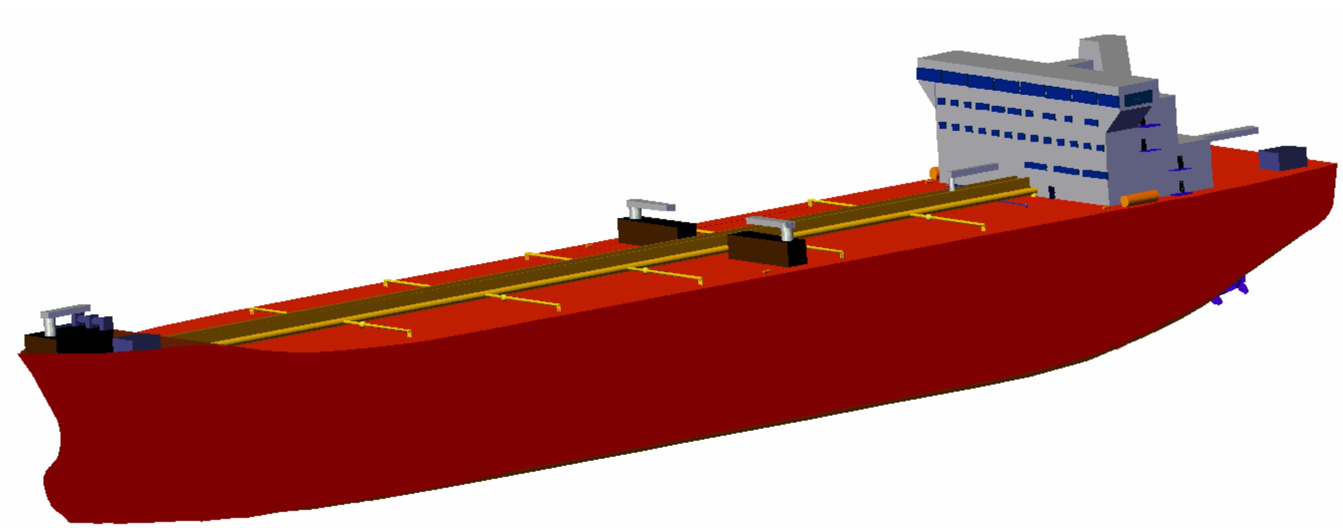
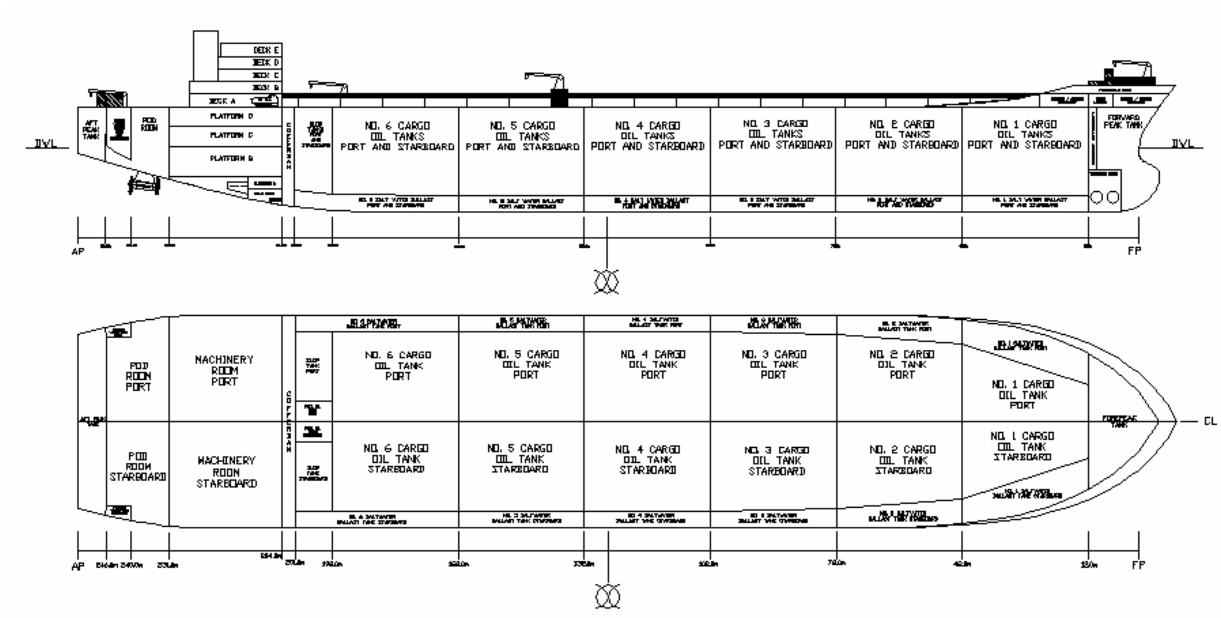
Virginia Tech Shuttle Tanker



Team Leader - Korin Strome	14618
William Moon	16025
John Sajdak	15346
Andrew Quillin	16411
Jessica Hopper	14675

Ocean Engineering Senior Design Project

Executive Summary



The Hibernia Shuttle Tanker's mission is to load oil from an offshore loading platform located in the Hibernia Oil Field and deliver the oil to either a Trans-Shipment Terminal in Newfoundland, Canada or to various ports on the East and Gulf Coasts of the United States. To optimize the design of the Shuttle Tanker, a structured design approach is utilized. First, trade-off studies are performed to analyze possible design solutions to meet the general requirements. Then, a Pareto Genetic Algorithm is used to identify a variety of feasible ships on a non-dominated frontier to optimize the effectiveness vs. cost of the baseline concept design.

In order to operate in the harsh conditions of the North Atlantic, the Shuttle Tanker has a dynamic positioning system that allows bow-loading capabilities in Sea State 6. The Shuttle Tanker also utilizes an integrated power system with podded propulsion to increase efficiency and maneuverability. To increase safety and decrease the risk of oil outflow, the Shuttle Tanker is ice strengthened and meets the structural requirements of both the American Bureau of Shipping and the Canadian Arctic Shipping Pollution Prevention Regulations. In fact, the ship is designed to survive a collision at 15 knots with a 10,000 tonne iceberg without shell rupture.

One of the main focuses in the design of the Shuttle Tanker is to reduce environmental impact. Hull coatings are chosen to reduce the seepage of heavy metals into the water. In addition to the fuel tanks being placed within the double hull, a large double bottom height and double side width are utilized to reduce oil outflow in the event of a collision or grounding. The ship is designed with crew safety as one of the top priorities and there is ample life saving and rescue equipment on board. The deckhouse is arranged to optimize convenience for the 28-member crew and be a highly producible structure. Cargo, ballast, bow loading and inert gas are the Shuttle Tankers four main mission systems. The Shuttle Tanker meets or surpasses all the general requirements and does so at a low total ownership cost.

Principal Characteristics

Characteristics	Baseline Value
LBP [m]	252.77
Beam [m]	50.55
Draft [m]	14.87
Cp	0.824
Cx	0.995
Lightweight [MT]	32832
Full load displacement	159832
FL Vertical CG [m]	14.118
Cargo [MT]	125920
Sustained speed [knt]	15
Lead Ship BCC [$M]	144.2
TOC [$M]	210.9
Manning	28
Cargo Divisions	6 x 2
OMOE	0.9473

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1 Requirements and Plan

1.1 Owner's Requirements

This report chronicles and documents the design process of a Hibernia Shuttle Tanker. The Shuttle Tanker must onload oil from a submerged Offshore Loading System (OLS) in the Hibernia Oil Field, located off the East Coast of Canada in the Grand Banks. It must then transport the oil to either the specially built Whiffen Head Trans-Shipment Terminal in Newfoundland, Canada or to various ports on the East and Gulf Coasts of the United States. In order to load the oil from the OLS, the Shuttle Tanker must have a bow-loading and a dynamic positioning system. Operating in the harsh environmental conditions of the Grand Banks requires the consideration of hull ice strengthening. Due to the Grand Banks being an extremely sensitive environmental area, technical solutions must be found to accommodate the many environmental restrictions of the area. The Shuttle Tanker also has many system operational requirements including a cargo, ballast and inert gas system. Finally, the ship must comply with ABS Class Rules, U.S. COFR and port regulations. The owner's requirements are explicitly defined in Appendix A.

1.2 Design Philosophy and Process

This project uses a total systems approach to the ship design process - eliminating many informal design decisions. Figure 1.2.1 provides a flow chart of the project's design process (circled in green) that includes concept exploration and concept development. The concept exploration phase, described in Chapter 3, incorporates a structured mathematical search of a generated design space based on multi-objective considerations of cost and effectiveness. This methodology replaces a more traditional "ad hoc" ship design process, based upon experience, rules of thumb and design lanes. Concept development, described in Chapter 4, follows the design spiral illustrated in Figure 1.2.2.

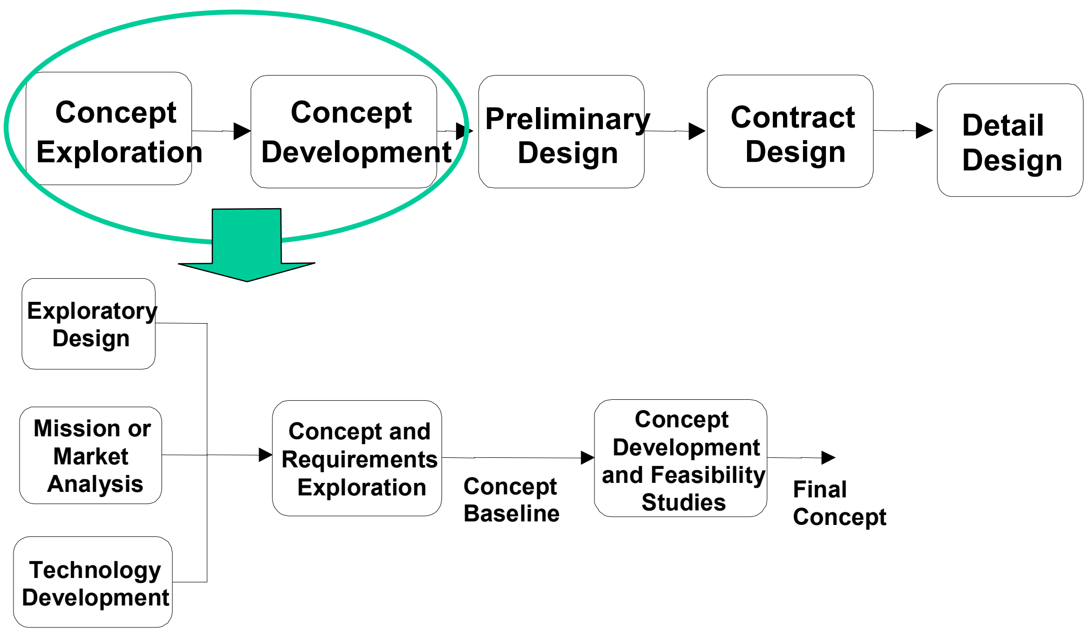

Figure 1.2.1: Concept Exploration [1]

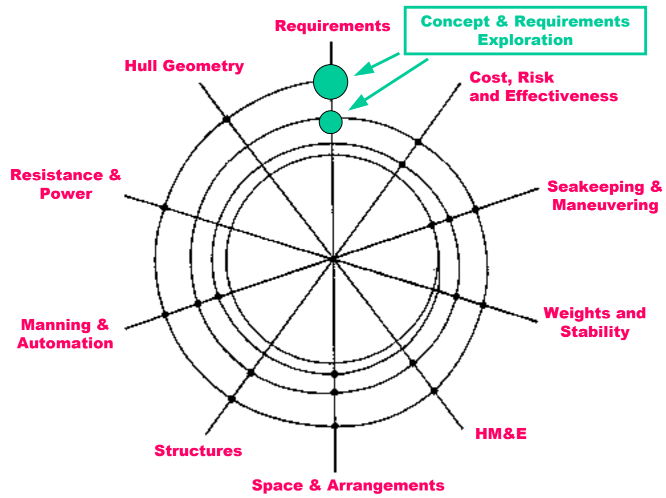

Figure 1.2.2: Design Spiral [1]

In concept exploration, the general requirements are formulated based upon the customer's need to transport oil from the Hibernia oil fields to the Whiffenhead Trans-Shipment Terminal and possibly to United States ports on the East and Gulf Coasts. Technical research is performed to identify trade-off options to fulfill the general requirements. Research is concentrated in the areas of stationkeeping, environment, structure and propulsion. Each area is given a measure of performance which are then used to calculate one overall measure of effectiveness (OMOE) for the ship. This information is then input into a genetic optimization algorithm to obtain a non-dominated frontier. A non-dominated frontier is the result of the evolutionary process performed by the genetic optimizer to maximize the cost/effectiveness. The customer then chooses an optimized design based on cost and effectiveness. The best choices are identified using the shape of the frontier. The non-dominated frontier (Figure 1.2.3) may contain a "knee," a region of sharp discontinuity within the curve. A baseline concept chosen at the top of a "knee" relates to a "best buy" design where effectiveness would be maximized for a relatively small increase in cost.

Having completed the concept exploration and chosen a specific ship from the non-dominated frontier as the concept baseline, concept development is initiated. Concept development is discussed in detail in Chapter 4.

Non-Dominated Frontier

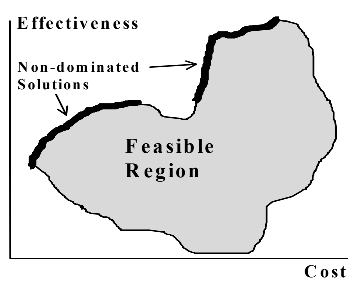

Figure 1.2.3: Non-Dominated Frontier [1]

1.3 Work Breakdown

The Virginia Tech Tanker Team is composed of five students. During concept exploration the entire team works cohesively to formulate the general requirements, OMOE and cost model. During concept development, each student specialized in various areas shown in Table 1.3.1. To facilitate organization and project management, a team leader was chosen to oversee the project.

Table 1.3.1: Work Breakdown

Name	Specialization
Korin Strome (Team Leader)	Editor/Arrangements/DPS
William Moon	Hullform/Intact Stability/Subdivision/Machinery Arrangements
John Sajdak	Hullform/Structures/Seakeeping/Manuevering/DPS
Jessica Hopper	Weights/Personnel/Cost
Andrew Quillin	Power/Propulsion/Resistance/Damage Stability

1.4 Resources

To facilitate the design process, various tools are utilized. In the concept exploration phase, the OMOE is developed in Expert Choice. Major modifications and improvements are made to an existing ship synthesis model in MathCad and then coded into a FORTRAN optimization program. Table 1.4.1 shows the software packages used in the project. The software listed in Table 1.4.1 was used solely to expedite the design process. A full understanding of basic methods and fundamental principals is gained prior to use.

Table 1.4.1: Software

Analysis	Software Package
Arrangement Drawings	AutoCAD
Hullform Development	FastShip
Hydrostatics	HecSalv
Resistance/Power	NavCADD
Ship Motions	SMP
Structures	ABS SafeHull
Ship Synthesis Model	Expert Choice, MathCad, FORTRAN 95

2 General Owners Requirements

The Shuttle Tanker is developed to transport oil from the OLS in the Hibernia Oil fields to the Whiffenhead Trans-Shipment Terminal.

2.1 Concept of Operations

The Hibernia Oil field is expected to recover a minimum of 600 million barrels (bbls) of oil during its 20 year life, which began in late 1997. Once the Hibernia Oil Field is depleted, the Shuttle Tanker can be utilized in surrounding oil fields, which are currently under development. These include: Tera Nova, which lies 25 miles east of Hibernia and is expected to recover over 300 million bbls of oil and Whiterose, which lies 25 miles northeast of Hibernia and is expected to recover 250,000 bbls of oil.

While loading at the OLS, the shuttle tanker must keep position in Sea State 6, with waves greater than five meters and winds that may exceed 27 knots. The tanker is expected to travel approximately 300 nautical miles (nm) from the OLS to Whiffenhead to offload oil. Depending on the economy and the amount of oil available, the tanker could be expected to travel as far as the Gulf Coast of the United States, which is approximately 2770nm from Hibernia. The shuttle tanker must be escorted by two tugs upon arrival to Whiffenhead. The estimated time of travel from the OLS to Whiffenhead is 20 hours traveling at 15 knots. Loading requires approximately 24 hours and offloading takes 14 hours. Figure 2.1.1 shows a timeline of the Shuttle Tanker's route.

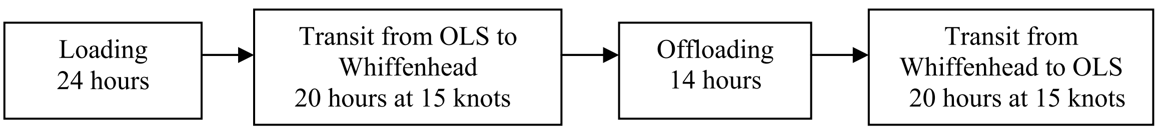

Figure 2.1.1: Timeline of Operations

2.2 Required Operational Capabilities and Projected Operational Environment

Required operational capabilities (ROC) are the minimum capabilities the tanker needs to perform its mission. These are as follows:

- Transport crude oil in incident free, year-round operation complying with International Safety Guide for Oil Tankers and Terminals (ISGOTT) and all IMO regulations.
- Provide capability of loading through submerged OLS while vessel maintains position by dynamic positioning. Systems must offload cargo alongside harbor piers, offshore facilities and lighter within the bounds of port regulations.
- Provide Inert Gas System (IGS) to minimize the risk of explosion in the cargo tanks. Provide Crude Oil Washing (COW) capabilities to remove deposits and wax buildup in cargo tanks.
- Provide precise navigation systems to minimize the risk of collision with icebergs and maximize dynamic positioning performance.

The projected operational environment for the shuttle tanker is the North Atlantic. The normal route will be from the Hibernia oil field to the Whiffenhead Trans-Shipment Terminal in Newfoundland, Canada with the capability to transport oil as far as the Gulf Coast of the United States. This ship is designed to operate in severe conditions of up to 5m significant wave height and 27 knot winds. Icebergs and visibility are also significant factors.

2.3 Goals and Thresholds

To evaluate the effectiveness of the ship, a hierarchy of performance parameters, shown in Figure 2.3.1, is created. A threshold and goal value for each measure of performance (MOP) is determined. Threshold values are absolute minimum performance requirements. Goals are either points of diminishing marginal value or technology limitations. Each performance parameter is rated on a merit index from zero to one with the threshold value representing a MOP value of zero and the goal a value of one. Table 2.3.1 shows the MOPs, their goal and threshold values and their motivation.

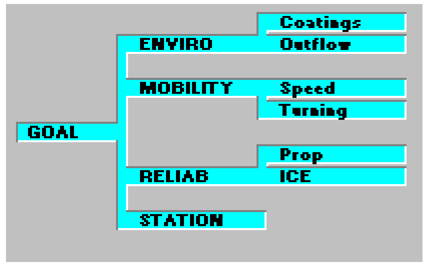

Figure 2.3.1: Effectiveness Hierarchy

Table 2.3.1: Measures of Performance

Measures of Performance	Goals	Thresholds	Motivations
Turning Radius	200 meter	Two Ship Lengths	Maneuvering Around OLS
Propulsion Reliability	Total Redundancy (two of everything)	No Redundancy	Continued Operation Despite Technical Difficulties
Speed	17 Knots	15 Knots	Delivery Time
Hull Coatings	72% solid Content, 2.12 ppm VOC's, 60 month	40% solid Content, 6.0 ppm VOC's, 24 month	Grand Banks Fishing Area environmental protection
Ice Strengthening	Bow strengthening at CAC4, Midbody strengthening at ABS 1A-	No bow or Midbody strengthening	10,000 tonne bergy bit
Mean Oil Outflow Given Accident	Mean Outflow = 0.01m^3	Mean Outflow = 0.02m^3	MARPOL Regulations
Dynamic Positioning	Integrated Power System with pods	Single Diesel with one rudder	Maneuvering Around OLS

2.4 Design Objective Attributes

2.4.1 Cost

The cost model used in the total ownership cost analysis considers the components shown in Figure 2.4.1. Cost components that did not depend on the ship design parameters were not considered in the model and are assumed to be constant for all designs. The cost model is shown in Appendix B, pg. B19.

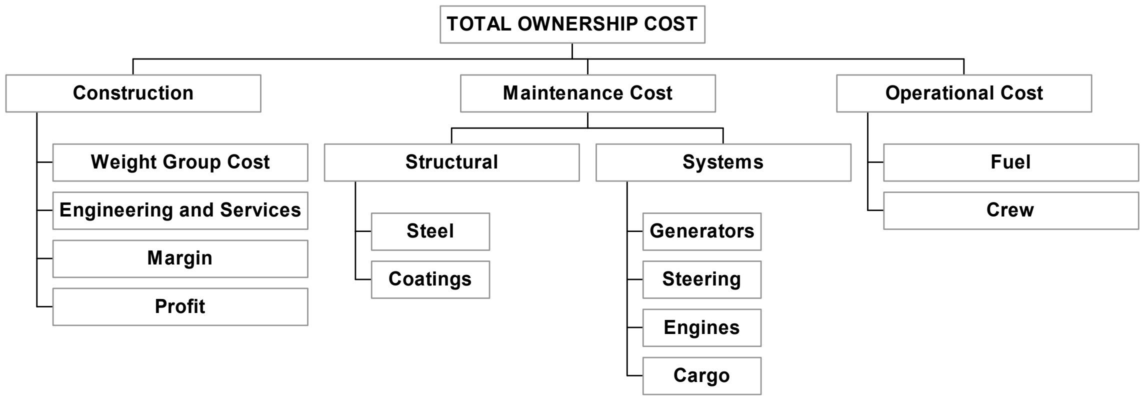

Figure 2.4.1: Cost Components [1]

The cost model calculates the total ownership cost (T_C) of the vessel by considering those costs directly associated with the design parameters and a lifetime cost estimate. To calculate the ship construction cost, the cost of each SWBS group is calculated and inflated using an 8% average inflation rate to the base year. The nine SWBS costs calculated are Structure, Propulsion, Electric, Command Control and Surveillance, Auxiliary, Outfit, Margin, Engineering and Integration, and Ship Assembly and Support. Each complexity factor, K_N, for these calculations is given in Table 2.4.1. The complexity factor is used to calculate the lead ship cost and is adjusted by calibration to recent tanker cost data.

Table 2.4.1: KN Values

Ship Component	KN Value	Choices
KN1	0.285	Mild/HT steel displacement hull
KN2	0.8 1.3	Diesel Diesel integrated power system
KN3	0.55	Conventional 60 Hz power, steam or diesel generator drive
KN4	2	Modest control systems, sophisticated electronics
KN5	0.15	Diesel propelled displacement ship
KN6	0.36	Conventional Displacement ship
KN7	2	Lead ship
KN8	2	Moderate tooling, moderate risks

Each SWBS cost is added together to produce the total construction cost and a shipyard profit of 8% of the total construction cost is added. An estimated annual cost for the vessel is calculated by incorporating a yearly fuel cost based on the three operating modes of the vessel, namely loading, offloading and in transit. Maintenance and manning cost estimates are also incorporated into the annual cost estimate. An overhaul cost is calculated based on the life expectancy of the chosen hull coating by incorporating the cost of dry-docking, painting and lost time at sea. Finally, a resale profit is calculated for the scrap value of the vessel at the end of its 30-year service life. The T_C is then calculated by using economic analysis to bring the resale profit, annual cost, overhaul cost and lead ship cost to the base year present worth.

2.4.2 Overall Measure of Effectiveness Model.

Design parameters and performance calculations within the ship synthesis model are used to calculate various measures of performance (MOP) listed in Table 2.3.1. Each MOP has a value between zero and one, that is based on the specific value of a design parameter or level of performance and its meaning relative to the ideal value for that same performance, as discussed in Section 2.3. As an example, one design parameter within the ship synthesis model is cargo tank subdivision. This design parameter, along with other parameters, influences the amount of oil outflow calculated using the MARPOL Annex I Regulations. The amount of oil outflow calculated is normalized to a value between zero and one against an allowable maximum outflow. This calculated number is the oil outflow MOP. Each MOP used within the ship synthesis model is listed in Table 2.3.1.

After MOP values are determined for each parameter, pair-wise comparisons are used to determine MOP hierarchy weights. Figure 2.3.1 shows the breakdown of performance parameters as they relate to the effectiveness goal. Figure 2.4.2 shows the resulting weighting factors used in determining the overall measure of effectiveness for the performance parameters.

Figure 2.4.2: Effectiveness Weighting

The effectiveness hierarchy has four main components. The most important operational requirement of the Hibernia Shuttle Tanker is station keeping during the loading process. Station keeping is assigned a discrete performance value determined by propulsor type. The mobility of the ship must also be considered. Speed and turning are the two main factors in mobility. Reliability is broken into two sections: propulsion and ice strengthening. The environmental aspects include hull coatings and oil outflow

The weighted sum of all the MOPs is the Overall Measure of Effectiveness (OMOE) of the ship, found in Appendix B, page B25. The following equation is the total effectiveness calculation for the ship. The coefficients in the equation are the weighting factors determined from Figures 2.4.2 and 2.4.3. The MOP values range from 0 to 1.0.

$$\text{EFFECTIVENESS} = 0.075\times MOP_{ETR} + 0.242\times MOP_{PREL} + 0.037\times MOP_{VS} + 0.036\times MOP_{COATING} + 0.08\times MOP_{ICE} + 0.18\times MOP_{OIL} + 0.350\times MOP_{DPS}$$

where:

MOP_{ETR} = Turning Radius
MOP_{PREL} = Propulsion Reliability
MOP_{VS} = Speed
$MOP_{COATING}$ = Hull Coating
MOP_{ICE} = Ice Strengthening
MOP_{OIL} = Oil Outflow
MOP_{DPS} = Dynamic Positioning System

3 Concept Exploration

3.1 Concept Exploration Model

3.1.1 Model Overview and Function

An existing tanker synthesis model is significantly altered to meet the requirements of the Shuttle Tanker concept exploration. The model is divided into various sections such as Resistance and Propulsion, Electrical Analysis, Oil Outflow and others. The complete ship synthesis model is shown in Appendix B. The principal characteristics, weight, volume, area, power and attainable speed are calculated for each possible ship. The ships are then compared to the required parameters and constraints to ensure they are balanced and are therefore feasible options. Figure 3.1.1 shows a flow chart of how the ship synthesis model balances each ship.

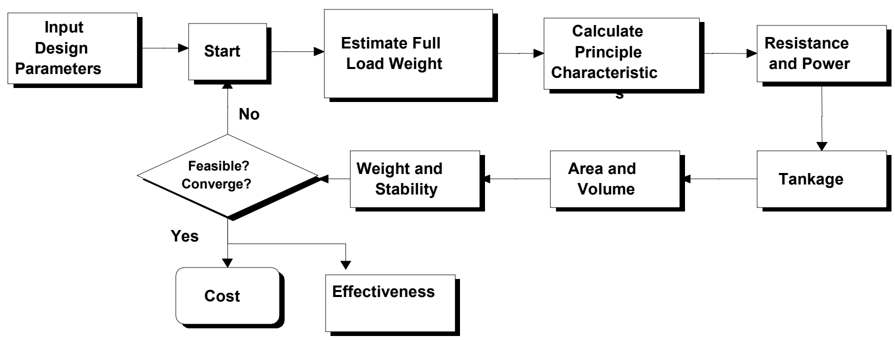

Figure 3.1.1: Flow Chart of Ship Synthesis Model

Some aspects of the ship that are not optimized are chosen based on available information from the M.T. "KOMETIK," a shuttle tanker currently servicing the Hibernia Oil Field. For instance, manning is calculated based on standard crew size for a shuttle tanker.

The eleven design parameters used to define each ship are listed in Table 3.1.1. The increments represent the number of steps analyzed between range values.

Table 3.1.1: Design Parameters

DP	Description	Range	Increments
1	Beam to Draft Ratio	2-4	40
2	Length to Beam Ratio	5-7	40
3	Block Coefficient	0.7-0.9	40
4	Depth to Draft Ratio	1.2-3.0	40
5	Height of Double Bottom	2-4	20
6	Width of Double Sides	2-4	20
7	BOW Ice Strength Selection	1-4	3
8	MIDBODY Ice Strength Selection	1-2	1
9	Hull Coating Selection	1-6	5
10	Number of Cargo Tanks	6-8	2
11	Propulsion System Type	1-9	8

3.1.2 Trade-Off Technologies and Sub-Models

3.1.2.1 Ice Strengthening

The Hibernia Oil Field on the Grand Banks of Newfoundland, Canada is filled with iceberg masses at times exceeding 10,000 tonnes. A 10,000 tonne glacial ice mass is the smallest ice mass that can be detected with standard navigational equipment in 4.6 meter significant wave heights. The maximum significant wave height of the Grand

Banks area often exceeds 4.6 meters and therefore prevention measures are taken to strengthen the hull to allow the transport of oil in a safe manner on a year round basis. Considering the requirement of “safe passage” from the Hibernia Oil Field to Canadian and U.S. Ports, the Chevron Shipping Company performed and published several analyses using the ICESHIP program. The ICESHIP program runs with the input variables shown in Table 3.1.2 for both a mid-body and bow iceberg impact using different ice strengthening levels as described in Table 3.1.3. In Table 3.1.3, the abbreviation, CAC4, refers to Canadian proposed regulations.

Table 3.1.2: ICESHIP Program Input Parameters

ICESHIP Input Variable	ICESHIP Input Value
Impact Rate	1 impact per year
Significant Wave Height	4.6 meters
Significant Wave Period	6 seconds
Ice Mass Velocity	2 knots
Tanker Length	240 meters
Tanker Depth	22.3 meters
Tanker Beam	48 meters
Tanker Capacity	895 Mbbls
Tanker Size	120,000 DWT
Full Load Draft	15 meters
Ballast Draft	9 meters
Wavelength	100 meters

Table 3.1.3: Ice Strengthening Considerations

Ice Class	Added Bow Weight	Added Bow Cost	Added Mid-body Weight	Added Mid-body Cost
	(Tons)	(U.S. $K)	(Tons)	(U.S. $K)
None	--	--	--	--
ABS 1A	50	215	800	2160
ABS 1A+	75	322.5	1175	3172.5
ABS 1A-	40	172	760	2052
CAC4	620	2666	3680	9936
CAC4-	500	2150	3550	9585
CAC4--	440	1892	3210	8667

The ICESHIP program outputs the maximum vessel speed and ice mass before shell rupture. The design variables input into the ship synthesis model are ice strengthening options for the bow and mid-body. They are represented by the variables A_{wb} and A_{wm}. Table 3.1.4 provides the simplified output of the ICESHIP program, which is the maximum vessel speed and ice mass before shell rupture. The values stricken in Table 3.1.4 are not included in the ship synthesis analysis due to their higher cost and lower resistance to impact compared to other options. Table 3.1.5 provides the output of the ICESHIP program that was considered in remaining analyses.

Table 3.1.4: ICESHIP Program Output

Classification	Added Cost	Maximum Vessel Speed	Maximum Ice Mass
	(U.S. $K)	(Knots)	(Tonnes)
Bow Strengthening			
None	--	9.7	10,000
~~ABS 1A~~	~~215~~	~~6~~	~~10,000~~
~~ABS 1A+~~	~~322.5~~	~~9~~	~~10,000~~
~~ABS 1A-~~	~~172~~	~~7.5~~	~~10,000~~
CAC4	2666	20	10,000
CAC4-	2150	13.5	10,000
CAC4--	1892	12	10,000
Mid-body Strengthening			
None	--	6	10,000
~~ABS 1A~~	~~2160~~	~~15~~	~~5,000~~
~~ABS 1A+~~	~~3172.5~~	~~17~~	~~7,000~~
ABS 1A-	2052	13	3,000
~~CAC4~~	~~9936~~	~~15~~	~~8,000~~
~~CAC4-~~	~~9585~~	~~13~~	~~5,000~~
~~CAC4--~~	~~8667~~	~~10~~	~~3,000~~

Table 3.1.5: Used ICESHIP Program Output

Classification	Added Cost	Maximum Vessel Speed	Maximum Ice Mass
	(U.S. $K)	(Knots)	(Tonnes)
Bow Strengthening			
None	--	9.7	10,000
CAC4	2666	20	10,000
CAC4-	2150	13.5	10,000
CAC4--	1892	12	10,000
Mid-body Strengthening			
None	--	6	10,000
ABS 1A-	2052	13	3,000

Analysis of the output is performed by pair wise comparison between the different ice strengthening classifications, based on their ice mass and vessel speed ratio. Each classification is given a weighting factor that is input into the overall measure of effectiveness equation as the ice MOP. The CAC4 classification for the bow is the most effective and is given the highest weighting factor. The CAC4- has the next highest weighting factor and the CAC4-- is the least effective and thus has the lowest weighting factor. The option of no ice strengthening is given a weighting factor of zero for both the bow and the mid-body. The complete ice strengthening calculations are shown in the Ice Strengthening Section, Appendix B, page B3.

3.1.2.2 Propulsion and Electrical

The design of the tanker's propulsion system and method of electrical power generation is extremely important. A poorly designed and matched propulsion plant can result in the loss of millions of dollars during the life of the ship. To avoid this, a trade-off study of different propulsion and electrical options is performed in the concept exploration phase of design.

Many tankers utilize a single slow speed diesel engine. Benefits of this system are its cost, fuel efficiency, simplicity and reliability. The single engine is directly coupled to the shaft, thus eliminating the need for reduction gears. Typically a controllable pitch propeller (CPP) is used to facilitate maneuvering and reversing. Electrical power is generated by engaging a generator connected to a power take off (PTO).

Another common propulsion system for tankers is two slow speed diesel engines coupled to two shafts and two CPPs. The main advantage of this system is the added redundancy due to two engines. This is particularly important when operating in an environment where engine failure could lead to grounding. The disadvantage to this system is the higher maintenance due to multiple engines, shafts and propellers. The electrical power is produced by two generators, one per engine, connected to two PTOs.

For ships that have high electric loads or dynamic positioning requirements, an integrated power system (IPS) is often chosen to produce power for both propulsion and electric requirements. The advantages of this system are reduced machinery box length, configuration flexibility, fewer prime movers/generators, lower maintenance, higher hydrodynamic efficiency and better maneuverability when coupled with podded propulsion systems. The disadvantages of IPS are high weight due to heavy electrical components and a high purchase cost. Medium speed diesel engines are chosen as the prime movers due to their higher power density. These are coupled to 6,600 Volt, AC generators to produce electric power. Pods are chosen as the propulsor and motor combination because of their high hydrodynamic efficiency, excellent maneuverability and crash stopping capabilities. Ship service power is taken from the main generators.

The ship synthesis model contains propulsion and electrical sub-modules. The design parameter for the propulsion system is an integer varying from one to nine. Each number represents a specific propulsion system as shown in Table 3.1.5. Design parameters 1-3 represent dual slow speed diesel engines. Parameters 4-6 represent single slow speed diesel propulsion system configurations and DPs 7-9 represent IPS systems. The original propulsion sub-module of the ship synthesis model contained only slow speed diesel engine combinations. This model was altered to include the IPS systems and specific slow speed diesel engines that have installed power ranging around an estimated ship required power. Various characteristics of the engines are input into the model including power, weight, length, width, height and specific fuel consumption.

Table 3.1.5: Propulsion and Electrical Trade-Off Options

Number	Engine Type	Engine Manufacturer	No. of Engines	MCR Each	Total MCR	Optimum RPM	Total Weight	Length	Width	Height	SFC
				kW	kW		tonnes	m	m	m	g/kWhr
1	6S50MC	MAN B&W	2	8.580E03	1.716E04	127	408.2	9.510	7.900	8.800	171
2	7S50MC	MAN B&W	2	1.001E04	2.002E04	127	462.6	10.40	7.900	8.800	171
3	8S50MC	MAN B&W	2	1.144E04	2.288E04	127	522.5	11.29	7.900	8.800	171
4	6L70MC	MAN B&W	1	1.698E04	1.698E04	108	476.2	11.50	9.684	10.85	174
5	7L70MC	MAN B&W	1	1.981E04	1.981E04	108	537.0	12.75	9.684	10.85	174
6	8L70MC	MAN B&W	1	2.264E04	2.264E04	108	605.0	13.99	9.684	10.85	174
7	12ZA40S	SULZER	2	9.000E03	1.800E04	514	204.0	9.650	5.464	4.185	183
8	14ZA40S	SULZER	2	1.050E04	2.100E04	514	238.0	10.61	6.190	4.185	183
9	16ZA40S	SULZER	2	1.200E04	2.400E04	514	264.0	11.39	6.190	4.185	183

All engine powers in the table are given at maximum continuous rating (MCR). A propulsion margin factor (PMF) is used to allow for added resistance due to heavy weather and marine fouling of the hull. A PMF of 0.9 was used for the sustained speed calculations. The mechanical efficiency of the slow speed diesel arrangement is assumed to be 98% and the mechanical efficiency of the IPS is assumed to be 93%. The detailed analysis can be seen in the Machinery section, Appendix B, Page B2.

To perform the electrical analysis the original electrical sub-module is altered to include calculations for IPS system alternatives, tunnel thruster power, dynamic positioning loads and three operating conditions. The three operating conditions are cargo loading/station keeping, cargo offloading and transit. The calculations are divided into cargo and non-cargo electrical loads. The purpose of the electrical analysis is to size the diesels and PTO generators and determine the fuel consumption for the three operating conditions. The electrical requirements of the ship are calculated based on principal characteristics and installed power. If the optimizer chooses an IPS system, no PTO generator is used. Tunnel thruster power is added to the load calculation of the PTO to account for thruster use during dynamic positioning while loading cargo.

For ship service power, electrical energy must be converted to the appropriate frequency and voltage. Traditionally, motor generator sets are used for this purpose. They have an efficiency of approximately 80%. With IPS, efficiency levels of 90% can be achieved using modern frequency and voltage converters. The electrical calculations can be seen in the Electrical Section, Appendix B, Page B8.

3.1.2.3 Dynamic Positioning

Dynamic positioning is a mandatory requirement for the ship. The tanker must be able to dynamically position within a 50m circle while loading cargo from the OLS buoy in harsh environmental conditions. A series of equations was developed to evaluate various dynamic positioning systems in a worst case scenario to determine how far the ship could drift in a sea with significant wave heights of 5.5m. This worst case scenario involved the ship oriented beam to the seas and only able to move in the sway direction. To determine the effectiveness of the dynamic positioning system, the forces from the wind and seas are calculated based on sail and submerged transverse area. Next, time required for the DPS to reach full power and generate enough thrust to overcome the wind and sea forces and stop drifting is computed. Thus, the inputs to the equations are the specific characteristics of the ship and dynamic positioning equipment and the output is the total drift distance. Therefore the dynamic positioning characteristics influence the choice of propulsion systems.

It was determined that this approach was too detailed for the concept exploration and development stages of the design process so a different approach was taken. Expert opinion was ascertained to perform a pair-wise comparison between the different propulsion systems considered on the basis of their dynamic positioning capabilities. Each propulsion system is given a weighting factor that is input into the OMOE equation. The integrated power system with pods, which can rotate 360 degrees, is the most effective and given the highest weighting factor. The one propeller, one slow speed diesel system is the least effective and thus has the lowest weighting factor.

3.1.2.4 Environmental

Due to the sensitive Grand Banks area, four environmental trade-off factors are considered; hull coatings, air pollution, acoustic pollution and ballast water exchange. Currently, there are no regulations on antifoulants. However, tri-butyl tin (TBT) is in the process of being banned for use as a hull coating so only TBT free antifoulants are considered. Six coatings are chosen and the solid content, fouling rate, cost and amount of volatile organic compounds (VOC) for each coating are included in the ship synthesis model for a trade-off analysis.

Air pollution was considered, however, there were not significant differences in the diesel options considered. It was decided to omit the air pollution parameter from the ship synthesis model.

The acoustic signature of the Shuttle Tanker was also considered. Research has shown that acoustic noise from ships has been linked to the beaching of whales [2]. However, after further research and lack of data, the team discovered that acoustic noise from just one ship was not enough to cause this phenomena and the design parameter was abandoned.

The ballast water exchange system of the Shuttle Tanker was the final environmental trade-off considered. Currently, the International Maritime Organization (IMO) has a voluntary exchange policy to minimize the transfer of harmful organisms, however it is likely that this voluntary policy will soon be mandatory. The policy states that ships traveling through various bodies of water should exchange or otherwise cleanse ballast water before entering destination ports. After finalizing the route of the Shuttle Tanker, this policy was no longer applicable to the ship. The primary route of the Shuttle Tanker covers a relatively short span of the North Atlantic Ocean and therefore there is no need to exchange ballast water to prevent the transfer of harmful organisms. In addition, due to the highly subdivided ballast tanks, ballast water can be exchanged by emptying and refilling each tank individually. This exchanges nearly 100% of the ballast water while not adversely affecting stability.

3.1.2.5 Resistance

Resistance is calculated using the Holtrop and Mennon Method and can be found in the Resistance and Power section, Appendix B, page B5. Viscous drag is calculated using the 1957 International Towing Tank Conference (ITTC) method. A residuary drag coefficient is found based on wave making drag for a hull with a bulbous bow for various beam to draft ratios. The viscous resistance and residual resistance are then used to find the bare hull resistance. Three different propulsor types are considered and an appendage drag is estimated for each system and added to the bare hull resistance. This is accomplished by approximating the additional drag as a percentage of the bare hull resistance.

3.1.2.6 Machinery Box

It is necessary that the main propulsion engines fit in the machinery box and that enough volume is available for the various systems. In the original synthesis model the required length, width, height and volume of the machinery box were calculated in the machinery section of the ship synthesis model. Width and height were taken directly from the engine size while a constant shaft length was added to the length of the engines to determine required machinery box length. The available machinery box dimensions were calculated assuming a parallel midbody machinery box such as on a destroyer. This is inaccurate for the tanker application and a better approach was deemed necessary. Due to the geometric complexity of the stern section of displacement hull forms, an approach based solely on the machinery box length is used to ensure fit of the main engines in the machinery box. In a wide and full tanker hull, volume and height are less critical constraints.

Required width and height are calculated based on engine dimensions. These calculations are shown in the Machinery Box section, Appendix B, page B11. Length is calculated using a triangular floor plan to fit the engine(s) footprint. Slow speed diesel engines are configured lengthwise for connection to the shafts. The forward effective breadth of the engine floor plan, or base of the triangle, was determined by assuming the engines are placed on the lowest level of the machinery box. The ($0.67 \times C_B \times B$) part of the following equation corresponds to this level. The below equation is developed by studying the machinery room arrangements of tankers and deducing a correlation between the beam and the block coefficient.

$$B_e := \text{if}\left[PSYS_{TYP} \geq 7, \left(0.91 \cdot C_B \cdot B\right), \left(0.67 \cdot C_B \cdot B\right)\right]$$

The required length is determined by connecting points A and B in the below footprint figures with a straight line and calculating the bisector length of the triangle. See Figures 3.1.2 and 3.1.3 for twin and single slow speed engine arrangements.

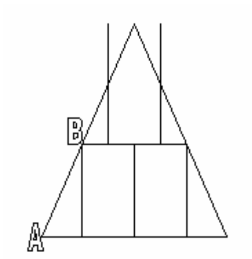

Figure 3.1.2: Two Slow Speed Diesel Engine Arrangement

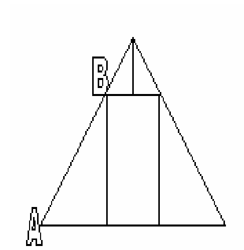

Figure 3.1.3: Single Slow Speed Diesel Engine Arrangement

The same calculation is used for an IPS arrangement except the engines are arranged sideways and the forward width of the engine floor plan, or base of the triangle, is determined by the ($0.91 * C_B * B$) part of the following equation.

$$B_e := \text{if}\left[PSYS_{TYP} \geq 7, \left(0.91 \cdot C_B \cdot B\right), \left(0.67 \cdot C_B \cdot B\right)\right]$$

A simple picture of the IPS engine arrangement is given in Figure 3.1.4.

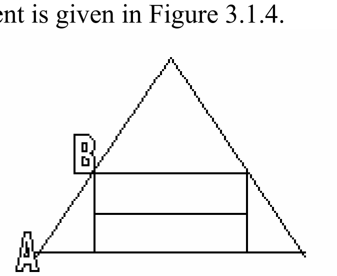

Figure 3.1.4: IPS Diesel Engine Arrangement

These calculations can be seen in the Machinery section, Appendix B, Page B3. The available machinery box length, L_{MB}, is calculated by what is left over in all ship length calculations. The following equation comes from the regression of a number of similar tanker designs and can be seen in the Machinery Box section, Appendix B, Page B11.

$$L_{MB} := LWL - 0.05 \cdot LWL - L_{CB} - 3\ \text{m} - 0.062 \cdot LWL$$

The available height of the machinery box is set equal to the depth of the hull. For feasibility, the required machinery box length and breadth must be less than available.

3.1.2.7 Weights

The original weight sub-module calculates ship weight by SWBS groups. Each section contains equations to calculate the SWBS group weight based on principal characteristics and propulsion plant data. Modifications to the original sub-module are made to include the IPS and to improve the accuracy of the model. The largest modification to the ship weight analysis is the bare hull weight. Originally it was a simple equation based on principal characteristics, but this proved inaccurate away from the 150k dwt calibration point. Due to the lack of regression curves and a detailed midships section at this stage in the design process, American Bureau of Shipping (ABS) rules are used to calculate the minimum scantlings of all major components in the midship section. The equations taken from ABS are based on principal ship characteristics. This allows for a simple minimum thickness check at the initial concept design phase.

The double hull tanker design is modeled as a rectangular box with no camber, zero bilge radius and no hoppers or stools. 1995 ABS Steel Vessels, Hull Construction and Equipment (Part 3) [3] and 1992 ABS Steel Vessels, Specialized Vessels and Services (Part 5) [4] are used to calculate minimum scantlings.

Inner and outer bottom, inner and outer side shell, horizontal stringers, longitudinal bottom girders, transverse webs, centerline bulkhead and transverse bulkhead minimum scantlings are calculated. The minimum calculated plate thickness is multiplied by a smearing ratio to account for stiffener material. These values are calculated using the standard IMO double hull 150,000 DWT tanker. A corrosion allowance is then added to the smeared plate thickness that is determined from 1995 ABS Steel Vessels, Specialized Vessels and Services (Part 5) [5]. These values range from 1mm to 2mm for various parts of the hull plating.

Longitudinal steel volume per unit length is calculated for each plate section and multiplied by the density of steel to give the weight per unit length. Upper and lower transverse web volume and weight are also calculated per unit length and added to the longitudinal weight to give a total weight per unit length. A weight distribution equation based on midship weight per unit length is used to extrapolate this weight to a full bare hull weight minus transverse structure [6]. The equation shown below and other bare hull weight calculations can be seen in the Weight section / SWBS 100, Appendix B, Page B12.

$$W_s = (0.715 C_B + 0.305) L W_{pm}$$

where:

W_s = weight of structure for full ship minus transverse structure
C_B = block coefficient
L = length between perpendiculars
W_{pm} = weight per unit length of structure

Transverse bulkheads are given the same thickness as the inner side shell. Transverse bulkhead weight is calculated based on plate thickness, an assigned smearing ratio, and principal characteristics. This is added to the longitudinal bare hull weight to give a total bare hull weight. A correlation factor of 1.17 is used to account for brackets and other miscellaneous structure.

To account for the significant weight difference of an IPS compared to a traditional slow speed diesel engine arrangement, the following equation was added to calculate the basic machinery weight of an IPS. [6]

$$W_{BM} := \mathrm{if}\left[PSYS_{TYP} \geq 7, \frac{KN_{200} \cdot 0.72\,MT}{kW^{0.78}} \cdot \left(P_I\right)^{0.78}, W_{BM}\right]$$

The KN_{200} is used as an adjustment factor to account for improvements in technology such as solid-state electronics. This equation can be found in the Weight section / SWBS 200, Appendix A, Page A11.

When using an IPS, shafts are not needed due to the podded propulsion. Therefore, shaft length is set to zero and shaft weight is also zero. Shaft length was determined using the geometry of the engine room. The length of the shaft is the length of the machinery box minus the length of the engine(s). Weight of the pods for propulsion with an IPS is included in the propulsor section of the weight sub-module.

The weight of the electrical generating equipment is calculated in SWBS 300 using simple regression curves to calculate the weight of diesel generating sets and PTO generators. Additional weight is added for electrical cable and equipment for ship service power alteration and filtering. At the end of the weight section, all SWBS groups are summed to give a lightship weight. This weight is added to the deadweight tonnage to give the full load weight.

The weight portion of the ship synthesis model is validated using two specific ships; the M.T. “KOMETIK”, a Hibernia Oil Field shuttle tanker, and the M.T. “POLAR ENDEAVOR”, a Trans-Alaskan Pipeline oil tanker. The principal characteristics of the “KOMETIK” were entered into the model and the lightship weight matched almost exactly with the weight calculated in the ship synthesis model. The full load weight was approximately 1.6% overweight. The “POLAR ENDEAVOR” also produced a close match.

3.1.2.8 Oil Outflow

The oil outflow was estimated using a simplified MARPOL Annex I Regulation method in both side and bottom damage cases assuming the occurrence of an accident. Calculations given in the Oil Outflow section, Appendix B, page B21 consider the size of the cargo and slop tanks, the pressure within the tanks, the tidal draft and the oil captured within the ballast tanks. The outflow for both a grounding and collision are multiplied by a probability factor of occurrence and summed into a total outflow amount.

3.1.3 Concept Design Feasibility

For a design to be feasible, it must meet both the thresholds and constraints. Thresholds are design parameters that are optimized such as speed, volume and power (Table 3.1.6). These parameters can range from minimum values (thresholds) to maximum values (goals) and be considered feasible. The closer a parameter is to the goal value, the higher weighting factor it receives, which increases the design's effectiveness. A constraint is a design parameter that is not optimized (Table 3.1.7), such as station keeping and maneuverability. The design either meets the criteria or does not, no increase in effectiveness occurs if the parameter exceeds the constraint. For example, our ship must be able to dynamically position and keep the bow within 50m of the OLS in Sea State 6. This is a constraint, meaning the ship must meet this criteria but if it can dynamically position and keep the bow within 30m of the OLS in Sea State 6 no extra effectiveness is gained. If the design meets the criteria it is feasible, if the design does not meet the criteria it is not feasible and is not included in the non-dominated frontier. The design is considered unfeasible if even one constraint or threshold is not met. A check is performed at the end of the optimization process in the Design Balance/Summary section in Appendix B, page B18, where all available parameters are checked against ship requirements to ensure feasibility.

Table 3.1.6: Design Parameter Thresholds

Design Parameters	Threshold	Goal
Height of Double Bottom	2	4
Width of Double Sides	2	4
BOW Ice Strength Selection	None	400 MT Added Weight
MIDBODY Ice Strength Selection	None	690 MT Added Weight
Hull Coating Selection	None	60 Month Lifetime Minimize VOC
Number of Cargo Tanks	6x2	8x2
Propulsion System Type	Single Shaft Slow Speed Diesel	Redundant Podded Propulsion

Table 3.1.7: Design Parameter Constraints

Design Parameters	Constraints
DPS	Stay within a maximum 50 meter radius in Sea State 6
Dead Weight Tonnage	Minimum 127,000 MT
Personnel	Maximum of 28 crew members

3.2 Multi-Objective Optimization

3.2.1 Pareto Genetic Algorithm (PGA) Overview and Function

A Pareto Genetic Algorithm (PGA) is used to optimize the ships within the design space. Using the design parameters shown in the Input: Design Parameters section, Appendix B, page B1, the optimizer randomly creates 200 balanced ships. The ships are compared to one another based on cost and effectiveness and penalized for infeasibility and niching, which is a cluster within the design space. The optimizer prefers even spacing of points along the non-dominated frontier to allow a greater spectrum of ships and thus penalizes ships that bunch-up in the design space. From the initial population the optimizer randomly chooses a second generation of designs. The probability of particular designs being chosen for the second generation depends on the design's cost versus effectiveness. Of this second generation, twenty-five percent are selected for cross-over of design parameters. A small percentage of randomly selected design variables are replaced with new random design variables. As each generation is created, the ships are spread across a cost and effectiveness frontier. The optimizer runs for 100 generations, after which a non-dominated frontier can be defined and used to select a ship. Figure 3.2.1 shows the optimization results for the Shuttle Tanker. The non-dominated frontier consists of ships that represent the highest effectiveness associated with the lowest costs.

3.2.2 Optimization Results

The non-dominated frontier for the feasible shuttle tanker designs is shown in Figure 3.2.1 Overall Measure Of Effectiveness (OMOE) is plotted against the Total Ownership Cost (TOC) for generations 1, 30, 80 and 100. As the generations progress, the design space is explored and optimized for the highest effectiveness and the lowest cost. The OMOE of the feasible designs ranges from 0.369 to 0.963 and from 196.277 to 241.254 millions of dollars for the TOC. Four ships are chosen based on a high ratio of effectiveness to cost where "knees" occur in the non dominated frontier. These four options are Low, Best Buy Low, Best Buy High and High.

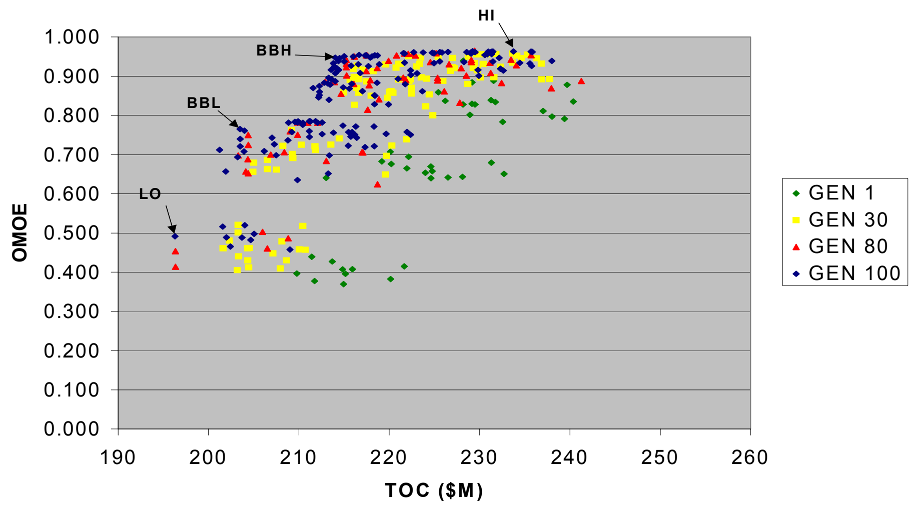

Figure 3.2.1: Shuttle Tanker Non-Dominated Frontier of Feasible Designs

The three bands of feasible ships shown in Figure 3.2.1 represent the three types of propulsion system options. The top band represents the ships with an integrated power system with pods. The middle band shows the twin screw ships and the lower band represents the single screw ships. A weight of 35% was given to the stationkeeping performance, the largest of any single MOP because of the necessity to dynamically position the ship during loading. The three bands represent IPS with pods, which receives a full 100% for stationkeeping performance, twin screw and single screw designs.

The objective of the optimization is to maximize effectiveness vs. cost while satisfying all constraints and thresholds. Table 3.2.1 describes the four chosen ships and the M.T. "KOMETIK", a shuttle tanker currently servicing the Hibernia Oil Field. A very large driver for the TOC is the propulsion plant. In order to drive cost down the designs are optimized for the lowest possible resistance. This allows the optimizer to choose the smaller propulsion plant while maintaining full effectiveness for the endurance speed. This trend can be seen as TOC decreases. In order to reduce resistance, block coefficient is reduced, draft is reduced, breadth is increased and length is optimized for the best speed to length ratio.

Increasing the freeboard gives a high hydrostatic head in the event of grounding. Thus the largest possible double bottom height is chosen for the high freeboard ships to increase effectiveness. Having a lower freeboard reduces this effect and a smaller double bottom height is chosen while still maintaining full effectiveness for oil outflow in a grounding situation.

The width of the double sides in all optimized designs is four meters due to the criteria used to analyze oil outflow in the event of a collision. All oil is assumed to outflow and thus to gain effectiveness each design chose the largest possible double side width, being four meters.

A minimum endurance speed of 15 knots was met in all four ship designs with additional speed given in the HI design due to the larger propulsion plant. It is clear that the additional speed gives a slightly higher effectiveness but at a very high cost.

Additional effectiveness is given to the HI design for having an 8 x 2 cargo block arrangement. This gives the highest effectiveness possible in the oil outflow calculations but also at a very high cost. The three low-end ship designs all have a 6 x 2 cargo block arrangement.

Table 3.2.1: Ship Characteristics

Parameter	HI	BBH	BBL	LO	KOMETIK
DP1 – Beam to Draft Ratio	3.15	3.4	3.45	3.3	3
DP2 – Length to Breadth Ratio	5	5	5.05	5	5.6
DP3 – Block Coefficient	0.805	0.82	0.76	0.755	0.825
DP4 – Depth of Hull	1.52	1.68	1.66	1.62	1.42
DP5 – Height of double bottom [m]	2.1	3.9	4	4	2
DP6 – Width of double side [m]	4	4	4	4	2
DP7 – Bow Ice Strengthening	2	2	2	2	0
DP8 – Midship Ice Strengthening	2	2	2	2	0
DP9 – Hull Coating	6	6	6	5	3
DP10 – Cargo Tank Subdivision	8	6	6	6	6
DP11 – Propulsion System Type	9	8	2	5	2
LBP [m]	247.33	252.77	261.67	255.98	256.83
Beam [m]	49.47	50.55	51.82	51.2	45.86
Draft [m]	15.7	14.87	15.02	15.51	15.29
D10 [m]	23.87	24.98	24.93	25.13	21.71
Prismatic Coefficient	0.809	0.824	0.764	0.759	0.829
Midship Section Coefficient	0.995	0.995	0.995	0.995	0.995
Number of Prime Movers	2	2	2	1	2
Lightweight [MT]	31645	32832	31759	30465	26490
Full load displacement	158645	159832	158759	157465	153490
FL Vertical CG [m]	13.698	14.118	15.069	15.17	12.656
Cargo [MT]	125861	125920	125934	125974	125936
Sustained speed [knt]	15.74	15	15.01	15.03	15.02
Lead Ship BCC [$M]	159.2	144.2	135.5	132	153
TOC [$M]	233.7	210.9	203.5	196.3	221.1
Manning	28	28	28	24	28
Mean Oil Outflow / Capacity (Om/C)	0.0096	0.0105	0.011	0.0105	0.0158
OMOE	0.9632	0.9473	0.765	0.4913	0.6649

3.3 Baseline Concept Design

The ship chosen for the baseline design from the non-dominated frontier is the best buy high ship (BBH). The ship is chosen because it lies at the top of a "knee" in the non-dominated frontier. A "knee" in the curve is a sharp discontinuity and at the top of the "knee" a large increase in effectiveness is attained with a minimal rise in cost. The overall characteristics of the best buy high ship are shown in Table 3.2.1. The principal characteristics of the ship are shown in Table 3.3.1. This ship has the shallowest draft and the highest block coefficient of the four optimized ships. The ship utilizes a mid-size integrated power system that produces 28,162 Hp and has a sustained speed of 15.01 knots. The Shuttle Tanker has a 28 member crew and has a full load weight of 159,832 MT. The ship has 6×2 cargo block arrangement. The overall measure of effectiveness (OMOE) of the ship is 0.9472 out of 1.000.

Table 3.3.1: Baseline Design Characteristics

Characteristics	Baseline Value
LBP [m]	252.77
Beam [m]	50.55
Draft [m]	14.87
Cp	0.8240
Cx	0.9950
Lightweight [MT]	32832
Full load displacement	159832
FL Vertical CG [m]	14.12
Cargo [MT]	125900
Sustained speed [knt]	15.00
Lead Ship BCC [$M]	144.2
TOC [$M]	210.9
Manning	28
Number of Cargo Divisions	6 x 2
OMOE	0.9473

4 Concept Development

The concept exploration phase of the design is complete and now the second loop in the design spiral, concept development, begins. Chapter 4 details the feasibility study performed for the baseline concept design Shuttle Tanker.

4.1 Hullform and Hydrostatics

The concept design hull form is created using the FastShip software program. A parent 70,000 DWT tanker is chosen by selecting it from the Hull Library in FastShip. Adjustments to the length, beam and depth are made to the hull prior to importing it to the FastShip modeling space. This modified hull is shown in Figure 4.1.1.

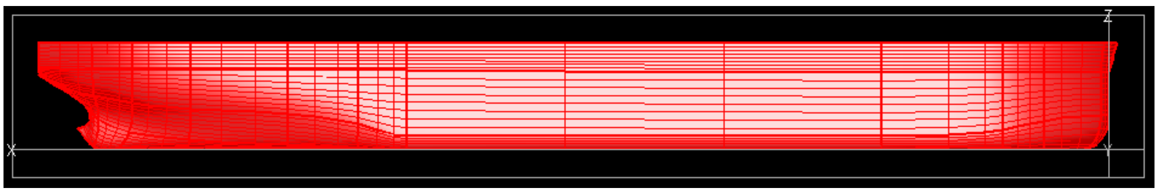

Figure 4.1.1: Hull Library 70,000 DWT Tanker

With the new hull form in the modeling space, the midship coefficient is increased slightly to the value of 0.995 by creating a smaller bilge radius in the parallel midbody. Due to the integrated power system and podded propulsion, a standard tanker stern is not applicable. Using expert opinion and various examples of ships with podded propulsion, a ramped stern section is created by pulling the net in FastShip. The net controls the shape of the hull. The stern section is shown in Figures 4.1.2 and 4.1.3. The ramped stern section increases producibility and maximizes the efficiency of flow into the propellers. By keeping the beam large in the aft section of the ship, greater volume is given to the machinery box and producibility is increased. The stern is brought to 1.04 meters below the design waterline at the transom and the flat section extends approximately 50 meters forward of the transom. The ship has an aft prismatic coefficient of 0.696.

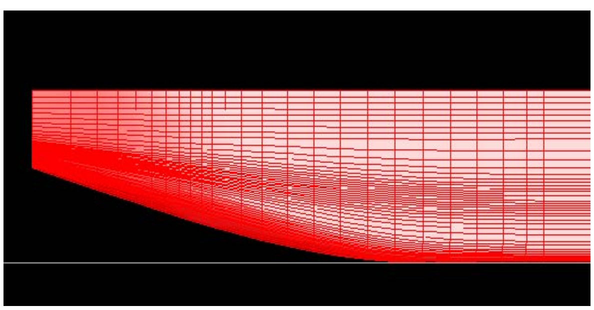

Figure 4.1.2: Concept design stern section, profile view

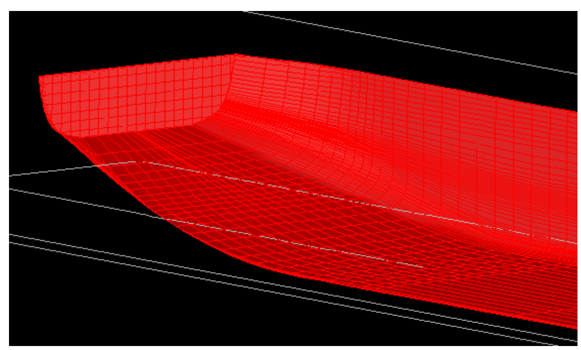

Figure 4.1.3: Concept design stern section, isometric view

The hull form imported from the Hull Library of FastShip does not have a bulbous bow. The main purpose of adding a bulbous bow is to reduce wavemaking resistance at the design speed in full load and ballast conditions. The height of the bulb center and transverse profile area are calculated in the Wave Making Drag part of the Resistance and Power section, Appendix B, page B5. These are used as a baseline bulb design and are fine-tuned later. Bulbous bow design parameters are calculated using the paper, "Design of Bulbous Bows", by Alfred M. Kracht [7].

Three bulbous bows are presented in the paper, Δ-type, O-type and ∇-type. These are shown in Figure 4.1.4. The ∇-type is chosen due to its common use in industry and its favorable seakeeping properties. The Δ-type was not chosen due to its high slamming characteristics in large sea-states. The suggested bulbous bow profile is shown in Figure 4.1.5. The top bulb height (H_B shown in Figure 4.1.5) is set equal to an initial ballast draft estimate of 10.225 meters to ensure decreased resistance in full load and ballast conditions.

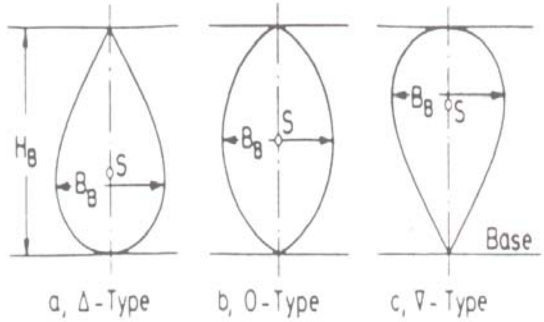

Figure 4.1.4: Bulbous Bow Types [7]

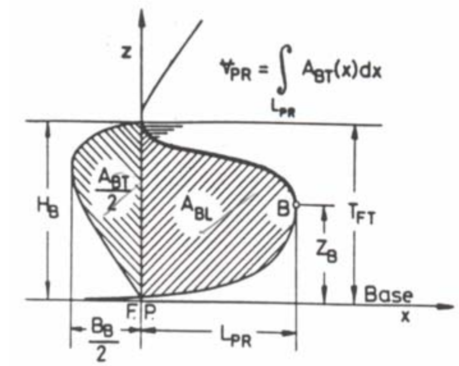

Figure 4.1.5: Bulbous Bow Profile [7]

The following equations and parameters are used to describe the dimensions and shape of the bulbous bow shown in figure 4.1.5.

$B_B = C_{BB} \times B_{MS}$ where C_{BB} is the breadth parameter chosen to be 0.11 and B_{MS} is the breadth at midships, giving a bulb breadth at the forward perpendicular (FP) of 5.56 meters.

$L_{PR} = C_{LPR} \times L_{PP}$ where C_{LPR} is the length parameter chosen to be 0.02 and L_{PP} is the length between perpendiculars, giving a maximum bulb length from the FP of 5.05 meters.

$Z_B = C_{ZB} \times T_{FP}$ where C_{ZB} is the depth parameter chosen to be 0.6 and T_{FP} is the draft at the forward perpendicular, giving a bulb depth of 8.92 meters.

$A_{BT} = C_{ABT} \times A_{MS}$ where C_{ABT} is the cross-section parameter chosen to be 0.05 and A_{MS} is the midship area, giving a bulb cross-sectional area of 37.40 meters^2.

$A_{BL} = C_{ABL} \times A_{MS}$ where C_{ABL} is the lateral parameter chosen to be 0.06 and A_{MS} is the midship area, giving a bulb lateral sectional area of 44.87 meters^2.

Design lanes based on block coefficient and Froude number given in the paper determine the "C" coefficients in the above equations. The final concept bulb section is shown in Figures 4.1.6, 4.1.7 and 4.1.8.

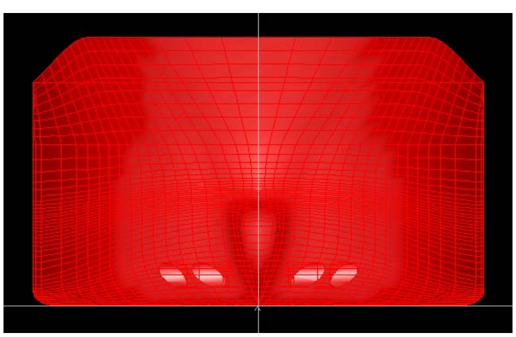

Figure 4.1.6: Concept Bulb and Forecastle Design

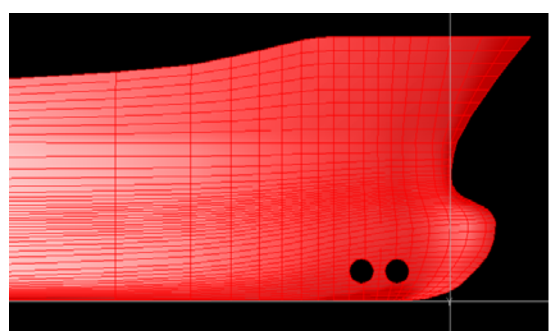

Figure 4.1.7: Concept Bulb and Forecastle Design

A forecastle is added to the concept hullform to create a dryer working area in heavy seas and to provide protection for the bow loading system. The forecastle is pulled 9.0 meters forward of the forward perpendicular, approximately 4 meters beyond the forward tip of the bulbous bow. This is done to reduce interference with the bulbous bow during bow loading. The forecastle is shown in Figures 4.1.6, 4.1.7 and 4.1.8.

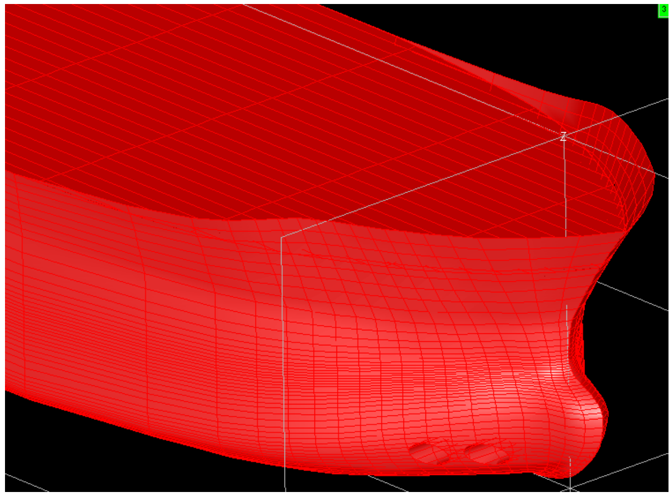

Figure 4.1.8: Forecastle and Bulb Isometric View on Concept Design

Two bow tunnel thrusters are added to the hullform. A 2.8 meter diameter tunnel is chosen for both thrusters with an approximate power rating of 2100 kW. Bow thruster placement is chosen based on available width and accessibility, while keeping them forward of the collision bulkhead. The placement of the bow tunnel thrusters is shown in Figure 4.1.7.

Final additions to the hullform in FastShip are a flat deck and vertical transom. A deck with zero camber is chosen to increase producibility and is located at the design depth of 24.98 meters. The vertical transom is located at the design waterline length of 252.77 meters. The finished concept hullform is shown in Figures 4.1.9, 4.1.10 and 4.1.11. The lines drawing can be seen in the attached Drawing 1.

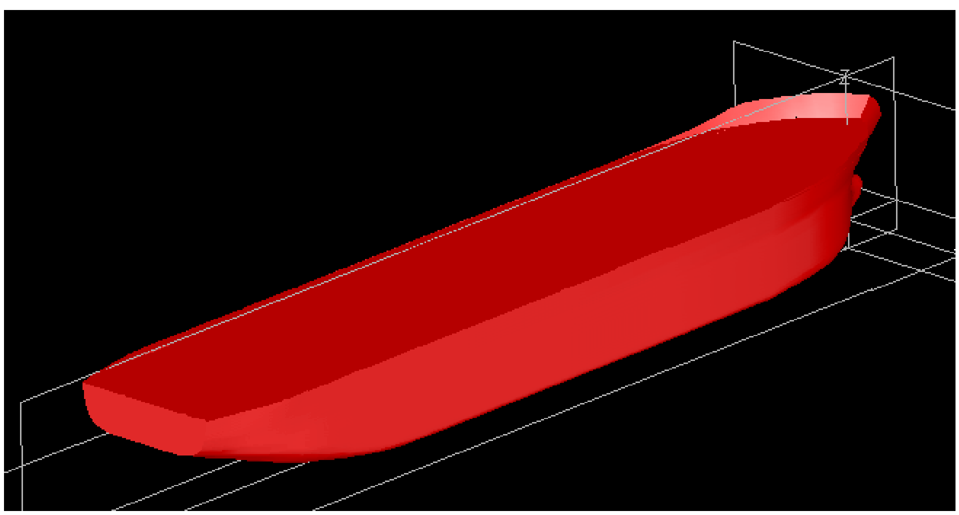

Figure 4.1.9: Concept Hullform with Deck, Transom, and Bow Thrusters

Figure 4.1.10: Concept Hullform with Deck, Transom, and Bow Thrusters

Figure 4.1.11: Concept Hullform with Deck, Transom, and Bow Thrusters

After the hullform is completed, a hydrostatic calculation is performed. Table 4.1.1 lists the principal characteristics of the final concept hullform and Figure 4.1.12 is the curves of form. Figure 4.1.13 is the cross curves for the final concept hullform and Figure 4.1.14 is the Bonjean curves. A table of molded offsets is provided in Appendix C.

Table 4.1.1: Hydrostatic Results for the Final Concept Hullform

Principal Characteristics	
LWL	252.8 m
LOA	261.8 m
Depth (molded)	24.98 m
Beam	50.55 m
Design Draft	14.87 m
Cb	0.82
Cp	0.824
Aft Cp	0.695
Forward Cp	0.887
Cm	0.995
Cwp	0.911
Wetted Surface	18,370 m^3
Displacement	159,700 tonnes

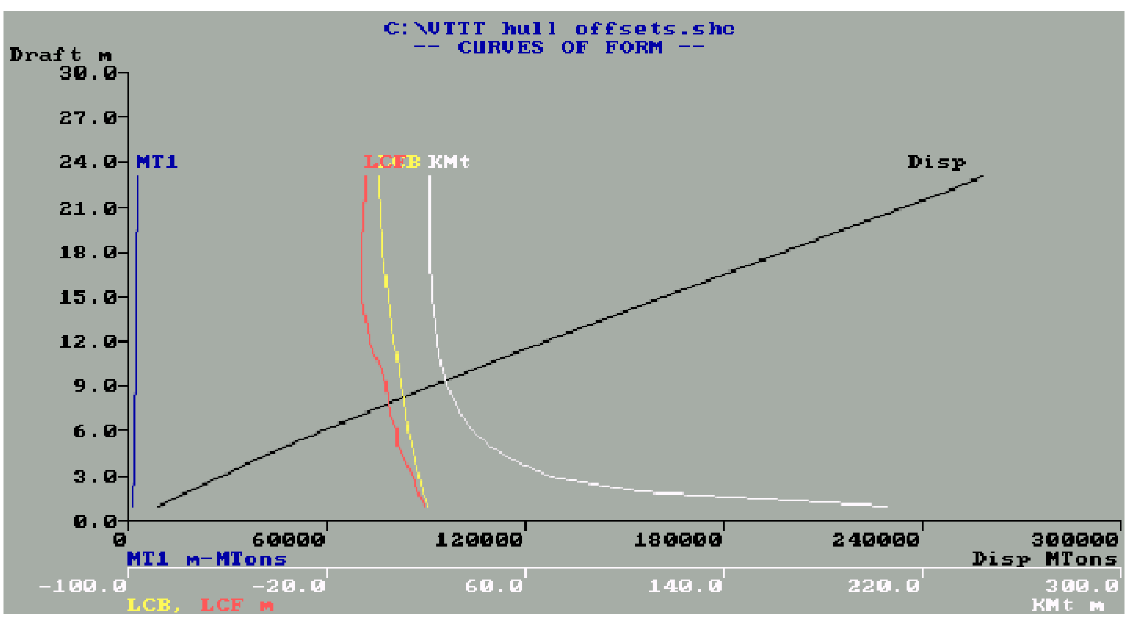

Figure 4.1.12: Curves of Form for the Concept Design

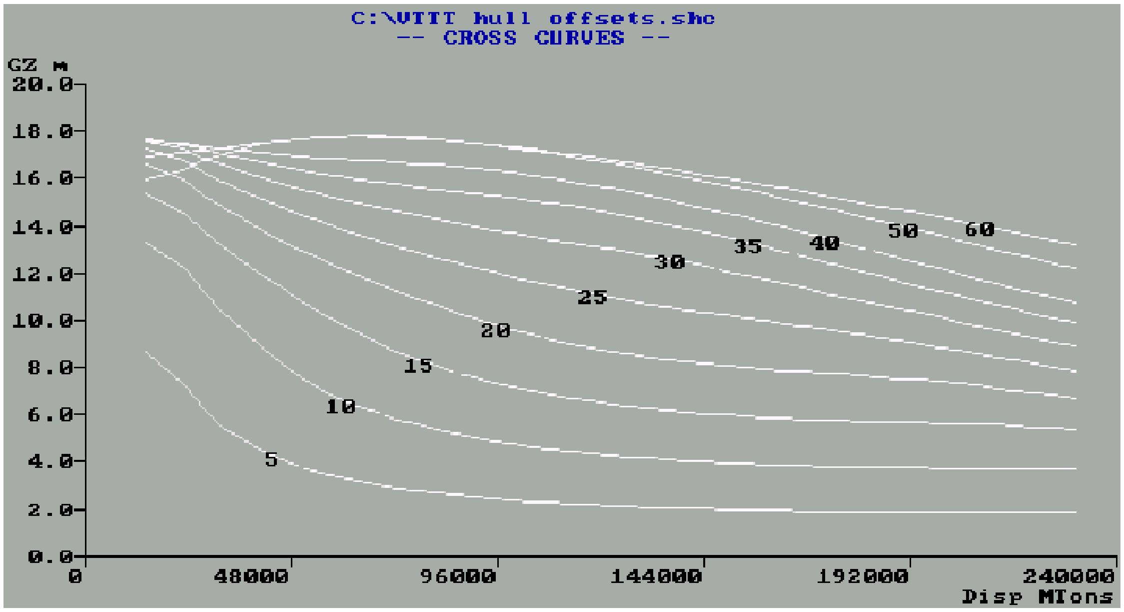

Figure 4.1.13: Cross Curves for the Concept Design

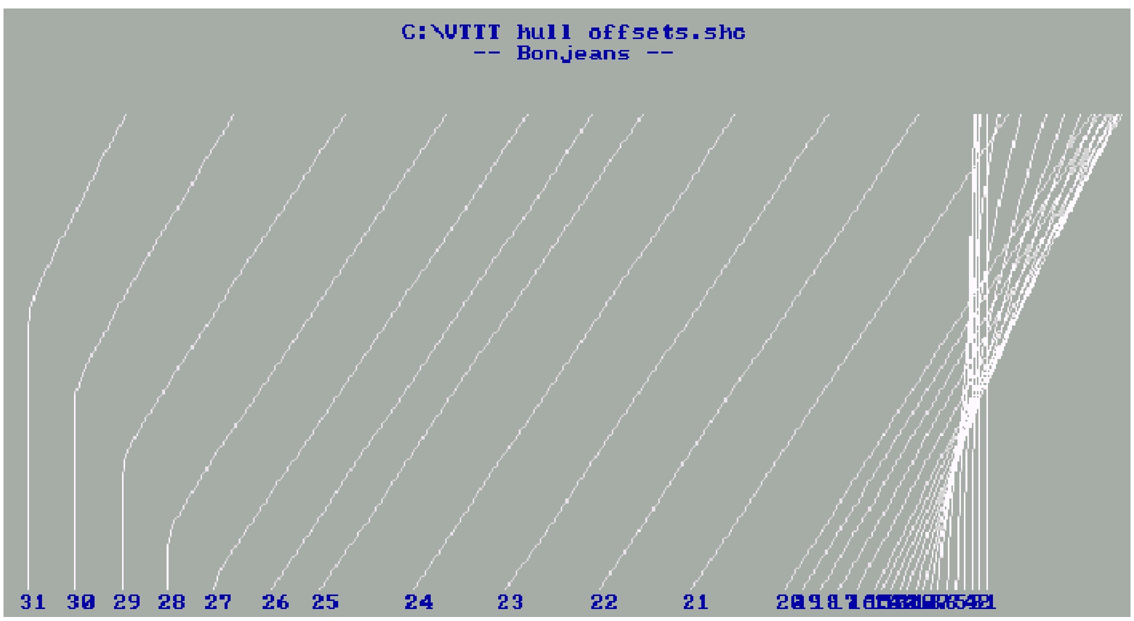

Figure 4.1.14: Bonjean Curves for the Concept Design

4.2 Structural Design and Analysis

Safehull 7.01 Phase A is used to perform preliminary structural design and analysis. Safehull allows the user to generate a midships section and evaluate a corresponding vessel to ensure it meets all American Bureau of Shipping (ABS) 2001 tanker requirements. Phase A strictly analyzes the ship based on the 2001 tanker rules as published by ABS. To perform a more detailed analysis of the ship, Phase B contains a Finite Element Analysis method for predicting structural failure. Phase B is not used in this concept design and is replaced by an analysis of the vessel using the Arctic Shipping Pollution Prevention Regulations (ASPPR) for ice class CAC4 as previously described in Chapter 3.

4.2.1 Safehull Phase A

To initiate the use of Safehull Phase A for the design of the Shuttle Tanker, basic ship dimensions are entered into the software. A double bottom, double side hull configuration is chosen.

The general midships geometry is entered into Safehull through the software prompt seen in Figure 4.2.1. A value of three meters for the web and floor spacing is used to provide even spacing within the thirty-meter cargo tank length. Zero camber and zero gunwale radius is chosen to simplify production. Increased strake and deck plating thickness compensate for stress concentrations at the gunwale. The bilge radius of 2.96 meters is determined to achieve the required section coefficient using Equation 4.2.1 where T is the draft, B is the breadth, C_x is the midship coefficient and r is the bilge radius.

$$TBC_x = TB - 2r^2 + 0.5\pi r^2 \qquad (4.2.1)$$

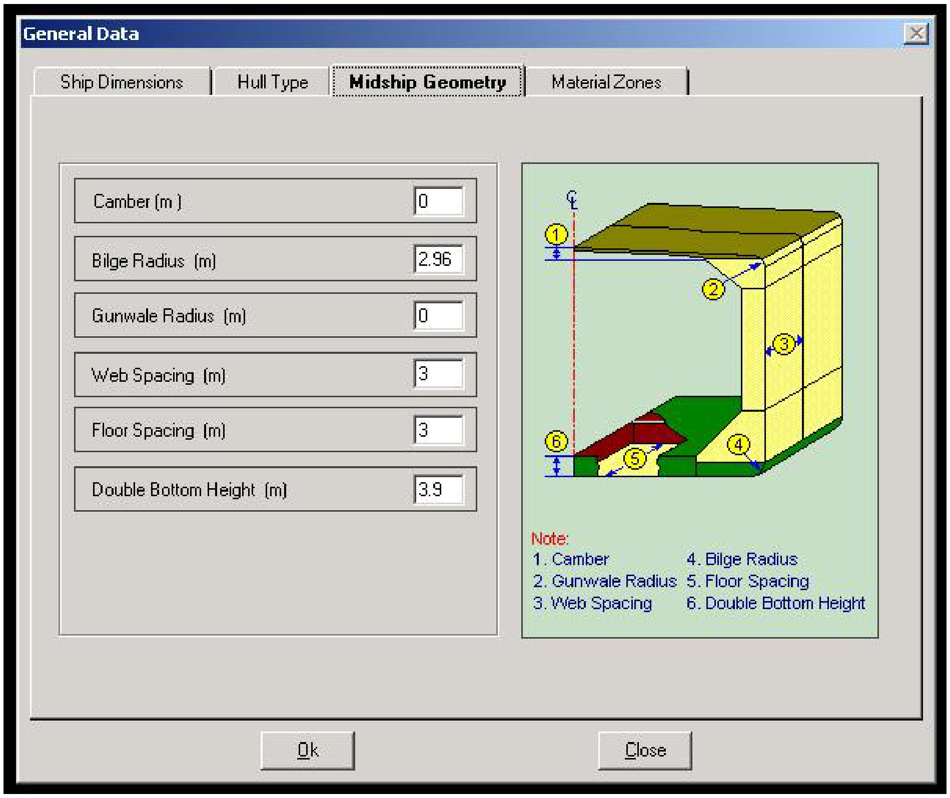

Figure 4.2.1: Safehull Midship Geometry

Plates and stiffeners are next defined within Safehull. The average stiffener spacing for the ship is 750 mm. Initially mild steel was used to meet the ABS requirements, however, the chosen material was changed in the sideshell, deck and other plating to meet ASPPR CAC4 local section modulus requirements. Material changes are specifically used only to increase yield strength or section modulus values as required by ASPPR and though desirable, are not used to decrease member thickness or total weight based on ABS requirements. A reduction in scantlings due to high strength steel was avoided to maintain fatigue strength.

Stiffeners used in the design are Bulb type stiffeners as shown in Figure 4.2.2. These stiffeners are chosen for: the lack of sharp corners, which promotes better paint adhesion, improved maintainability and reduced life-cycle cost.

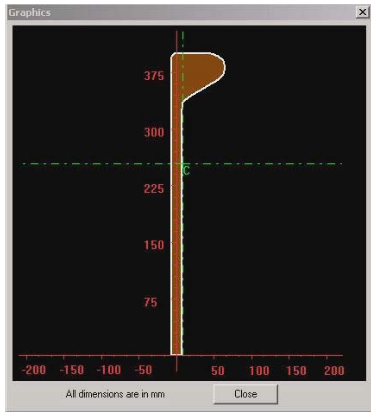

Figure 4.2.2: Example Bulb Type Stiffener

The design has three non-tight bottom girders and three non-tight stringers to provide access through the J-Shape ballast tanks for inspection. A minimum one-meter square access is provided in each longitudinal member. Initial offered scantlings were selected based on expert opinion and "Kometik" scantlings.

After all plates and stiffeners are defined and a complete midship section is produced, values of maximum bending moments are input into Safehull. Figure 4.2.3 is a plot of the full load sagging bending moment. Maximum bending moment in this condition is 273,113 MTm, which occurs at midships. The maximum shear force at station 8 is -5213 MT.

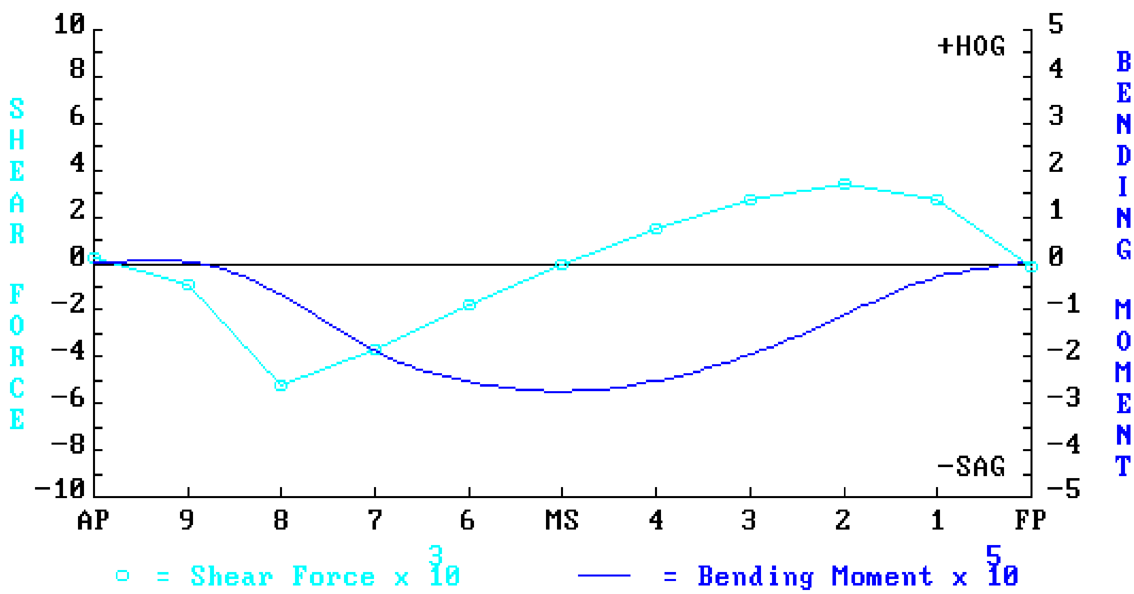

Figure 4.2.3: Full Load Sagging Bending Moment

Figure 4.2.4 shows a plot of the ballast load hogging bending moment. The maximum bending moment in this condition is 435,453 MTm and occurs at Station 6. The maximum shear force at station 8 is 7,282 MT.

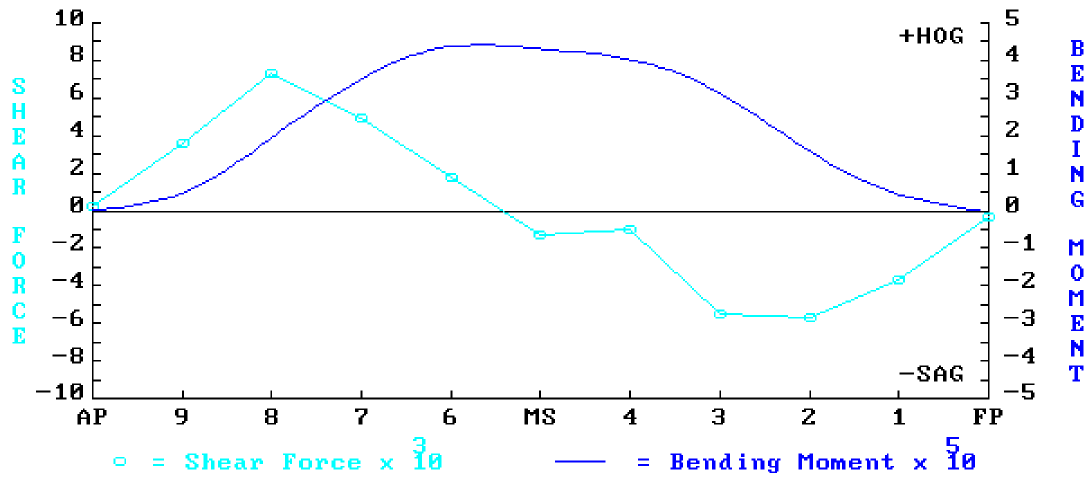

Figure 4.2.4: Ballast Load Hogging Bending Moment

Figure 4.2.5 is a plot of the light ship hogging bending moment. The maximum bending moment is 411,550 MTm and occurs at Station 6. The maximum shear force at station 8 is 8732 MT.

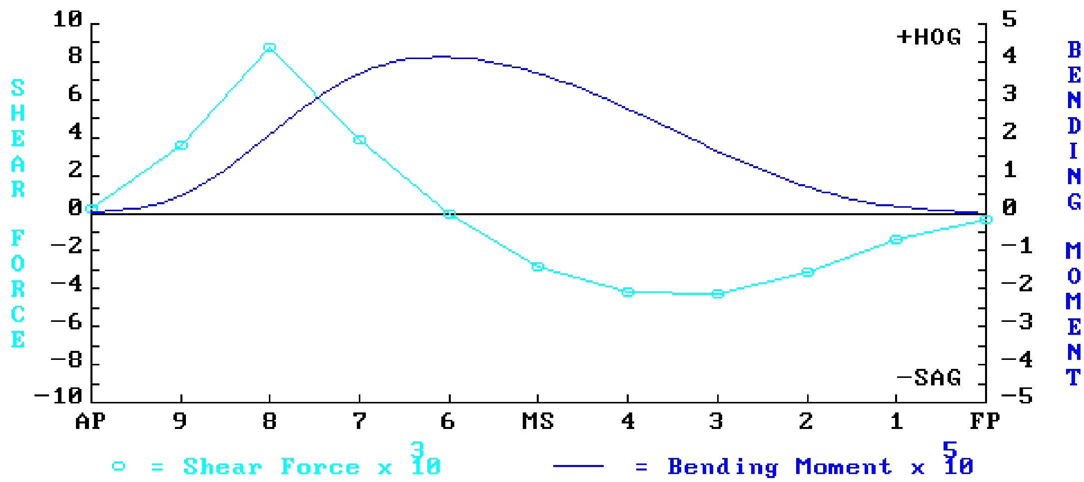

Figure 4.2.5: Light Ship Hogging Bending Moment

Following the specification of the bending moments, tanks are defined. A tank length of thirty meters is chosen to provide even three-meter frame spacing. After the tanks are defined, each plate and stiffener are examined to ensure that the minimum ABS thickness and section modulus have been met. In Appendix D, a graphical representation for each longitudinal member is shown. The green line indicates minimum ABS required thickness or section modulus while the blue lines indicate ASPPR CAC4 regulated values that are also the actual values provided for each member in the final concept design.

After the longitudinal members are examined for compliance with both the ABS and ASPPR regulations the transverse members of the ship are input into Safehull. The floor spacing matches the web spacing of three meters and the girders are placed every six meters measured from the centerline.

The transverse rings occurring at each three-meter web position consist of the above-defined floor, the side web frame, the deck transverse beam and the vertical web on the longitudinal bulkhead. The analyzed effective width of the side transverse is three meters as opposed to the four-meter double side width due to one-meter wide cuts for inspection. Figure 4.2.6 provides a graphical view of the deck transverse member.

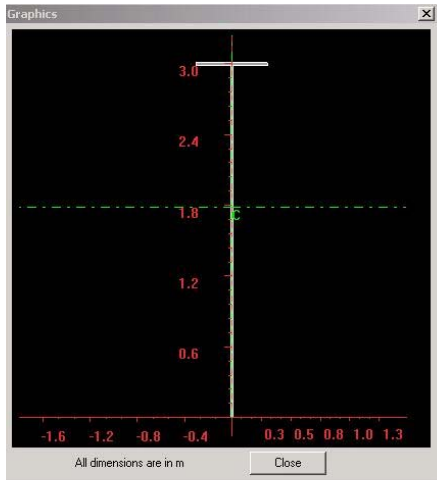

Figure 4.2.6: Safehull Deck Transverse

The web and flange thickness and width of the vertical web on the longitudinal bulkhead are similar to those of the deck transverse beam with only the web depth increasing from the three meter value to approximately four meters where the transverse meets the inner bottom.

Three horizontal girders are defined on the transverse bulkheads at heights corresponding with the non-tight stringers previosly described. The web thickness and depth are 22 mm and 3.3 m respectively and the flange depth and thickness are one meter and 22 mm respectively.

Simalarly, three vertical webs on the transverse bulkhead are defined at corresponding floor girder locations. The web thickness and depth are 15 mm and 3.7 m respectively and the flange depth and thickness are 0.5 m and 20 mm respectively.

The transverse bulkhead plates and stiffeners were designed to meet the ABS regulations yeilding an average stiffener spacing of 500 mm and an average plate thickness of approximately 15 mm. The midships drawing, shown in Drawing 5, provides exact dimmensions for all members discussed.

4.2.2 ASSPR Calculations

Five specific calculations are performed on eleven principal members of the ship in order to comply with the requirements set forth in the ASPPR CAC4 regulations. The first calculation is on the stringers and side transverse webs. The requirement is that the applied stress on these members is less than or equivalent to the yield stress of the member given a loading of 163.8 tonne/m. Equation 4.2.2 shows the calculation of the applied stress where, P is the loading, g is the gravitational acceleration, t is the member thickness and A_s is the applied stress.

$$A_s = Pg/t \qquad (4.2.2)$$

Table 4.2.1 provides the results of the above calculation with the difference being the check for compliance with ASPPR CAC4 requirements. If the difference has a negative value then the members meet the imposed regulations.

Table 4.2.1: ASSPR CAC4 Yield Stress Requirement

	Loading (tonne/m)	Provided Yield Stress (Mpa)	Provided Member Thickness (mm)	Calculated Applied Stress (Mpa)	Difference
Stringers	163.8	235.0	13.50	119.0	-115.9
Side Transverses	163.8	313.5	12.50	128.5	-184.9

The second ASSPR requirement is on the shell plating thickness. The thickness must be greater than a calculated thickness based on a given pressure of 4.55 Mpa. Equation 4.2.3 shows the calculation of the required thickness where P is the given pressure, Y is the yield stress of the material, F is the provided frame spacing and t is the required thickness.

$$t = FP/3Y \quad (4.2.3)$$

Table 4.2.2 provides the results of the above calculation with the difference being the check for compliance with ASPPR CAC4 requirements.

Table 4.2.2: ASSPR CAC4 Plate Thickness Requirement

	Pressure (Mpa)	Yeild Stress (Mpa)	Frame Spacing (mm)	Required Thickness (mm)	Smallest Provided Thickness (mm)	Difference
Shell Plating	4.550	313.5	3000	14.51	17.50	-2.986

The third ASSPR requirement is on the strake plate thickness. The thickness must be greater than a calculated thickness based on a given factor. Equation 4.2.4 shows the calculation of the required thickness where K is the given factor, S is the adjacent shell plate thickness and t is the required strake thickness.

$$t = 0.75KS \quad (4.2.4)$$

Table 4.2.3 provides the results of the above calculation with the difference being the check for compliance with ASPPR CAC4 requirements.

Table 4.2.3: ASSPR CAC4 Strake Plate Thickness Requirement

	Adjacent Shell Plate Thickness (mm)	K-factor	Thickness (mm)	Required Strake thickness	Provided Thickness (mm)	Difference
Stem Plate	18.50	1.300	24.05	18.03	18.50	-0.4625

The fourth ASSPR requirement is on the main transverse member section modulus. The section modulus must be greater than a calculated section modulus based on a given pressure. Equation 4.2.5 shows the calculation of the required section modulus where P is the given pressure, S is the member span, F is the member spacing, Y is the yield stress of the material and M is the required section modulus.

$$M = PF(S-400)/8Y \quad (4.2.5)$$

Table 4.2.4 provides the results of the above calculation with the difference being the check for compliance with CAC4 requirements.

Table 4.2.4: ASSPR CAC4 Transverse Member Section Modulus Requirement

	Span (mm)	Spacing (mm)	Pressure (Mpa)	Yield Stress (Mpa)	Calculated Section Modulus (cm^3)	Provided Section Modulus	Difference
Side Transverses	17950	3000	6.890	313.5	144600	151600	-6941
Deck Transverses	15980	3000	4.550	353.0	75310	76010	-701.9
Lng. Blkhd Vertical Webs	19070	3000	4.550	353.0	90240	90540	-300.2
Horizontal Girder	20070	5267	2.280	313.5	94220	135600	-41340
Trn. blkhd Vertical Webs	18770	5318	2.280	313.5	88820	91950	-3129

The final ASSPR requirement is on the main longitudinal members section modulus where offered section modulus must be greater than the calculated section modulus based on a given pressure and factor. Equation 4.2.6 shows the calculation of the required section modulus where P is the given pressure, S is the member span, F is the member spacing, Y is the yield stress of the material, K is the given factor and M is the required section modulus.

$$M = 2SFPK/Y \quad (4.2.6)$$

Table 4.2.5 provides the results of the above calculation with the difference being the check for compliance of the ASPPR CAC4 requirements.

Table 4.2.5: ASSPR CAC4 Longitudinal Member Section Modulus Requirement

	Span (mm)	Spacing (mm)	Pressure (Mpa)	Yeild Stress (Mpa)	K-factor	Calculated Section Modulus (cm^3)	Provided Section Modulus	Difference
Girder	30000	3000	6.890	235.0	5.38E-05	283.7	969.0	-685.2
Stringers	30000	3000	6.890	235.0	5.38E-05	283.7	469.0	-185.2

After ASPPR CAC4 compliance is obtained, a bare hull weight estimate is performed to compare the concept structural weight to the ship synthesis model predicted weight. The estimate incorporates the summation of all longitudinal and transverse member sizes provided by Safehull multiplied by the density of the specific steel of each member. The summed weight is substituted into the bare hull weight calculation in the Weight Section of the ship synthesis model shown in Appendix B, page B16. By substituting this weight into the calculation a more accurate lightship weight is obtained as opposed to using the equations based on ship principal characteristics. As shown in Table 4.2.6 the final structural weight is approximately 6% greater than the predicted ship synthesis model value. This difference is due to the low initial ASPPR CAC4 weight estimates discussed in Chapter 3.

Table 4.2.6: Bare Hull Weight Comparison

Longitudinal weight estimate	14590	tonnes
Longitudinal and transverse estimate	19600	tonnes
Estimate w/ bow & transom	25480	tonnes
Math model estimate	23930	tonnes

4.3 Resistance, Power and Propulsion

4.3.1 Resistance Analysis

To assess the resistance and powering feasibility of the Shuttle Tanker, NavCAD is used as a tool. Analyzing the resistance of the hull and knowing engine characteristics allows calculation of the optimum propeller design. Once found, this propeller design is used to perform a complete system analysis for the concept ship. Outputs of resistance, power and fuel consumption rates are calculated for the concept ship design.

The hull characteristics in Table 4.3.1 are specified in NavCAD. In addition to these characteristics, bow and stern shapes of the U-shape are specified. The ship operates in saltwater and is analyzed at a range of speeds. Bow thrusters and the podded propulsion units are included in the appendage resistance. Since the ship has two bow thrusters and only one diameter can be input into NavCAD, a single diameter of 3.96 m was entered with a drag coefficient of 0.0075. The area of this single thruster is equivalent to the area of the two thrusters of 2.8 m diameter with which the ship is equipped. Pods are considered as rudders in the analysis, as there is no option to enter pods. The total wetted surface area for two pods is estimated at 122 square meters with a drag coefficient of 1.5. Environmental data contributing to the performance of the ship corresponds to Sea State 6 as listed in Table 4.3.2. Additional resistance due to seas is estimated as ten percent of the total ship resistance. This is done because the method by which NavCAD estimates resistance due to seas is for a NavSEA ship, of which the Shuttle Tanker is far outside the limits. To find bare-hull resistance predictions, the Holtrop-1984 method is used. The friction coefficient is found using the ITTC method, and the Holtrop method is used with a correlation allowance of 0.00014 and a 3-D form factor of 1.3729. Resistance calculations for the full load design case can be found in Table 4.3.3. Comprehensive resistance data for the full load design case are found in the Appendix E.

Table 4.3.1: NavCAD Hull Form Parameters

	Design	Ballast
Length Between PP (m)	252.8	252.8
WL bow pt aft FP (m)	0	0
Length on WL (m)	252.8	252.8
Max beam on WL (m)	50.55	50.55
Draft aft mid WL (m)	15.01	8.584
Displacement bare (tonnes)	161700	87010
Max area coefficient	0.995	0.995
Waterplane coefficient	0.911	0.911
Wetted surface area (m^2)	18450	14210
Trim by stern (m)	0.004	0.574
LCB aft of FP (m)	122.2	117.6
Bulb ext fwd FP (m)	5	5
Bulb area at FP (m^2)	42.6	42.6
Bulb ctr above BL (m)	8.9	8.9
Transom area (m^2)	14.4	14.4
Half entrance angle (deg)	40	40

Table 4.3.2: Environmental Data

Parameters	
Wind speed	37.5
Angle off bow	0
Tran. hull area	613
VCE above WL	7.567
Tran. superst. area	817.7
VCE above WL	16.5
Total longl. area	4527
VCE above WL	8.37
Wind location	Free Stream
Hull type	Tanker/Bulk

Table 4.3.3: Resistance Summary for the Full Load Design Case

Velocity (kts)	Rbare (kN)	Rapp (kN)	Rwind (kN)	Rseas (kN)	Rtotal (kN)	Petotal (kW)
4	99.52	2.340	257.6	9.950	369.4	760.1
6	223.3	5.180	286.7	22.33	537.5	1659
8	381.5	9.120	317.4	38.15	746.1	3071
10	577.0	14.14	349.6	57.70	998.5	5137
12	810.1	20.24	383.5	81.01	1295	7993
14	1090	27.42	418.8	109.1	1646	11850
14.5	1171	29.38	427.9	117.1	1746	13020
15	1257	31.41	437.1	125.8	1858	14290
15.5	1350	33.50	446.4	135.0	1965	15670
16	1451	35.66	455.8	145.1	2088	1718

Where: Rbare = bare hull resistance Rapp = appendage resistance Rwind = wind resistance
Rseas = resistance due to seas Rtotal = total resistance PEtotal = total effective power

The two main engines chosen in the optimization of the ship are Sulzer 14ZA40S diesel generators. This engine/generator combination produces 9,750 kW at 60 Hz and 6.6 kV. After further analysis of power requirements, the larger Sulzer 16ZA40S are chosen as the main propulsion engines. These engines produce 11,210 kW of electrical power at 514 rpm, 60 Hz and 6.6 kV. This allows for a greater service speed margin and improves reliability of the engines by operating below the MCR. The specific fuel consumption of the engines is 183 g/kWh, as specified by the manufacturer.

The podded propulsor chosen for the concept design is the SSP10 Propulsor made by Siemens and Schottel delivering up to 10,000 kW of propeller power each. Two of these propulsors are used which totals four propellers. The steerable azimuth drive system provides an increase in efficiency largely due to its twin propeller technology, combined with the hydrodynamically optimized propulsion module, and the permanently excited 60 Hz synchronous propulsion motor [8]. These pods are characterized by two propellers per pod, one in front of the propulsion module and one behind, both on a common shaft. Each propeller has three blades and rotates in the same direction. Two fins are mounted between the propellers on the propulsion module to help regain swirl energy and increase efficiency. A propeller diameter of 4.75 m and a propeller speed of 160 RPM are standard. Figure 4.3.1 shows the pod and its main components.

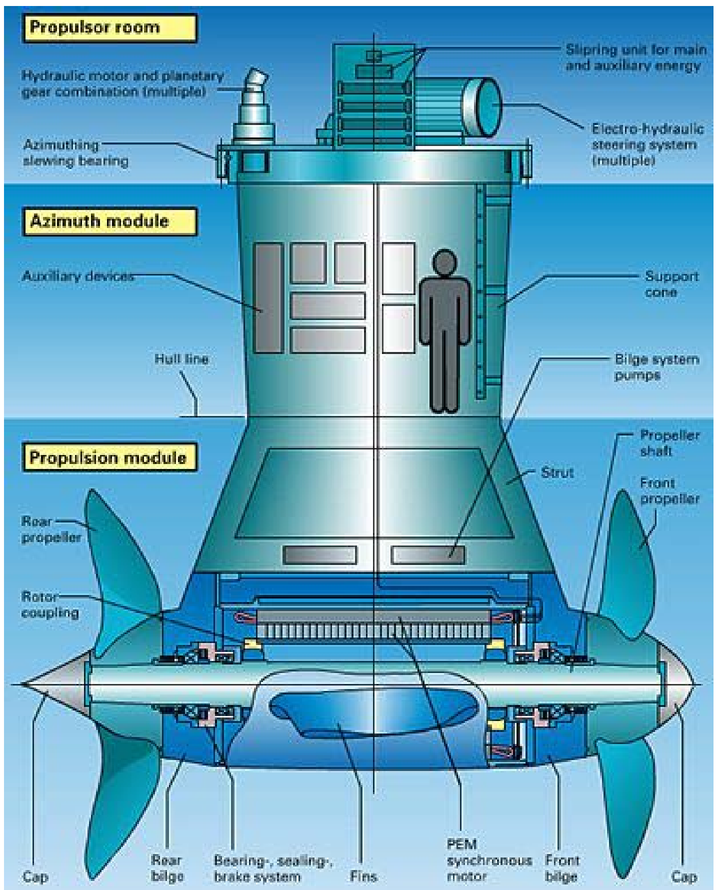

Figure 4.3.1: SPP Propulsor [8]

Since the propulsion units are equipped with three bladed propellers, only a three bladed fixed pitch propeller was optimized. The data defined in the propeller section is found in Table 4.3.4. The expanded area ratio input in this section is a generic value and is optimized in the analysis. The values of Kt and Kq are estimates for commercial ships. Cavitation breakdown is not applied to the propeller. The minimum and maximum diameters for the propellers are values around the given diameter for the SSP10 and were defined as 4.5 m and 5.5 m respectively. These values are flexible due to the fact that the height of the strut can be lengthened or shortened accordingly to allow for proper clearance.

Table 4.3.4: Propeller Characteristics

Parameters	3-blade FPP
Series	B-series
Blades	3
Exp area ratio	0.65
Min diameter	4.5 m
Max diameter	6.25 m
Pitch type	FPP
Scale Correlation	B-series
Kt multiplier	0.97
Kq multiplier	1.03
Blade t/c ratio	0
Rouchness	0
Propeller cup	0

The design speed of 15 knots and minimum and maximum speeds of 10 knots and 16 knots respectively are used in the propeller analysis. The Keller equation is used for the cavitation criteria. Since an overall efficiency of 93% is specified for the entire IPS system, the shaft efficiency is taken as 1. Propeller immersion is estimated at 6 meters.

Optimizing the propellers is iterative. In the first iteration, expanded area ratio, diameter, and pitch are all optimized. Each consecutive iteration optimizes pitch only. The expanded area ratio is gradually increased on each consecutive run to obtain acceptable pressure limits on the propeller. After this iterative process is complete, the expanded area ratio is found to be 0.5045, the diameter 5.4271m and the pitch 4.0537m. Appendix E contains complete results for the propeller optimization.

After the propeller has been optimized, a complete system analysis is performed. Resistance, power, and propeller data is generated for the speeds shown in Table 4.3.3. At endurance speed, the brake power is 20176.7 kW and the fuel rate is 3958.95 lph. Total resistance, brake power, and fuel consumption are each plotted against ship velocity in Figures 4.3.2 through 4.3.4.

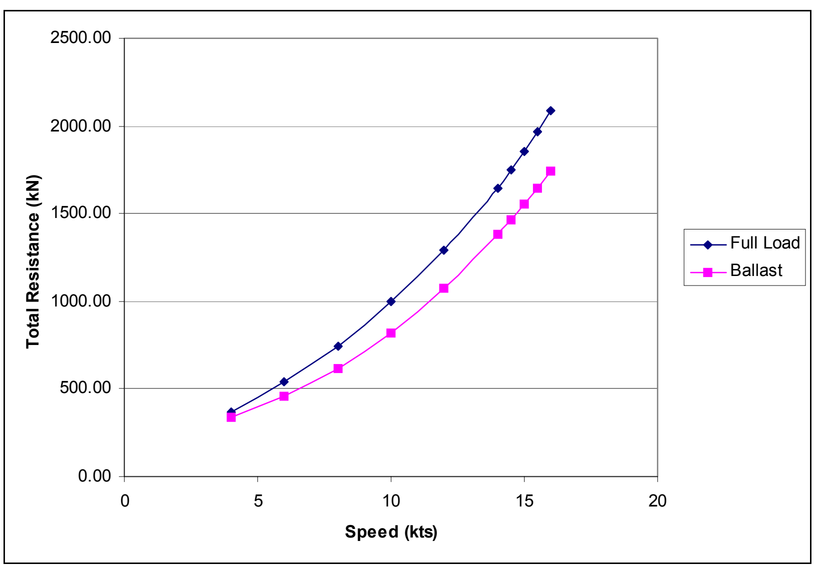

Figure 4.3.2: Total Resistance vs. Ship Speed

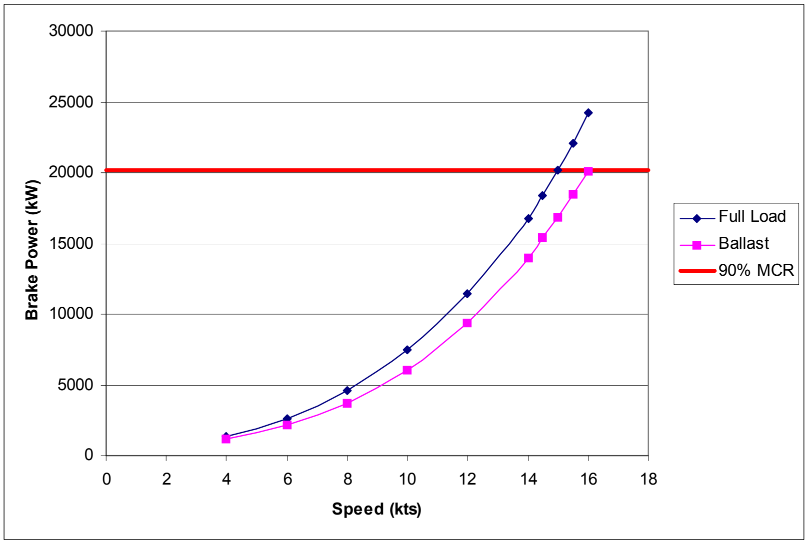

Figure 4.3.3: Brake Power vs. Ship Speed

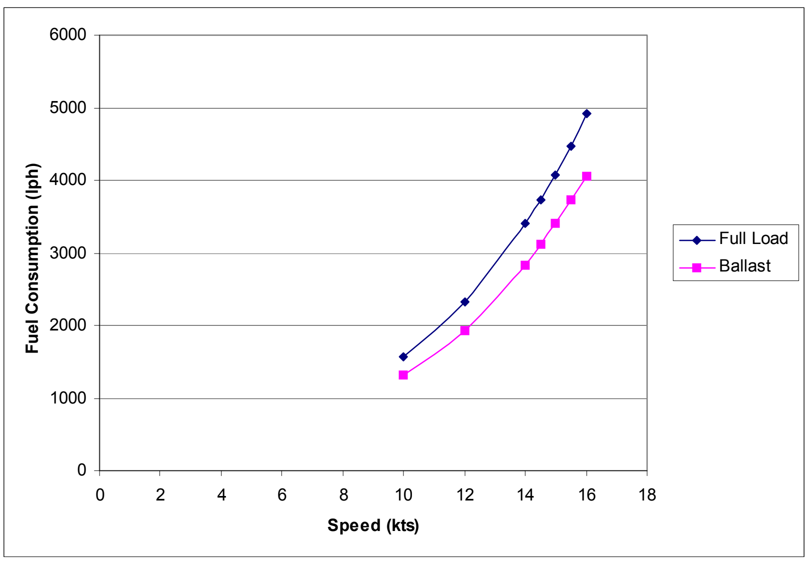

Figure 4.3.4: Fuel Consumption vs. Ship Speed

Once the system analysis is completed, an additional ballast case is analyzed using the optimized propeller. All hull parameters remain the same except the draft, displacement, wetted surface area, and trim. These values are found in Table 4.3.1. Appendix F contains complete ballast case results.

As defined in the owner requirements, the sustained speed must be calculated at 90 % maximum continuous rating (MCR) to allow for any biofouling, seas and wind. The two Sulzer engines are rated at 11,210 kW each, giving a total of 22,420 kW. This gives 20,178 kW at 90 % MCR. Further analysis in NavCAD gives maximum sustained speed at BHP_{max}, as well as fuel consumption at this speed. These values, along with fuel consumption at endurance speed are shown in Table 4.3.5.

Table 4.3.5: Maximum Load Cases

Case	Sustained speed at BHPmax (kts)	Fuel rate at sustained speed (lph)	Fuel rate at endurance speed (lph)
Design	15.58	4516	4057
Ballast	16.63	4526	3394

4.3.2 Endurance Electrical Power Analysis

The average electrical load required to service the ship over one full trip is needed to determine fuel weight and volume. The electrical loads taken into account are propulsion, steering, lighting, firemain, heating, ventilation, bow thrusters, crude oil washing, and the inert gas system. Electrical load estimates for each condition are shown in Table 4.3.6. One trip is defined as a 300 mile trip for ballast and full load each as well as a 24 hour on-load period and 14 hour off-load period. The full analysis is found in the Electrical Load section, Appendix B, Page B8. The Electrical and Propulsion schematic can be seen in Appendix G.

Table 4.3.6: Endurance Analysis Summary

Condition	Power (kW)	Endurance Fuel (m^3)
Offload	6121	22.30
Onload	14150	88.40
Steam - Ballast	16700	76.90
Steam - Full	20890	92.00

4.3.3 Endurance Fuel Calculation

An endurance fuel calculation is performed for one full round trip for the Shuttle Tanker using fuel consumption data obtained from NavCAD as well as the electrical load analysis found in the Electrical Load section, Appendix B, Page B10. There are four different states in which the Shuttle Tanker will be operating; on-load, off-load, full load steaming and ballast steaming. Incorporating the fuel rates with the electrical loads, the required fuel weight for one trip was calculated. Since refueling cannot be completed at the offloading site at Whiffenhead, a special trip needs to be made to St. Johns, Nova Scotia to refuel. Because of this, our total required tank volume was defined for 5.5 round trips, which allows enough fuel to make the trip to refuel without being too inconvenienced by having to refuel after every trip. Table 4.3.6 shows the required amount of fuel for each condition for one roundtrip. The maximum range of the Shuttle Tanker is 6,150 nm at 15 knots.

4.4 Mechanical and Electrical Systems

The equipment list used for the concept design is determined by the optimization process, equipment from similar ships and expert opinion. A complete mechanical and electrical system list, including approximate dimensions and capacities, is shown in Table 4.5.4.

4.4.1 Mechanical Systems

The main propulsion components included in our design consist of two main engine/generator sets, bow thrusters, fuel, diesel and lube oil purifiers. The main engines are described in Section 4.3.1. Two 2000 kW bow thrusters are incorporated into the design in order to maximize stationkeeping ability while loading. The bow thrusters have controllable pitch propellers allowing them to produce variable and reversible thrust. The bow thrusters can produce up to 300 kN of thrust each.

Fuel oil, diesel oil, and lubrication oil purifiers, as well as two fuel oil heaters condition the fuel and lube oil. These are sized according to fuel consumption. Two fuel oil and lubrication oil purifiers service the two main engines. The diesel oil purifier services the auxiliary and emergency generator.

One auxiliary boiler rated at 23,473 kg/hr of steam and one heat recovery boiler rated at 15,648 kg/hr of steam supply steam for hotel services, evaporators, and fuel oil heating. The heat recovery boiler takes advantage of the latent heat in the exhaust gasses. The auxiliary boiler provides additional output when the heat recovery boiler cannot provide enough output alone. The auxiliary boiler can also be used when the main engines are shut down.

4.4.2 Electrical Systems

The electrical loads are analyzed using the results from the Electrical Load section of Appendix B, page B8. This analysis is used to determine the power requirements during the four operating conditions; load, offload, full load steaming and ballast steaming. A summary table of these conditions can be seen in Table 4.4.1. The loads are divided into cargo and ship service for each case. The integrated power system is designed so that the two main engines can power all electrical needs in each operating state. The full load steaming condition requires the most power.

Table 4.4.1: Operating Condition Power Requirements

Operating Condition	Required (kW)	Available (kW)	Engines Online
Offloading in Port	6120	10700	1
Onloading at Sea	14100	21500	2
Full Load Transit	20900	21500	2
Ballast Transit	16700	21500	2

The main engines use an isochronous mode of load sharing where the engines are held at a constant rpm producing a constant frequency of 60 Hz. Electronically governing the engines and constantly monitoring the load carried by each main engine accomplishes this. The fuel supplied to each unit is adjusted to ensure that the load is proportionally shared between each prime mover [9].

The main generators also have load sharing capabilities using crosscurrent compensation. This system is similar to the main engine load sharing mentioned above. Voltage regulators for both generators have a common current sensor that is used to adjust the field current in each generator to achieve a proportional current division [9].

Automatic load shedding is used to maintain the load within the system capabilities in the event of an abnormal overload condition. The removal or reduction of previously selected loads, such as the propulsion motors, prevents a total generator shutdown due to activation of the under-frequency protective system [9].

Power limiting of the main engines by an automatic power limiting regulator compensates in the event of a sudden loss of generating power. This is to prevent either main engine from being overloaded. The same regulator is used to limit the power delivered to the propellers in the event that the throttle is advanced to a power greater than what is available [9].

Transformers are used to match the higher bus voltage of the main engine generating sets (6.6 kV) to the lower voltage required by the static power converters (3.3 kV). Power converters of the cycloconverter type are used to change the frequency of the power delivered to the synchronous propulsion motors. The combination of the cycloconverter and the AC synchronous propulsion motor allows the propeller speed to be varied smoothly over the entire speed range. The power converters also control cargo and ballast pumps using 3.3 kV power [9].

Harmonic currents can adversely affect the quality of the ship service distribution power and the operation of instrumentation and protection features. For this reason, harmonic filters are added to the power system. These filters provide a low-impedance path for harmonic currents, which flow mostly through the filter, bypassing other system components [9].

Ship service distribution transformers are used to supply 440V power from the 6.6 kV main switchboard to the ship service distribution switchboard.

4.5 Space and Arrangements

4.5.1 Internal Space and Arrangements

Tank arrangements are defined using the HecSalv software program as an aid. The ship synthesis model is used as a starting point for the dimensions and volumes of internal tanks and structure. Table 4.5.1 shows required and actual values for internal arrangement.

Table 4.5.1: Required and Actual Internal Arrangement Characteristics

Parameter	Required	Actual
Length of Machinery Box (m)	19.02	27.00
Height of Machinery Box (m)	6.30	23.35
Cargo Subdivision	3 x 2	6 x 2
Cargo Weight (MT)	127,000	128,200
Cargo Block Length (m)	182.3	189.0
Height of Double Bottom (m)	3.9	3.9
Width of Double Side (m)	4.0	4.0
Fuel Weight (MT)	837.0	1672
Potable Water Weight (MT)	212.5	244
Lube Oil Weight (MT)	18.8	53
Aft peak Ballast Tank Weight (MT)	2516	3247
Forepeak Ballast Tank Weight (MT)	6143	3232

The double bottom height of 3.9 meters and the double side width of 4.0 meters define the general shape of the cargo and ballast tanks. A stool of 3.275 meters (base and height of stool) is added for structural purposes. The outboard longitudinal bulkhead in way of Tanks No. 1 and 2 is kept vertical and not sloped with the flare of the hull to increase producibility and to move the cargo center of gravity aft. The port cargo and ballast longitudinal bulkhead is shown in Figure 4.5.1.

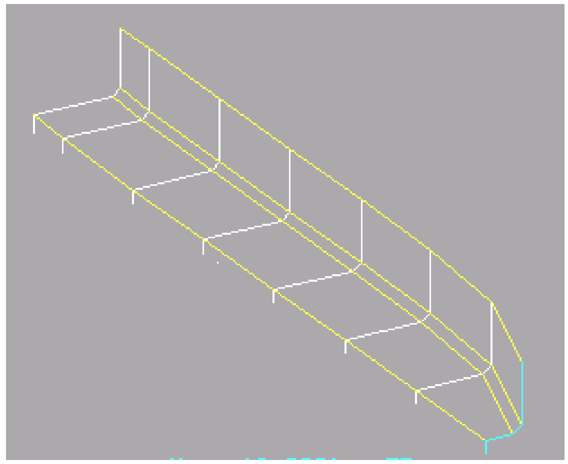

Figure 4.5.1: Port Cargo/Ballast Bulkhead

A transverse frame spacing of 3 meters is chosen to give an integer number of frames in each tank while not exceeding the cargo block length recommended by the ship synthesis model by more than 5%. The collision bulkhead is located at 12 meters aft of the FP in accordance with ABS Rules for Building and Classing Steel Vessels [10]. This allows enough room for the two bow tunnel thrusters and other equipment forward of the collision bulkhead. Each cargo and water ballast "J" tank is 30 meters long, except for the #6 port and starboard ballast tanks, which are 36 meters long. Table 4.5.2 lists all tank and compartment boundaries along with total volume minus internal structure and the capacity where applicable. Saltwater ballast and freshwater capacity is computed assuming 100% filling. Cargo oil capacity is computed assuming 98% filling and a cargo density of 0.853 MT/m^3. Fuel oil capacity is computed assuming 98% filling and a cargo density of 0.95 MT/m^3. Total cargo and ballast capacities are given in the bottom of Table 4.5.2.

Table 4.5.2: Compartment boundaries, volume and capacities

Compartment Name	Forward Bound m-FP	Aft Bound m-FP	Volume m^3	Capacity MT
Forecastle	-9.00	42.00	NA	NA
Thrust Room	-4.04	12.00	NA	NA
Forepeak	-5.80	12.00	3153	3232
No. 1 WBT S	12.00	42.00	5897	6044
No. 1 COT S	12.00	42.00	8355	7058
No. 1 WBT P	12.00	42.00	5897	6044
No. 1 COT P	12.00	42.00	8355	7058
No. 2 WBT S	42.00	72.00	5760	5904
No. 2 COT S	42.00	72.00	12240	10340
No. 2 WBT P	42.00	72.00	5760	5904
No. 2 COT P	42.00	72.00	12240	10340
No. 3 WBT S	72.00	102.0	5436	5572
No. 3 COT S	72.00	102.0	13010	11000
No. 3 WBT P	72.00	102.0	5436	5572
No. 3 COT P	72.00	102.0	13010	11000
No. 4 WBT S	102.0	132.0	5451	5587
No. 4 COT S	102.0	132.0	13160	11120
No. 4 WBT P	102.0	132.0	5451	5587
No. 4 COT P	102.0	132.0	13160	11120
No. 5 WBT S	132.0	162.0	5495	5632
No. 5 COT S	132.0	162.0	13160	11120
No. 5 WBT P	132.0	162.0	5495	5632
No. 5 COT P	132.0	162.0	13160	11120
No. 6 WBT S	162.0	201.0	7081	7258
No. 6 COT S	162.0	192.0	13070	11040
No. 6 WBT P	162.0	201.0	7081	7258
No. 6 COT P	162.0	192.0	13070	11040
Slop Tank S	192.0	201.0	2906	2455
Slop Tank P	192.0	201.0	2906	2455
Fuel Tank S	192.0	201.0	898	836
Fuel Tank P	192.0	201.0	898	836
Cofferdam	201.0	204.0	NA	NA
Engine Room S	204.0	231.0	NA	NA
Engine Room P	204.0	231.0	NA	NA
Pod Room S	231.0	246.0	NA	NA
Pod Room P	231.0	246.0	NA	NA
Fresh Water S	240.0	246.0	122	122
Fresh Water P	240.0	246.0	122	122
Aft Peak	246.0	252.8	3168	3247
Total COT			151800	128200
Total WBT			76560	78480

The slop tanks are designed to have volume equal or greater than 2% of the total cargo volume. The slop tanks are used for cargo storage in normal operating conditions and are required to be within the double hull to reduce oil outflow in the event of a collision or grounding.

Fuel tanks of sufficient size are also placed within the double hull. Fuel is present during 100% of the operation time where the slop tanks contain oil roughly only 50% of operation time. Therefore the two tanks are placed on the centerline just inside the slop tanks.

A 3-meter gastight cofferdam is placed just aft of the slop and fuel tanks to isolate cargo and fuel vapors and to provide space for pipe runs. The cofferdam is connected to the pump room located just below Platform A in the engine room.

The engine room is divided by a centerline bulkhead to allow for added reliability and redundancy. The port and starboard engine rooms extend 27 meters and span half the width of the hull. The pod room is also divided by a centerline bulkhead and extends 15 meters longitudinally. The aft peak tank is given sufficient volume based on the ship synthesis model to allow for trim adjustment. The final compartment boundaries are shown in Figure 4.5.2 and attached Drawings 2 and 3.

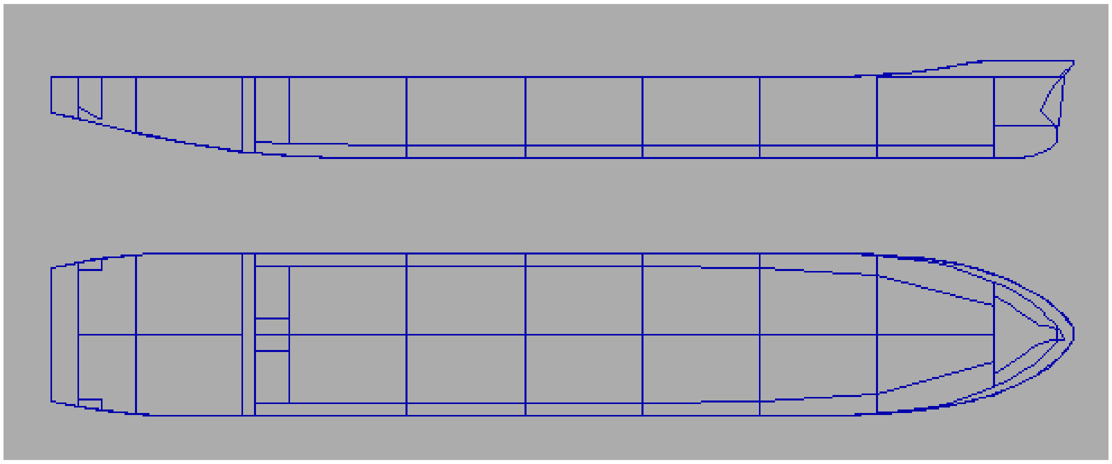

Figure 4.5.2: Concept Hullform with Internal Tanks and Compartments

4.5.2 Machinery Space

Spacing and arrangements in the machinery box are modeled using AutoCAD. The machinery space includes the port and starboard engine rooms and the port and starboard pod rooms. This division allows for added redundancy and reliability. [11] The engine rooms start at 204 meters aft of the FP and extend 27 meters. From here the pod rooms start and extend 15 meters aft. Four platforms are placed within the port and starboard engine rooms to allow arrangement of equipment. The platform elevations and associated areas are listed in Table 4.5.3.

Table 4.5.3: Engine Room Platform Locations and Areas

Engine Room Platform	Elevation from Baseline (m)	Area (m^2)
A	5.447	400
B	8.447	1200
C	15.45	1320
D	20.19	1350

All components in both port and starboard engine rooms are placed at a minimum distance of 1 meter from any bulkhead to allow for service room and auxiliary systems. Many heavy components are placed aft in the engine rooms to compensate for the full displacement of the stern section.

All large pumps are placed as low as possible to allow for maximum suction. As seen in Figure 4.5.3, the space above Platform A is reserved for electric pump drive motors. The space below Platform A is directly connected to the cofferdam and is reserved for the pump housings. This division achieves a gas seal preventing cargo fumes where electrical sparks are possible. Platform A can be seen in Figure 4.5.4 with the item numbers referenced in Table 4.5.4. Contained just aft of Platform A are the lube oil and waste lube oil tanks shown in Figures 4.5.3 and 4.5.4.

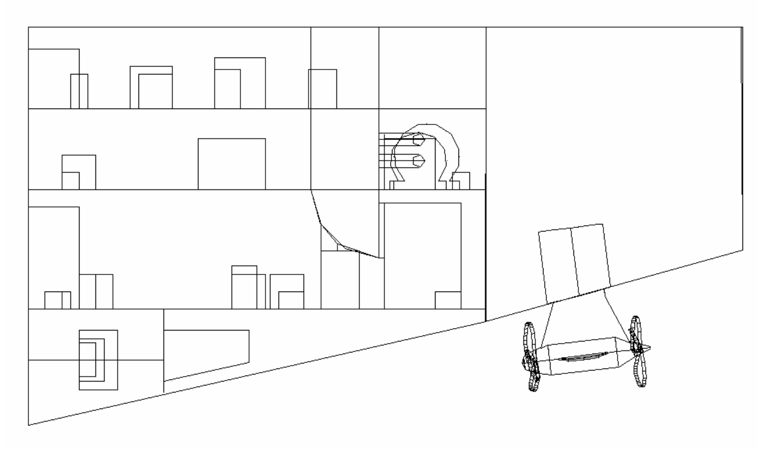

Figure 4.5.3: Engine and Pod Room Profile View

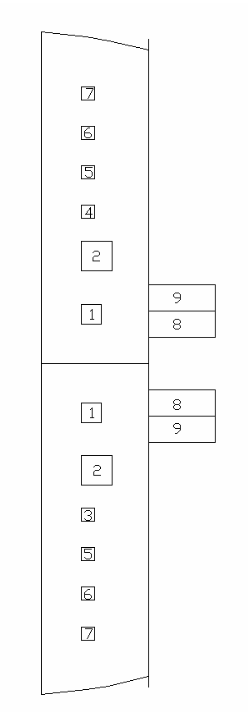

Figure 4.5.4: Platform A Plan View with Labeled Components

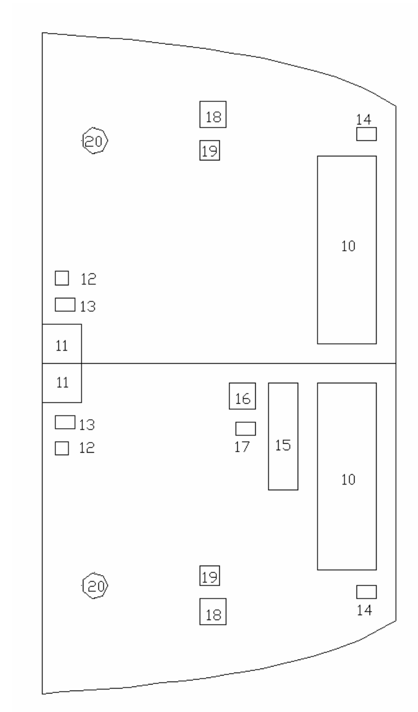

Figure 4.5.5: Platform B Plan View with Labeled Components

Platform B is the first full-length platform. Both main engine/generator sets are placed on this level. The auxiliary diesel generator is also placed on this platform and joined with the exhaust of the main engines. All fuel oil, diesel oil and lube oil components are placed on this level to be in close proximity to the engines. Item locations can be seen in Figure 4.5.5 with number references in Table 4.5.4.

The main boiler and heat recovery boiler are placed on Platform C directly aft of the inlet and exhaust stack. The potable water pumps and fresh water generators are placed outboard and aft in the engine rooms to be in close proximity to the potable water tanks. These are located in the port and starboard pod rooms. Platform C can be seen in Figure 4.5.6.

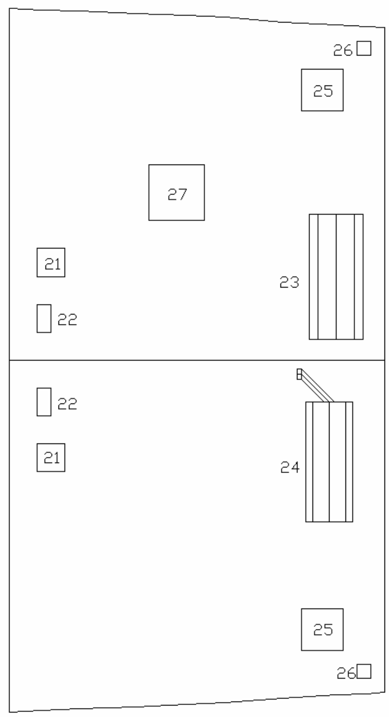

Figure 4.5.6: Platform C Plan View with Labeled Components

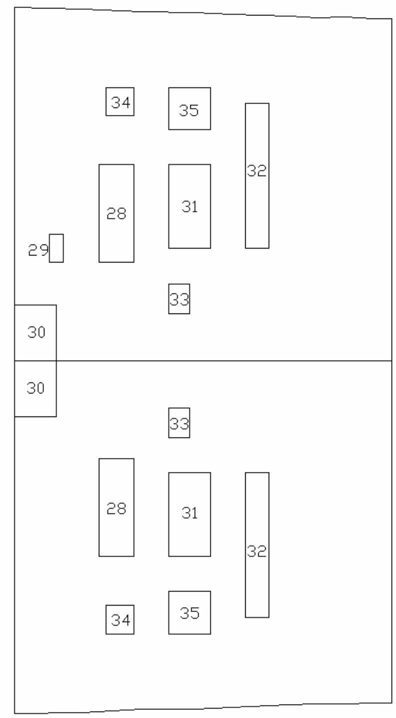

Figure 4.5.7: Platform D Plan View with Labeled Components

All sensitive items such as electronic equipment is placed on Platform D, which is located over 5 meters above the design waterline. Figure 4.5.7 shows component locations on Platform D. An equipment list is provided in Table 4.5.4, showing the location of each equipment item, item number, quantity, capacity and gross dimensions. The 2-D machinery and deckhouse arrangements are shown in Drawing 4.

Table 4.5.4: Equipment List for Port and Starboard Engine Rooms

Equipment	Item Number	Quantity	Capacity	Gross Dimensions (m) lxwxh
Platform A				
Ballast pumps	1	2	2500 m^3/hr	1.5x1.5x2.5
Cargo pumps	2	2	6000 m^3/hr	2.5x2.5x3.5
Cargo stripping pump	3	1	300 m^3/hr	1x1x2
Crude oil washing pump	4	1	1000 m^3/hr	1x1x2
Main seawater	5	2	1500 m^3/hr	1x1x2
Central freshwater cooling	6	2	1500 m^3/hr	1x1x2
Fire pumps	7	3	300 m^3/hr	1x1x2
Lube oil tanks	8	2	24 m^3	5x3x2
Waste lube oil tanks	9	2	24 m^3	5x3x2
Platform B				
Main engine/generator	10	2	11210 kW	14.3x4.5x6.2
Fuel oil day tank	11	2	50 m^3	3x3x6
Fuel oil heaters	12	2	NA	1x1x1
Fuel oil purifiers	13	2	NA	1.5x1x1
Lube oil purifiers	14	2	NA	1.5x1x1
Diesel generator	15	1	1400 kW	8.179 x 2.252 x 3.374
Diesel oil day tank	16	1	8 m^3	2x2x2
Diesel oil purifiers	17	1	NA	1.5x1x1
Ship service air receiver	18	2	150 ft^3	1.5x1.5x2.5
L/P air compressors	19	2	150 ft^3/m	0.85x1.52x1.355
Central SW/FW heat exchanger	20	2	NA	2x1.5x2
Platform C				
A/C units	21	2	75 ton	2x2x2
Refrigeration units	22	2	3 ton	1x2x1
Auxiliary boiler	23	1	23,473 kg/hr	8.9662x3.88x4.191
Heat recovery boiler	24	1	15,648 kg/hr	8.5852x3.3528x3.6322
Fresh water generator	25	2	30 tonnes/day	1.32x1.5x1.87
Potable water pumps	26	2	25 m^3/hr	1x1x1
Sewage treatment plant	27	1	40 persons	4x4x3
Platform D				
Main switchboard	28	1	24,000 kW	10x2x3
Emergency switchboard	29	1	1,000 kW	2x1x2
Propulsion control room	30	1	NA	8x3x3.5
Power converter	31	4	6,000 kW	4x1.5x2.3
Power transformer	32	8	3,000 kW	2.591 x 1.673 x 2.286
Distribution transformer	33	2	750 kW	2.134 x 1.524 x 2.286
Control and excitation module	34	2	9000 kW	4x1.5x2.3
Harmonic filter	35	2	12,000 kW	4x1.5x2.3
Other				
Pods		2	10,000 kW	8.38x4.75x9.907

Figures 4.5.8 through 4.9.11 show the completed three-dimensional concept machinery room design. The colors are used to designate different platforms.

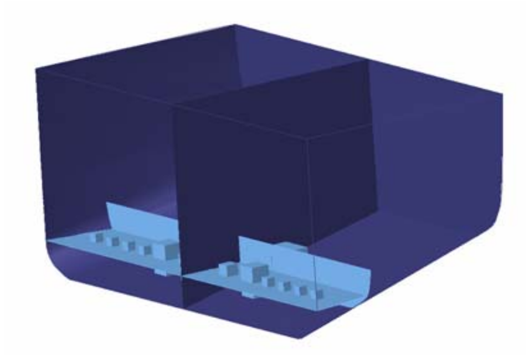

Figure 4.5.8: Platform A

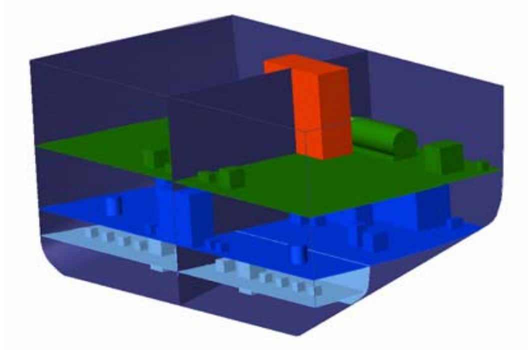

Figure 4.5.10: Platform A, B, and C

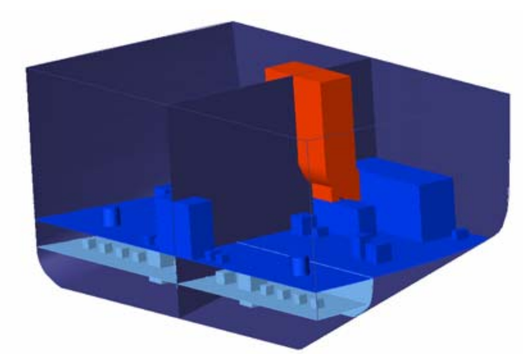

Figure 4.5.9: Platform A and B with exhaust in red

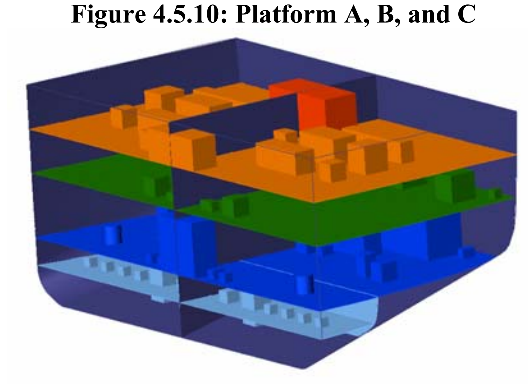

Figure 4.5.11: Platform A, B, C, and D

The pods are located 10 meters from centerline and approximately 236.5 meters aft of the FP, or 5.5 meters aft of the forward pod room bulkhead. Based on expert opinion and current designs they are tilted 7 degrees down to help align the pods with the water streamlines while not exceeding extreme angles. The lowest point of the propulsor is the forward propeller that has a clearance of 2.95 meters above the baseline. This clearance provides added survivability in the event of a grounding situation. The pod location can be seen in Figures 4.5.3 and 4.5.12.

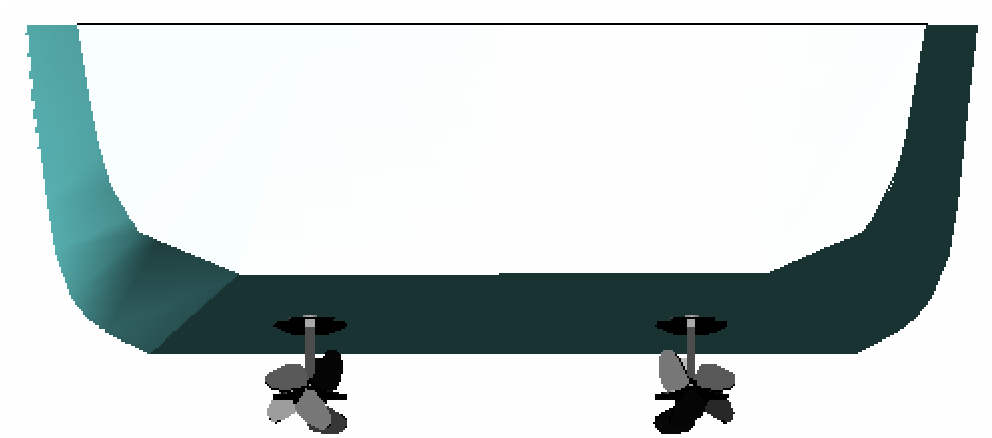

Figure 4.5.12: Stern View of Pods Mounted On Hull

4.5.3 Deckhouse

The deckhouse is designed using calculations found in the Manning and Deckhouse Volume section of Appendix B, page B4 and expert opinion. The deckhouse starts at 204 meters aft of the FP, which is the beginning of the machinery box, and has a length of 22 meters. It is 42.55 meters wide, which leaves a 4 meter walkway on each side of the deckhouse. The height of the deckhouse is determined using USCG regulations for visibility for cargo carrying vessels. These regulations specify that the navigation height must allow visibility of a length 500 meters forward of the FP of the vessel. To meet this requirement the deckhouse must have a minimum height of 14 meters. In order to have five equal deck heights, a deck height of 3 meters is chosen thus giving an overall deckhouse height of 15 meters. This gives an overall navigation height of 31 meters from the waterline. To increase visibility, the navigation bridge, Deck E, extends the full breadth of the hull, 50.55 meters.

The inlet/exhaust casing extends 3m above the height of the deckhouse to ensure that the gases do not blow back into the deckhouse. The overall shape of the deckhouse is rectangular to maximize producibility. A 3D view of the deckhouse is shown in Figure 4.5.13.

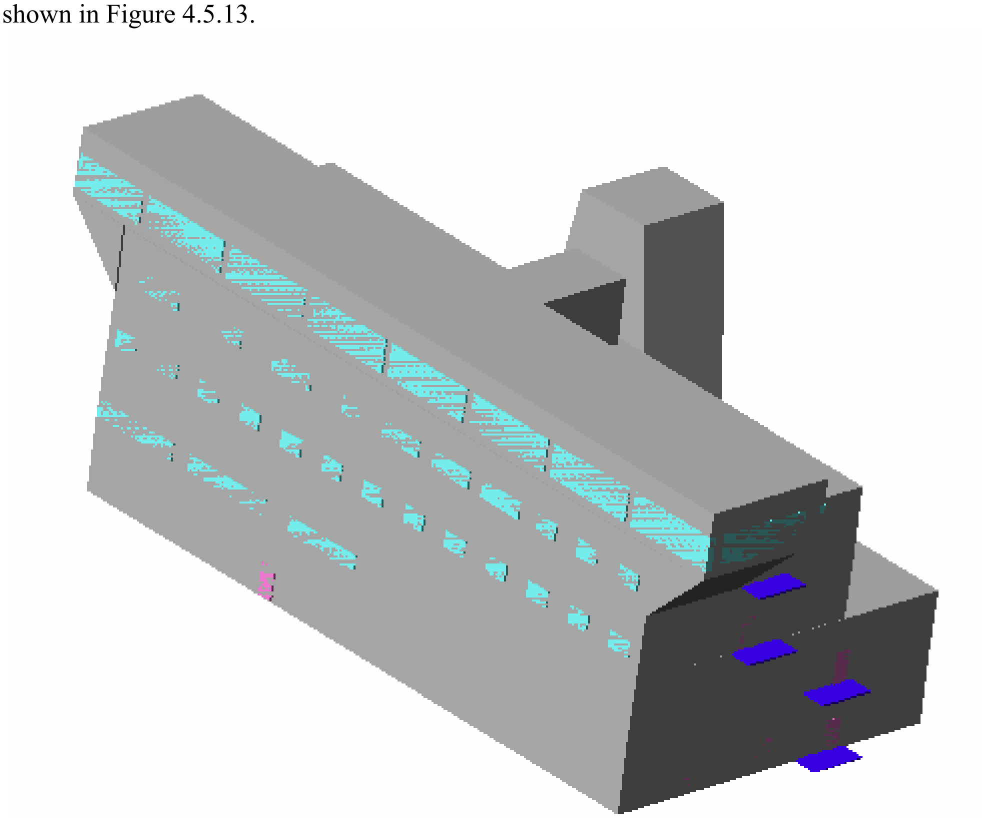

Figure 4.5.13: 3D Deckhouse

The deckhouse is separated into two sections. The aft portion of the deckhouse contains machinery spaces and is located on Deck A and B with the inlet/exhaust casing extending above Deck E. The forward portion of the deckhouse contains all crew living spaces, workshops, stores and the navigation bridge. It is composed of five decks, A through E. The two portions of the deckhouse are joined together to increase producibility and decrease structural weight and cost. There is a central passageway on Deck A and B to connect the two portions of the deckhouse. There is also a main ladder and an elevator that connect Deck A through E. There are also two sets of external ladders, one port and one starboard, connecting Deck A through D. On the aft portion of the deckhouse, one more external ladder connects Deck A and B. In Figure 4.5.13, the blue landings indicate where the external ladders allow entrance into the deckhouse.

The aft portion of the deckhouse contains machinery spaces. Deck A includes the lower inert gas room, the fan room which includes the inlet/exhaust casing, two shore power connection rooms and the CO_2 room. Deck B contains the upper inert gas room, the fan room with casing, the foam room, the emergency generator room and the garbage room. The foam room is positioned on Deck B so if a fire did occur the foam could easily be sprayed down on either deck. The garbage room is positioned next to the galley to allow for easy waste removal. The internal layouts for Deck A and B are shown in Figures 4.5.14 and 4.5.15.

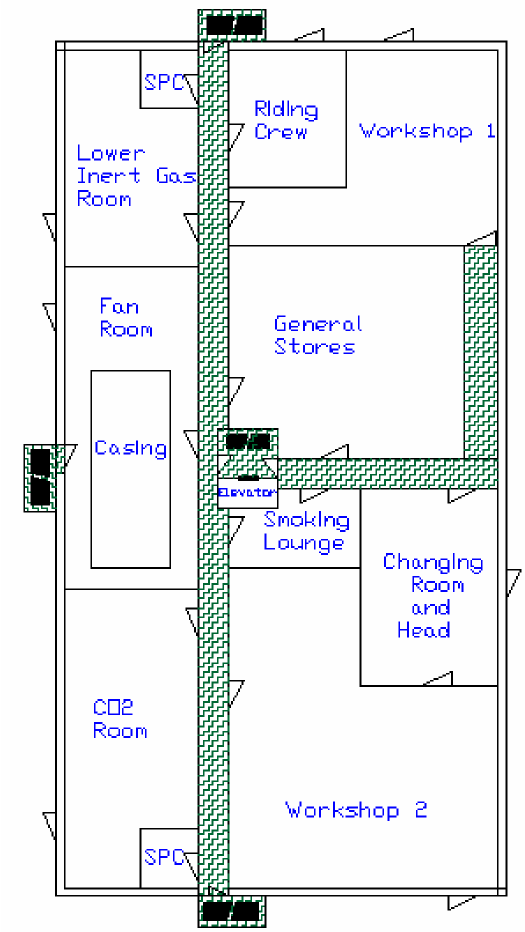

Figure 4.5.14: Deck A

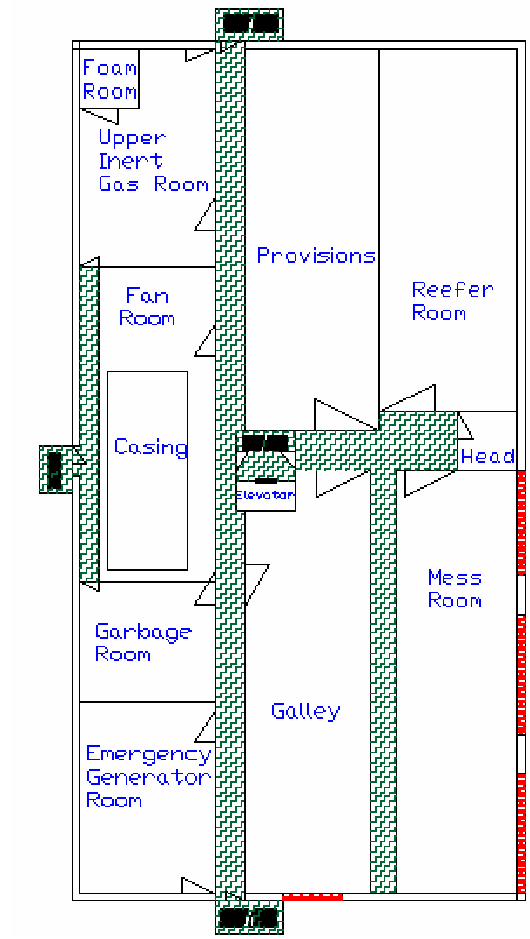

Figure 4.5.15: Deck B

The forward portion of Deck A contains the riding crew room, two workshops, general stores, and a changing room with a head. Deck B includes the galley, mess room, refrigerator room, provisions and a head. The galley and mess room are connected via a passageway to allow for easy food service and handling. The reefer room and provisions room are also easily accessible to the galley via passageways. A complete listing of the deckhouse areas for Deck A and B is shown in Table 4.5.5.

Table 4.5.5: Deck A and B Layout Areas

AFT PORTION OF DECKHOUSE	
DECK A	**AREA (m^2)**
Lower Inert Gas Room	66.2
Shore Power Connection Room (STB)	9
Shore Power Connection Room (Port)	9
Fan Room	71.8
Inlet / Exhaust Casing	40
CO2 Room	95
DECK B	
Foam Room	9
Upper Inert Gas Room	75.2
Fan Room	69
Inlet / Exhaust Casing	40
Garbage Room	41
Emergency Generator Room	66

FORWARD PORTION OF DECKHOUSE	
DECK A	**AREA (m^2)**
Riding Crew Room	42
Workshop 1	93.7
General Stores	126.5
Smoking Lounge	24.2
Changing Room and Head	70
Workshop 2	180.8
Elevator / Stairs	12
Total Passageways	96.8
DECK B	
Galley	128.4
Mess Room	130
Referigator Room	126.8
Provisions	129.9
Head	9
Elevator / Stairs	12
Total Passageways	100

Deck C contains 23 staterooms, each with an individual head. Figure 4.5.16 shows an individual stateroom arrangement. There is also a linen locker, luggage locker and lounge located on Deck C. Deck D includes six staterooms, the laundry, the gym and the locker room. The LAN room, cargo control room, hospital and medical supplies, training library and conference room are also located on Deck D. The cargo control room is located in the forward, central portion of Deck D to allow for easy visibility over the cargo deck. The Master's and Chief Engineer's stateroom and office are also on Deck D but are separated from the other crew spaces by the conference room and training library. Deck E is the navigation bridge and also contains the map room, the Master's dayroom and a head. There are large windows all around Deck E to increase visibility in all directions. A complete listing of the areas for Deck C, D and E is shown in Table 4.5.6. Figures 4.5.17, 4.5.18 and 4.5.19 show the layouts of Deck C, D, and E respectively.

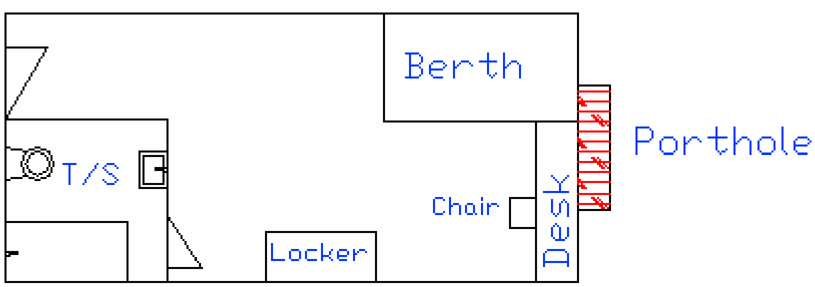

Figure: 4.5.16 Stateroom Arrangement

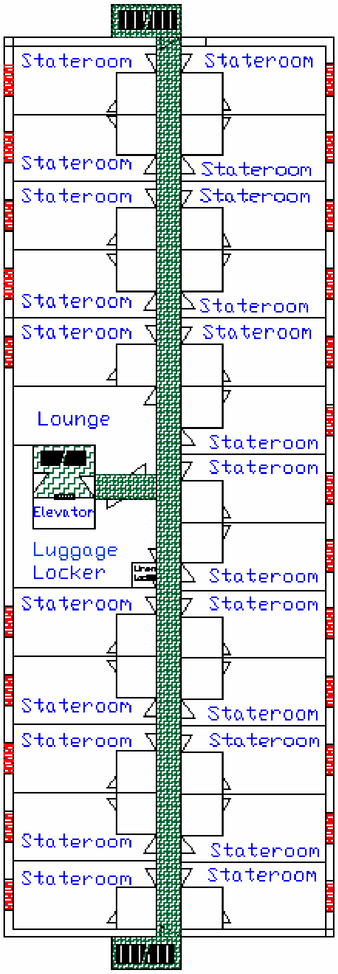

Figure 4.5.17 Deck C

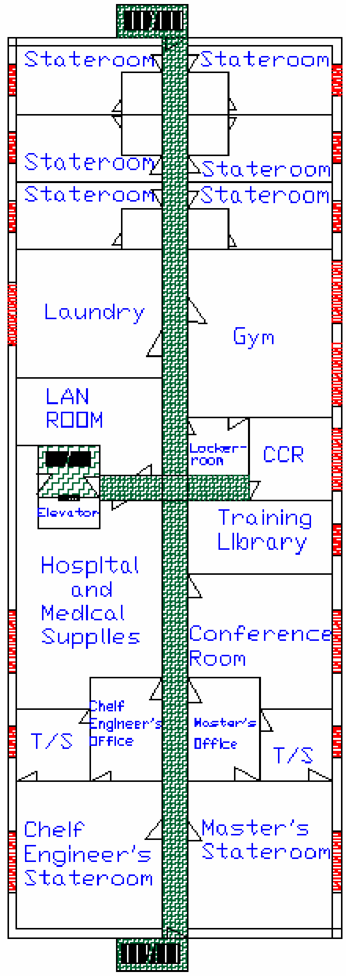

Figure 4.5.18: Deck D

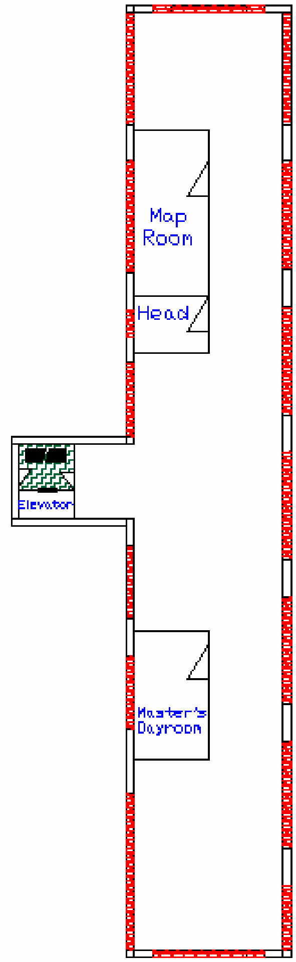

Figure 4.5.19: Deck E

Table 4.5.6: Areas for Deck C, D and E

FORWARD PORTION OF DECKHOUSE	
DECK C	AREA (m^2)
23 Staterooms	23
Stateroom Head	4
Linen Locker	1.6
Luggage Locker	28.5
Lounge	24.5
Elevator / Stairs	12
Total Passageways	50.3
DECK D	
6 Staterooms	23
Stateroom Head	4
Laundry	43
LAN Room	27.4
Hospital and Medical Supplies	63.5
Gym	56
Lockerroom	8.5
Cargo Control Room	16
Training Library	24.9
Conference Room	40.3
Master's Stateroom	50
Master's Head	12.3
Master's Office	17.5
Chief Engineer's Stateroom	50
Chief Engineer's Head	12.3
Chief Engineer's Office	17.5
Elevator / Stairs	12
Total Passageways	56.7
DECK E	
Bridge	238.6
Map Room	36
Master's Dayroom	28
Head	12
Elevator / Stairs	12

4.6 Mission Systems

4.6.1 Bow Loading System

To perform cargo-loading operations at the Hibernia Offshore Loading System (OLS), shown in Figure 4.6.1, the ship requires a bow-loading system. The bow-loading system, seen in Figure 4.6.2, includes an extendable nine meter, five tonne hose crane that assists in the lifting of the OLS flexible loading hose. The OLS hose is then attached to the hose winch and bow-loading manifold. To retrieve the flexible loading hose the crane operator is assisted by a Hibernia support vessel. The bow manifold is equipped with a twenty-inch inlet and booster pump that assists in maintaining required loading pressure. From the bow-loading manifold, the cargo oil flows through a riser through the forecastle deck to the main cargo piping and is distributed to each cargo tank.

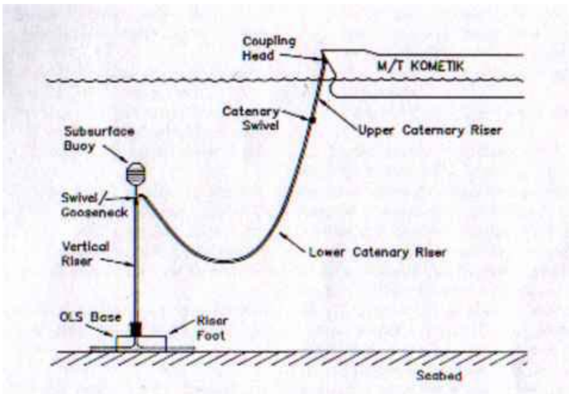

Figure 4.6.1: Offshore Loading System

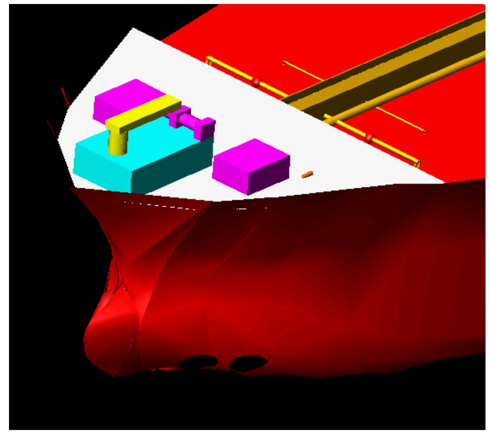

Figure 4.6.2: Bow Loading System

4.6.2 Cargo System

From the bow-loading manifold, the cargo oil is distributed to each cargo tank through one-meter diameter cargo piping. The cargo system has six cargo subdivisions yielding a total of twelve cargo tanks and two slop tanks arranged symmetrically about the centerline. The tanks are capable of being filled individually or simultaneously through a series of stop valves on the piping. A schematic for the cargo-oil system is provided in Drawing 6. The two midship deck manifolds connect with two risers that drop to two cargo mains located port and starboard. The system piping is designed in a circular crossover manner so any tank can be loaded or unloaded from any of the three manifolds.

During an off-loading procedure, two segregated suction mains remove the cargo oil from the tanks. Two 6000 m^3/hr cargo pumps are attached to the suction mains, which are designed in a circular crossover manner for redundancy. Either pump is capable of removing oil from any tank in the event of pump failure. To reduce the risk of deck spills the cargo piping is all located within the cargo-oil tanks. In the event of piping damage, no oil is discharged into the sea. Figure 4.6.3 shows an isometric representation of the Cargo-Oil system including Cargo Stripping and Crude Oil Washing systems.

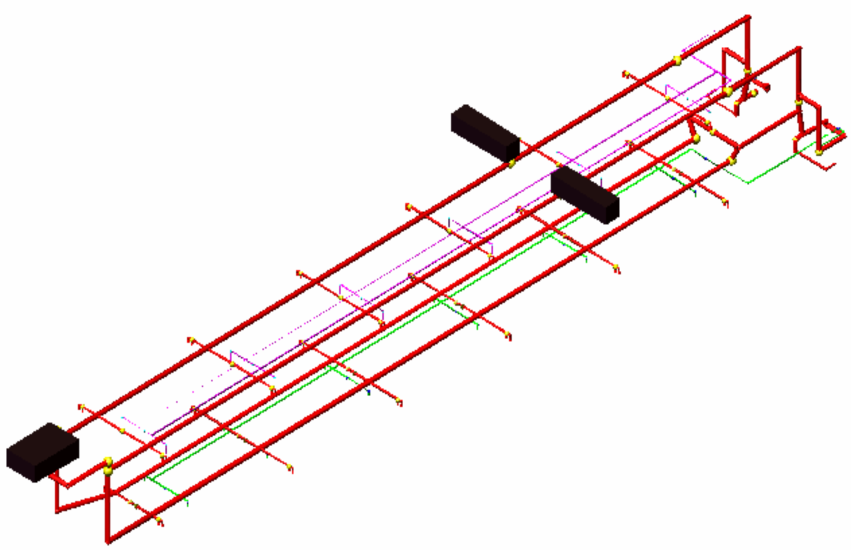

Figure 4.6.3: Isometric Representation of Cargo System

4.6.3 Crude Oil Stripping System

When the oil content within each tank becomes low, the cargo pumps will begin to intake air causing possible damage to the pumps. The Cargo Stripping system is comprised of a smaller 300 m^3/hr Cargo Stripping Pump (CSP) and piping system that is capable of removing the remaining oil from each tank. The removed oil is discharged through the deck manifolds. The Cargo Stripping Pump and system is also capable of removing oil-water mixtures after Crude Oil Washing operations.

4.6.4 Crude Oil Washing System

As per IMO regulations, the Shuttle Tanker is required to have a Crude Oil Washing (COW) system. The COW system is comprised of rotating nozzles capable of reaching above ninety percent of internal tank structure, piping, and a 1000 m^3/hr COW pump. The COW pump enables cleaning of the tanks to be independent of the cargo and ballast systems.

4.6.5 Inert Gas System

As per IMO regulations, the Shuttle Tanker is also required to have an Inert Gas System (IGS). The inert gas system maintains a non-explosive environment inside the cargo tanks of the ship. The space inside the tanks not filled by cargo is filled with a vapor displacing inert gas. The inert gas system strips gases from the exhaust manifold. The exhaust gases are treated in a scrubber unit to remove any SO_X, NO_X, and particulate matter. The remaining inert gases are blown through a deck water seal and then through designated piping to each tank. The water seal prevents any sparks from entering any of the possibly explosive tanks. A schematic of the inert gas system is provided in Appendix H. Figure 4.6.4 shows an isometric rendering of the inert gas system.

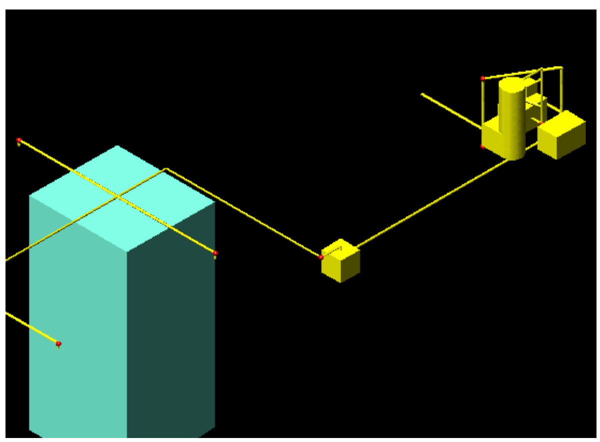

Figure 4.6.4: Inert Gas System

4.6.6 Ballast System

The ballast system services the forepeak trim/ballast, the aft peak trim/ballast and twelve "J" shaped ballast tanks. The system is comprised of two 2500m^3/hr. ballast pumps and designated piping. The system intakes and discharges seawater from two sea chests located below the pump room on the port and starboard sides of the ship. Oil control monitors test the effluent of the ballast system to ensure no oil is discharged into the sea. A schematic of the ballast system is provided in Drawing 6. Figure 4.6.5 shows an isometric rendering of the ballast system.

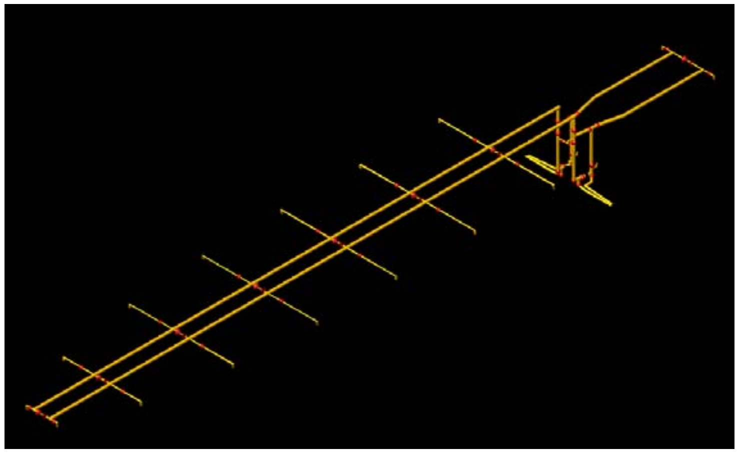

Figure 4.6.5: Ballast System

4.7 Personnel

The Shuttle Tanker crew is divided into two main departments: deck and engineering. Deck officers include the Master, Chief Mate, Second Mate and Third Mate. Engineering officers include Chief, First Assistant, Second Assistant and Third Assistant Engineers. Both departments must comply with the Oil Pollution Act of 1990 (OPA 90) work regulations. These regulations dictate no crewmember will work more than 15 hours in a 24 hour period and no more than 36 hours in a 72 hour period except during emergencies or drills. Work regulations are in place to protect against collisions, grounding and other accidents that could result in an oil spill. Limitations make it necessary for senior officers to delegate responsibility, which in turn requires a competent crew.

Manning levels are determined by the Canadian Shipping Act (CSA). Size of the ship, route and trade characteristics, levels of ship automation and propulsion type determine minimum levels of manning. Based on this information, equations found in the Manning and Deckhouse Volume section of Appendix B, page B4, are used to calculate a personnel level of 28 with an allowance for three additional crewmembers. Table 4.7.1 shows a breakdown of the crew for the Shuttle Tanker. Every Master, First Mate, Chief Engineer and Second Engineer must hold an Oil Tanker, Level 2 Certificate in accordance with CSA Part 1, Division 5. In addition, any crewmember assigned specific oil operation duties must have an Oil Tanker Level 1 Certificate and assistants to these persons need a Proficiency in Oil Tankers Certificate.

Table 4.7.1: Personnel Levels

Department	Rank	Shuttle Tanker
Deck	Master	1
	Deck Officers	3
	Radio Officer	1
	Seamen	7
Engine	Chief Engineer	1
	Engineer Officers	3
	Technicians	4
	Uncertified	3
Steward	Cooks	5
	Total	**28**

A Master's primary roles are to be the ship's commander, chief pilot/navigator and to be responsible for ship operations throughout the voyage. The Master manages personnel and is the primary contact with company representatives and port authorities. The Master also monitors ship progress and navigation, over-seeing safe cargo loading, discharge and ballast operations. Conducting maneuvering while entering and exiting port, supervising the radio officer, monitoring crew safety and health as well as union or legal concerns of the crew are all responsibilities of the Master of the ship.

The Chief Mate is the deck department manager and the ship's cargo officer, for which an additional Supervisor of an Oil Transfer Operation Certificate is required. The cargo handling responsibilities include insuring safe handling, containment and transportation of the cargo by directing crew and delegating duties. The Chief Mate is responsible for the vessel should the Master be absent or incapacitated. Supervision of deck maintenance, tank cleaning and preventive maintenance as well as administration duties regarding logs and company forms are included in the job description.

The Second and Third Mate are both ship watchstanders. The Second Mate is the ship's navigation officer and maintains a full chart inventory, assists the Master in the wheelhouse and supervises all uncertified crew. The Third Mate is the ship's safety officer maintaining all lifesaving safety equipment. This officer assists the Chief Mate in cargo handling and administrative duties.

A Radio Officer maintains communications at port and sea and is responsible for maintenance and repairs to the electronic equipment and navigation system. Seamen on the ship are responsible for cargo and line handling on the deck, deck machinery operation, and mooring and anchoring duties. Cooks are required for meal preparation and maintaining the galley and mess areas.

The Chief Engineer is the director of the Engine Department and the primary engineer on board. The responsibilities include the overall management, supervision and efficient operation and maintenance of the engine room. This officer establishes maintenance schedules, serves as the ship's technical expert, plans and directs department operations and ensures compliance with all safety requirements and environmental regulations. The Chief Engineer reports to the Master on the condition of the engine spaces and power supplies, coordinates with the Chief Mate for ballast and fuel oil transfer requirements and for engineering support during tank cleaning, deck repairs and maintenance. Maintaining records of repairs, expenditures and fuel use is also the Chief Engineer's responsibility.

The First Assistant Engineer's primary role is the implementation of the Engine Department maintenance as directed by the Chief Engineer in a safe and timely manner. The First Assistant is responsible to the Chief Engineer for maintenance, administration, supervision and safe operation of the Engine Department. Should the Chief Engineer be absent or incapacitated, the First Assistant assumes responsibility of the engine space. This officer supervises, schedules and assesses the work assignments for all uncertified personnel. Responsibilities also include supervising engine start up and inert gas systems, maintaining the machinery control system and the Engine Department shop repair and storage areas.

The Second Assistant Engineer operates the boiler systems and the diesel fuel/fuel oil systems. This officer administers and supervises watchstanding and assists the Chief Engineer in taking on bunker fuel while in port. The Third Assistant Engineer maintains the ship's electrical, lube oil, sanitary systems and distilling plant. This officer also stands watch in the engine space if needed. In addition, there are four technicians and three uncertified crewmembers to assist in maintenance, operation and repair.

4.8 Weights and Loading

4.8.1 Weights

After the arrangements of all permanent equipment in the Shuttle Tanker are made, the lightship center of gravity is found to check stability and trim calculations. To do this, weights of all equipment are found from manufacturer catalogs and expert opinion. These weights are divided into SWBS groups and shown in table 4.8.1. Measurements of distances from baselines to each piece of equipment's center of gravity are then taken. Measurements for the longitudinal and vertical centers of gravity (LCG, VCG) are taken from the forward perpendicular and keel respectively. For the transverse center of gravity (TCG), measurements are taken from the centerline with starboard positive and port negative. The center of gravity for the bare hull is estimated to be 6.5 percent of the overall length aft of midships. These centers are each multiplied by their respective weights and summed in a table shown in Appendix I. The total LCG and VCG are divided by the total weight to find the center of gravity of the lightship.

Table 4.8.1: Lightship Weight Summary

SWBS Group	Weight (MT)	VCG (m)	LCG (m)
100	26560	8.661	146.2
200	1513	9.676	216.9
300	386.5	17.33	217.7
400	7.8	35.92	208.4
500	5344	14.25	209.4
Total	33810	9.696	160.2

4.8.2 Loading

Using the lightship center of gravity, centers of gravity for full load and ballast conditions are calculated. Full load assumes all cargo tanks filled to capacity, fuel oil tanks filled to 98% of their total capacity and full potable water tanks. The individual weights are shown in Table 4.8.2 and the trim summary is in Table 4.8.3.

Table 4.8.2: Weight Summary: Full Load Condition

Item	Weight (MT)	VCG (m-BL)	LCG (m-FP)
Light Ship	33807	9.696	160.2
Cargo Oil	127000	14.61	111.1
Fuel Oil	1,673	14.80	196.5
Fresh Water	245	20.59	242.6
SW Ballast	0	0	126.4
Totals	162700	13.60	122.4

Table 4.8.3: Trim Summary: Full Load Condition

Item	
FP Draft	15.08 m
AP Draft	15.12 m
LCF Draft	15.10 m
LCB (even keel)	122.3 m-Aft
LCF	132.6 m-Aft
MT1cm	2,180 m-MT/cm
Trim	0.041 m-Aft
Prop Immersion	238%
List	0 deg

Ballast condition calculations are performed with cargo tanks empty, fuel oil tanks filled to 10% of capacity, and potable water tanks 50% full. The weight breakdown is shown in Table 4.8.4 and the trim summary in Table 4.8.5. Stability of the Shuttle Tanker is calculated at all three conditions and is described in Section 4.9.

Table 4.8.4: Weight Summary: Ballast Condition

Item	Weight (MT)	VCG (m-BL)	LCG (m-FP)
Light Ship	33807	9.696	160.2
Cargo Oil	0	0.000	126.385
Fuel Oil	171	14.796	196.470
Fresh Water	122	20.589	242.591
SW Ballast	57,409	8.838	90.905
Totals	91,509	9.182	116.912

Table 4.8.5: Trim Summary: Ballast Condition Stability

Item	
FP Draft	8.953 m
AP Draft	9.003 m
LCF Draft	8.979 m
LCB (even keel)	116.81 m-Aft
LCF	123.158 m-Aft
MT1cm	1,726 m-MT/cm
Trim	0.054 m-Aft
Prop Immersion	110%
List	0.00 deg

4.9 Stability

4.9.1 General

Intact and damage stability for the concept design Shuttle Tanker are analyzed using the HecSalv software program. Based on previously calculated Hydrostatic Curves, Cross Curves and Bonjean Curves shown in Section 4.1, various loading conditions in both intact and damaged cases are analyzed. Intact and damage stability are analyzed for full load and ballast loading conditions. All criteria is met for both intact and damage stability.

4.9.2 Intact Stability

In full load and ballast loading conditions, a stability summary is calculated along with a graphical representation of the static righting arm. The static stability curves are compared to the requirement for oil tankers greater than 5,000 DWT from MARPOL 73/78 Annex 1, Regulation 25A.

For the intact stability to be satisfactory, many conditions must be met. In port, GM corrected must be greater than 0.15 m without the use of operational methods in all loading or unloading conditions. Due to loading at sea, GM must also be greater than 0.15 m while loading at sea. At sea, the area under the GZ curve up to 30 degrees must be greater than 0.055 m-rad. In addition, the area up to 40 degrees must be greater than 0.09 m-rad and between 30 and 40 degrees the area must be equal to 0.03 m-rad. The GZ curve must at least reach 0.2 m above 30 degrees and the maximum GZ value must occur at an angle greater than 25 degrees.

For the full load condition, 127,000 MT of cargo oil with a density of 0.853 MT/m^3 is loaded into the cargo tanks. This and other weights can be seen in Section 4.8.2. The stability summary for the full load condition is shown in Table 4.9.1.

Table 4.9.1: Stability Summary at Full Load Condition

Item	(m)
KMt	22.125
VCG	13.597
GMt	8.528
F.S. Correction	1.006
GMt Corrected	7.522

All requirements for the full load condition are met as seen in Figure 4.9.1.

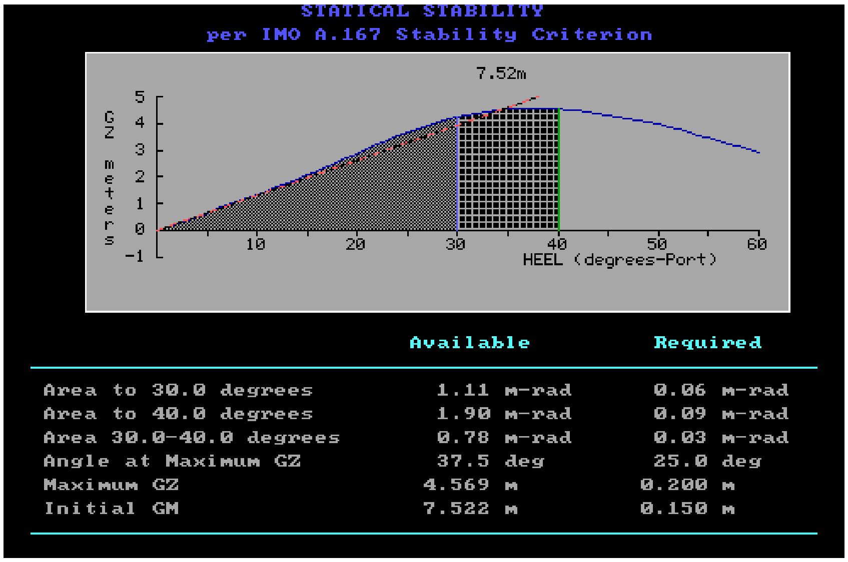

Figure 4.9.1: Righting Arm Curve with Required Values for Full Load

For the ballast condition, 57,400 MT of seawater is loaded into the water ballast tanks. This and other weights can again be seen in Section 4.8.2. The stability summary for the ballast condition is shown in Table 4.9.2.

Table 4.9.2: Stability Summary in Ballast

Item	(m)
KMt	28.165
VCG	9.182
GMt	18.983
F.S. Correction	3.111
GMt Corrected	15.872

All requirements for the ballast condition are met as seen in Figure 4.9.2.

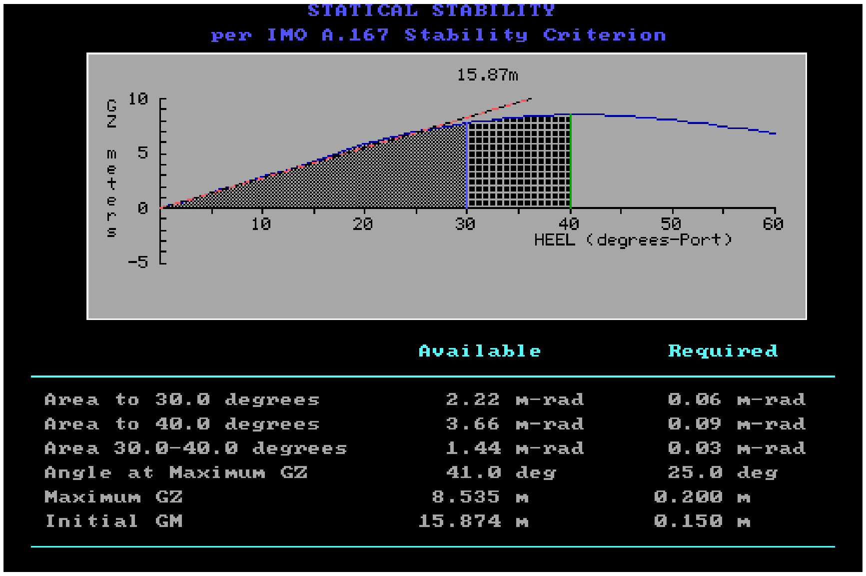

Figure 4.9.2: Righting Arm Curve with Required Values in Ballast

4.9.3 Damage Stability

The full load condition and the ballast condition are checked for damage stability. Extents of damage areas were calculated using the MARPOL 73/78 - Annex I - Regulations for the Prevention of Pollution by Oil (Regulation 25, Section 2) and are found in Table 4.9.3. To analyze side damage, the calculated damage area was applied longitudinally along the ship such that the maximum number of tanks are damaged. This is done from the forward perpendicular to the aft perpendicular such that all possible damage combinations are considered. This gives a total of eight damage cases for each loading condition, with three critical cases; one for the ballast load and two for full load.

Table 4.9.3: MARPOL 73/78 Annex I Regulations for the Prevention of Pollution by Oil

Extent	Side Damage	
Longitudinal	$1/3\ L^{2/3}$ or 14.5 m; whichever is less (13.33)	
Transverse	B/5 or 11.5 meters; whichever is less (10.11)	
Vertical	From molded bottom at centerline upwards with-out limit	
	Bottom Damage	
	.3L from FP	Any Other Part
Longitudinal	$1/3\ L^{2/3}$ or 14.5 m; whichever is less (13.33)	$1/3\ L^{2/3}$ or 5 m; whichever is less (5)
Transverse	B/6 or 11.5 meters; whichever is less (8.425)	B/6 or 5 meters; whichever is less (5)
Vertical	B/15 or 6; whichever is less (3.16)	B/15 or 6; whichever is less (3.16)

In addition to this damage analysis, an oil outflow analysis is done using a simplified probabilistic approach. This method corresponds with MARPOL 73/78 - Annex I - Appendix 8 - Approval of Alternative Methods of Design and Construction. Although not currently required, this method is analyzed in anticipation of future requirements. For this method, the probability of zero outflow and mean oil outflow are calculated for every compartment and group of compartments possible. These individual values are used to find the probability of zero outflow and mean outflow for the whole ship. A simplified pollution prevention index, which is required to be greater than or equal to one, is calculated using these values and reference values for a 150,000 dwt ship. These calculations yield a pollution prevention index of 1.23 and are found in the Oil Outflow section, Appendix B, Page B21.

The ballast condition results are shown in Table 4.9.4. These results are for damage on the starboard side in each case. The worst case is the condition in which the aft section of the ship, specifically number six cargo and ballast tanks, slop tank, cofferdam and engine room, are damaged. Figures 4.9.3 and 4.9.4 correspond to this worst-case condition. This case yields a heel of 4.7 degrees, a maximum GZ of 7.912 m, a maximum GZ angle of 41.4 degrees, a trim of 12.47 m down by the stern and a maximum bending moment of 523,979 m-MT in hog. The Shuttle Tanker meets all stability requirements in all ballast damage conditions considered.

Table 4.9.4: Ballast Damage Results

		Intact	Forpeak, Thrust, #1	#1, #2 Cargo and Ballast	#2, #3 Cargo and Ballast	#3,#4 Cargo and Ballast	#4, #5 Cargo and Ballast	#5, #6 Cargo and Ballast	#6 Cargo/Ballast Slop, Coffer, Engine	Engine, Pod, Fresh Water, Aftpeak
Draft @ FP	(m)	8.955	7.732	8.837	10.27	10.09	9.31	8.84	7.726	8.444
Draft @ AP	(m)	9.003	9.629	9.225	8.72	8.994	9.756	10.96	12.47	10.05
Trim	(m)	0.048A	1.897A	0.838A	1.544F	1.096F	0.446A	2.117A	4.744A	1.606A
Static Heel	(deg)	0	1.5P	2.7P	1.2S	1.5S	1.4S	3.5S	4.7S	1.4S
Total Weight	(MT)	91510	88340	89900	97700	98210	58090	102100	104800	94750
GMt	(m)	15.87	20.3	17.79	16.1	16.43	17.92	15.7	13.8	17.11
Max GZ	(m)		8.424	7.093	8.187	8.469	8.809	8.997	7.912	8.008
Max GZ Angle	(deg)		40.9P	40.8P	42.3S	42.1S	41.8S	42.6S	41.4S	40.6S
GZ Pos. Range	(deg)		>58.5	>57.3	>58.8	>58.5	>58.6	>56.5	>55.3	>58.6
Outflow	(MT)		0	0	0	0	0	0	0	0
Flooded Water	(MT)		5304	10340	14880	15080	14980	17790	17690	3344
Shear Force	(MT)		6596	6962	7327	6975	6004	-5401	7438	9071
Bending Moment	m-MT		378900H	414700H	430500H	393600H	325800H	361200H	524000H	508200H

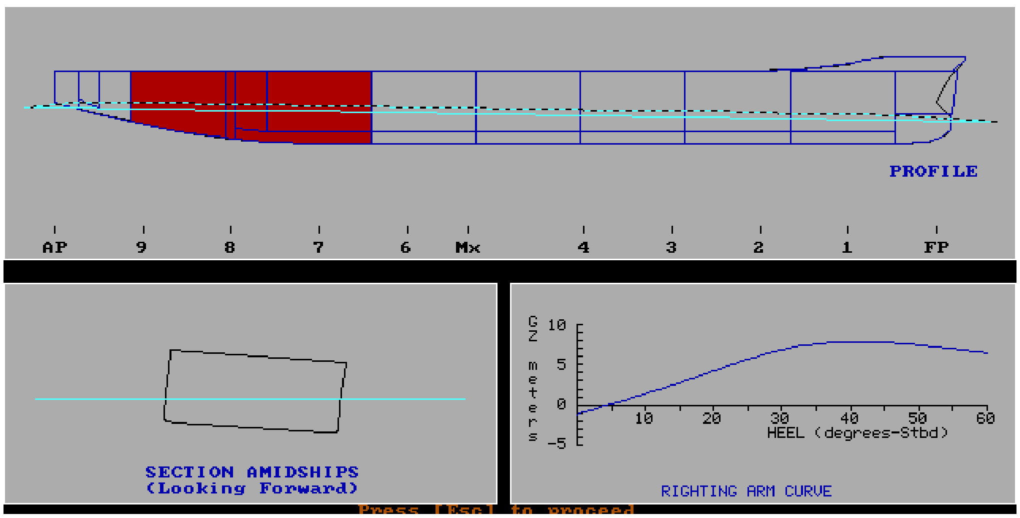

Figure 4.9.3: Ballast Damage Summary

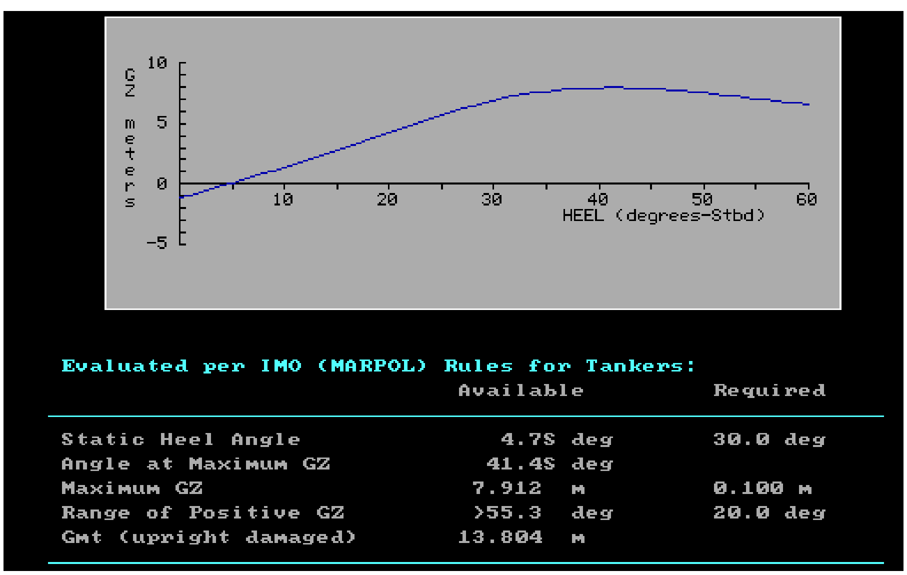

Figure 4.9.4: Ballast Damage Righting Arm Curve

The full load condition results are shown in Table 4.9.5. For this condition, there are two worst-case scenarios. The worst stability case is the condition in which the aft section of the ship, specifically number six cargo and ballast tanks, slop tank, cofferdam and engine room are damaged. Figures 4.9.5 and 4.9.6 correspond to this case. This case yields a heel of 8.7 degrees, a maximum GZ of 2.634 m, a maximum GZ angle of 40.4 degrees, a trim of 5.352 m down by the stern and a maximum bending moment of 187,001 m-MT in sag. The oil outflow on this case is 13,361 MT. The other worst-case scenario is the case in which maximum oil outflow occurs, and is shown in Figures 4.9.7 and 4.9.8. This occurs when the number 4 and number 5 cargo and ballast tanks are damaged. Oil outflow in this case is 22,002 MT and stability conditions are not an issue. For all full load damage cases considered, the Shuttle tanker exceeds all requirements.

Table 4.9.5: Full Load Damage Results

	Intact	Forpeak, Thrust, #1	#1, #2 Cargo and Ballast	#2, #3 Cargo and Ballast	#3,#4 Cargo and Ballast	#4, #5 Cargo and Ballast	#5, #6 Cargo and Ballast	#6 Cargo/Ballast Slop, Coffer, Engine	Engine, Pod, Fresh Water, Aftpeak
Draft @ FP (m)	15.1	16.99	16.57	15.77	15.43	15.25	14.99	13.63	13.31
Draft @ AP (m)	15.11	14.12	14.47	14.52	15.13	15.29	15.89	18.98	13.66
Trim (m)	.037A	2.812F	2.1F	0.849F	.299F	.033A	.89A	5.352A	5.348A
Static Heel (deg)	0	2.1S	4.1S	3.6S	3.2S	3.3S	4.4S	8.7S	6.2S
Total Weight (MT)	2E+05	168000	167600	165700	165100	165100	167400	179200	175400
GMt (m)	7.522	7.72	7.521	7.308	7.425	7.466	7.37	6.506	6.373
Max GZ (m)		4.117	3.828	3.979	4.103	4.108	3.856	2.634	2.802
Max GZ Angle (deg)		37.7S	38.9S	39.1S	39.2S	39.2S	39.7S	40.4S	38.1S
GZ Pos. Range (deg)		>57.5	>55.9	>56.4	>56.8	>56.7	>55.3	>51.3	>53.8
Outflow (MT)		6989	17230	21130	21900	22020	21940	13360	0
Flooded Water (MT)		-12280	22090	24130	24270	24390	26630	29370	12760
Shear Force (MT)		-3846	-4409	-5101	-5436	-5730	-6959	-4176	1896
Bending Momen (m-MT)		135300S	195000S	275500S	315300S	327400S	321300S	187000S	103100S

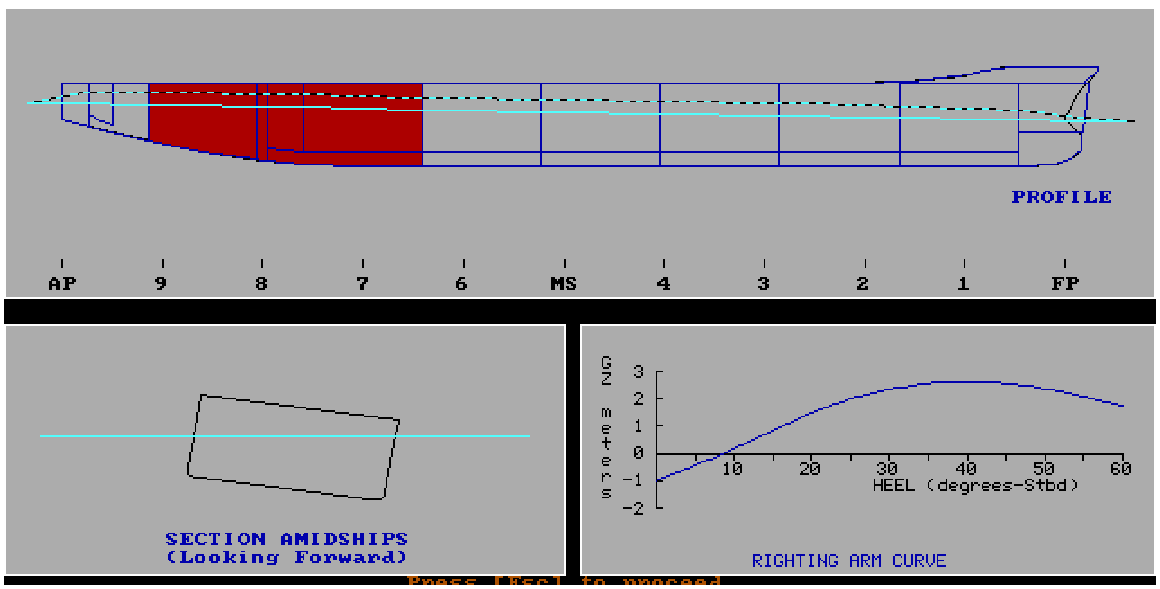

Figure 4.9.5: Full Load Worst Case Summary

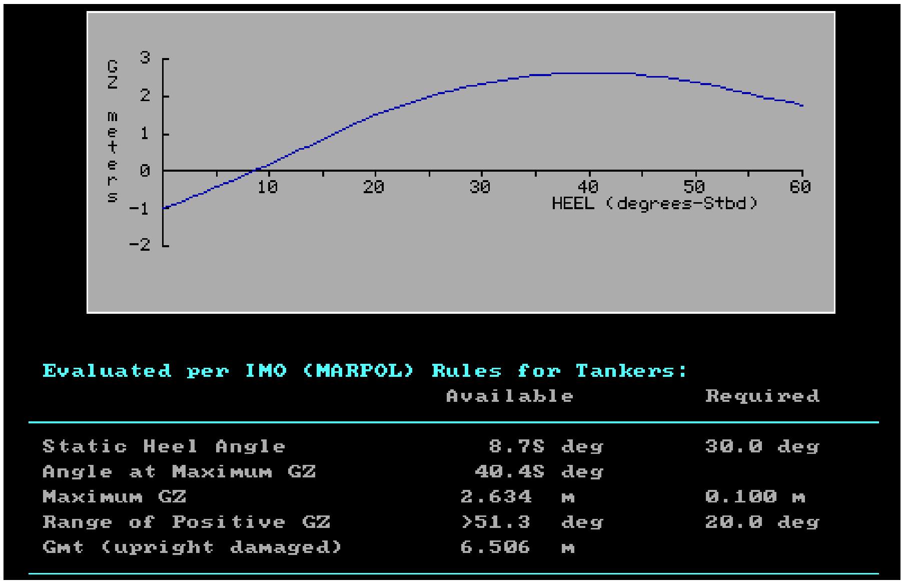

Figure 4.9.6: Full Load Worst Case Righting Arm Curve

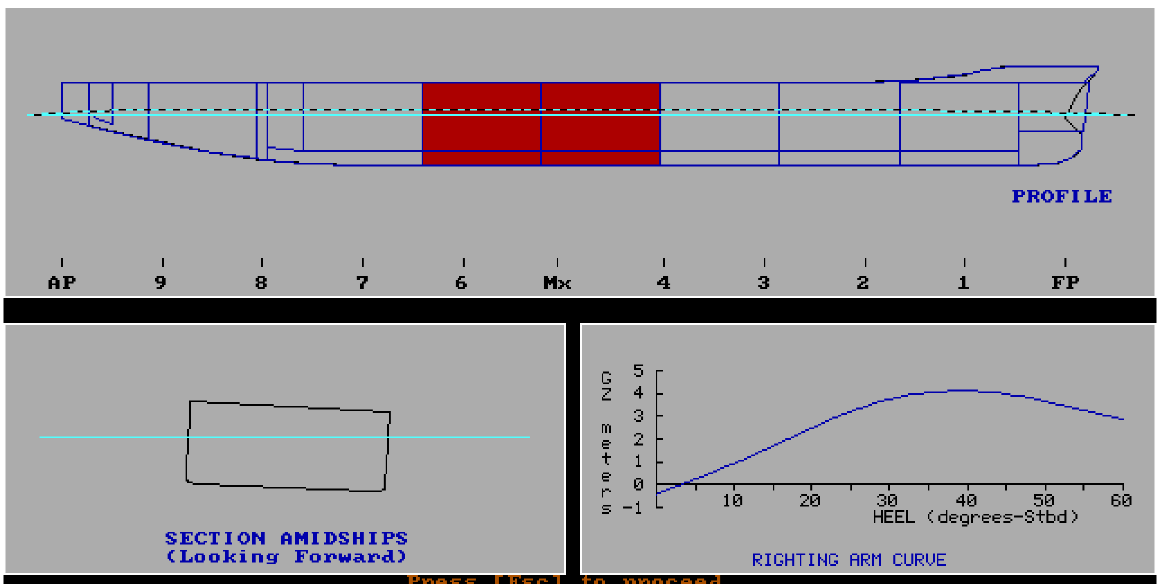

Figure 4.9.7: Full Load Max Outflow Summary

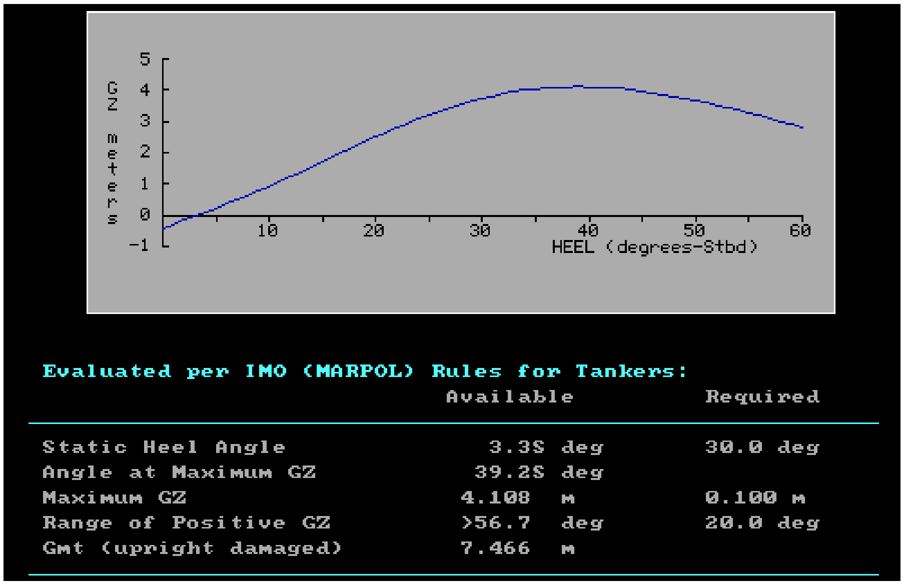

Figure 4.9.8: Full Load Max Outflow Righting Arm Curve

A traditional floodable length curve, shown in Figure 4.9.9, is created for the full load condition. When number one and two ballast and cargo tanks, and when number two and three ballast and cargo tanks, are flooded, the high permeability floodable length curves are intersected. This means the deck is submerged for the given permeability. Due to cargo tanks containing 98% oil, the permeability would be significantly below the levels shown in Figure 4.9.9. Therefore, all flooding conditions are acceptable.

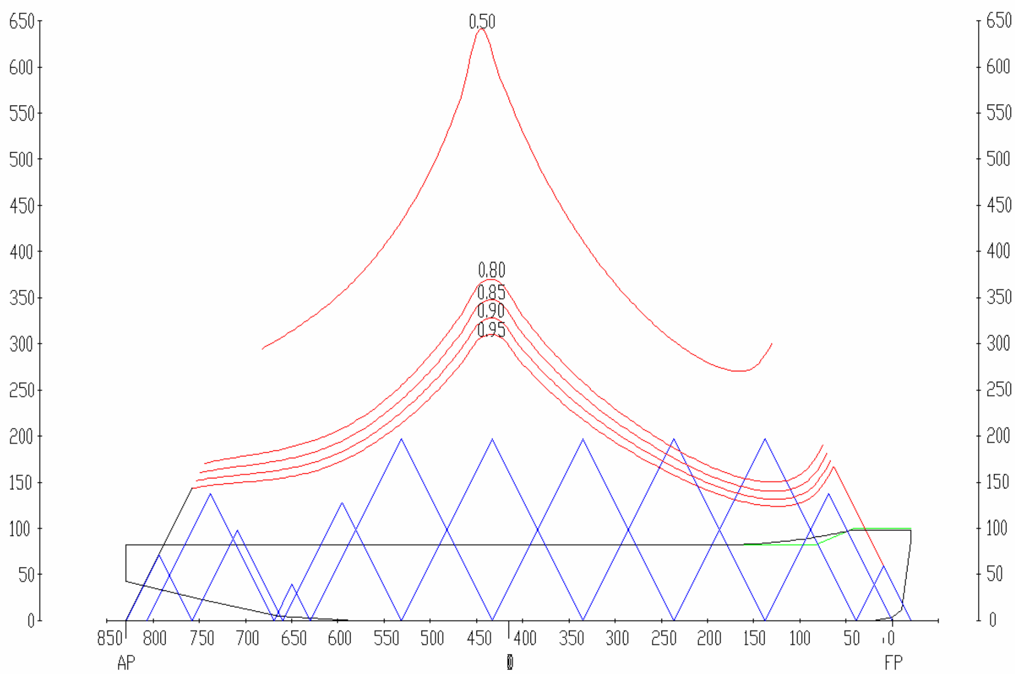

Figure 4.9.9: Full Load Floodable Length Curve

4.10 Seakeeping, Maneuvering and Dynamic Positioning

4.10.1 Seakeeping

The seakeeping analysis is performed by applying strip theory and Lewis forms to the full load condition of the Shuttle Tanker. The full load condition is analyzed because it is the least stable condition. The fully loaded shuttle tanker is divided into eleven stations equally spaced between the forward and aft perpendicular. At each station the breadth, draft and submerged area are determined. These values are used to determine the section area ratios and breadth to draft ratios that determine the Lewis form coefficients. The Lewis form coefficients are used to determine the added mass, added damping, added stiffness, and coupled values for five degrees of freedom; sway, heave, pitch, roll and yaw. With these values, equations of motion for the aforementioned five degrees of freedom are constructed.

The sea keeping analysis for the Shuttle Tanker is performed using a sea state six (significant wave heights of 7.5 meters) Ochi wave spectrum. Ship headings of 0, 45, 90, 135 and 180 degrees are analyzed at three locations on the ship. The locations analyzed are the intersection of the forward perpendicular and the baseline, the intersection of the forward perpendicular and the forecastle deck where the bow-loading manifold is located and at the center of the Navigation Bridge.

At the intersection of the forward perpendicular and the baseline the vessel endurance and sustained speeds are analyzed to determine the number of emergences and slams of the bow. Maximum requirements for the number of slams and emergences of the bow are twenty per hour.

At the location of the bow-loading manifold, the ship is analyzed at a drift speed of one knot to determine loading operational capability. To be able to load oil and utilize the bow-loading system, the vertical motion must not exceed one half the full load draft.

At the center of the Navigation Bridge the Shuttle Tanker's endurance speed is analyzed to determine motion sickness accelerations. The requirement for the motion sickness acceleration is acceleration not exceeding 0.4g's.

By equating the aforementioned equations of motion to the wave amplitude, velocity or acceleration transfer functions and response amplitude operators (RAO) are created for relative velocity, motion or acceleration where applicable. The RAO's for each speed, heading angle, location and response are shown in Appendix J, Figures 1 through 7.

Multiplying the RAO's by the Ochi wave spectrum, shown in Figure 4.10.1, creates response spectra for the relative velocity, motion or acceleration where applicable. The response spectras are shown in Appendix J, Figures 8 through 14. These spectras are used to determine the moments of response for the ship that determine the RMS and characteristic vertical velocity, motion and accelerations.

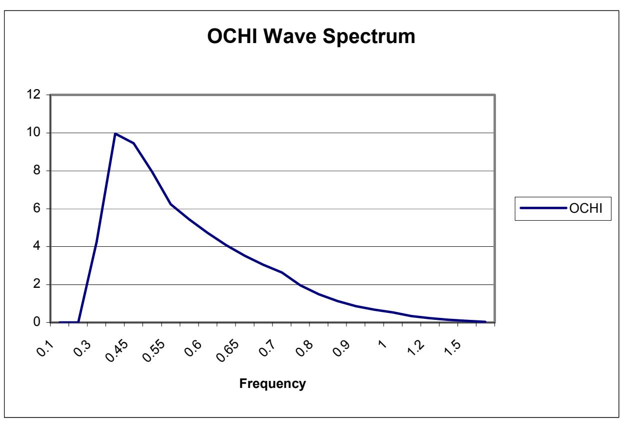

Figure 4.10.1: Sea State Six Ochi Wave Spectrum

The occurrences of emergence and slamming for the bow at both the forward perpendicular locations, the occurrences of exceeded motion for cargo handling at the bow-loading manifold and the motion sickness accelerations on the Navigation Bridge are calculated using this response spectra. Tables 4.10.1 through 4.10.4 provide the results of the sea keeping analysis performed on the Shuttle Tanker. According to the results of the analysis shown in Table 4.10.3, bow-loading manifold operation is not suggested in any following sea condition.

The initial connection of the bow loading system is not recommended in Sea State 6 however if the system is already connected loading may continue in Sea State 6.

Table 4.10.1: Results at Vessel Speed of 15 knots and at Intersection of FP and Baseline

Heading (degrees)	RMS Relative Velocity (m/s)	Characteristic Relative Velocity (m/s)	RMS Relative Motion (m)	Characteristic Relative Motion (m)	# Emergences Per Hour	# Slams Per Hour	Critical Velocity (m/s)	Maximum Allowed Emergences Per Hour	Maximum Allowed Slams Per Hour
0	0.58	2.62	1.998	9.082	4.19	0.286	4.63	20	20
45	0.751	3.396	1.963	8.876	4.335	0.268	4.63	20	20
90	0.839	3.876	1.179	5.362	0.19	8.55E-05	4.63	20	20
135	1.745	8.019	2.494	11.291	7.013	1.252	4.63	20	20
180	2.004	9.23	2.878	13.078	8.932	2.449	4.63	20	20

Table 4.10.2: Results at Vessel Speed of 14.5 knots and at Intersection of FP and Baseline

Heading (degrees)	RMS Relative Velocity (m/s)	Characteristic Relative Velocity (m/s)	RMS Relative Motion (m)	Characteristic Relative Motion (m)	# Emergences Per Hour	# Slams Per Hour	Critical Velocity (m/s)	Maximum Allowed Emergences Per Hour	Maximum Allowed Slams Per Hour
0	0.595	2.689	1.999	9.095	4.444	0.304	4.63	20	20
45	0.766	3.464	1.963	8.878	4.364	0.27	4.63	20	20
90	0.843	3.893	1.186	5.392	0.202	9.91E-05	4.63	20	20
135	1.73	7.951	2.487	11.26	6.982	1.234	4.63	20	20
180	1.984	9.138	2.873	13.052	8.912	2.432	4.63	20	20

Table 4.10.3: Results at Vessel Speed of 1 knot and at Bow-Loading Manifold

Heading (degrees)	RMS Relative Velocity (m/s)	Characteristic Relative Velocity (m/s)	RMS Relative Motion (m)	Characteristic Relative Motion (m)	# Emergences Per Hour	# Slams Per Hour	Critical Velocity (m/s)	Maximum Allowed Emergences Per Hour	Maximum Allowed Slams Per Hour	Maximum Allowed Relative Motion (m)
0	0.952	4.383	1.828	8.319	3.028	0.123	4.63	20	20	7.55
45	1.025	4.699	1.795	8.117	2.921	0.105	4.63	20	20	7.55
90	0.955	4.39	1.392	6.298	0.775	3.06E-03	4.63	20	20	7.55
135	1.319	6.015	2.224	10.01	5.806	0.664	4.63	20	20	7.55
180	1.29	5.907	2.346	10.631	6.357	0.906	4.63	20	20	7.55

Table 4.10.4: Results at Vessel Speed of 15 knots and at Navigation Bridge

Heading (degrees)	RMS acceleration (m/s^2)	Characteristic Acceleration (m/s^2)	Maximum Allowed Acceleration (m/s^2)
0	0.073	0.334	3.924
45	0.12	0.546	3.924
90	0.438	1.968	3.924
135	0.746	2.151	3.924
180	0.42	1.913	3.924

4.10.2 Maneuvering

The Shuttle Tanker must maneuver in both open water and in port. The ship uses a podded propulsion system, which can direct the thrust force in any direction. For slower speeds (like those seen in port), the dynamic positioning systems (DPS) allows zero radius turning in calm seas and very little turning radius, less than one ships length, in larger sea states.

The turning radius for the ship is important for high speed maneuvering. The structure of the podded propulsors cannot withstand stresses of vectoring thrust at large angles and high speeds. To analyze the performance of the pods and the maneuverability of the ship a motion prediction analysis is performed using calculations found in Principles of Naval Architecture, Volume III, page 209 [12]. To perform this analysis on the Shuttle Tanker, an effective rudder area is calculated to provide the equivalent yaw moment as produced by the podded propulsors. The effective rudder area analysis is based on flat plate theory and is presented in Appendix K.

The bow thrusters are not included in this analysis because the focus is on the pods' performance. It is however estimated that the use of bow thrusters in high speed maneuvering reduces the turning radius. Figure 4.10.2 is a chart of the resulting turning diameter with an approach velocity of fifteen knots at various pod angles. At zero degrees, the pod is parallel to the centerline of the vessel.

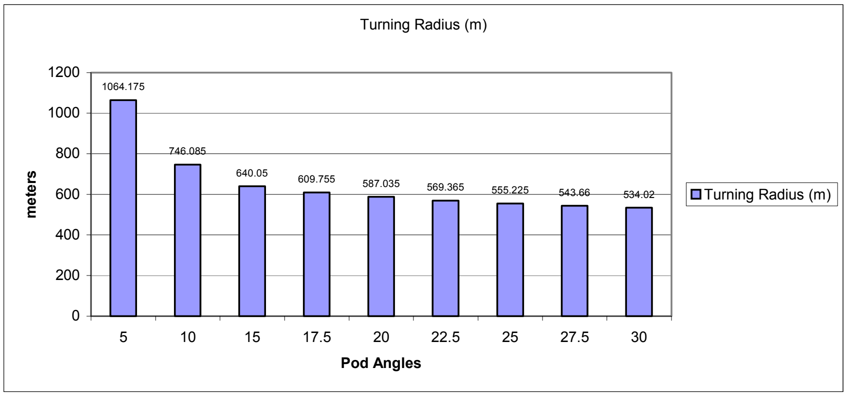

Figure 4.10.2: Turning Diameter vs. Pod Angle

4.10.3 Dynamic Positioning

To analyze the dynamic positioning system of the Shuttle Tanker, a simplified, worse case, closed form calculation is performed on both the ballast and full load conditions. The worse case analysis accounts for the ship in full beam seas and winds. The vessel is evaluated in Sea State 6 with a significant wave height of 5.5 meters and wind speeds of 35 knots. The closed form calculation, shown in Appendix L, determines the sway drift distance of the ship prior to a dynamic positioning correction. The sway drift distance of the Shuttle Tanker must be less than fifty meters, as stated by the general requirements. This requirement is necessary for safe bow loading operations. Figure 4.10.3 shows an illustration of the DPS analysis.

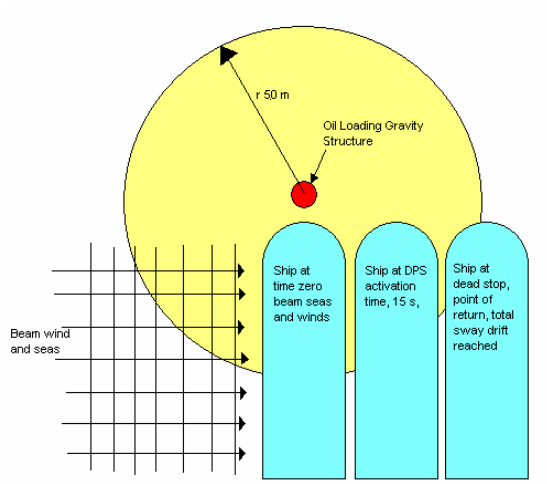

Figure 4.10.3: DPS Analysis Illustration

To calculate the sway drift distance, the wind velocity and stokes drift velocity of the ship are used. These velocities are used to calculate the transverse pressure acting on the ship. These pressures are applied to approximate windage and submerged areas respectively and a total transverse force is determined. Using the total force and the lag time for the DPS activation, 15 seconds, a momentum is calculated for the ship. Using this momentum and the total mass in sway for the Shuttle Tanker, the sway velocity is determined. Using the transverse forces produced by the DPS components, which are the pods and bow thrusters, an acceleration constant for the dynamic position correction is determined. Dividing the vessel sway velocity by the acceleration constant provides the time until the dynamic positioning system overcomes the sway motion. Using the acceleration, velocity and time values, a total drift distance is calculated for the ship. The Shuttle Tanker meets all dynamic positioning criteria and the results of the DPS analysis are provided in Table 4.10.5.

Table 4.10.5: DPS Analysis Summary

	Approximate Windage Area (m^2)	Approximate Submerged Area (m^2)	Wind Pressure (Pa)	Sea Pressure (Pa)	Sea Force (N)	Wind Force (N)	Sway Velocity (knots)	Time Until Dead Stop (sec)	Total Distance Traveled (m)
Ballast Condition	4700	2300	199.1	4.404	10100	936000	0.449	53.30	9.64
Full Load Condition	3700	3800	199.1	4.404	16700	737000	0.339	71.57	8.876

4.11 Cost and Producibility

Cost calculations for the Shuttle Tanker are based on weight and producibilty. These calculations are shown in the Simplified Tanker Cost Model, Appendix B, pg. B19. The weight of the ship is broken down into SWBS groups, which represent the lightship condition. The cost is analyzed for each group by weight and multiplied by a producibility factor. This factor is determined based on the hull curvature in the stern section. Other producibility factors were not affected directly by the concept design parameters and were not included in the cost model. However, they were considered in the design. Because the Shuttle Tanker utilizes podded propulsion instead of a conventional shaft and propeller, it is not as important for the stern to be constructed to induce flow through the propeller. The Tanker has a single curvature stern that rises gradually. This is a very producible hull and contributes to the reduction of machining and labor costs. The deckhouse also contributes to the ship's producibility. Each deck is three meters high and rectangular. The staterooms can be built modularly, separate from the construction of the deckhouse and inserted when necessary. There is a cost margin included in the analysis that accounts for additional cost due to design error, added equipment and added cost due to production. Ice strengthening of the hull contributes to a higher than normal structural cost but is imperative for the mission. A breakdown of costs is shown in Table 4.11.1. Note that in Table 4.11.1, the fuel, manning and maintenance costs are annual costs. The total ownership cost includes the present value of these costs.

The Shuttle Tanker has a low cost, schedule and technology risk. The ship utilizes industry standard equipment to minimize technology risk. To minimize setbacks in the building schedule, common building materials are used to ensure availability and the use of double curvature plates are minimized. The only areas with increased risk are the integrated power system and podded propulsion, which are not commonly used in oil tankers.

Table 4.11.1: Cost Analysis

Type	Cost ($ million)
Lead Ship Cost	141.6
Fuel Cost	2.46
Manning Cost	2.8
Maintenance Cost	1.3
Total Ownership Cost	210.94

5 Conclusions and Future Work

5.1 Assessment

The Shuttle Tanker meets and in many cases exceeds the owner requirements. Table 5.1.1 shows the actual and required specifications. The use of a podded propulsion system gives the vessel high mobility and dynamic positioning ability. It also allows for a ramped and single curvature stern section that greatly increases producibility. The deckhouse is a very producible, rectangular structure. It incorporates the upper machinery space, exhaust stack and the crew living quarters into a single structure to decrease structural weight and manufacturing time. The engine room has many automated features that increase safety and decrease maintenance and monitoring. Cargo and ballast piping systems are based off of current systems known to be reliable.

Table 5.1.1 Actual and Required Specifications

Requirement	Minimum Specification	Shuttle Tanker
Cargo Capacity	127,000 MT	127,000 MT
Minimum Sustained Speed	15 knots at 90% MCR	15.14 knots at 90% MCR
Endurance Range	3,000 nm at 15 knots	6,150nm at 15 knots
Maximum Full Load Draft	15 meters	15 meters
Cargo Segregation	3x2	6x2
Dynamic Positioning	50 meter radius in sea state 6	Worst case: travel 9.64m in ballast condition
Minimum Double Bottom Height	2 meters	3.9 meters
Minimum Double Side Width	2 meters	4 meters
Minimum Deck Height	3 meters	3 meters
Manning	NA	28

5.2 Recommended Improvements

There are a few major changes that should be made to the Shuttle Tanker in the next iteration through the design spiral. The following areas merit further consideration.

5.2.1 Hull Form

The next iteration through the design spiral should consider some changes in hull form. Further analysis of the aft waterlines should be performed to ensure there is no potential flow separation or turbulence. The extreme flare in the forward most stations should be analyzed to reduce the possibility of bow slamming. The stern section of the ship is very flat and broad. In performing the sea keeping analysis, it was discovered that this flat stern shape gives the ship too much dynamic lift. It causes the ship to ride up on the wave until the bow slams back into the sea. To help this problem, a stern section with more of a V-shape should be utilized. This will slightly reduce the Shuttle Tanker's dynamic positioning ability but, it will also decrease the effect of dynamic lift by reducing the tendency to ride the wave, thereby lessening slamming of the bow. This change will also reduce the chance of stern slamming in the ballast condition. The addition of a skeg in the stern section should be considered to improve directional stability and support the stern section in drydock.

5.2.2 Space and Arrangements

There is extra space in both the machinery and pod rooms of the Shuttle Tanker. This is due to the shape of the hullform, which is at an optimal length to reduce wave-making resistance. Eliminating the pod room and moving the machinery room aft after shortening it three meters will better utilize this space. This will open up an additional 18 meters of longitudinal space. Consequently, the deckhouse will also be moved 18 meters aft to keep the forward bulkhead of the deckhouse inline with the forward bulkhead of the machinery room. Additionally, the fresh water tanks should be moved in close proximity to the deckhouse to reduce pipe length. The move of both the machinery room and the deckhouse will improve the LCG location of the bare hull, which is presently too far aft. The cargo block is then lengthened by increasing the length of each cargo tank by three meters. The additional cargo space will increase the cargo volume, which will lead to increased profit at little or no additional cost. This is preferable to shortening the ship length, which would increase the total resistance and thus increase the total ownership cost.

Consideration should be made to extend the double bottom through the engine room to increase reliability in the event of outer shell rupture. This would also allow space for bottom stiffeners. If this change is included in the next design iteration, the pump room needs to be resized to allow for the added bottom structure and to incorporate more maintenance space. In the resized machinery room, the diesel generators would be mounted fore and aft to reduce thrust forces on the engines and generators. In addition, larger maintenance space would be included in the area around the generators to allow for rotor removal.

5.2.3 Weights and Loading

While data was collected for many of the SWBS groups, there were some weights that had to be estimated due to lack of data. However, in the future, more research could be done to improve estimates. By finding more accurate weight estimations, a better estimate for the center of gravity of the lightship could be computed. This will lead to more accurate loading and damage scenario estimates. The new arrangement of the cargo block will also change the loading and damage scenarios.

5.2.4 Resistance, Seakeeping and Maneuvering

The seakeeping analysis for the Shuttle Tanker would have to be rerun for the new hull form. The maneuvering analysis could also be improved. A commercial program for dynamic positioning should be found that would represent a more accurate estimate of the chosen dynamic positioning system capabilities instead of worse case scenario calculations such as were used during the first iteration through the design spiral. In addition, a better analysis for a podded propulsion system, as opposed to a conventional system, needs to be found or created. This could give more accurate results for the maneuvering capabilities of podded propulsion.

Model testing of the ship in a towing tank will further improve estimates of the Shuttle Tanker. Conventional calculations were used for seakeeping and maneuvering as well as dynamic positioning. They were adjusted to represent the unconventional Shuttle Tanker as accurately as possible. Model testing would greatly improve accuracy of resistance and seakeeping data for this ship and could be used as data for other ships with positioning systems and unconventional propulsors. The most effective angle to mount the pods to both align the flow and minimize the vertical component of thrust can also be tested with the model, as well as the V-shape stern section.

5.2.5 Cost

Because cost is directly based on weight, improved weight estimates will improve cost estimates. Production and producibility costs also need to be more accurately represented in cost estimates. The highly producible hullform should decrease costs more drastically than represented in the cost calculations. Increased cargo volume will also increase profits over the lifetime of the ship. These factors lead to diminishing the total ownership cost over the 30 year lifetime of the Shuttle Tanker.

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Appendices

Appendix A: General Requirement

Appendix B: Ship Synthesis Model

Appendix C: Table of Molded Offsets

Appendix D: Structures Analysis

Appendix E: Resistance Data for Full Load Case

Appendix F: Resistance Data for Ballast Load Case

Appendix G: Electrical and Propulsion System Schematic

Appendix H: Inert Gas System Schematic

Appendix I: Weights Table

Appendix J: Seakeeping Results

Appendix K: Maneuvering

Appendix L: Dynamic Positioning System Analysis