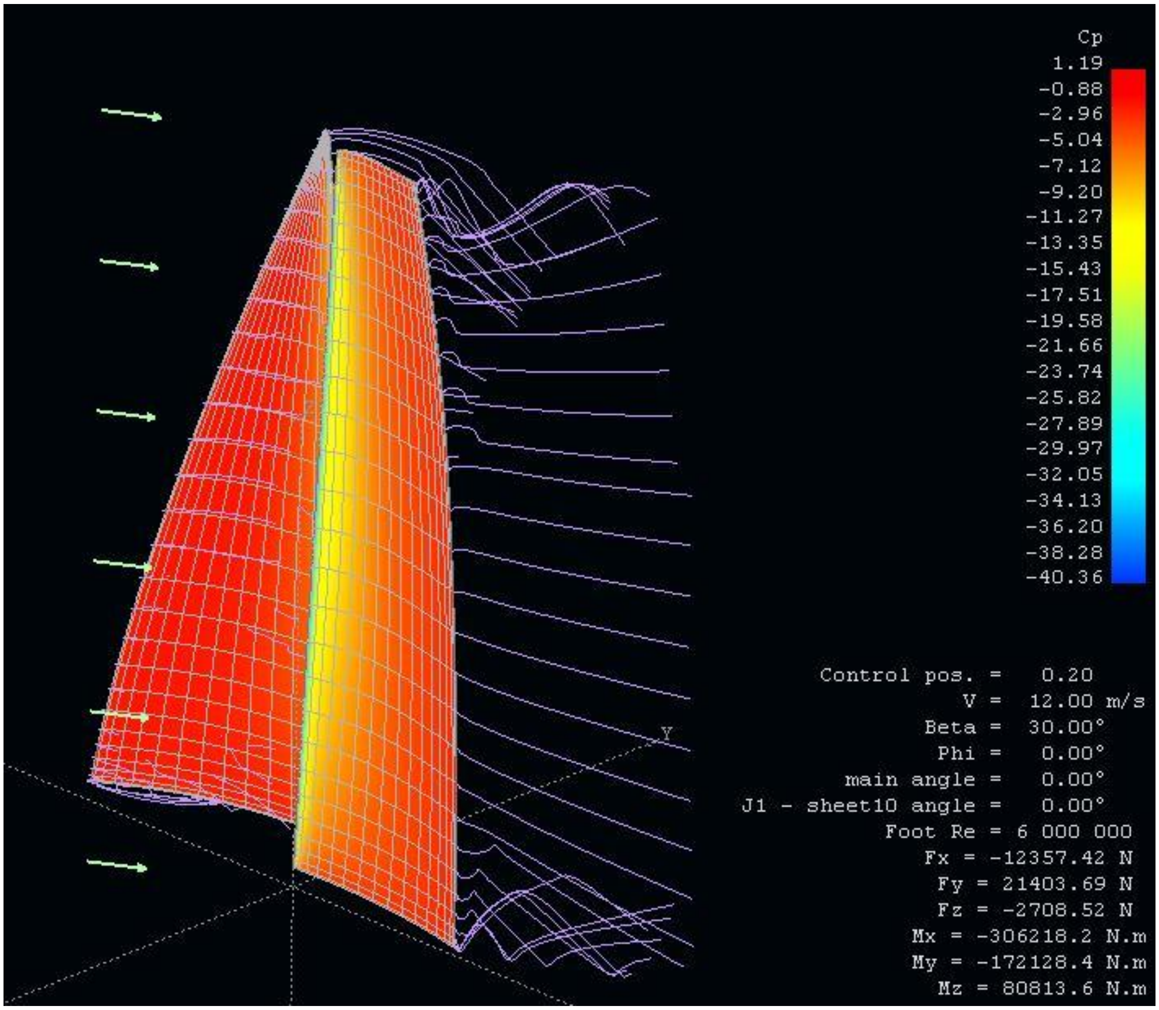


Figure 38: Sail7 Example: Mainsail and Jib Close Hauled

Sail7 was also used to qualitatively evaluate the performance of each sail by observing how the flow around the sails behaves. Shown below is a figure for the J1 jib and the mainsail with streamlines showing the flow around the sails. The mainsail produces low amounts of turbulent flow with the exception of the vortices at the head and foot of the sail. The jib produces much more turbulent flow on the leeward side and the streamlines break free from the sail before reaching the trailing edge.

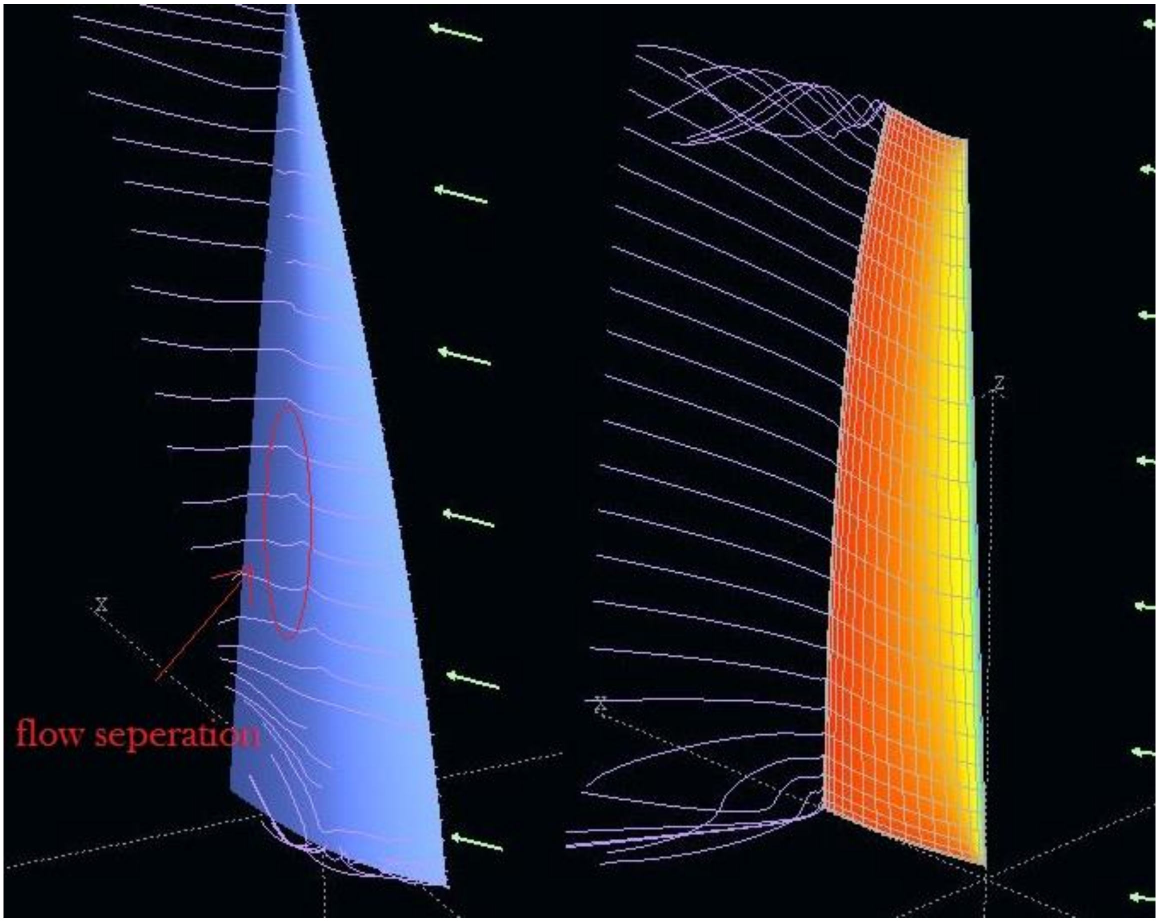


Figure 39: Streamlines Around the Mainsail and Jib

11.5.2. Center of Effort

The center of effort (CE) is estimated to be approximately 30 percent of the chord line from the leading edge of the sails. For the same conditions as Figure 38, close hauled with wind speeds of 43 km/h using the J1 and the mainsail, the lateral center of effort is calculated by taking an average of the CE, weighed by sail area, for each sail.

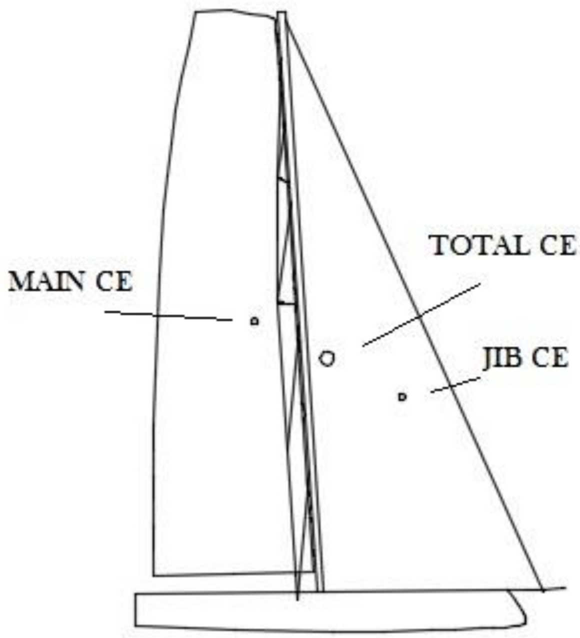


Figure 40: Center of Effort

The CE is calculated to be 550 mm forward of the mast step.

For running condition, due to the driving force being almost entirely drag, the center of effort is simply the center of area for the sails.

11.5.3. Lead

To calculate the center of lateral resistance (CLR) of the vessel, the forward resistance of the bow is assumed to be cancelled out by the rudder, and the CLR is calculated using the center of projected area for the keel and foils.

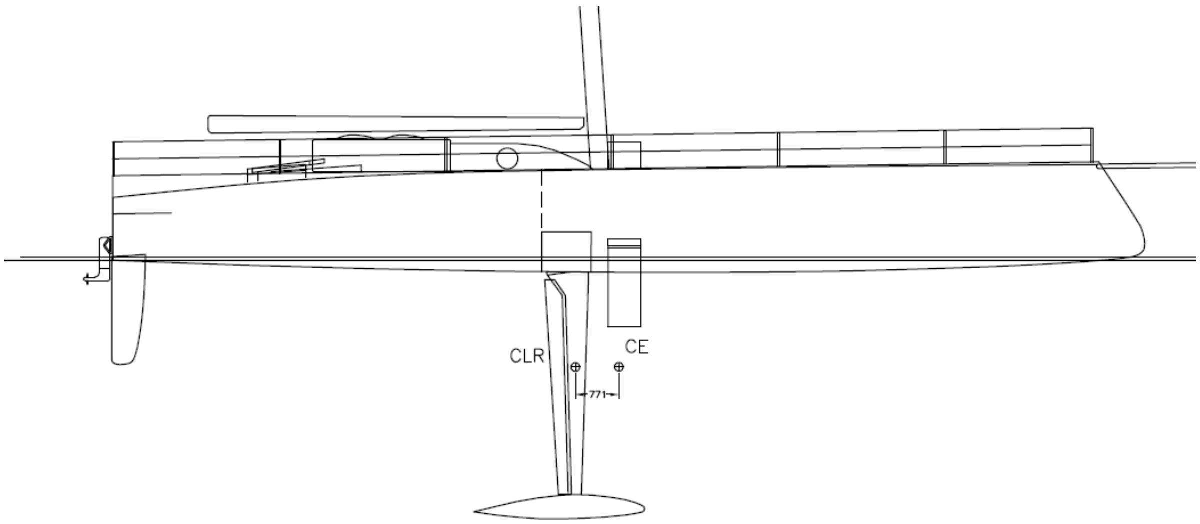


Figure 41: Lead

The lead shown in Figure 41 is 771 mm which is 4 percent of the L_{WL}. It is recommended that for a vessel with a beam to length ratio of 0.25, the lower limit and upper limit for lead should be 5 and 11 percent respectively of the L_{WL} (Fassati, 2009).

11.5.4. Weight

During an interview on *Youtube* with a skipper from the 2012 VG race, it was stated that the sails weigh approximately 60 kg (Vendee Globe TV, 2015). If it is assumed that this is referring to the main sail, the weight of all other sails can be approximated. The main sail area is 167 m^2 giving a weight to area ratio of 0.36 kg/m^2. Applying this ratio to all of the sails gives the following weights:

Table 17: Weight Estimate of Rigging

Sail	Area	Weight	X-Centroid	Z-Centroid
[-]	[m^2]	[kg]	[m]	[m]
Main	167	60	4	13
J1	162	58	11	9
J2	112	40	11	8
J3	53	19	11	6
Gennaker	343	123	unknown	unknown
HWG	257	92	unknown	unknown
Storm Sail	20	7	unknown	unknown
Total		**400**		

11.6. Line arrangement

Each foresail is required to have the following lines:

- One (1) Halyard line
- Two (2) Furler lines
- One (1)Cunningham
- Two (2)Sheets

An additional two (2) halyards are run down the mast, one for the main sail and one for the skipper in the event that it is necessary to climb the mast. All of the halyards (7 lines) travel to the port side of the cockpit and into designated cleats. Reefing lines for the clew and the tack of the mainsail (6 lines) are passed through the boom and come down the mast then travel to the port side of the cockpit into their designated cleats. The furler lines and cunninghams (15 lines) all act at the appropriate stay, and run back to the cockpit along centerline, and along the starboard side of the cockpit into a set of cleats. The sheets for each forestay run to the cockpit along port and starboard.

The boom swings along a traveler running through the cockpit. The mainsheet runs through this traveler and is accessible on the port and starboard side of the cockpit. There are also 3 lines on each side of the cockpit controlling the tension of each runner. This results in twenty two (22) lines on the port side and twenty four (24) lines on the starboard side. Approximately 50 percent extra cleats are added to each side as a safety measure, resulting in thirty two (32) lines at port and thirty six (36) lines at starboard.

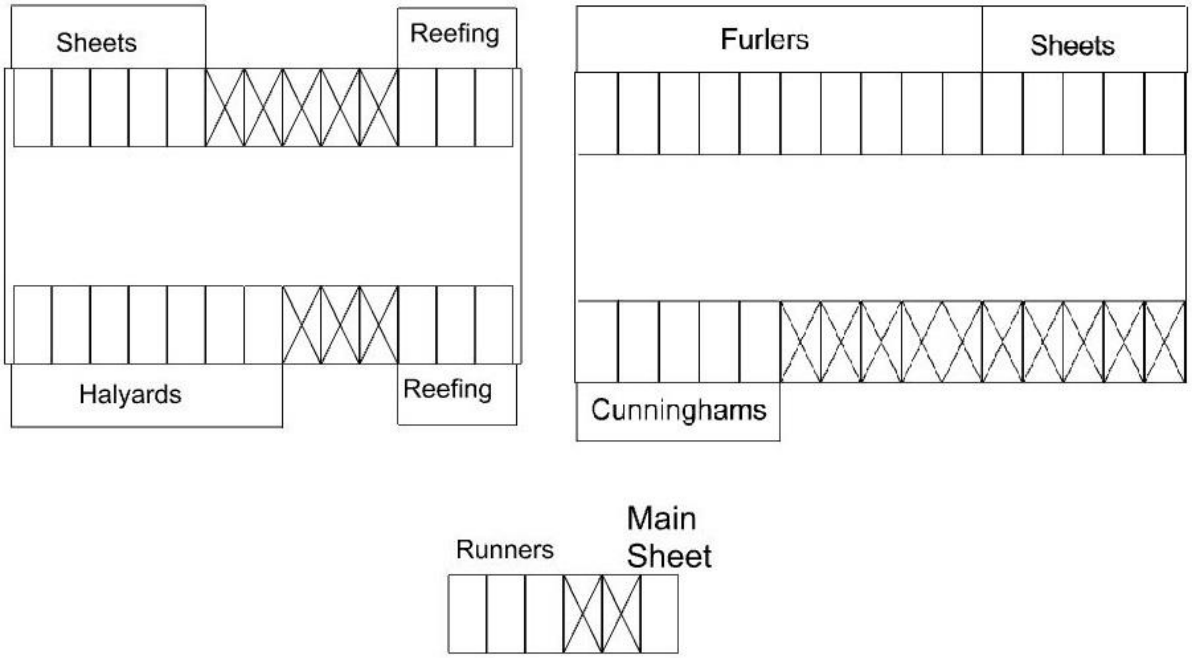


Figure 42: Arrangements of Cleats, top left is port, top right is starboard, bottom center is aft port and starboard, X implies spare cleats

Within the cockpit there are two (2) winches on the port side and two (2) winches on the starboard side which are used in combination with the grinder to tension lines. The full arrangement of the lines and cleat placement can be seen in the Drawing Package.

11.7. Survivability

The survival of the skipper is the most important aspect of this design. The major sail and rigging related hazards to the skipper are outlined below:

11.7.1. Climbing the mast

Possible reasons for the skipper to climb the mast could include failure with any electronics such as the radar on the mast, jamming of any lines on the mast, or the need to replace any lines or equipment. This is a very dangerous task due to the height and motions of the vessel. If the skipper is required to perform this task, he should first depower and lower the sails completely to prevent any sudden changes in HM. The Type 600 Vessel is equipped with a personal halyard which the skipper can use to hoist themselves up the mast. The VG racing rules state that the skipper must always be securely clipped onto a fixed point on the vessel while climbing the mast or climbing along any part of the vessel. For this reason, there are attachment points placed at strategic locations along the mast.

11.7.2. Dismasting

One of the most common reasons for the early retirement of skippers is the breaking of the mast. If the mast were to break, there are several possible dangers such as the mast falling on other equipment or the skipper as well as the danger of mast shrapnel.

11.7.3. Breaking of Structural Equipment

If a cable such as a runner or stay should break while under tension, the cable could cause catastrophic damage to the skipper or another part of the vessel. The cables for the rigging are carbon fiber which has a high stiffness. It is assumed that this large stiffness will prevent a large amount of stretch in the cables, and therefore less 'kickback' if the cable should break. Appropriate safety factors have also been applied to the dimensioning of the cables to prevent this failure.

11.7.4. Collision with the Boom

During a tack or gybe, the boom will swing along the cockpit. The boom is 700mm above the deck and therefore the skipper, and any equipment in the boom's path, is at risk of collision. However, the risk of the boom hitting the skipper is very low due to the amount of times that the skipper is expected to tack or gybe the vessel. Along the VG route, the contestants spend only approximately 20% of the time close hauled or reaching, which is where most of these manoeuvres will take place, and during this time the skippers will only tack or gybe once or twice a day. Therefore it is highly unlikely that the skipper will ever be unaware of the incoming boom.

11.7.5. Lightning

Lightning striking the mast is a major hazard to the skipper and any equipment onboard. Luckily, this problem can be solved by grounding the mast into the ocean. The skipper should resist climbing the mast during a lightning storm.

12. Foils

Historically the keel of a sailboat has played a dual role in vessel performance providing righting moment and lateral lift. The effectiveness of the keel in these roles however is inverted, such that when it is acting as an effective lifting device, it is providing the least righting moment. The inverse is true as when the vessel is heavily heeled the keel is fully engaged and providing its maximum righting moment, its ability to provide lateral lift is significantly compromised.

In response to this inverted dynamic, many high-performance sailing vessel have separated the tasks of the keel into their unique contributions: stability and lift. In doing this, the righting moment of the keel can be optimized through the introduction of a canting keel (section 0) while the introduction of asymmetric daggerboards maximum the lateral lift that can be produced. This has allowed vessels to achieve significantly superior upwind performance compared to their predecessors. These advantages are only seen upwind, as on downwind legs or at speeds above 14 kt the foils are typically retracted (Merfyn Owen, 2013).

Over the duration of the next VG, the IMOCA Class will witness the coming of another performance technology, Dynamic Stability System (DSS) foils that will shift the focus from upwind to off the wind performance. While the IMOCA class vessels have been developing their abilities in off the wind conditions for some time using water ballast and planing oriented hull forms, the implementation of DSS foils will take this many steps forward.

Figure 43: Wilds Oats with DSS Deployed

Over the past decade DSS foils have emerged as impressive means of improving off-the-wind performance. Recently the outfitting on *Wild Oats XI* (above), a 100ft maxi offshore racer has been attributed to her latest win of the Sidney-Hobart race (Infiniti Yachts, 2014). The crew also noted improved motions and longer surfing runs (Infiniti Yachts, 2014). Whereas the primary concern of a daggerboard is to provide lateral lift, a DSS foil is designed to provide horizontal lift. Its primary role is to create a dynamic righting moment from the foil lift although the vertical force also lifts the bow (300 mm on Wild Oats (Infiniti Yachts, 2014)) which in turn further improves performance. With a DSS foil deployed on the leeward side of the vessel, the vessel's stability becomes directly related to its speed, creating a stiffer vessel as the wind speed increases. Their deployment also reduces pitching moment (Dynamic Stability Systems, NA).

Applied to an IMOCA 60 where at least 50 percent of the sailing time is spent off the wind, this technology was identified as able to provide large performance gains. The motions improvements are also important as an IMCOA 60 is exposed to some of the worst weather on the planet.

12.1. Initial Concept

As DSS foils are have only recently been applied to IMOCA 60s, their application is still in a state of evolution. In response to this, *Crash Box Design Ltd*. decided early in the design stage to keep the DSS foil arrangement as close to the existing daggerboards as possible without significant impact on DSS performance. This decision was made to minimize the number of new variables introduced into the systems, both to reduce the skipper's learning curve impact on general arrangement. It should be noted here that the IMOCA class rules currently limit the number of appendages to 5; one (1) keel, two (2) rudders and a set of foils, therefor it was not an option to add DSS foils in addition to the existing conventional daggerboards.

The design process began with developing two potential geometries. The geometry present below were developed to maximize immersed area and horizontal attitude of the foil while at the same time fitting within the imposed geometry of the hull. It was also that the both foils be capable of being retracted at the same time, thus they could not cross centerline in their retraced mode.

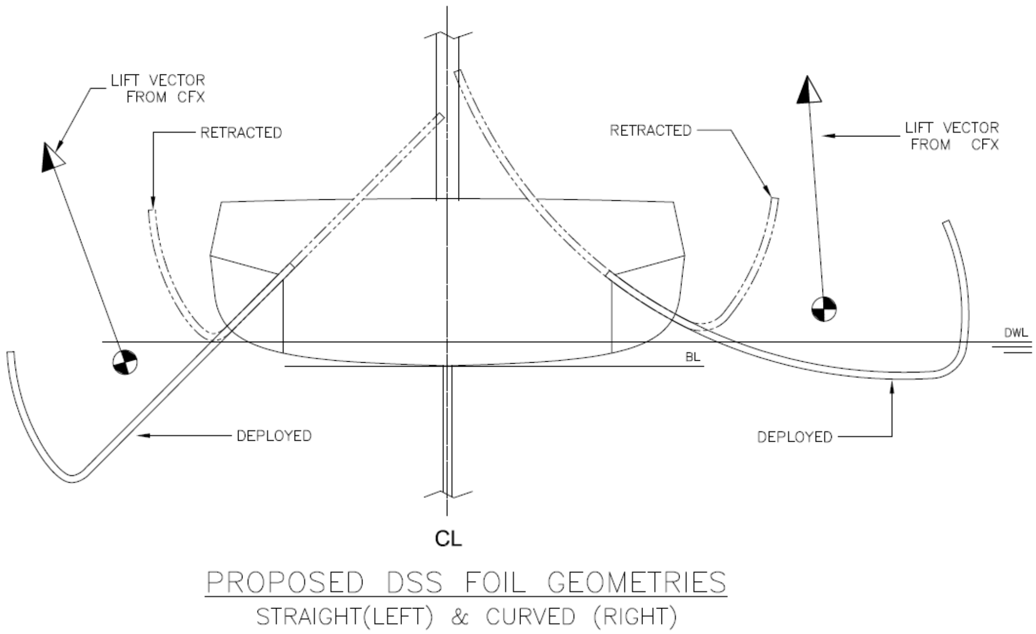


Figure 44: Proposed DSS Foil Geometries

12.2. Foil Selection

The performance of the geometry is strongly affected by the 2D foil cross-section. As there is a vast array of foil sections available, the following criteria were established to aid in the selection:

- High lift to drag ratio
- Adequate thickness for structural requirements
- Asymmetric -provide lift at zero angle of attack (α_0)
- Buildable – no excessive camber or thin trailing edges
- Performance over a range of attack angles and Reynolds numbers
- Cavitation Resistance

Based on these criteria, three foil sections in Figure 45 were selected: NACA 64-412, Eppler 68 and H105.

The NACA 64-412 foil is a laminar foil (6-series) and has been used with good results in many smaller dingy daggerboards. It provides high levels of lift with moderate drag. However, as can be seen in Figure 45, its geometry is the most exotic of the three presented.

The Eppler 68 was developed by Richard Eppler for low Reynolds numbers however it has proven as a good reference for asymmetric daggerboard on other vessels. It has a far higher theoretical lift to drag ratio, though at the cost of a reduced lift coefficient and the possibility of cavitation at higher Reynolds numbers.

The H105 was developed by Thomas Speer as a hydrofoil section and has similar properties as the NACA 64-412 although its geometry is more favorable for structural purposes.

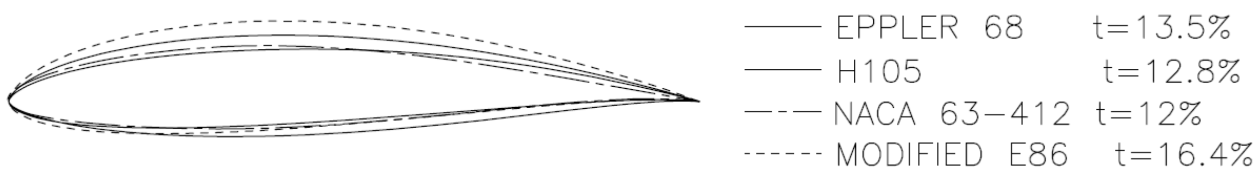


Figure 45: Foil Sections

In comparing the three sections lift and drag characteristics it was decided that variation was smaller enough to not be a driving factor at this stage of design. The Eppler 68 was thus chosen as the cross section due to its large thickness and simple shape. It should be noted that cavitation resistance is not an attribute of the Eppler 68 and future development of the section should investigate this.

12.3. Computational Fluid Dynamics

Following the selection of sectional shape, a Computational Fluid Dynamic (CFD) campaign was launched to evaluate the effect of geometry on drag, lift, and produced moment. While empirical formulas were used at the early stages of design, the subtly between the curved and straight DSS foil geometry in Figure 46 was such that CFD was chosen as the best method. There was also uncertainty regarding the effect the upturned tip would have on the overall performance of the foils and the lift vector.

Beyond the two foil geometries presented earlier a third conventional daggerboard was added to the CFX campaign. It was used as a baseline to assess that the results from CFX were sufficient for a comparative analyses. The results also established a baseline for the performance of the other two DSS foil.

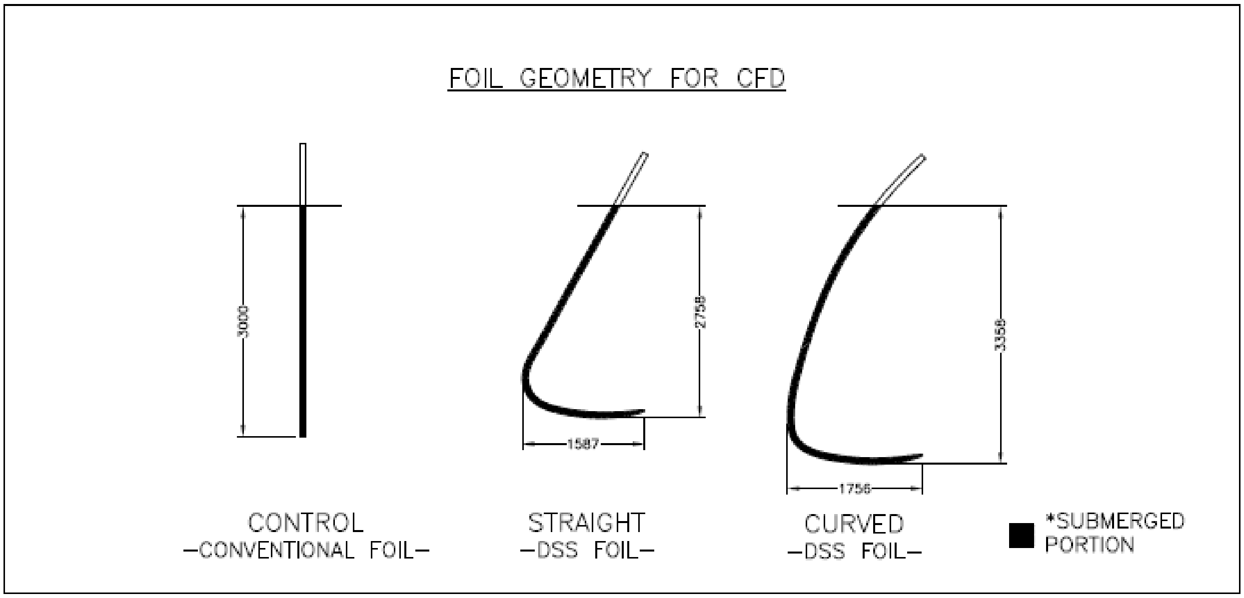


Figure 46: Three Foil Types Analyzed in CFX

12.3.1. CFD Solver, Geometry and Mesh

ANSYS CFX was chosen as the solver due to both its availability and prior user experience. Three velocities were modelled: 8, 14 and 20 knots; representing the lower end of the speed range, the typical speed at which conventional foils are lifted, and the upper end of the vessel's range, respectively. The foils were set in the control volume such that foil angle of attack was zero.

The geometry for the CFD was developed in Rhino and exported to ANSYS Design Modeler© to merge the solid bodies before being meshed in ANSYS Meshing©.

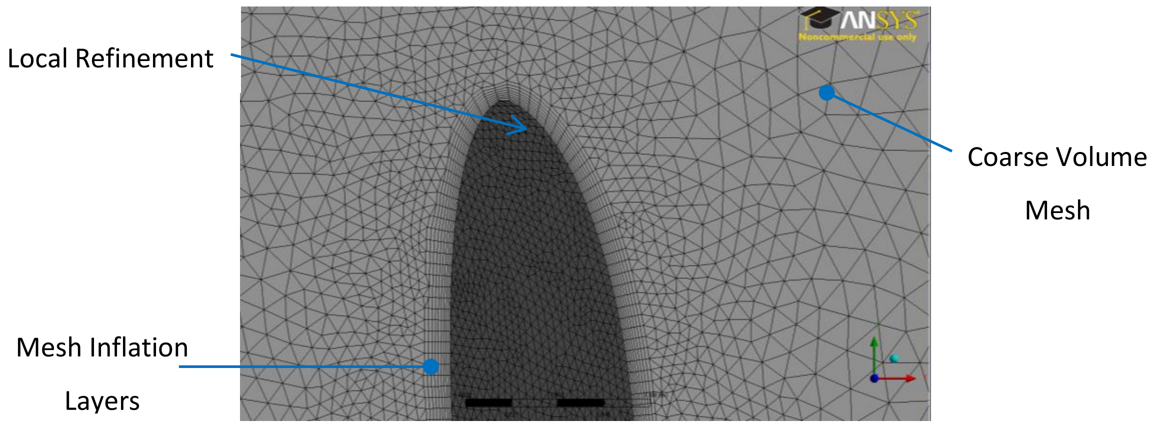


Figure 47: Local refinement and inflation layers

The control volume dimensions are based loosely on control volume guidelines for vessel CF, 5 ship lengths long and 2 ship lengths wide, where ship length was applied to foil dimensions. This equated to a cross section measuring 2 times the maximum width of the curved foil when suspended vertically (see Figure 46) and twice the depth. The length was taken as twice the depth of the foil. The control volume is 4m wide, 5m tall and 10m long.

A coarse mesh throughout the control volume with local refinement and inflation layers was defined in way of the foil (see above). The refinement allowed for higher resolution along the foil surface, while the inflation created better definition of the boundary layer of the foil. It produced valid results as they were used primarily for comparative purposes. Mesh statistic can be found in Appendix 15.

Boundary conditions were established to be robust and realistic. Fluid velocity was set at the inlet with relative pressure governing the outlet. The bottom and sides of the control volume were set to 'free-slip' while the top was set to a symmetric condition. The foil surface was specified as a 'no-slip' with the roughness set to 'smooth'. The model set up can be seen in figure 48.

Most of the default settings of the CFX model were left unchanged. The standard Root Mean Square (RMS) values were set as the termination criteria of 10^{-4} while monitors were created for lift, drag and y-moment to track how stable the desired variables were.

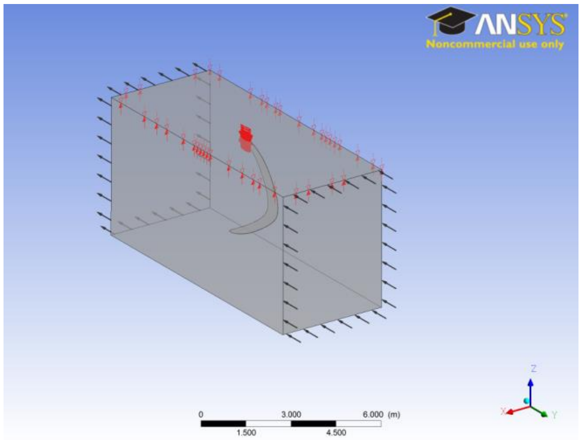


Figure 48: Model Set-up

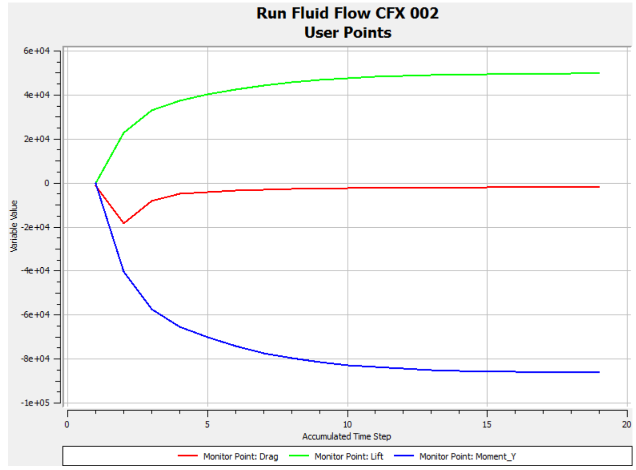


Figure 49: User Monitor Points

12.3.2. CFD Results

The first evaluation of the CFX results was undertaken once the control foil had been run at the three specified speeds. To assess the model set-up and mesh quality, the CFX RMS levels and monitor values were reviewed to ensure the solver was converging and the monitor values were stable (See Figure 49). Furthermore the results for lift and drag were compared with empirical values. From Table 18, good agreement WRT lift between the CFX results and the calculated values can be seen. The drag results however are not as agreeable, with a consistent 40 percent difference. It is presumed that the drag was due to assumptions of infinite span, and thus no correction for aspect ratio or tip vortexes, of the empirical formula.

Table 18: Calculated vs. CFX Results

Speed	Calculated Values		CFX Results		**Percent Different	
	Lift	Drag	Lift	Drag	Lift	Drag
[kts]	[kN]	[kN]	[kN]	[kN]	[Percent]	[Percent]
8	6.0	0.15	5.4	0.21	10	-40
14	18.45	0.46	17.78	0.64	4	-39
20	37.70	0.94	36.32	1.26	4	-34
$** \ Percent \ Differemce = \frac{Calculated\ Value - CFXResult}{Calculated\ Value}$						

From the above results, it was concluded that the simulation of the two DSS geometries could be carried out with reasonable levels of certainty in the results.

The figure below show typical pressure contours of the analyzed foils. It can be seen the loss of pressure as one moves towards the tips as well as the low pressure zone located in the turn of both DSS foils. One can also see the effect of symmetry vs. free slip boundary conditions for the top of the control volume. The control pictured below was free-slip boundary conditions and it can be seen that the low pressure in the center of the foil extend all the way to the top. Compare this with the DSS foils were the low pressure field is reduced as one moves upwards.

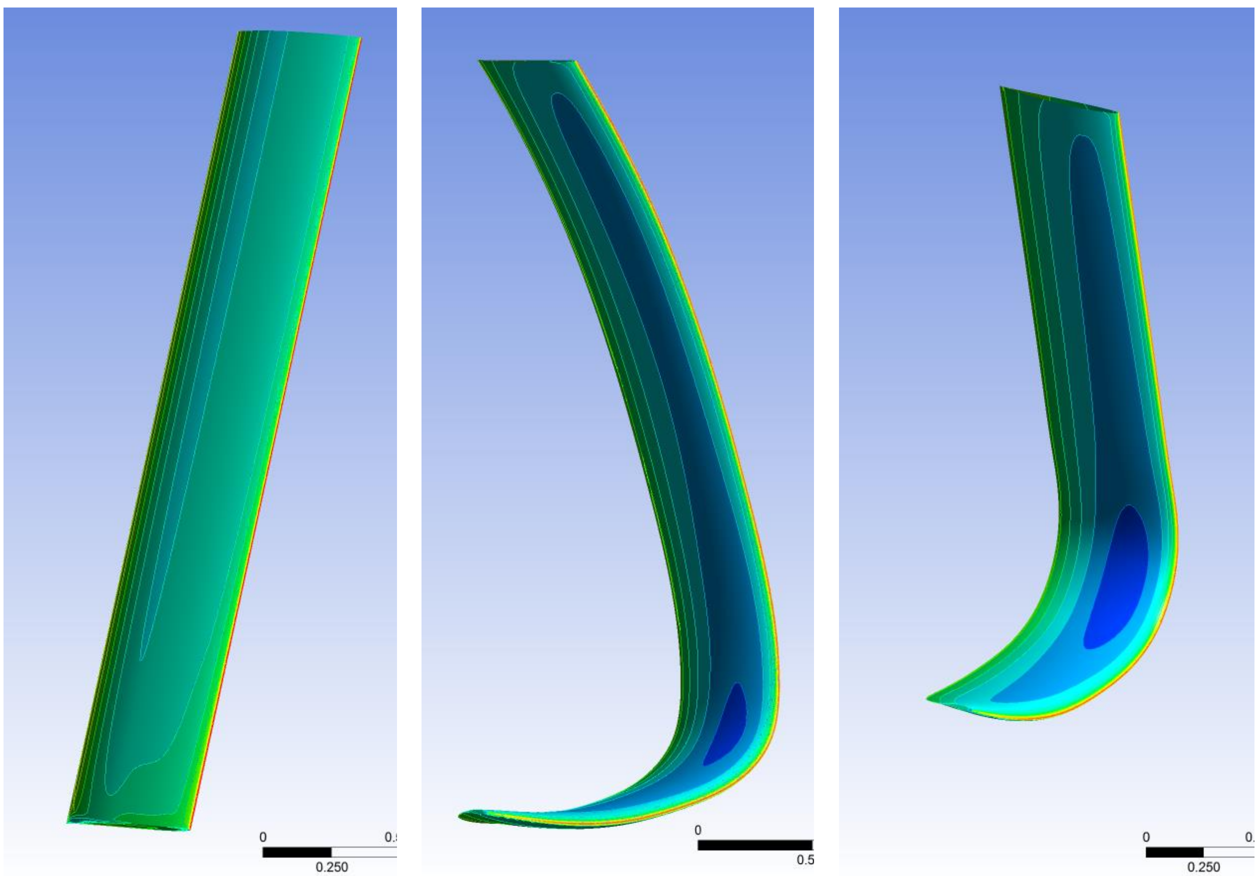


Figure 50: Control, Curved and Straight (L-R) Pressure Contours

The coefficient of lift in the figure 51 is similar to what is stated for the 2D properties of the Eppler section. It is clear that the control foil is a more effective lifting device through the range of speed simulated. This is assumed to be a function of the tip lift of the two DSS foils counteracting some of the main body lift. However, when compared with C_H[1], the curved foil and the control both yield similar results, indicating that although the curved DDS foil generates less lift; it generates it farther down the foil and thus results in a greater righting arm (figure on pg.77).

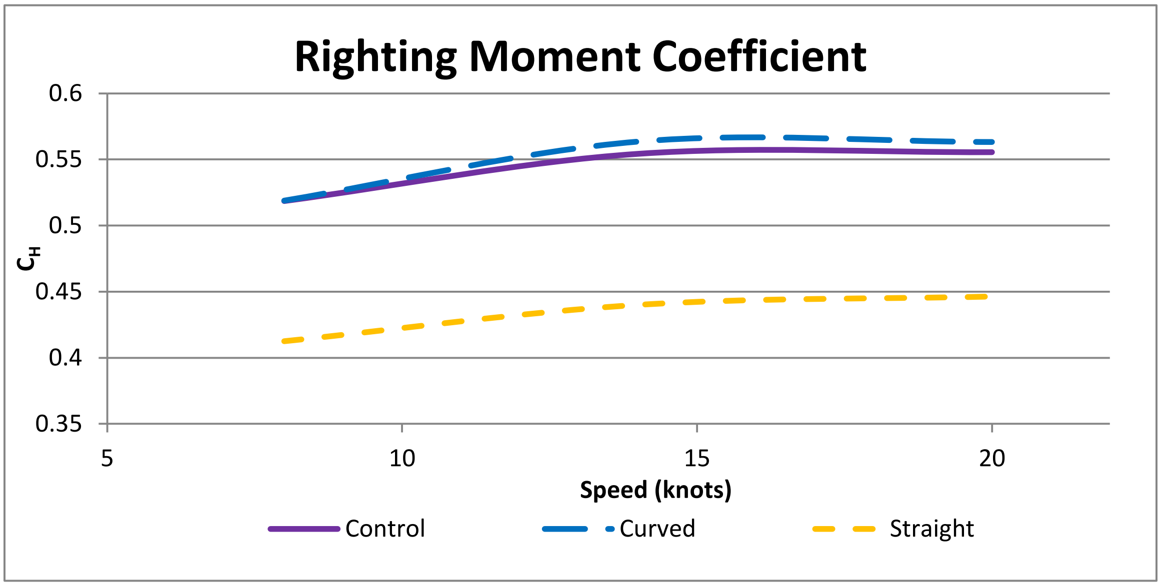


Figure 51: Righting Moment Coefficient vs Vessel Speed

Seen in the Figure 52 plot, the curved DSS foil generates the most effective moment (most moment per unit drag). The straight DSS foil was the least suitable for the application. Based on these results, it was decided to move forward with the curved DSS foil. This selection was based primarily on the higher values of C_H/C_D and higher overall values of righting moment as a curved geometry could be made bigger than a straight DSS foil while still fitting within the hull geometry. The figure below shows the difference in righting moment between the curved and straight DSS foils. As the speed increases beyond 14 kt, the speed where conventional dagger boards are being raised, the DSS foils are creating upwards of 25 percent of the static righting moment. As vessel speed increases to 20 kt, the curved foil is capable of generating almost half the static righting moment. This indicates the potential performance enhancements this system will provide while also drawing attention to the high levels of loading these foils will experience.

---

[1] $C_H = \frac{RM}{.5\rho V^2 A}$

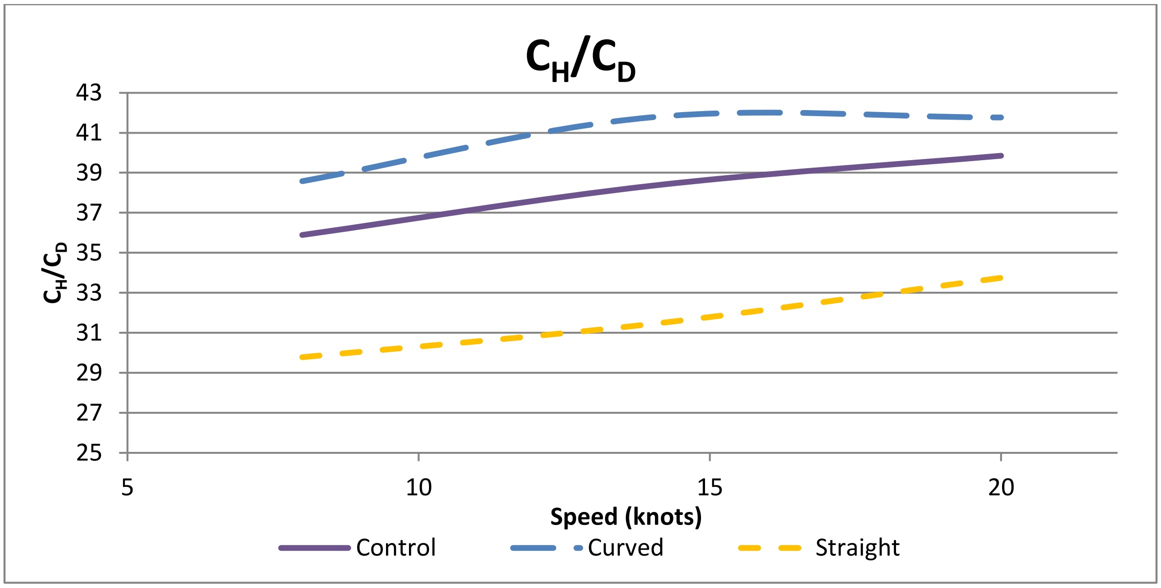


Figure 52: C_H/C_D vs Speed

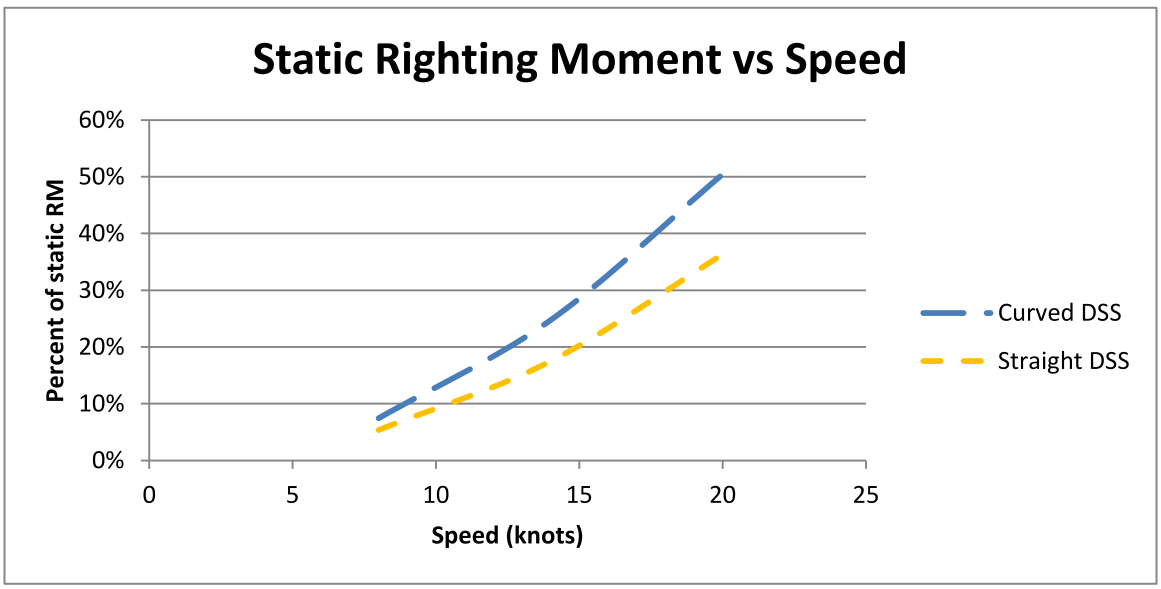


Figure 53: Righting Moment vs. Speed (10 degree Heel)

The implementation of the foil selection can be seen in the figure below.

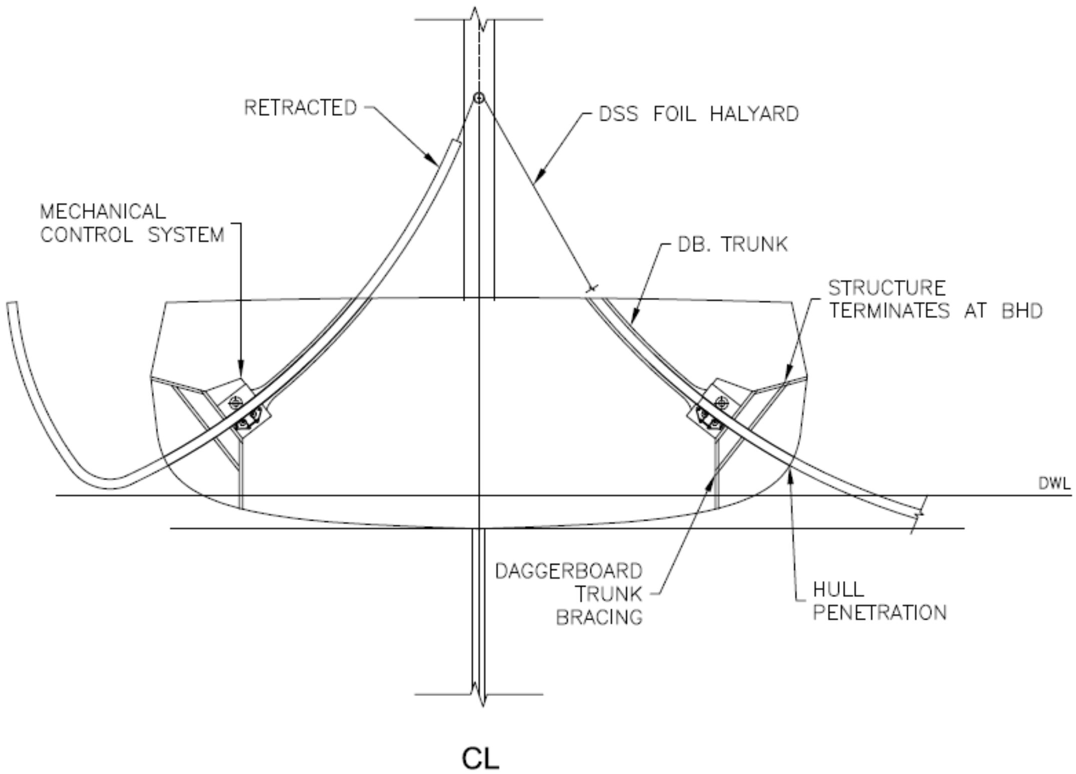


Figure 54:DSS Foil, Body Plan

12.4. Mechanical System and Structure

The control of the DSS foils will be through a continuous loop driven roller system. The system is comprised of an active top roller controller by lines leading aft to the cockpit. On the opposing side of the foil are two passive rollers; their purpose is to control the compression imposed by the roller system on the foil. Both sets of rollers are accessible through access hatches from above and below in case of failure or jamming. The system will be integral to the structural DSS trunk though not designed to act as a portion of the primary structure. All control lines will lead aft to the cockpit through aft line chases.

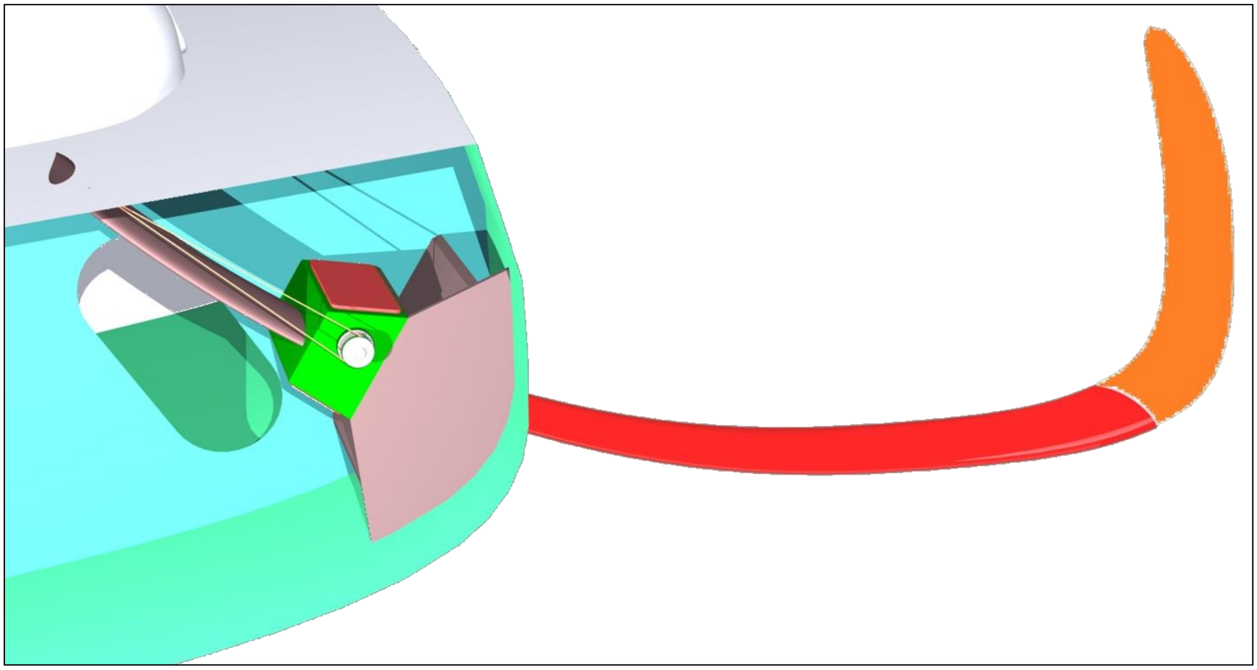


Figure 55: DSS Foil Arrangement

Although not part of the standard operation, a halyard will lead from the top of the foil out the DSS trunk to the mast for emergency retraction of the foil should the rollers fail. The halyard can also be employed to suspend the foil should maintenance or repair be necessary on the foil or roller system. The DSS foil lifting system is shown in Figure 56.

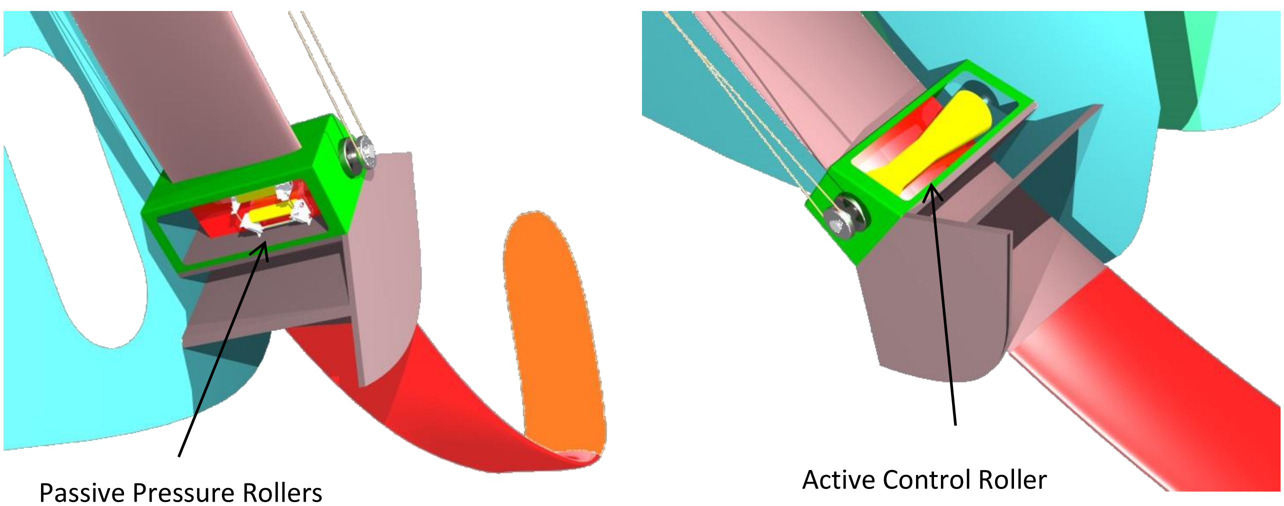


Figure 56: DSS Foil Lifting System (Top and Bottom Views)

12.4.1. Foil Strength

The structure of the foils is again a balance of strength and weight. In order to meet the structural requirements while also reducing weight, a tapered lay-up is implemented. The force of the foil was estimated as the projected area of the foil multiplied by the stagnation pressure of a flow as 25 knots. This represents a conservative approach, but is one that has been applied to conventional foils in the past (Merfyn Owen, 2013). Upon investigation it was found that the thickness of the Eppler 68 was not sufficient to handle the imposed bending moment. In response the thickness was increased by 20 percent- see *Modified E68* in Figure 45 .

The final weight of the foils was 110 kg, see Appendix 11 for workings. The foil trunk structure was estimated at approximately 30 kg (see section 9 for more details).

12.4.2. Collision Contingency

One of the major concerns with the DSS foil is its increased body profile when compared with a conventional daggerboard. Although the addition of a daggerboard also increases the body profile of the vessel compared to no daggerboard, it presents a much more compact profile as it is located more or less below the hull. A DSS foil on the other hand extends laterally beyond the hull thus increasing vulnerability. In an attempt to mitigate the effects of an impact it is proposed that the foils are built in two full enclosed sections -main foil and foil tip - and then joined together. This joint will act as a failure point that will engage prior to loss of the entire foil, allowing the foil to still be effective, although at a reduced level. Employing this construction method would also reduce the complexity and cost of construction.

12.5. Operational Profile

As discussed prior, the advantages of DSS foils are only seen as the vessel begins to operate off the wind, beam reach and greater. In the case of an IMOCA 60 competing in the VG, this yields "...gains 60% of the time and slight losses 20% of the time." (Vendée Globe -, 2015). The slight losses are due to the loss of lateral resistance that the daggerboards would have provided. This loss has to be made up by the keel which in turn reduces its cant angle and the sail carrying ability of the boat. Sailing close hauled represents less than 30 percent of the race conditions. In many cases, specifically light air conditions where righting moment is not an issue, the loss of lateral lift will be negligible.

These advantages do not come without operational complications. As with all elements of a sailing vessel, how hard and fast a vessel is driven directly impacts the risk of failure. Immersed length will have to be varied by the skipper in response to sea conditions and overall variability of wind, waves, and autopilot of the system. In heavy running conditions it is advisable that the boards are semi-retracted to reduce exposed area in case of a heavy broach. This is the decision of the skipper, based on their understanding of the designed strength and their willingness to push the vessel.

12.6. Final Remarks

While the concept and initial evaluation of this system has been established, there remains significant Research and Development (R&D) prior to completion of the design. Items such as tip section and the amount of load that they should be expected to carry as well as the effect of thickening the foil with respect to cavitation will need to be further analyzed. The tip does providing some lateral lift while sailing to windward, however its overall contribution may be minimal and thus a region where drag could be reduced. Cavitation at higher speed should also be investigated as it is a large problem on hydrofoils and this vessel will be reaching well up into the 20 kt -plus region were cavitation becomes a concern .

With regards to the structure of the foil and the foil trunk, there are limits to how much force the trunk structure can absorb before failure will occur. As a rule of thumb, a daggerboard/DSS trunk should be 7 times stronger than the foil (Merfyn Owen, 2013). In this regard, it may be found that the strength of the foil needs to be reduced in order to fail (without a doubt) before the hull and trunk structure. A second option would be to reduce the projected area of the foil, either though reducing the foil length or by shortening the chord. These options would have to be evaluated against the performance trade-offs before coming to a final decision.

13. Keel

The canting keel concept was pioneered in the 90's as a form of movable ballast to enhance and control righting energy of the vessel, ultimately allowing for larger sail area and greater power. It is especially useful on a single handed sailing vessel, as there is no extra crew onboard to act as movable ballast; which is seen in traditional crewed racing where extra hands will sit as far outboard on the windward gunwale as possible. In the IMOCA 60 rules, the range for a canting keel is limited to 38 degrees port and starboard.

13.1. New IMOCA 60 Structural Rule

The keel fin has been a common source of failure for the IMOCA 60 class, failing in extreme conditions and having very dangerous results. In the latest revision of the IMOCA rules the keel structure has been standardized as solid forged stainless steel construction. This results in a much heavier structure than some types used in the past such as composite construction, titanium, and manufactured steel, however some weight savings can be made by having only the forward half of the keel section constructed of stainless steel, with the trailing edge constructed of carbon fiber acting only as a fairing to achieve a NACA foil section shape.

13.2. Canting Mechanism

Three options were considered for the canting mechanism;

1. A fixed keel
2. Two (2) hydraulic rams on either side of the keel head
3. A single hydraulic ram starboard with a locking pin to port

Option 3 was selected for its simplicity and reliability in locking mode. With a dedicated locking pin, at least some control over the system can be maintained if there is a hydraulic or power failure. The canting mechanism will be housed in a keel box at the forward end of the navigation station, with transverse floor framing added IWO the mechanism seating. The keel arrangement is displayed in Figure 57.

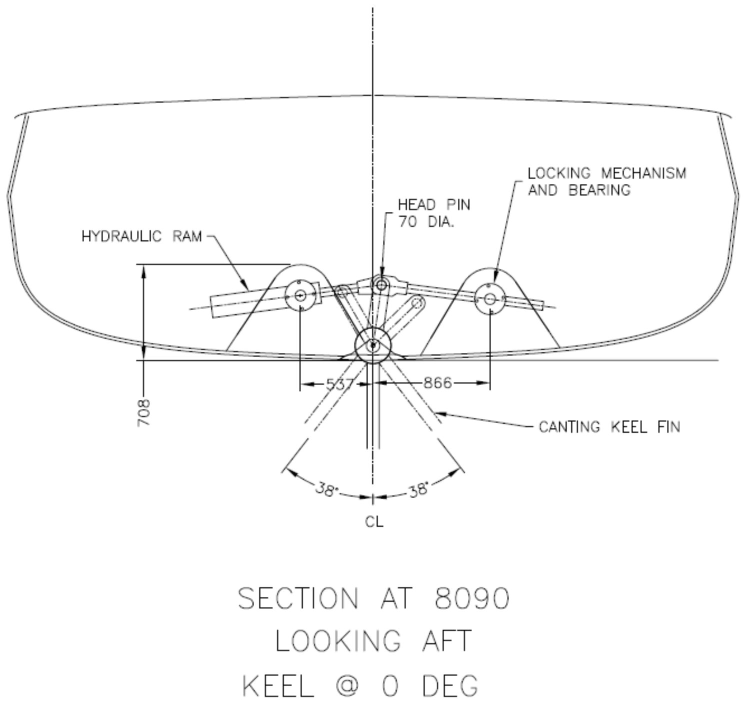


Figure 57: Canting Keel Mechanical Arrangement

13.3. Keel Fin Geometry

Different from a fixed keel or centerboard, the design drivers for the canting keel fin geometry have more emphasis on strength and drag reduction than on lift generation. There were some strength requirements laid out in the class rules, as well as some transverse self-imposed strength requirement of max righting moment transverse loading.

Initially a NACA 0012 section was chosen for the keel fin, the forward half chord to form the forged stainless steel portion of the fin. A simple beam bending calculation proved that this section at desired chord length did not have adequate transverse strength. The section was increased to a NACA 0018 and transverse strength was found to be ample. This section profile was then tested against the extreme loading conditions of grounding required by class rules. In both transverse and longitudinal cases of grounding the keel was found to be of adequate strength. Given the nature of the keel and bulb arrangement, there are also a minimum natural frequency requirements of the assembly, these were calculated and found to be well above the required values. There are additional keel assembly strength rules relating to the bearing capacity at the pin, and rams, these were expected to be met as the canting mechanical outfit was designed based on recommended materials and outfit sizes.

The type 600 project has opted to house a saltwater intake within the composite fairing of the keel. This is a convenient arrangement, which although complicates the keel box slightly, it guarantees salt water (SW) ballast availability at all times that the vessel is upright. An emergency portable SW intake hose will be onboard which can be used to ballast-up in a capsized condition. The keel fin geometry and SW intake is shown in Figure 58.

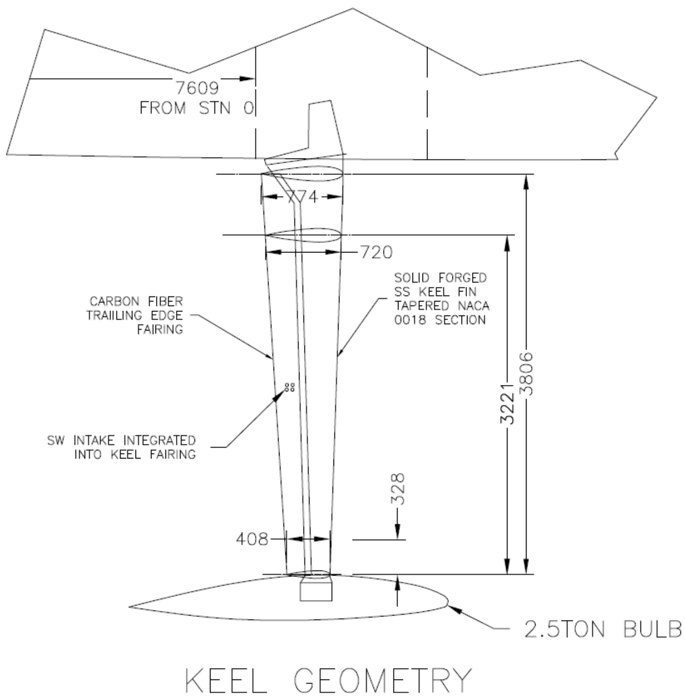


Figure 58: Keel Fin geometry with SW intake

13.4. Contingency Plan

Due to the arrangement of the canting mechanism, a failure within the hydraulic system or electrical system will severely reduce control of the canting keel. However the mechanical locking allows for the system to be manually locked in place, or released in an attempt to let the keel swing to a more favorable position. This small amount of control provides some reassurance when considering system failure.

13.5. Final Remarks

Upwind sailing, especially in lighter winds would be a case where it may be favorable to not cant the keel, or cant it to leeward. In these cases, where strong righting moment is not needed, and vessel speeds are slower, keeping the keel fin vertical can assist in providing lateral lift, preventing leeway loss. Canting to leeward in extremely low wind speeds can also help to bring shape back to the sails by using their own weight to fill them. Although this is not the primary function of the canting feature, it can be very helpful in extreme conditions sometimes seen through the doldrums. See Appendix 12 for strength and natural frequency calculations.

14. Electric Load Analysis

With the aim of The Type 600 Project being to navigate around the world single handedly, it is important that the vessel has enough resources to complete this goal. Since safety is a key element in successful design many of the major systems on the vessel such as the rigging, steering and foil are purely mechanical which allows for easier repair and maintenance. Other essential systems such as the communication and navigation equipment and the hydraulic ram for the canting keel rely on electric power. As one of the innovative goals of The Type 600 Project is to use renewable energy to power to vessel, it is crucial that there is enough power generated to run all of the electrical systems. IMOCA 60 Class rules list mandatory navigation and other electrical equipment. Since communication and navigation is the main draw from the electric load, the preliminary electric load analysis is focused on these pieces of equipment.

The Type 600 Project selected a B&G navigation package to fulfill the requirements of the IMOCA 60 Class rules. The navigation equipment was selected based on the navigation package supplied to Volvo Ocean Racing competitors which has a mission statement to sail around the world, similar to the VG. B&G equipment was used on over half of the 2012 VG entrants. The preliminary electric load analysis in Appendix 13 is based on the mandatory electrical equipment specified in the class rules. The preliminary Electric Load Analysis requires 12.7 kW power required during the day and 14.3 kW of power required during the night. These loads were multiplied by a safety factor of 2 to take into account additional equipment which will be considered in the next phase of design.

A goal of The Type 600 Project is to power the vessel with renewable energy. Wind turbines, hydro generators and solar panels have been selected to generate enough power for 24 hours of the worst case expected power consumption of the vessel. The expected power generation from renewable energy is broken down in Table 19.

Table 19: Expected Renewable Energy Generation

Renewable Energy Source	Day		Night	
	Use [h]	Power [kW]	Use [h]	Power [kW]
Thirty Three Solar Panels	8	8	0	0
Two Wind Turbines	6	3.24	8	4.32
Two Hydro Generators	10	9.6	10	9.6
Total Power Generated	-	20.8	-	13.9
Total Power Generated per Day	**35**			

Two (2) Racing Hydro Generators from Watt & Sea (Figure 59) were selected to be fitted to the transom inboard of the rudders. Each will produce 600W (20A at 24V) of power while the vessel travels at its average expected speed of at 12 knots. To reduce drag, the hydro generator will be lifted out of the water for speeds less than 5 knots.

Two (2) RDK Marine-Grade Wind Turbines with three blades spanning 1.17 m (Figure 60) were selected and will be fitted at the stern of the vessel where the wind energy system will not interfere with the sails and rigging systems. Each turbine can create a maximum output of 400W to a 12V system. The turbines require a minimum of 11.27 km/h winds to generate power and there is a regenerative electromagnetic break system for over speed protection.

Thirty four (34) 50W, 12V semi-flexible RDK solar panels (Figure 61) have been selected to be fitted to the aft deck of the vessel and along the port and starboard tumblehome with a total surface area of 13.4 m^2. They are very thin, lightweight, weather resistant, water resistance and corrosion resistant and can mold to the curved surface of the deck and side. They are made of high-efficiency monocrystalline solar cells and are strong enough to be walked on. The solar panels will continuously charge the battery.

The arrangement of the renewable energy is displayed in Figure 62 and the attached Drawings Package. The power generated from the renewable energy sources is displayed in the ELA in Appendix 13. Specification for the renewable energy sources are in Appendix 14.

Figure 59: Watt & Sea Racing Hydro Generators

Figure 60: RDK Wind Turbine

Figure 61: RDK solar panels

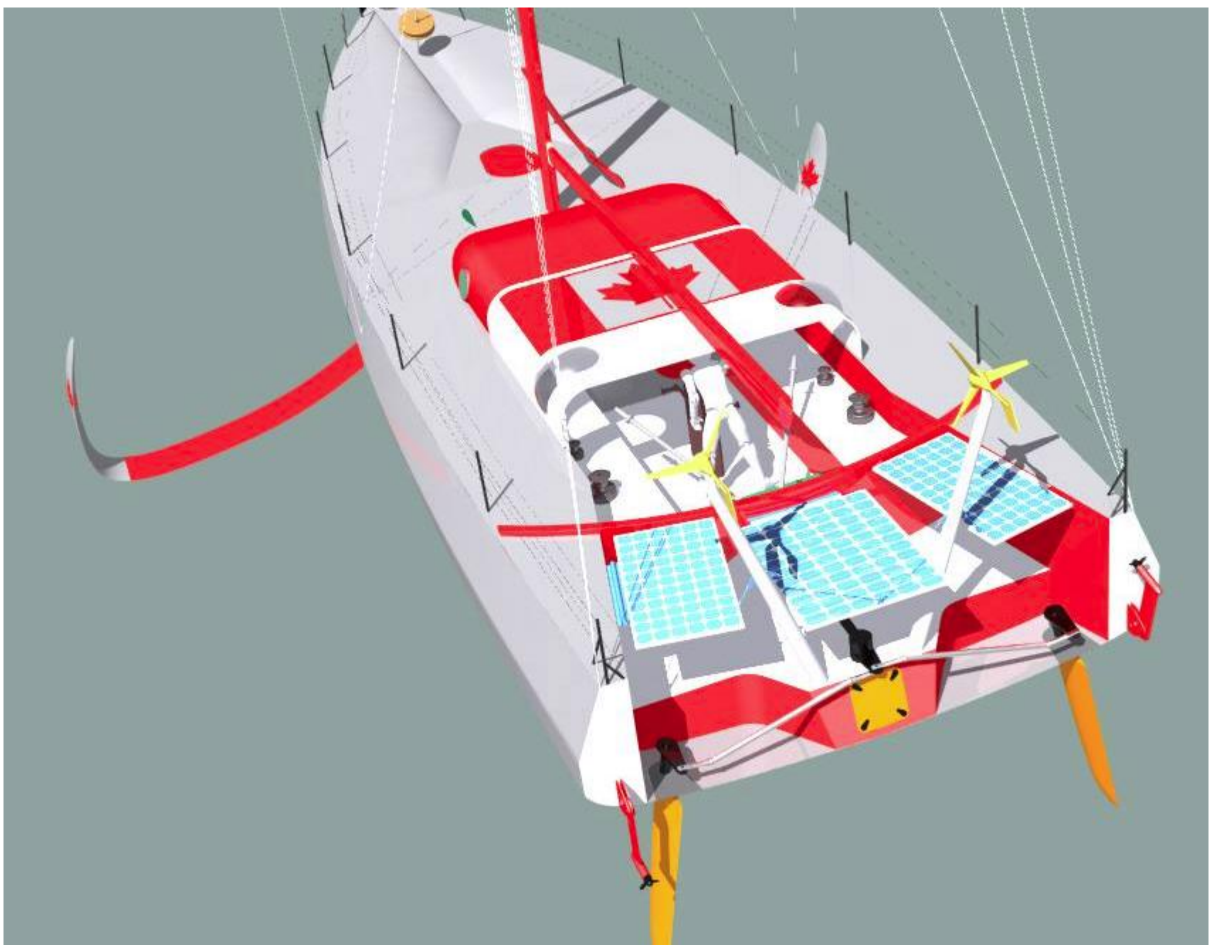


Figure 62: Renewable Energy Arrangement

Three (3) MVG 24 V Gel batteries bank has been selected for the vessel. It is common practice for vessels under 60 ft to have 12 V house battery banks, but for The Type 600 Project, there is a combination of 12 V and 24 V devices. A 24 V system will use smaller wire have a lower voltage drop which will be beneficial for the weight and efficiency of the system. Assuming the batteries remain close to fully charged at all times, there is a 5-6 hour reserve of power to the vessel.

Four (4) Buck-Boost DC-DC converters will be required. Three (3) 12 V-24 V DC-DC converters for each of the wind turbines and the solar panel systems, and one (1) 24 V-12 V DC-DC converter to provide the capacity for the 12 V devices from the 24 V battery bank. It is expected that there will be 90 percent efficiency from each converter. A block diagram of the electrical system is in Figure 63.

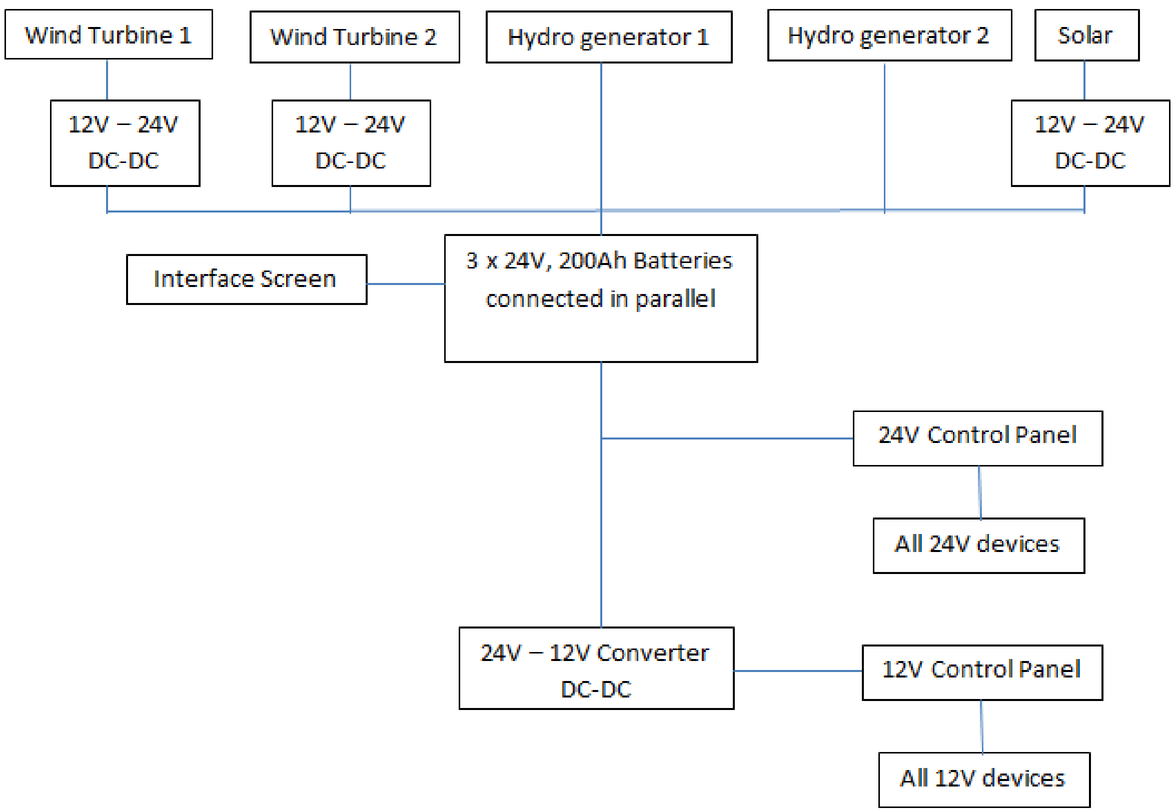


Figure 63: Electrical System Block Diagram

Assuming that during the day solar panels are collecting power for 8 hours, wind turbines are collecting power for 6 hours and Aqua Energy is being collected for 10 hours and at night wind turbines are collecting for 8 hours and Aqua Energy is collected for 10 hours, the boat will generate 20.8 kW in the day and 13.9 kW in the night. Comparing this to the power required, the vessel will be able to function on renewable energy alone. A summary of the Electric Load Analysis (ELA) with the renewable energy generators is in Table 20.

Table 20: Electric Load Analysis Results

	Day	Night	Units
Safety Factor	2	2	[-]
Power Required	12.7	14.3	[kW]
Power Generated	-20.8	-13.9	[kW]

15. Engine Selection

From the IMOCA 60 class rules, the vessel must have an emergency engine with a minimum of 37 hp (27.6 kW). The rules allow for either combustion or electric engines. Although The Type 600 Project is aiming to reduce CO2 emissions, the losses from the increased weight of an electric engine compared to a diesel engine selection and the complexity of the systems were considered. Since the engine is only going to be used in case of an emergency which would force an early retirement from the race, an electric engine is out of the scope of this phase of design.

IMOCA 60 class rules state that engine must be checked to meet the following requirements at the end of a race:

1. Power the vessel at 5 knots for duration of 5 hours
2. Create a bollard pull value of 280 daN[2] at a fixed boat for 15 minutes (boat tied to the dock, bollard pull measured with a dynamometer)

The NavCad Delft 2/3 prediction method was validated for low speeds in section 8.6. From the resistance and power curves in Figure 65 and Figure 66 to overcome 280 daN of force, the vessel would be going 8.5 knots in calm seas. This corresponds to 16.5 kW of power. With a 37 hp engine, we will meet requirement 2 above. From the power curve in Figure 66, the power required to propel the vessel at 5 knots in calm seas is 2.7 kW. However, it is unlikely that the conditions which will require the engine to be used will be calm. The fuel capacity will be designed for the maximum rpm and power consumption. From Figure 64, the maximum rpm is 3600, which corresponds to 27 kW power and 320grams/KWh fuel consumption. With 5 hours duration, and 1/3 reserve, an 18 gal fuel tank is required.

[2]DecaNewton = 10 Newtons

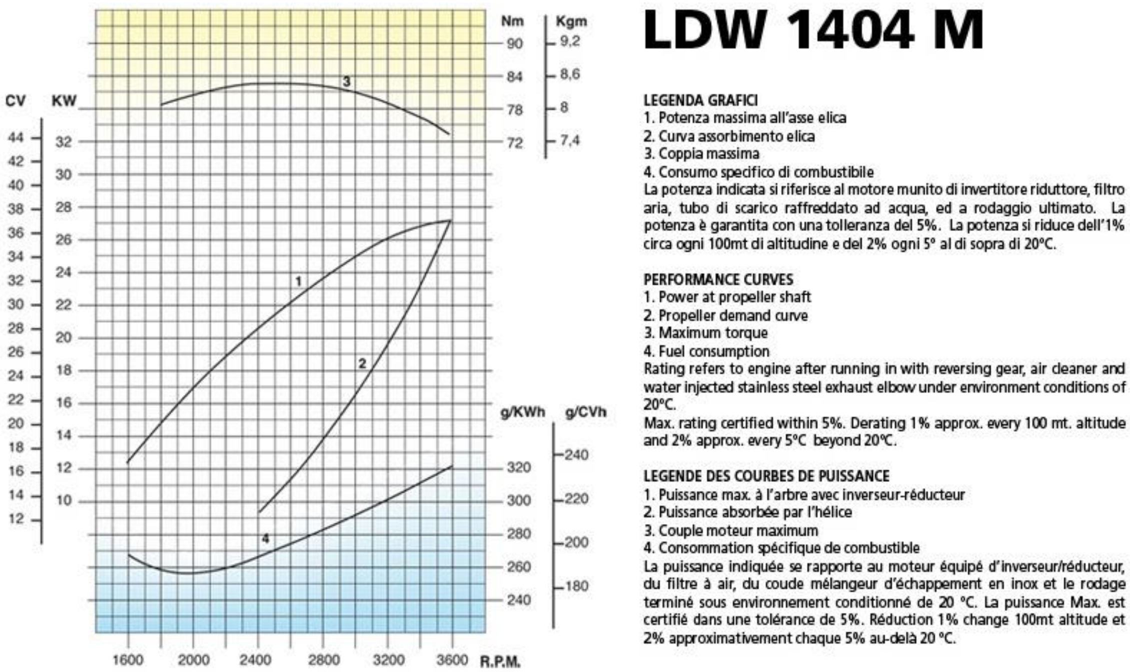


Figure 64: LDW 1404M 37 hp engine powering curves (ref: http://www.lombardinimarine.com/download/curva1404m.jpg)

As per IMOCA rules, the propeller is not foldaway or located on a movable appendage. It is positioned permanently in the water flow running under the hull with zero heeling angle. A 4-stroke, 37hp LDW 1404 M inboard engine was selected in combination with a low drag hydro drive ¾ to 1 ton Saildrive. The saildrive propeller is located below the waterline on the centerline of the vessel. This arrangement is common amongst current IMOCA 60 vessels and can be seen in the GA drawing in the attached Drawing Package.

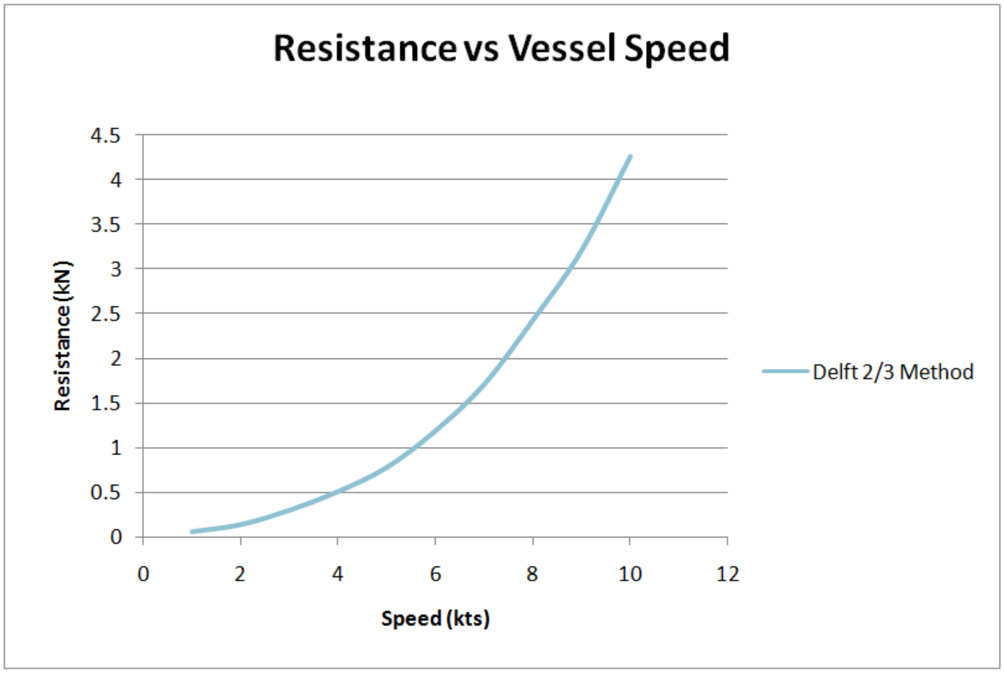


Figure 65: Resistance vs Ship Speed

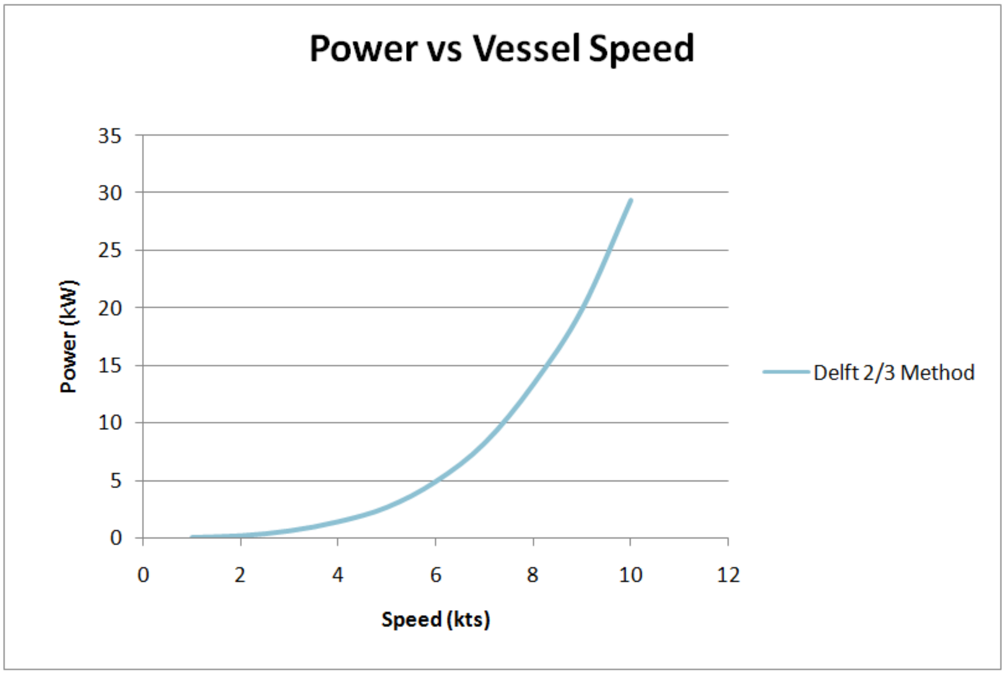


Figure 66: Power Required vs Ship Speed

16. Safety

Safety is a high concern for a solo racing event because there can be no outside assistance without major waiting periods. It is important that each aspect of the design, especially the systems which have been identified as having a high failure rate historically, have contingency plans. These involve having backup equipment onboard, designing failure points to prevent catastrophic damage, and carrying tools and material for maintenance or repair. The contingency plan for each major system has been described in more detail in the sections above.

16.1. Loss of Electrical Power

Backup navigation equipment which does not require electrical power must be included on all IMOCA 60 vessels. As per the class rules, two (2) marine magnetic compasses and navigation charts for the VG race route will be on board.

16.2. Escape Hatch

In the event of a capsize, the rules state that the boat must have at least two (2) emergency exit escape hatches of $0.2m^2$. One is located on the deck, forward of the mast. The second hatch is located on the transom, above the water line. This hatch is accessible when The Type 600 Vessel is in upright or inverted position.

16.3. Equipment

The IMOCA 60 rules also outline the following mandatory safety equipment which must be carried onboard the vessel:

- Stowage plan marking the location of key safety equipment
- Draining system – Two (2) permanently installed manual bilge pumps, one from above deck, one from below deck (min outflow of 1.3 litres/cycle). Electric draining system (minimum outflow of 2400 litres/hour). Two (2) stout buckets (minimum capacity of 9 litres).
- Emergency drinking water
- Lifebuoys
- Pyrotechnic signals

- Lifejackets
- Safety harness and safety line
- Soft wood plugs for through hull fittings
- Two (2) fire extinguishers
- Fire blanket
- Foghorn
- First aid manual and first aid kit
- Heaving line readily accessible for use from the cockpit
- Diving equipment
- Immersion suite
- Life rafts – Two (2) shall be carried on board.
- Watertight emergency container containing communication equipment, navigation equipment, flash light, knife, fluorescein sea markers, pyrotechnic signals, high energy food, survival blanket and a radar.
- Emergency attachment points – For the life raft, watertight container, individual grab bags and distress beacons.

17. Environmental Impact

Environmental impact is becoming more important in the offshore racing industry. Elements of the design which are prone to failure are being designed with this in mind.

17.1. Food waste

The VG racing rules state that the skippers are prohibited from throwing any garbage into the ocean. This rule comes into play mostly regarding the waste from food. According to an interview with a previous VG competitor, the daily intake of calories can approach 6000, and therefore special care must be taken when selecting the food onboard the vessel to ensure that the skipper gets all the nutrition he needs while minimizing the extra weight and waste. This interview states that all of the meals are sealed in bags and labeled so the skipper will know at which point throughout the race he should eat certain meals (Vendee Globe TV, 2015).

Figure 67 Brian Thompson's prepackaged meals (Thompson, 2008)

Possible solutions for this problem include using a more efficiently packed food system, such as using reusable containers without carrying extra weight, or using products that are packaged in bio-degradable materials which would be harmless to the ocean or marine life if thrown overboard. Another option could be to use edible packaging materials such as Monosol packaging products which are designed to be dissolved in water and eaten with the food. Unfortunately these products are not yet developed enough to be of use, since the edible packaging requires a secondary packaging to prevent contamination.

17.2. Renewable Energy

Only one vessel in the past, Acciona 100% Eco-Powered, has attempted to complete the race on only renewable energy resources including wind, hydro, and solar energy. With an attempt to create zero waste and The Type 600 Project design will be almost completely powered by alternative energy. With two hydro generators and two wind turbines at the stern, and strong, flexible solar panels on the deck; the vessel is expected to produce an average of 2500 Ah per day. With an estimated Electric Load Analysis requiring 2000 Ah, the renewable energy system will generate enough energy to power the electronics on the vessel. The reason for selecting a propane stove rather than relying on renewable energy is the major weight savings that were seen when comparing electric stoves with fuel burning stoves as well as a very high electrical load associated with an electrical stove.

The emergency backup engine which is a requirement of the IMOCA class has been selected as a 37 hP Diesel sail drive. The reason a diesel selection has been made is to reduce weight while still operating on zero carbon emissions, since using the emergency engine results in retirement from the race.

18. Cost & Economics

While most projects in the commercial sector are driven by direct financial return on the investment, as one shift towards sectors that serve purposes beyond direct applications such as research and public relations, the financial return becomes harder to quantify and the time-line of return indefinite. These types of projects are of value both to the development of the corresponding commercial sector and in generating public interest, both in the short term though product recognition and long term by sowing the seeds of inspiration from which the future work force will grow.

This project represents an opportunity to develop all three of the indirect returns associated with sponsorship of a high performance project

VG campaigns are funded primarily through corporate sponsorship although there are notable campaigns that have been made possible through public donations. Corporate support is driven by the promotional aspects of the race; either as an exhibition of their products capabilities or by ad campaigns that relate the inspirational aspects of the race with the company's product.

The latter is best demonstrated in the partnership between Hugo Boss© and Alex Thompson Racing. Alex Thompson presents the inspiring dream (if you're from England) of being the first British citizen to win the Vendée Globe, a race typically dominated by Frenchmen. Hugo Boss on its part has capitalized on Alex Thompson's many sailing stunts suited in Hugo Boss apparel and the national pride aspect of Alex Thompson's campaign. The two most notable promotions Mr. Thompson has performed for Hugo Boss have been the '*Keel Walk*' and '*Mast Walk*', the former in Figure 68 having been viewed close to 2.5 million times on *YouTube*. These stunts are unique with respect other skippers on the IMOCA circuit, but they demonstrate the promotional returns on such an investment.

Figure 68: Alex Thompson: 'Keel walk' (elks, 2012)

Another other aspect to sponsorship is companies who can use the vessel to promote their capabilities and products. Interests of this nature typically come from aerospace and defense contractors. As an example, the vessel Safran and her skipper Marc Guillemot's primary support came from Safran S.A. A company that specializes in aircraft, rocket-engines and aero-space components that through its involvement with exotic materials saw the 2008 VG Safran fitted with a titanium keel fin which is unique in the IMOCA class. Safran's case is but one example of how the integration between sponsor and race team is carried out. There are various other teams sponsored by composite manufacturers or other high-performance component manufacturing companies that integrate their product into the vessel or, as smaller sponsors, the race outfit such as Harken winches or Gill foul weather gear

18.1. Past Canadian Campaigns –IMOCA and Other

Past Canadian entrants have had mixed luck. Gerry Roufs, tragically lost in the 1996 VG edition, was sponsored by Groupe LG, a specialized forestry product company from Quebec. Derek Hatfield's 2008 entry on the other hand was supported by over 6000 individuals. Mr. Hatfield's campaign demonstrates the high level public support that can be found within Canada for offshore racing.

Another project with similar financial goals and non-commercial organizational structure is Sail And Life Training Society (S.A.L.T.S) campaign to build an offshore training vessel. Their vessel construction costs are comparable to an IMOCA 60 at approximately 5.5 million USD and they will rely heavily on public donations to fund the construction costs (Sail And Life Training Society).

Prior to commencing the construction phase of the project, S.A.L.T.S is looking to secured 80 percent of the total construction cost through donations of 10,000 USD or more. Once the required capital is established, they will begin construction while continuing their fundraising. This graduated fundraising scheme represents a template for the proposed IMOCA 60 campaign.

18.2. Current Campaign and Cost

From reviewing past Canadian entrants and taking into account other non-profit build campaigns it was decided that the best method would be a combination of both corporate and private donations. For this campaign it was assumed that the sponsorship break down would be similar to the S.A.L.T.S. campaign. In their case, they have projected their fundraising goals based on value and percentage.

Table 21: Campaign Budget (Gain, 2012), (Vendée Globe, 2012)

Initial Cost Estimate		
Vessel Construction	4.2	[Million USD]
R&D (Pre-Build) - (~ 30% of vessel cost)	1.3	[Million USD]
Approximate Total Vessel Cost	5.5	[Million USD]
Campaign Cost Estimate		
Team Optimisation	2/18	[Million USD/months]
Approximate Total Costs - 18 month campaign	7.5	[Million USD]

For a VG campaign the overall costs are around 7.5 million USD over 18 months and up to a year of design and research work. The majority of these costs are associated with the design, construction and outfit of the vessel, approximately 5.5 million USD. This is a large capital requirement and would be required prior to starting the detailed design. The other 2 million USD would be drawn out over the 18 months leading up to the VG race. The initial costs of vessel's design work would rely on large private or commercial sponsorship. This would provide the capital injection to initiate the project while also ensuring that there was enough capital to see the vessel construction through to completion.

Once the vessel is complete, smaller private donations would fund the race portion of the campaign, in a similar fashion to Derek Hatfield. The benefit of combining corporate and private donations is the reduction of the financial responsibility of the corporate contributors while also engaging supporters who would like to contribute, but do not have the deep pockets to make sizable donations. These smaller donations also feed back into the breadth of exposure that the corporate sponsors are looking to receive in exchange for their investment.

18.3. Demand

There is currently an initiative, “Canada Offshore Sailing”, that is looking to put a Canadian in the 2016 VG. To date they have secured a vessel, Derek Hatfield's *O- Canada*, and funding to launch the vessel and sustain their campaign in the short term. While this initiative has many hurdles to cross prior to entry into the 2016 VG and even more for a 2020 entry, there is the support available within Canada to make it a reality.

Assuming a successful entry in the 2016 VG, this Canadian team will need a world class vessel if it is to continue. The vessel presented by Crash Box Design Ltd above would compete as a world class entrant. A Canadian designed vessel, for a Canadian Skipper, funded by the people of Canada.

18.4. Vessel Design Features –Multi-Media

From the standpoint of all parties involved with the public image, it is the human vs. nature element that engages people. The enduring length, through some of the worst weather on the planet, augmented by the fact that it is completed alone, captures the attention of the public to understand what motivates people to this level of insanity. To bring the human element of this race to the public, the Type 600 Vessel has been designed to incorporate multiple media stations. The stations have been placed around the vessel to capture the weather conditions, daily activities and sentiments of the skipper. To capture the skipper’s thoughts, video stations have been installed in the navigation station and on deck above the companionway hatch (see Drawing Package). Both stations allow the skipper to record their reflections without having to deviate from their daily activities.

To capture the shipboard life, three cameras have been installed in the high use areas. One is located on the mast facing forward to capture the fore-deck activities, primarily sail handling; one is located on the aft rail to capture the cockpit activities from a broad stance, and the final camera is mounted within the coach roof facing aft to bring a sense of closeness to the footage.

Weather conditions are captured via a camera on the underside of the bow sprit with fisheye lens facing aft, as well as an aft facing camera located on the coach roof to captures following seas.

The entire video system will be integrated through the central communication system and the raw footage will be sent to headquarters for processing. The operation of the system could be set to either manual or automatic. In the automatic setting, footage could be captured at regular intervals with minimal energy of the skipper while when set to manual, important events, such as changing a head sail could easily be captured and recorded.

While there is an increase in construction cost associated with integrating this system, the increase in footage and stills captured would be a huge advantage to any sponsor interest in promotional media and developing a public following.

18.5. Identified Canadian Corporate Sponsors

The following briefly covers the type of companies that may be approached for sponsorship. They have been identified in this report as prospects due to the following elements and divided based on potential interest – either sector based or public image:

- Involvement in the marine or engineering sector
- Strong relation between sales and public support
- Canadian owned or heavily involved in Canadian operations

18.5.1. Sector Related Companies

Secunda Canada : Manages harsh-weather fleet of offshore support vessels. Primarily focus is on Eastern Canadian waters servicing the offshore industry (Secunda Canada, 2013). In the past, the company owned the Schooner *Highlander Sea* which was used as a promotional vessel and competed in races against the Bluenose II.

Seaspan Marine Corporation: "Provides coastal and deep-sea transportation, bunkering, ship repair and shipbuilding services in Western North America." Their Vancouver yard secured the non-combat portion of the National Shipbuilding Procurement Strategies. They employ large numbers of naval architects and marine engineers. By supporting this project they would demonstrate their commitment to the Canadian marine industry.

18.5.2. Public Image Companies

The other type of companies that could benefit from supporting this campaign would be those who leverage a large portion of their sales through advertisement and popular public opinion. Their interest in this project would be much more related to the generation of multi-media that can be used in advertisement and public campaigns.

Tim Hortons: Large public image typically thought of as 'Canadian Icon' which could tie in with the Canadian image of the vessel.

The Bay: A company with a long Canadian history and historically strong links to the maritime industry. In the past they have been known to sponsor high-performance sporting events, most notably the 2010 Olympics.

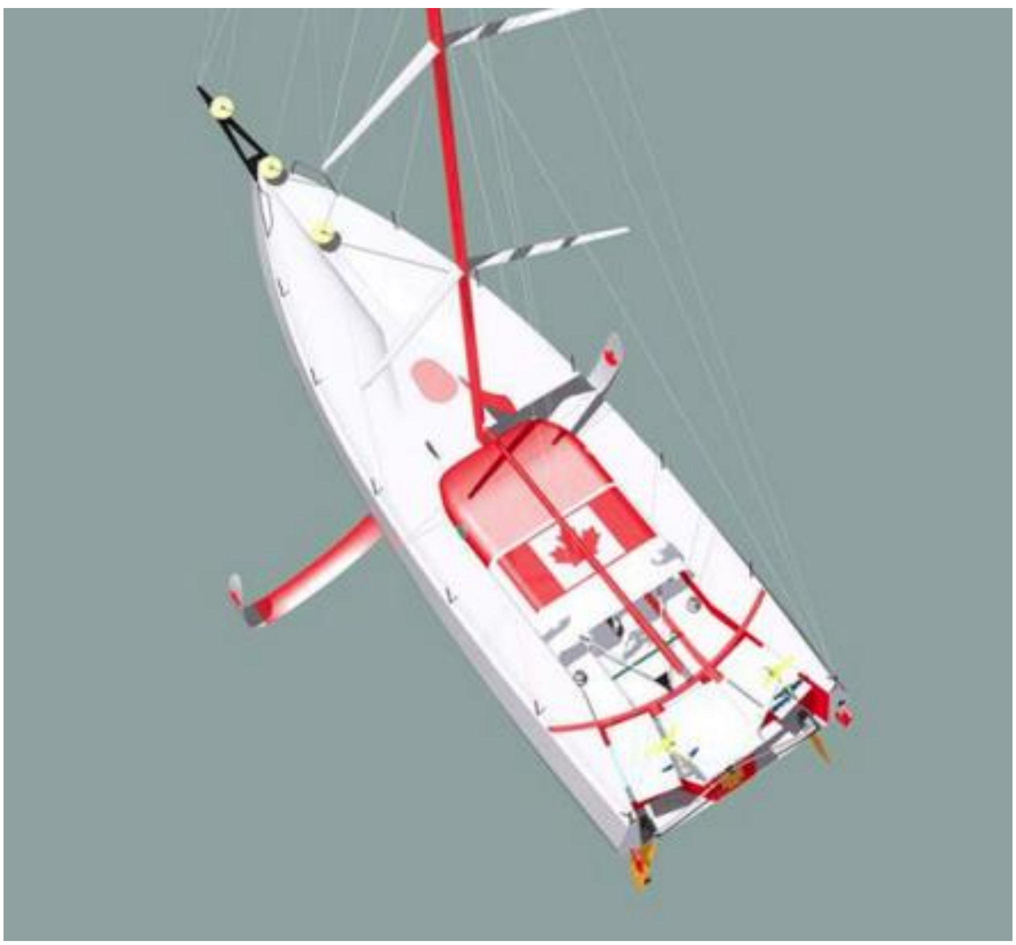


Figure 69: Rhino Rendering of The Type 600 Vessel, Crazy Horse

References

Almeida, R. (2015, January 15). *Next Vendée Globe to Feature Foiling Monohulls* . Retrieved 03 27, 2015, from gCaptain: http://gcaptain.com/next-Vendée-globe-feature-foiling-monohulls-video/

ANSYS CFX. (2015). Retrieved March 25, 2015, from ANSYS: http://www.ansys.com/Products/Simulation+Technology/Fluid+Dynamics/Fluid+Dynamics+Products/ANSYS+CFX

Colbourne, B. (2012). *Engineering 4011 Ship Resistance and Propulsion Course Notes.*

Dynamic Stability Systems. (NA, NA NA). *Dynamic Stability Systems - Technology*. Retrieved 03 19, 2015, from Dynamic Stability Systems: http://www.dynamicstabilitysystems.com/contact/

elks, S. (2012, February 01). *Daredevil walks along keel of a boat in James Bond-style stund*. Retrieved March 20, 2015, from Metro: http://metro.co.uk/2012/02/02/daredevil-walks-along-keel-of-boat-in-james-bond-style-stunt-306255/

Fassati, F. (2009). *Aero-Dynamics and the Performance of Sailing Yachts.* International Marine/McGraw Hill.

Gain, B. (2012, March 05). *Opinions vary on vendée globe one-design*. Retrieved 03 20, 2015, from Sailing World: http://www.sailingworld.com/racing/opinions-vary-vende-globe-one-design

General Hydrostatics. (2012). Retrieved from Creative Systems Inc.: http://www.ghsport.com/home.htm

Infiniti Yachts. (2014, January 01). *DSS Foil Lifts Wilds Oats XI to Seventh Rolext Sidney-Hobart Race*. Retrieved March 27, 2015, from Infinit Yachts: http://www.infinitiyachts.com/dss-foil-lifts-wild-oats-xi-to-7th-rolex-sydney-hobart-line-honours/

Lars Larsson, R. E. (2014). *Principles of Yacht Design.* Camden, Maine: International Marine/McGraw-Hill Education.

Merfyn Owen, I. C. (2013). Daggerboard evaluation for an IMOCA 60 yacht. *Innov' Sail Conference* (p. 16). Lorient: Owen Clarke Design.

Monthly/Seasonal Climate Composites. (n.d.). Retrieved from National Oceanic and Atmospheric Administration: http://www.esrl.noaa.gov/psd/cgi-bin/data/composites/printpage.pl

NACA 63-015A AIRFOIL. (2015, March). Retrieved March 15, 2015, from Airfoil Tools: http://airfoiltools.com/airfoil/details?airfoil=n63015a-il

NavCad. (2014). Retrieved from Hydrocomp Inc.: http://hydrocompinc.com/software/navcad

Sail And Life Training Society. (n.d.). *New Schooner Project -Profect CAse Statement.* Retrieved March 23, 2015, from SALTS: http://www.salts.ca/site/our-tall-ships/new-schooner-project.html

Sail7. (2012, November). Retrieved March 25, 2015, from Sail7: http://www.xflr5.com/sail7/sail7.html

Secunda Canada. (2013, January 01). *Our Company: Secunda Canada*. Retrieved March 20, 2015, from Secunda Canada: http://www.secunda.ca/vessel-ryanleet.php

The Route. (2012). Retrieved 2015, from Vendée Globe: http://www.Vendéeglobe.org/en/the-route.html

Thompson, B. (2008, December 9). Brian Thompson's on board meal plans during the Vendée Globe World Race. (T. Telegraph, Interviewer)

Vendée Globe. (n.d.). Retrieved from Vendée Globe: http://www.Vendéeglobe.org/en/

Vendée Globe -. (2015, January 15). *Armel Le Cléac'h reveals his new Banque Populaire VIII*. Retrieved March 22, 2015, from Vendée Globe: http://www.Vendéeglobe.org/en/news/article/14654/armel-le-cleac-h-reveals-his-new-banque-populaire-viii.html

Vendée Globe. (2012, October 07). *How much is a Vendée Globe budget*. Retrieved March 20, 2015, from Vendée GLobe, Les Sables d'Olonne: http://www.Vendéeglobe.org/en/news/article/1535/how-much-is-a-Vendée-globe-budget.html

Vendée Globe TV. (2015). Retrieved 2015, from Vendée Globe Channel: https://www.youtube.com/user/VendéeGlobeTV/videos

VG. (2013, February 26). *Statistics and Figures*. Retrieved June 2014, from Vendée Globe: http://www.Vendéeglobe.org/en/news/article/13669/statistics-and-figures.html

XFLR5. (2014, 02 01). *XFLR5- Home page*. Retrieved 03 20, 2015, from XFLR5: http://www.xflr5.com/xflr5.htm