UNIVERSITY OF OXFORD

Sailing Simulator

M.Eng. 3rd Year Design Project

Paul Bossward
Max Jaderberg
Tim Thornton
Anna Whayman

Trinity Term 2011

FINAL HONOUR SCHOOL OF
ENG / EEM (delete as appropriate)

DECLARATION OF AUTHORSHIP

Each candidate should complete this certificate. It should be bound into the third year project report, immediately after the title page. Three copies of the report should be submitted to the Chairman of Examiners for your Honour School, c/o Clerk of the Schools, Examination Schools, High Street, Oxford.

Name (in capitals):

College (in capitals): **Supervisor:**

Title of project (in capitals):

Page count: _____

Please tick to confirm the following:

I am aware of the University's disciplinary regulations concerning conduct in examinations and, in particular, of the regulations on plagiarism. ☐

The project report I am submitting is entirely my own work except where otherwise indicated. ☐

It has not been submitted, either wholly or substantially, for another Honour School or degree of this University, or for a degree at any other institution. ☐

I have clearly signalled the presence of quoted or paraphrased material and referenced all sources. ☐

I have acknowledged appropriately any assistance I have received in addition to that provided by my supervisor. ☐

I have not sought assistance from any professional agency. ☐

The project report, or my contribution to the report in the case of a joint report, does not exceed 30 pages (including all diagrams, photographs, references and appendices). ☐

I agree to retain an electronic version of the work and to make it available on request from the Chair of Examiners should this be required in order to confirm my word count or to check for plagiarism.

Candidate's signature: …….. Date: ……..

FINAL HONOUR SCHOOL OF
ENG / EEM (delete as appropriate)

DECLARATION OF AUTHORSHIP

Each candidate should complete this certificate. It should be bound into the third year project report, immediately after the title page. Three copies of the report should be submitted to the Chairman of Examiners for your Honour School, c/o Clerk of the Schools, Examination Schools, High Street, Oxford.

Name (in capitals):

College (in capitals): **Supervisor:**

Title of project (in capitals):

Page count: _____

Please tick to confirm the following:

I am aware of the University's disciplinary regulations concerning conduct in examinations and, in particular, of the regulations on plagiarism. ☐

The project report I am submitting is entirely my own work except where otherwise indicated. ☐

It has not been submitted, either wholly or substantially, for another Honour School or degree of this University, or for a degree at any other institution. ☐

I have clearly signalled the presence of quoted or paraphrased material and referenced all sources. ☐

I have acknowledged appropriately any assistance I have received in addition to that provided by my supervisor. ☐

I have not sought assistance from any professional agency. ☐

The project report, or my contribution to the report in the case of a joint report, does not exceed 30 pages (including all diagrams, photographs, references and appendices). ☐

I agree to retain an electronic version of the work and to make it available on request from the Chair of Examiners should this be required in order to confirm my word count or to check for plagiarism.

Candidate's signature: …….. Date: ……..

FINAL HONOUR SCHOOL OF
ENG / EEM (delete as appropriate)

DECLARATION OF AUTHORSHIP

Each candidate should complete this certificate. It should be bound into the third year project report, immediately after the title page. Three copies of the report should be submitted to the Chairman of Examiners for your Honour School, c/o Clerk of the Schools, Examination Schools, High Street, Oxford.

Name (in capitals):

College (in capitals): **Supervisor:**

Title of project (in capitals):

Page count: _____

Please tick to confirm the following:

I am aware of the University's disciplinary regulations concerning conduct in examinations and, in particular, of the regulations on plagiarism. ☐

The project report I am submitting is entirely my own work except where otherwise indicated. ☐

It has not been submitted, either wholly or substantially, for another Honour School or degree of this University, or for a degree at any other institution. ☐

I have clearly signalled the presence of quoted or paraphrased material and referenced all sources. ☐

I have acknowledged appropriately any assistance I have received in addition to that provided by my supervisor. ☐

I have not sought assistance from any professional agency. ☐

The project report, or my contribution to the report in the case of a joint report, does not exceed 30 pages (including all diagrams, photographs, references and appendices). ☐

I agree to retain an electronic version of the work and to make it available on request from the Chair of Examiners should this be required in order to confirm my word count or to check for plagiarism.

Candidate's signature: …….. Date: ……..

FINAL HONOUR SCHOOL OF
ENG / EEM (delete as appropriate)

DECLARATION OF AUTHORSHIP

Each candidate should complete this certificate. It should be bound into the third year project report, immediately after the title page. Three copies of the report should be submitted to the Chairman of Examiners for your Honour School, c/o Clerk of the Schools, Examination Schools, High Street, Oxford.

Name (in capitals):

College (in capitals): **Supervisor:**

Title of project (in capitals):

Page count: _____

Please tick to confirm the following:

I am aware of the University's disciplinary regulations concerning conduct in examinations and, in particular, of the regulations on plagiarism. ☐

The project report I am submitting is entirely my own work except where otherwise indicated. ☐

It has not been submitted, either wholly or substantially, for another Honour School or degree of this University, or for a degree at any other institution. ☐

I have clearly signalled the presence of quoted or paraphrased material and referenced all sources. ☐

I have acknowledged appropriately any assistance I have received in addition to that provided by my supervisor. ☐

I have not sought assistance from any professional agency. ☐

The project report, or my contribution to the report in the case of a joint report, does not exceed 30 pages (including all diagrams, photographs, references and appendices). ☐

I agree to retain an electronic version of the work and to make it available on request from the Chair of Examiners should this be required in order to confirm my word count or to check for plagiarism.

Candidate's signature: …….. Date: ……..

1. Table of Contents

2. Introduction 5

3. Project Management 6

3.1. Work Breakdown Structure 6

3.2. Network Diagram 6

3.3. Gantt Chart 8

4. Market targeting 9

4.1. Market Research 9

4.2. Arcade game 9

4.3. Sailing simulator for novices 11

4.4. Simulator for professional athletes 12

4.5. Summary Table 14

4.6. Market choice 14

4.7. Further market research 15

5. Design Specification 16

5.1. Computer Simulation 16

5.2. Physical Simulation & Structural Design 17

5.3. Control System 18

5.4. User Interface (Display & Sound) 19

6. CAD Hull Analysis 20

6.1. Submerged volume 20

6.2. Draft 20

6.3. Wetted surface area 23

7. Buoyancy 26

7.1. Vertical Forces 26

7.2. Roll and Pitch 26

7.3. Users Weight Distribution 28

8. Hydrodynamic Drag 30

8.1. Governing Physics 30

8.2. Finding the Drag Coefficient 31

8.3. Implementation 33

9. Sail, Rudder and Centreboard 36

9.1. Underlying Physics + Equations 36

9.2. Control 38

10. Waves and Wave Effects 42

10.1. Specification 42

10.2. Waves 42

10.3. Wave Forces 46

10.4. Conclusion 49

11. Computer Simulation 50

11.1. Specification 50

11.2. Simulator Structure 50

11.3. Prototype Implementation 55

11.4. Production Implementation 56

12. Boom and rudder actuators 58

12.1. Actuator Motors 59

12.2. Actuator Gearing 60

12.3. Actuator Mounting 61

13. Cradle and Hull 67

13.1. Hull 67

13.2. Actuator choice 70

13.3. Magnetic Spring 70

13.4. Creating the cradle 70

13.5. Rod end 71

13.6. Bracket 71

13.7. Connecting cylindrical rods to bracket and hull 74

13.8. Mounting cradle 79

13.9. Human 83

14. Display and sound 84

14.1. General Specification 84

14.2. Display 84

14.3. Sound 90

15. Graphics and Sound Rendering 92

15.1. Specification 92

15.2. User Interface 92

15.3. Three Dimensional Graphics 93

15.4. Graphics Engine 95

16. Electronics 98

17. Control 100

17.1. Specification 100

17.2. Cradle Control 100

17.3. Actuator Interface 102

17.4. Sail and Rudder Control 103

18. Systems Integration 104

18.1. Computer 104

18.2. Display 104

18.3. Sound 104

18.4. Linear Actuators 105

18.5. Sail and Rudder Actuators 105

18.6. System Diagram 107

19. Pricing 108

19.1. Price list of all components (starred components are optional) 108

19.2. Price of all manufacturing/service jobs 109

19.3. Packages (to buy) 110

19.4. Packages (to rent) 110

20. Summary 111

21. Appendix 112

22. Bibliography 115

2. Introduction

Sailing is ever increasing in popularity, with thousands of people each year in the UK alone taking to the water for competition or recreation. These rising numbers has led to sailing schools being established throughout the UK and with that comes a need to provide consistent training adhering to a timetable; something that the UK weather systems regularly prevent.

It is from here that the motivation for a sailing simulator stems; to produce a compact, transportable system that can provide a fully immersive learning experience with accurate responses mimicking the real world conditions. This could then be leased out or sold to schools across the country, providing a safe non-threatening environment for beginners to grasp the basics alongside acting as an accurate, responsive training aid for experience sailors to hone their skills on.

The only previous attempt at such a simulator is the Australian made Virtual Simulator VS-1. This was a progression of an old simulator developed to aid sports physiology decades ago by the Department of Anatomy and Physiology in the School of Medicine of the University of Tasmania. Although their version was heavily updated from the Tasmanian original design, it still lacked major key areas such as a centreboard and a boom. This, combined with a very small CRT display, made it a relatively poor training aid as due to the lack of key objects in the boat it was hard to recreate accurately the sailors movement; being watchful of the low boom etc.

The new simulator has been freshly designed from the ground up, with completely unique software and hardware. It has not used any of the VS-1 design and hence does not suffer from its limitations. It includes a full boom and rudder extension, combined with a large high definition projected display to give the user the most realistic and immersive experience on the market. Alongside this, electronic actuators are used to minimise noise, weight and cost by removing the need for any form of pre-actuator fluid compression and this paired with a well thought out design has led to the finished assembly being easily transportable.

This paper will describe the design of the new model and demonstrate that this new design provides a far more realistic and immersive sailing experience.

3. Project Management

3.1. Work Breakdown Structure

A list of all jobs that must be completed was grouped into distinct sections: detailed specification, governing equations of motion, mechanical design, electrical design, systems integration and business plan.

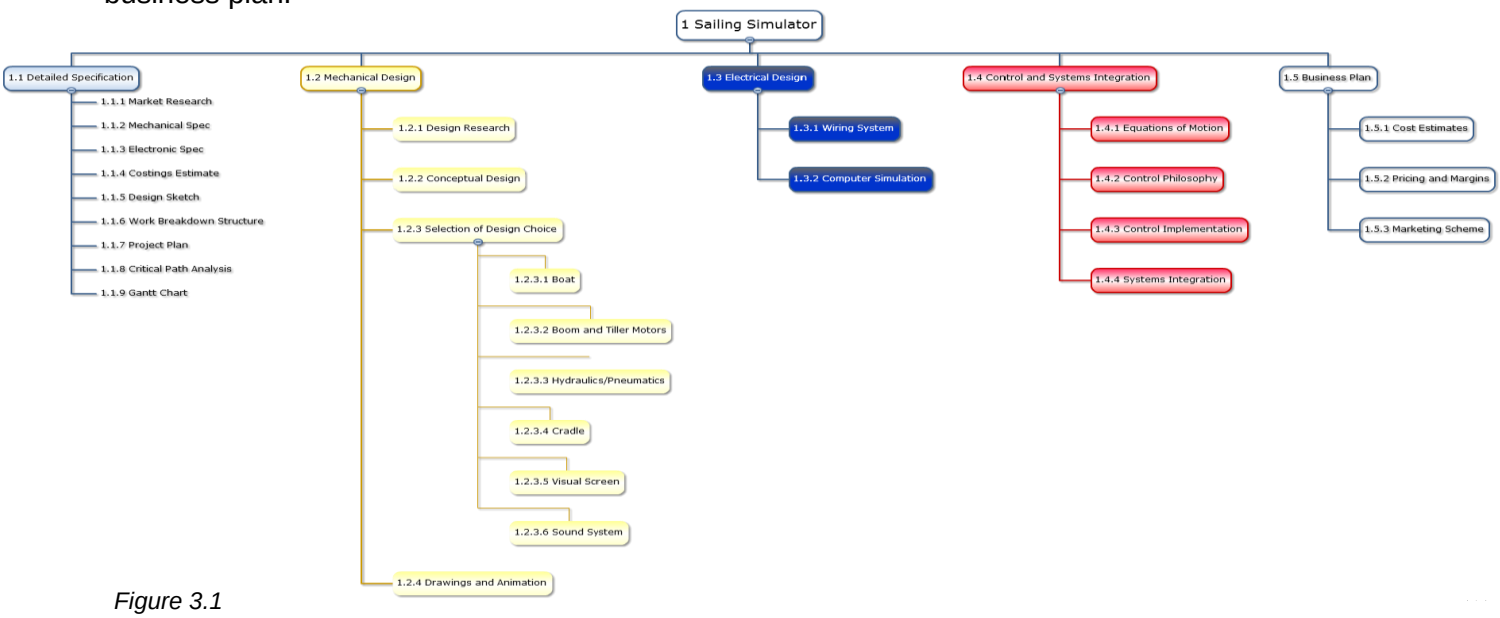


Figure 3.1

3.2. Network Diagram

The list of jobs in the work break down structure was then used to create a network diagram of completion times, and to find the critical path of jobs that had to be done without float.

Number	Job Name	Time to complete (days)	Jobs required to be completed before starting	Earliest Start Date	Float
1	Project Management	4	-	9/11/10	0
2	Market Research	4	1	13/11/10	0
3	Further Research	6	2	17/11/10	0
4	Business Plan	7	3, 19	23/11/10	123(add 6 weeks)
5	Design Specification	7	3	23/11/10	0
6	Costings Estimate	7	5	30/11/10	46 (add 6 weeks)
7	Laser Drawing	14	5	30/11/10	0
Break 6 weeks for Christmas holidays					
8	Sail and rudder force	14	6, 7	25/01/11	39
9	Wave exciting force	14	6, 7	25/01/11	39
10	Yaw force	14	6, 7	25/01/11	39
11	Buoyancy force	14	6, 7	25/01/11	39
12	Integration of forces	7	8, 9, 10, 11	08/02/11	39
13	Cradle and structure	28	6, 7	25/01/11	0
14	Tiller and boom	28	6, 7	25/01/11	0
15	Electronics	28	13, 14	22/02/11	0
16	Display and sound	7	13	22/02/11	21

17	Model simulation	7	12	15/02/11	39
18	Graphics	7	15, 16,17	22/03/11	11
19	Control	14	18	29/03/11	11
20	Systems Integration	21	19	12/04/11	11
21	CAD drawings	39	15, 16	22/03/11	0
22	Animation	14	21	30/04/11	0
23	Pricing	7	18	29/03/11	39
24	Report	20	4, 20, 22, 23	14/05/11	0
25	Printing	21	24	03/06/11	0

Table 3.1

Drawing Network Diagram

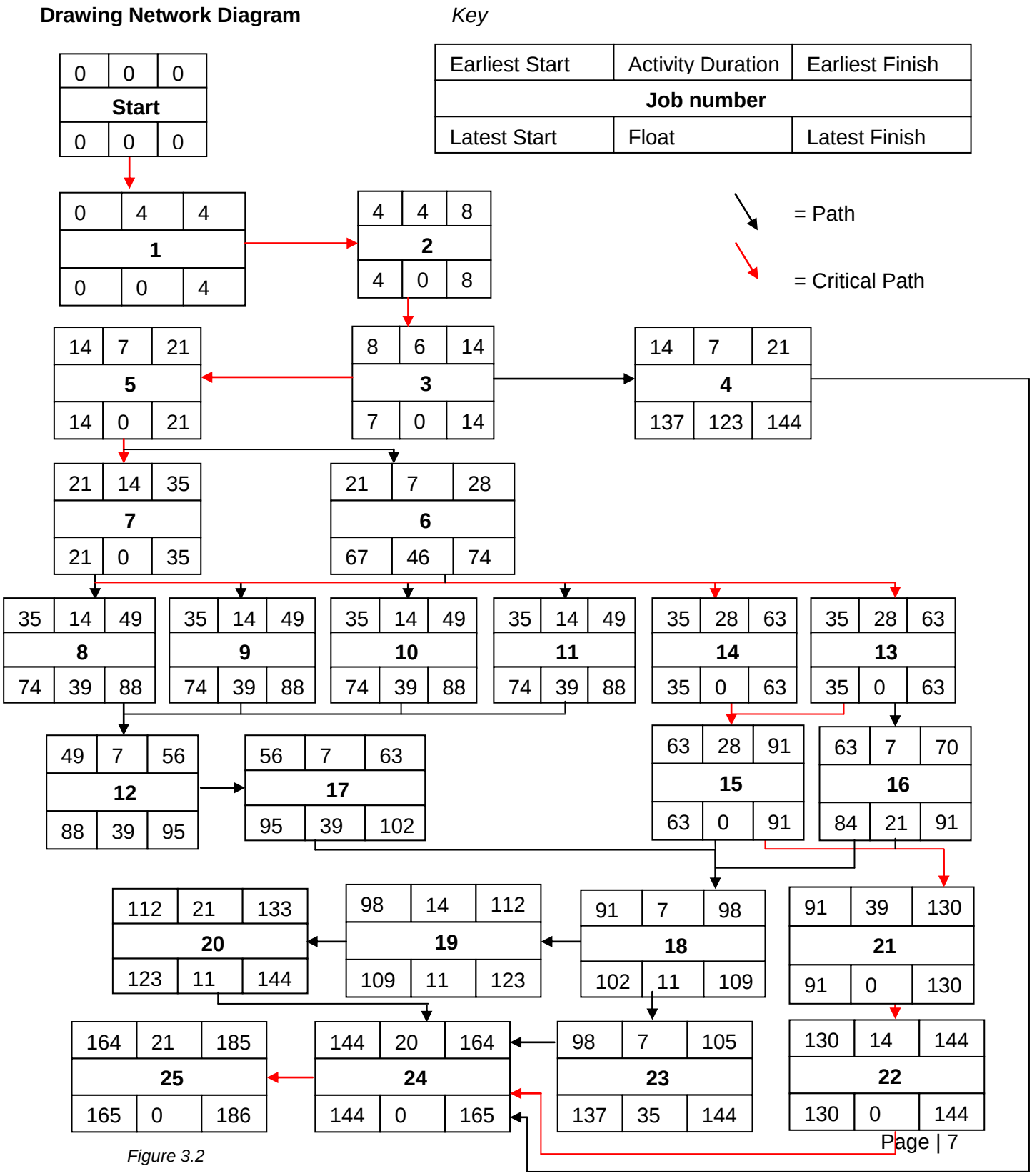


Figure 3.2

3.3. Gantt Chart

The precedence diagram (network diagram) was converted into a Gantt chart using the earliest possible start date for each task. The end of the float indicates the latest possible end date for each task.

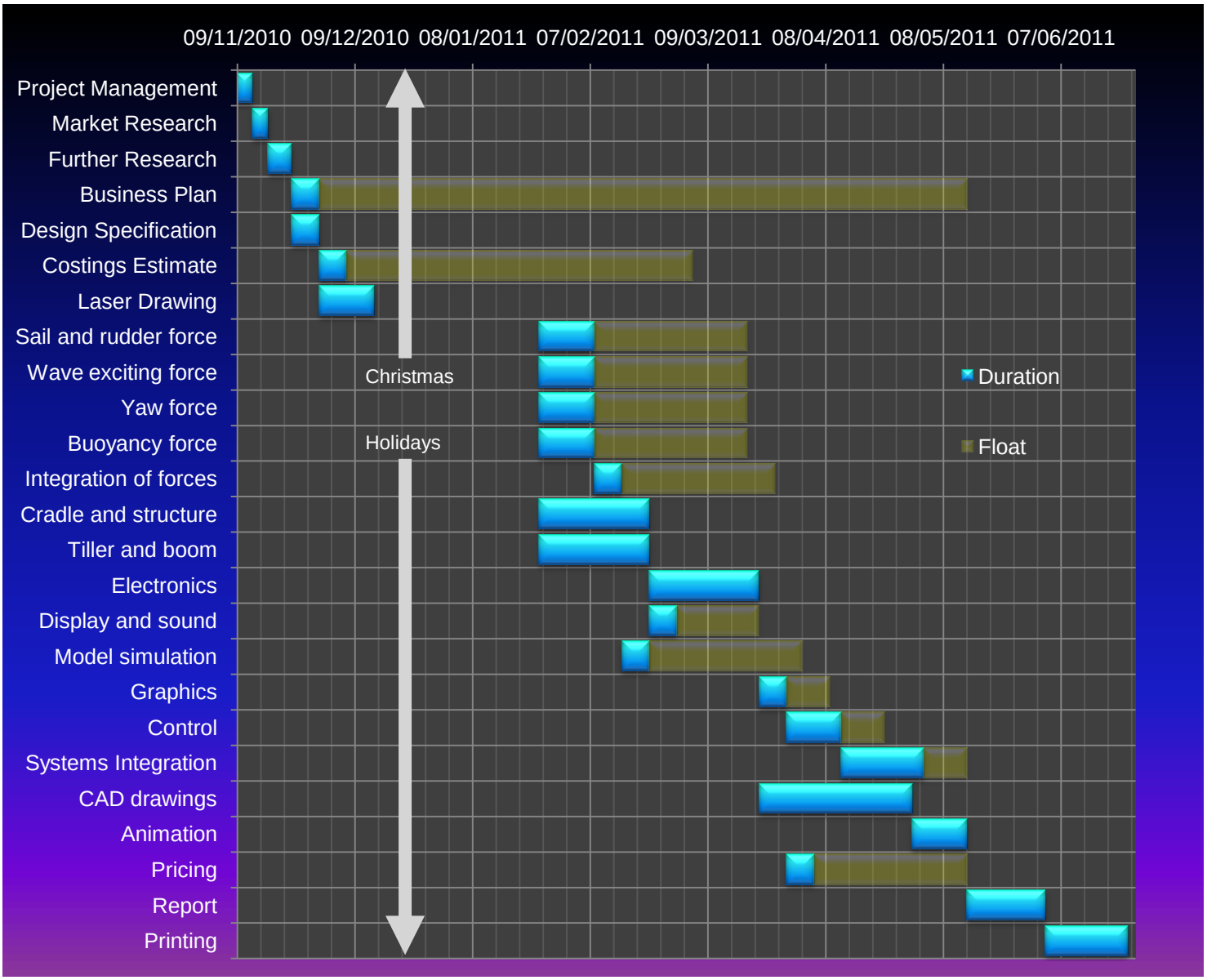


Figure 3.3

4. Market targeting

4.1. Market Research

The project aim is to design a form of sailing simulator. This could be done to target three potential markets: arcades, sailing schools for novices/amateurs, and professional athletes and medical research. Market research was conducted to see how viable each market would be to launch a new simulator, and to see where in the market we would place ourselves.

According to Porter's model of generic competitive strategies, there are five forces which determine the industry's overall competitiveness and profitability. These are: threat of new entry, intensity of rivalry among existing firms, bargaining power of buyers, bargaining power of suppliers, and threat from substitute products. Profitability is negatively related to these forces overall strength. We can use this analysis to help decide which market we will target with our sailing simulator. Using the first four forces, we will decide upon the target market(s), and with the last force we can decide if we will be providing the lowest cost, or the best product.

Figure 4.1

The first four forces represent how saturated a market already is, and how much power a firm would have within that market. We want to place ourselves in a market that has minimal competition, is easy to get into, and we are able to have high bargaining power compared to suppliers and buyers. This means that the simulator must be quite unique.

4.2. Arcade game

Whilst there is a huge market for gaming globally of around £10.5 billion in 2010[1] this is primarily in home consoles and online gaming. Over 67% of all American households own a games console[2]. Arcade machine gaming is a dwindling industry, only taking 0.3% of the total game market share,

[1], [2] The ESA 2010 review http://www.theesa.com/facts/pdfs/ESA_Essential_Facts_2010.PDF

so is worth approximately £31.5 million[3]. The main draw for arcade gaming is the interactive feature that consoles are unable to provide at home. However, the competitive nature of sports gaming (such as a sailing game) leads many people to play for free online, substituting the expensive cost of using arcade games, for free online competitiveness.

In spite of appearances and publicity hyperbole, it is an undisputed fact that the gaming industry as a whole is declining, with sales declining from 2009[4] (see Figure 4.2).

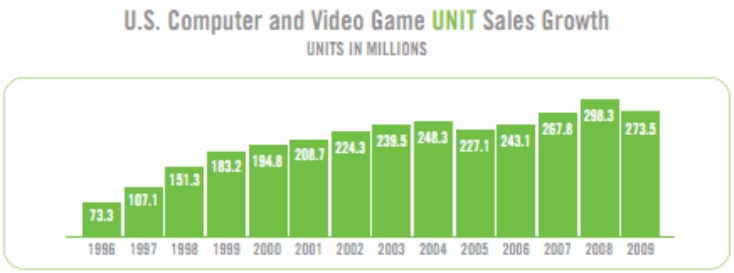


Figure 4.2 Source: The NPD Group/Retail Tracking Service

As such the buyer has a huge amount of power within the industry. This seems to occur cyclically until a new type of games station is released. As the arcade game has been around for decades, a sailing simulator theme game is a continuation of what has already been. The arcade game industry is also extremely volatile, with many firms not surviving their first year in business. There is intense rivalry between existing firms, with over 75 arcade game manufacturers[5].

Normally game makers need the backing of major publishers (such as Sega, EA games etc), and only around 17% of profits are actually received by the game developers[6]. This means that the supplier also has a large amount of power within the industry.

It would be very difficult to distinguish ourselves within this oversaturated market. There is difficulty in producing a low cost option, by reducing interactive features to be the cheapest arcade game. This can be provided at home using a console (such as interactive Wii) or even with no interactive features for free online. Low cost aquatic arcade games range in price from £3500 to £7500. It will be extremely difficult to create a product that is cheaper than £3500 without restricting an incredible amount of the simulator.

[3], [4] The ESA 2010 review http://www.theesa.com/facts/pdfs/ESA_Essential_Facts_2010.PDF

[5] http://www.bmigaming.com/mastercatalogselect.htm
[6] http://en.wikipedia.org/wiki/Video_game_publisher

The other option to go for an extremely high end option is also difficult, as the public is only willing to pay a certain amount to play in arcades, ranging from £0.50 to a maximum of £2. Therefore arcades looking to buy a new game station will not buy any extremely expensive games, with a limit of around £20000. This means we will have to produce something that is much better than such games at this price, which is difficult due to the already saturated market. The five market forces suggested by Porter are all high in the arcade game market, which suggests low or little profitability for a game developer.

4.3. Sailing simulator for novices

Currently in the market for a sailing simulator for teaching purposes is only one model, the Australian designed virtual sailing simulator[7]. The market for learning to sail is large, with around 10million people taking learn to sail holidays each year[8], costing approximately £80 per day per person[9]. This results in an £800 million industry in sailing holidays alone. The most likely candidates for a sailing holiday tend to be from the USA and the UK. There is also the new upcoming market of sailing holiday's taken by people from China. From the UK and the US, the two main sailing schools are Moorings and Sunsail, both owned by First Choice. First Choice can either be targeted to sell the simulator to, or we could perhaps look to selling to a rival holiday firm (such as Thompson) to provide sailing holidays with our simulators.

There are only two sailing schools in the whole of the UK that employ the Australian sailing simulator, out of a possible hundred schools that are active. This suggests that threat of new entry is relatively low, in such an untapped market there will be little to no restrictions upon entering. There is also very small rivalry among existing firms, with currently only the Australian simulator viable for purchase.

This also means that buyers' power is minimal, with so little choice the seller is able to charge as much as they desire for the simulator. The bargaining power of suppliers is also low, as the main costs will be on the actual hull, and the actuators. Actuators are readily available, with many

[7] See http://www.virtualsailing.com.au/ for more information
[8] http://www.onecaribbean.org/content/files/Sailing.pdf
[9] http://www.rya.org.uk/

different suppliers all over the world. There is also the ability to pick up second hand dinghy hulls for as little as £150.

Within the sailing school market, we then have to focus on whether we will design a high end or low cost model. The cost of buying the Australian sailing simulator outright is approximately £16,500. The cost to hire the existing simulators in the UK is approximately £500 per day (indoorsailing.com). This means that we have to produce a simulator that undercuts the current model, or is much better.

However, as there are only two sailing schools using the simulator, we must investigate why this is the case, if the cost is off putting or the limited simulation provided (see section 4.7).

Research using the existing sailing simulator by the Australian team was performed on a two day "Learn to sail" course for children in Tasmania in 2002[10]. The coaches reported that of the children who had been on the simulator, prior to using the dinghies, learned on-water skill more rapidly than those who had not been on the simulator. This suggests that a more realistic simulator would mean even better results for people that use it.

4.4. Simulator for professional athletes

Many professional athletes train by a combination of dry land and water based training. The dry land training consists of weights and cardio. A sailing simulator could be potentially used to bridge the gap between on water and dry land training.

Another main aim for professional sailors is to target how injuries occur, and how to prevent them. Currently, little is known about sailing injuries, despite organised sailing as a sport has been around for 150 years[11]. Most injuries occur by hiking (leaning over the edge of the boat to oppose heeling). The main issue with rehab for such injuries, and to investigate how they occur, is the ability to simulate hiking on dry land.

Hiking has been described as a contraction of the knee extensor and the hip flexor muscles, and is quasi-isometric due to small amplitude changes in the force production to counteract the changing

[10] http://www.virtualsailing.com.au/site/misc/virtualsailing/downloads/MooneyHabgoodSaundersBinns_RevF
[11] North American Journal of Sports injuries http://www.ncbi.nlm.nih.gov/pmc/articles/PMC2953336/

waves. Problems therefore occur in replicating the unpredictability of the waves. This is something that the existing sailing simulator cannot create, with only heel movement simulated by the machine. To be able to generate realistic simulation for medical research, a model with three degrees of freedom needs to be implemented, in order to simulate unpredictable wave conditions, and the effect that this has on the muscles and joints in the legs. There is currently no dry land test to predict and monitor hiking performance, which can dramatically improve world class sailing athlete's skill.

Additional factors that are involved in the rehabilitation of an elite sailing athlete include movements within the boat. While racing, athletes also need to reposition themselves to the other side of the boat with ease and speed. This requires the ability to squat fully while moving laterally under the swinging boom, during turns through the wind. Full hip and knee range of motion are required as well as exceptional balance and coordination. This position involves moving laterally while in a fully flexed position and good stability of the knee is essential. This motion also occurs under adverse circumstances including a wet and continually moving surface due to the activity of the waves. The requirements of this motion, along with the unpredictability of the waves, put the athlete at risk for injury if he or she does not possess adequate range of motion, strength, and balance. This also results in greater training in a simulator with at least three degrees of freedom, and suggests that there is a market for such a model to be made.

After an injury to an athlete, intense rehab is needed to ensure that they are fit for sailing. Dry land hiking exercises with increasing levels of instability are used to determine if the athlete could tolerate the activity. This could be further used with the sailing simulator, by gradually increasing wind speed, and so wave speed. Limits to sailing activity on post-rehab athletes are based upon wind conditions, as higher wind speeds are directly proportional to hiking demands. Therefore there are only certain days that the athlete could return to sailing, and recovery time could be impacted. Using a sailing simulator is an ideal alternative, as the wind speed can be input, and can be used any day of the week to ensure that injury is not further sustained.

Currently, UK Olympic standard sailing teams have a budget in excess of £2 million to spend per year, which is spent on boats, training and research. With over 204 Olympic teams, this leads to an enormous market of approximately £400 million pounds per year. There is scope to target many Olympic teams by providing a model with at least 3 degrees of freedom to simulate as real as possible sailing conditions, both for training and medical research.

The threat of new entry, intensity of rivalry among existing firms, bargaining power of buyers and suppliers are all low in this market, there is little to no simulators that are capable of producing adequate simulation. The only simulator available is the current Australian simulator, and while this can be used for training, a significantly better model will greatly improve professional athletic sailing teams. Therefore the market positioning would be at the high end of the scale, as opposed to providing the cheapest option.

4.5. Summary Table

Market	**Threat of entry**	**Intensity of rivalry**	**Bargaining power of buyers**	**Bargaining power of suppliers**	**Pressure from substitute products**
Arcade Game	High (licence needed)	High (many manufacturers)	High (large amount of choice available)	High (publishers used charge a large amount to produce a game)	High (option to use at home consoles)
Sailing schools	Low (no licences needed)	Low (only one simulator available)	Low (little choice)	Low	Low (Can either produce lower cost than Australian model, or much better model, scope for differentiation)
Athletes	Low	Low	Low	Low	Low (high end choice to provide model with at least 3 degrees of freedom)

Table 4.1

4.6. Market choice

The arcade game market is already over saturated, and with a current dwindling market for gaming and arcade games, it seems that it is an unviable option to design an arcade game. With all five of Porter's forces high, it is not suitable to make high profits in the industry. There seems to be a viable option to enter into the sailing school or high end athletic training sailing simulator.

Within the athletic training, it seems to be the best option to produce the most high end model that is currently in the market, superseding the current simulator available for purchase. It makes sense to also try to engage in the sailing school market, and to produce a high end product in this market. This is opposed to producing a cheaper costing product than the current model. Therefore we will design a simulator with at least three degrees of freedom, and has a sliding cost scale which can be sold in both novice and athlete markets.

4.7. Further market research

After conducting the preliminary market research, further questionnaires and surveys were conducted. The main focus was contacting sailing schools to see why the current simulator was not used by them, if it was cost, lack of simulation options, or if they simply had not heard about it. The questionnaire was then used to gauge interest in a new simulator, and what factors would be important in purchasing one.

The result was unequivocal, after contacting 20 sailing schools, there was very high levels of interest from 95% of those that had been contacted. The reason for not using the current simulator was due to lack of knowledge (75%) and lack of simulation options (20%) (see logbook for full breakdown pg19). The factors that affected whether or not to purchase a sailing simulator is a combination of cost and adaptability. All of the schools said they would not consider purchasing a sailing simulator unless it gave a realistic enough experience. On average, the maximum cost of a sailing simulator that schools would consider purchasing was £18,500. We will try and produce a simulator at a cheaper price to this. Estimated costs came to a build price of £16,500 (logbook pg9, slide 6).

5. Design Specification

5.1. Computer Simulation

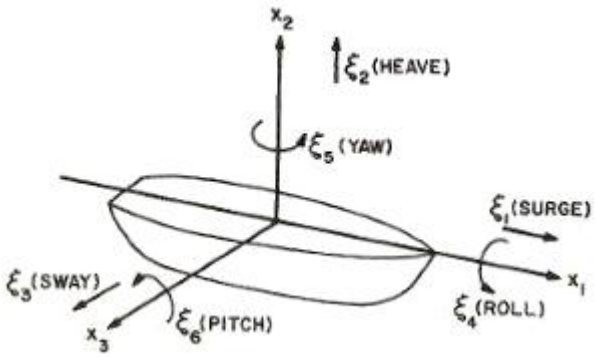


Figure 5.1 - The six degrees of freedom with reference to a boat

- The simulator will model a photorealistic 3D universe including boat and surrounding water with basic back drop surroundings.
- The simulation engine will account for the movement of the boat in all 6 degrees of freedom (Figure 5.1).
- Different locations will be able to be specified, e.g. natural lake, reservoir, open sea etc.
- A course may also be specified such that boys appear on the water defining a course.
- Wind speed (with a maximum of 40 knots) and direction will be able to be specified by the user and randomly vary over time if required.
- Water/wave conditions will be able to be specified in terms of amplitude and frequency.
- There should also be wind and wave presets ranging from calm to stormy for quick set up of conditions.
- Equations and algorithms should all prioritise speed over accuracy, so that the simulator responds as realistically as possible in real time.
- The transient response of sudden changes, as well as the steady state should also be modelled (for example the oscillations of the boat as someone jumps onto the simulator.)
- Sounds will also be generated depending on wind speed/direction and water/wave conditions.
- There should be the option to connect to Google's 'Google Earth' software, to allow virtual sailing in a real location anywhere in the world.

5.2. Physical Simulation & Structural Design

- The simulator will be built around an existing dinghy hull, (either a Topper or a Laser) including modified parts of the tiller, boom and rigging, to ensure the most realistic interface possible.
- Three degrees of freedom will be simulated physically; heave, roll and pitch (see Figure 5.1) leaving yaw, sway and surge to be experienced by the user via the visual display.
- The roll must have a range that allows the simulator to reach a point where an actual boat would capsize. At this point the user is notified via the visual display.
- The pitch must have a range of at least ±15° (allowing for extreme weather conditions, on still water the variation in pitch will be far less than this.)
- The heave must have a range of at least 1000mm when the boat is on an even keel to allow simulation of extreme weather conditions. At maximum roll the actuators must maintain the ability to simulate heave of up to 200mm.
- The simulator must support the weight of 2 adults (a trainee and an instructor for example) at an estimated weight of 95kg each, including appropriate safety factors.
- The main hull actuators must provide accelerations of at least 1G so that the movements of the boat feel natural. This is particularly important with regards to heaving motion which is heavily affected by gravity.
- The boom and tiller will be driven by constant torque motors to allow the user to feel the effect of the wind in the sail and the effect of the water resistance on the rudder.
- The boom and tiller actuators must be able to apply enough torque to simulate the weather conditions specified above. They must also be able to apply this torque through the full 180° of movement required.
- The boom and tiller actuators will require position sensors and the main hull actuators will require load cells, to feedback to the computer.

- The simulator must be transportable, with a maximum weight of 1100kg and maximum dimensions 2000x4100mm when packed down to fit onto the back of a Transit Box Van. However, the weight and packed down size must be minimised where possible.
- The simulator should be able to be set up or packed down in under 2 hours by a team of 2 people. This is to allow single day usage on location.
- Safety features must be included to account for someone falling off the simulator or being hit by the boom.
- All the electronic components, the computer, actuators etc. must be hidden from view so that the user is fully immersed in the simulation, seeing only the boat around them and the visual displays.

5.3. Control System

- The computer must output to the main hull actuators values for; heave, roll and pitch mapped to values of extension/rotation for each actuator.
- The main hull actuators must in turn feedback load values mapped to an overall centre of mass which can be fed back into the simulation.
- The computer must output to the boom and tiller actuators values of torque to be felt by the user.
- The boom and tiller actuators must in turn feedback position/angle information to allow the user to control the boat.
- If there are conflicting requirements in the demand on the main hull actuators (i.e. cannot roll 30° whilst heaving 1m, due to actuator limitations) roll will take precedence followed by pitch and then heave. There should be no conflict with the boom and tiller actuators.
- The speed of the control system needs to take priority there should be no noticeable time lag between the user shifting their weight or moving the boom/tiller and the boat responding.

- The accuracy of the control system should suffer if it means an improvement in response time. A basic degree of accuracy is required of around 10%.
- The system should also have no noticeable ‘overshoot’ if a sudden change is made by the user.

5.4. User Interface (Display & Sound)

- The Visual display should have HD and 3D capabilities for a fully immersive experience.
- The display should be large (at the very least large enough for the front of the boat to be displayed actual size).
- A curved panoramic display should be used, either partly or fully surrounding the user.
- On the display the user will see; the front of the boat in its photorealistic environment, some kind of heads up display (including a compass) and visual indicators of wind speed and direction in the form of 3D animated arrows.
- The visuals should be displayed at an acceptable frame rate, no less than 30 frames per second, for fluid, comfortable viewing.
- A surround sound system should be able to convey realistic directional sound effects to the user, corresponding to the simulation, allowing the user to get a sense of wind speed and direction.
- The volume of the audio system must be sufficient to overpower any mechanical noise, from the actuators etc.
- All the audio and visual equipment should be able to ‘pack down’ to allow the whole simulator to be portable.

6. CAD Hull Analysis

For certain aspects of the computer simulation information needed to be known about the following parameters;

- **Submerged volume** - the total volume of water that the hull is displacing, providing buoyancy
- **Draft** - the level of the water up the side of the hull
- **Wetted surface area** - the surface area of the hull which is under the water

6.1. Submerged volume

Submerged volume at an equilibrium point clearly depends on the mass of the user and boat combined and is simply calculated using a force balance (Equation 6.1).

$$mg = \rho g V \qquad \therefore\ V = m/\rho$$

Equation 6.1

6.2. Draft

Draft is dependent on the total mass of boat and user, but is also dependant on the roll and pitch of the boat and is a function of the hulls specific profile. For this reason, analysis was done on the CAD model of the Laser Sail Boat to generate data for Draft at various values of roll angle and occupant mass. The mass of a Laser with all its fixtures and fittings is 98kg. The physical components of the simulator are designed to take 2 users weighing a total of 190kg. However, with the simulator predominantly for 1 person this upper limit will rarely be approached. Also, at these higher loads the boat will sit so low in the water sailing would become impractical. So for this analysis the range used was roughly 60-160kg (occupants), 160-260 (occupants + boat).

The CAD model was cut at different levels and angles with planes that represented the surface of the water (Figure 6.1).

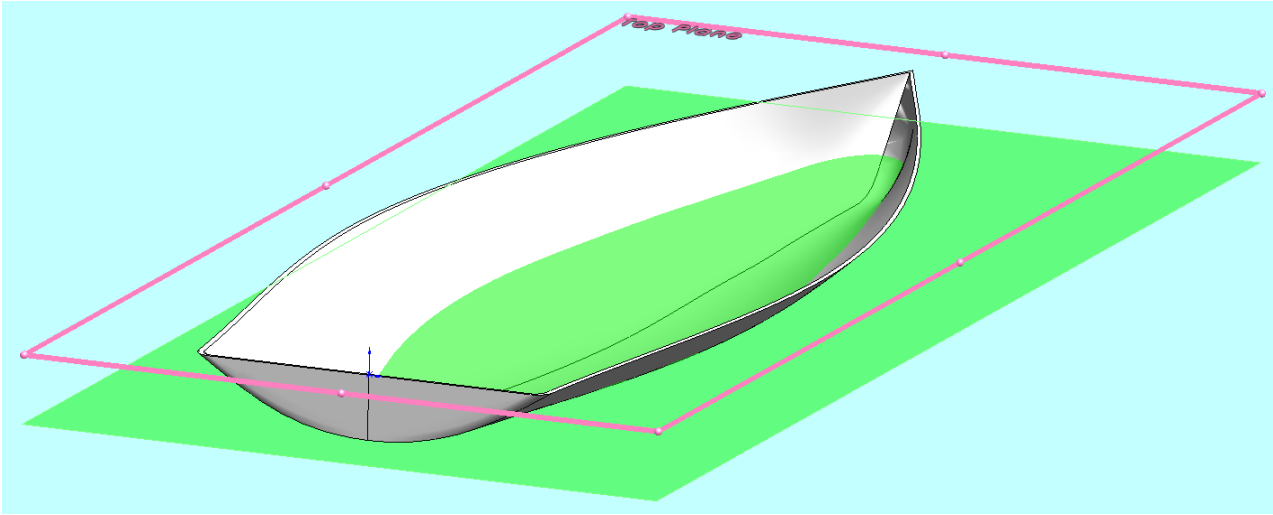


Figure 6.1 - Measurements being done on the hull CAD model using 'Solidworks'

The hull shell was then filled out to form a solid object so the CAD software could be used to calculate the volume of the hull under the plane (Figure 6.2).

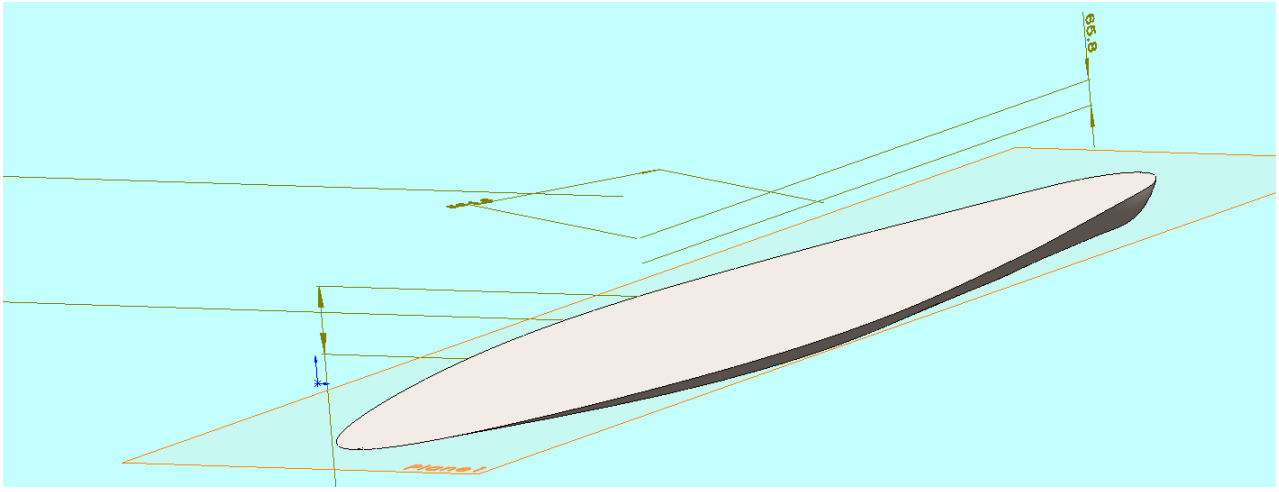


Figure 6.2 - Volume calculations being performed using 'Solidworks'

The raw data collected from the analysis is shown here (Table 6.1).

roll angle	Water Level	submerged volume	total mass	occupants' mass
0	200	0.1588	158.8	60.8
10	205	0.1512	151.2	53.2
20	210	0.1506	150.6	52.6
30	215	0.1579	157.9	59.9
0	190	0.1825	182.5	84.5
10	190	0.1866	186.6	88.6
20	195	0.1864	186.4	88.4
30	205	0.1826	182.6	84.6
0	180	0.2078	207.8	109.8
10	182	0.2068	206.8	108.8
20	188	0.2043	204.3	106.3
30	196	0.2058	205.8	107.8
0	170	0.2341	234.1	136.1
10	172	0.2335	233.5	135.5
20	178	0.2311	231.1	133.1
30	184	0.2393	239.3	141.3
0	160	0.2619	261.9	163.9
10	160	0.2672	267.2	169.2
20	166	0.2649	264.9	166.9
30	176	0.26234	262.34	164.34

Table 6.1 - Results for submerged volume and hence occupant's mass for varying input parameters

Draft is traditionally measured as the vertical distance from the keel to the waterline. In these results Water level is the vertical distance from the centre of the top plane (the deck of the boat) to the water line. A range of 30° was used for the roll angle because at this angle the water level began to come over the edge of the hull onto the deck. This range could be expanded at a later date if needs be.

Matlab's 'griddata' function was used to interpolate a surface from the non-uniformly distributed data (Figure 6.3). Matlab's 'interp' function was then used to interpolate values for water level given input values for occupants' mass and roll angle. This piece of code (Code 6.1) could then be integrated into the computer simulation.

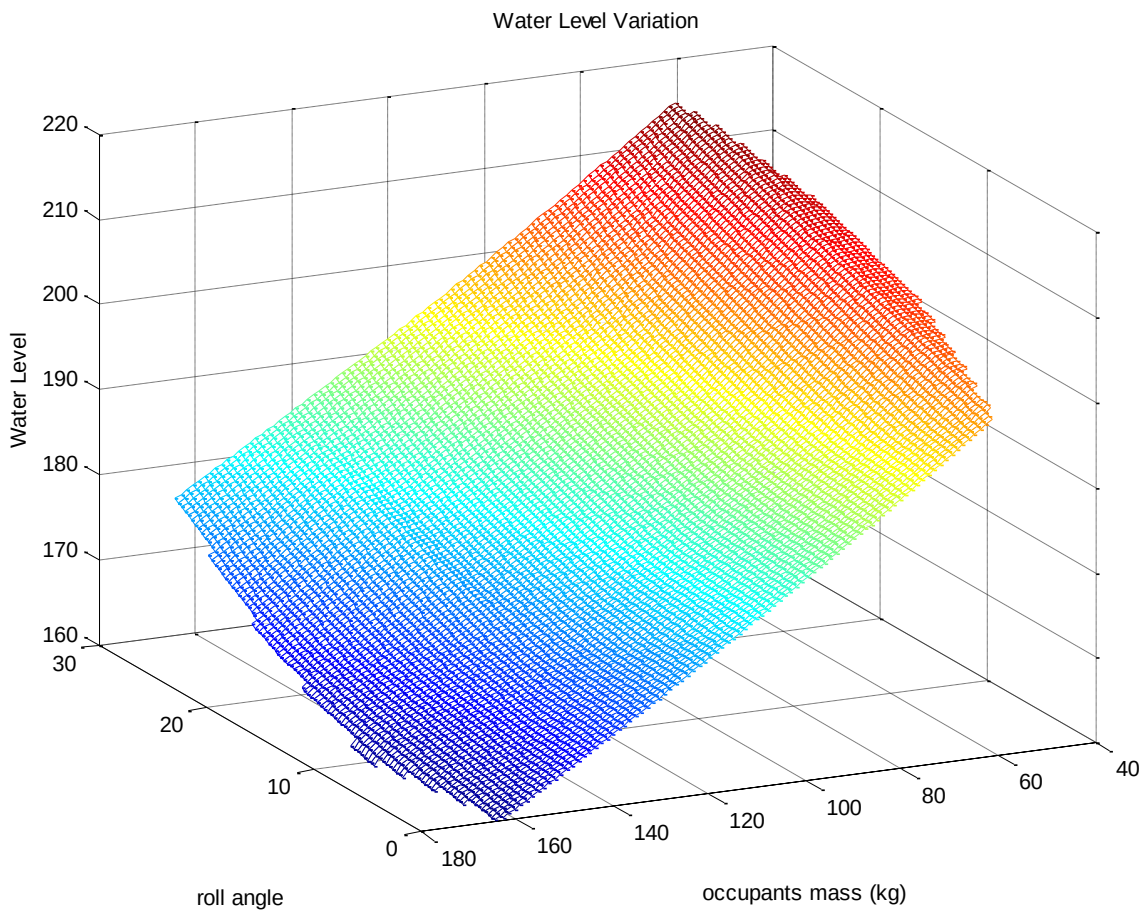


Figure 6.3 –Variation of the water level due to roll angle and occupants mass

```
r=[0 10 20 30 0 10 20 30 0 10 20 30 0 10 20 30 0 10 20 30];
m=[60.8 53.2 52.6 59.9 84.5 88.6 88.4 84.6 109.8 108.8 106.3 107.8 136.1 135.5 133.1
141.3 163.9 169.2 166.9 164.34];
wl=[200 205 210 215 190 190 195 205 180 182 188 196 170 172 178 184 160 160 166 176];

x=linspace(0,30);
y=linspace(52,170);

[X,Y]=meshgrid(x,y);

Z=griddata(r,m,wl,X,Y,'cubic');

WL=interp2(X,Y,Z,roll,mass);          % Water Level output
```

Code 6.1 - Implementation of measured values to calculate Water level for any given roll and mass

6.3. Wetted surface area

A similar process was carried out to obtain data for the wetted surface area which again depends on occupants' mass as well as pitch and roll angle. The same limits for mass and roll were used and the CAD software was used to calculate the surface area of the hull that was in contact with the water.

The raw data is shown here (Table 6.2) along with a mesh plot of the results (Figure 6.4)

occupants' mass	total mass	roll angle	wetted surface area
60.8	158.8	0	2.452
53.2	151.2	5	2.396
52.6	150.6	10	2.412
59.9	157.9	15	2.512
84.5	182.5	0	2.633
88.6	186.6	5	2.671
88.4	186.4	10	2.6905
84.6	182.6	15	2.703
109.8	207.8	0	2.804
108.8	206.8	5	2.8045
106.3	204.3	10	2.8085
107.8	205.8	15	2.859
136.1	234.1	0	2.96
135.5	233.5	5	2.9615
133.1	231.1	10	2.9685
141.3	239.3	15	3.0595
163.9	261.9	0	3.109
169.2	267.2	5	3.14
166.9	264.9	10	3.1525
164.34	262.34	15	3.1765

Table 6.2 - Results for wetted surface area for varying input parameters

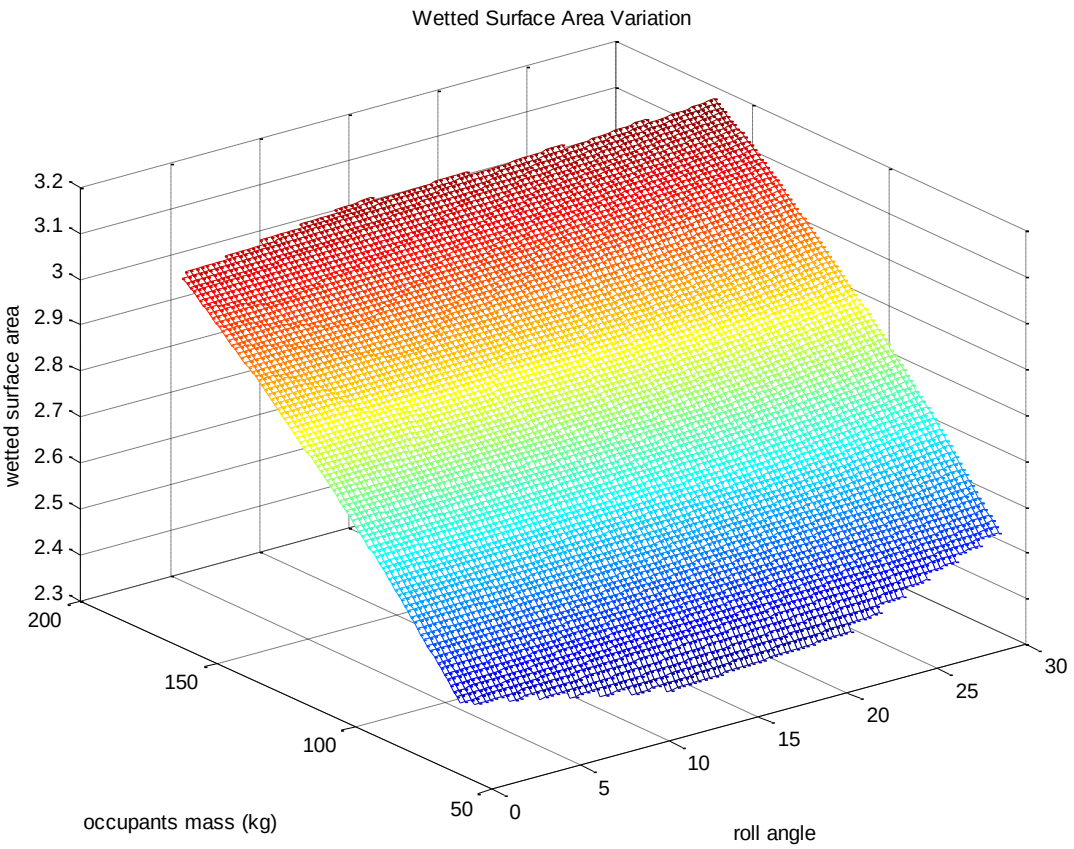


Figure 6.4 - Variation of the wetted surface area due to roll angle and occupants mass

Matlab's 'griddata' and 'interp' functions were again used to interpolate a surface from the non-uniformly distributed data and then interpolate values for surface area, given values of occupant's mass and roll angle. The following code (Code 6.2) which also includes the calculation for submerged volume and water level from the previous sections could then be integrated into the computer simulation, requiring inputs of 'roll', 'mass' and 'density' and outputting values for water level, wetted surface area and submerged volume.

```
% Raw data
r=[0 10 20 30 0 10 20 30 0 10 20 30 0 10 20 30 0 10 20 30];

m=[60.8 53.2 52.6 59.9 84.5 88.6 88.4 84.6 109.8 108.8 106.3 107.8 136.1 135.5 133.1
141.3 163.9 169.2 166.9 164.34];

s=[2.452 2.396 2.412 2.512 2.633 2.671 2.6905 2.703 2.804 2.8045 2.8085 2.859 2.96
2.9615 2.9685 3.0595 3.109 3.14 3.1525 3.1765];

wl=[200 205 210 215 190 190 195 205 180 182 188 196 170 172 178 184 160 160 166 176];

x=linspace(0,30);
y=linspace(52,170);

[X,Y]=meshgrid(x,y);

Z=griddata(r,m,wl,X,Y,'cubic');

Zs=griddata(r,m,s,X,Y,'cubic');

% Water level output
WL=interp2(X,Y,Z,roll,mass);

% Wetted surface area output
S=interp2(X,Y,Zs,roll,mass);

% Submerged volume output
V=mass/density;
```

Code 6.2 – Implementation of measured values to give outputs for Water level, Wetted surface area and submerged volume

The effect that roll had on these 2 outputs was minimal compared to the effects of the mass, especially in the results for wetted surface area where it had almost no impact. For this reason the effect the pitch has on wetted surface area and water level was not investigated. The simulator will only calculate these 2 outputs as functions of roll and mass. This will increase the processing speed of the simulation and have minimal effect on its accuracy.

7. Buoyancy

7.1. Vertical Forces

The resultant vertical force due to buoyancy is the difference between the weight of the boat and user and the weight of the displaced water. This force results in an acceleration that will bring the boat to an equilibrium point. Essentially, at equilibrium the mass of water displaced has to equal the mass of the boat and user combined. This allows the submerged volume and hence the draft to be found for any given weight. This process is described in the 'CAD Analysis' section (page 20).

$$F = mg - \rho g V$$

Equation 7.1

7.2. Roll and Pitch

A multitude of forces act on the boat to cause roll or pitch, such as the weight distribution due to the users position, wind in the sails etc. A restoring moment is provided by buoyancy such that the boat reaches an equalibrium position and is not an unstable system. This restoring force can be calculated by taking an integral over the submerged volume, either side of the centre of mass. This allows a moment balance to be generated (Equation 7.2).

$$Total\ moment = \iiint_{V_2} \rho g z - \iiint_{V_1} \rho g z$$

Equation 7.2

where z is the horizontal distance (always taken as positive) from the centre of mass. The moments due to self weight (W_1 & W_2) cancel because moments are taken about the centre of mass (Figure 7.1).

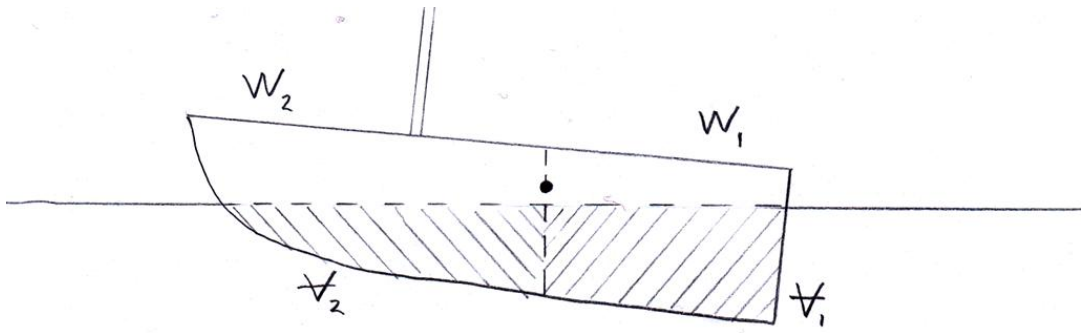


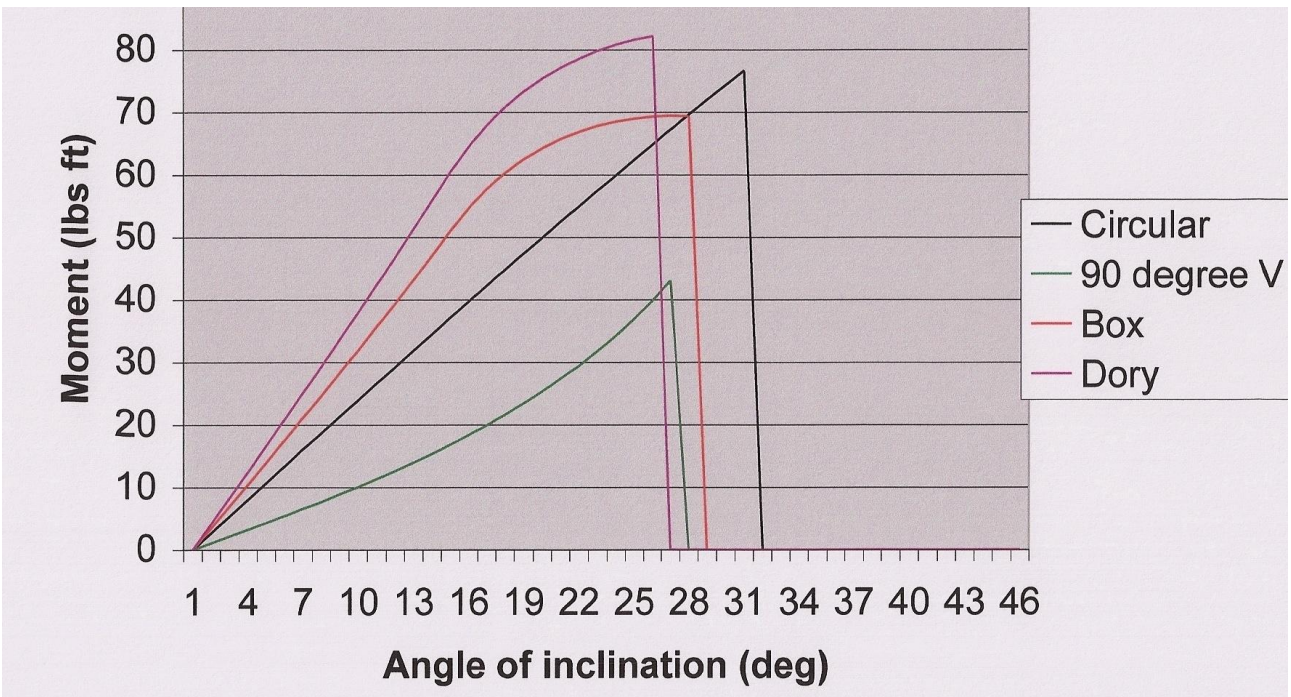
Figure 7.1 – Boat hull showing the volume integrals about the centre of mass

For the heeling motion, values have already been calculated for various hull shape approximations. The Dory is a similar hull shape to that of a Laser, so this approximation was used (Figure 7.2 – Restoring moment as a function or roll angle

(Kimball, John, The Physics of Sailing, CRC Press, New York, 2010). To implement this, data points from this curve will be stored in the simulator so that using interpolation the righting moment can be calculated for any roll angle. The units also need converting to allow integration with the rest of the simulator; $moment\ (N\ m) = 4.45 \times 0.305 \times moment\ (lbs\ ft)$.

Figure 7.2 – Restoring moment as a function or roll angle

(Kimball, John, The Physics of Sailing, CRC Press, New York, 2010)



To simplify the integration for the pitch, the hull was approximated to a half cylinder of height 3.961m (the length at water line of a Laser) and diameter 1.446m (the width of a Laser at its widest point). The centre of mass, found from the CAD model is; [2.57, 860.99, 2092.75] in millimetres and measured from the front tip of the hull, with the x direction being port/starboard, y being up the mast and z being down the length of the boat. Assuming that the centre of mass is on the centre line of the boat (reasonable as it is only 2.5mm out) the problem can be simplified still further so that you are integrating over an area rather than a volume (Figure 7.3). The submerged volume at any one time is known so the ratio of submerged volume to submerged area can be used to scale the results from the area integrals, giving a value for total restoring moment.

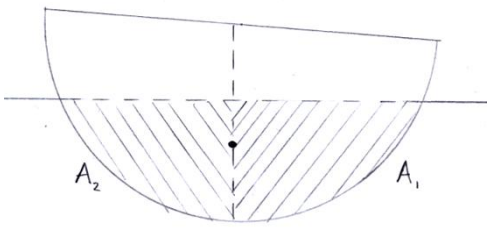


Figure 7.4 – Hull simplification, end view

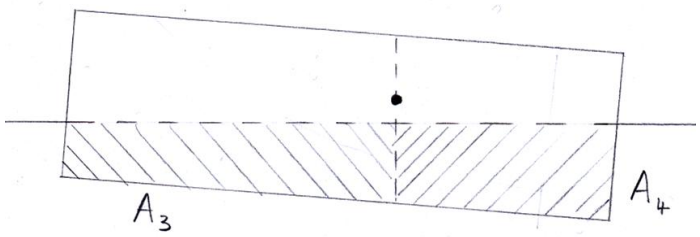


Figure 7.3 - Hull simplification, side view

7.3. Users Weight Distribution

As the wind tries to pull the boat over in one direction the sailor will lean out over the edge of the boat, counteracting the heeling motion. This is called hiking and is one of the main effects that the buoyancy has to counteract. So we needed a way for the simulator to find the position of the user, more specifically, the centre of mass of all the users weight (this may include 2 people in different positions). To do this, actuators were chosen for the main hull that had load cells built into them. The actuators are positioned in a rectangular lay out (Figure 7.5) so taking a force input from each actuator it is relatively straight forward to find the centre of mass (Equations 7.3).

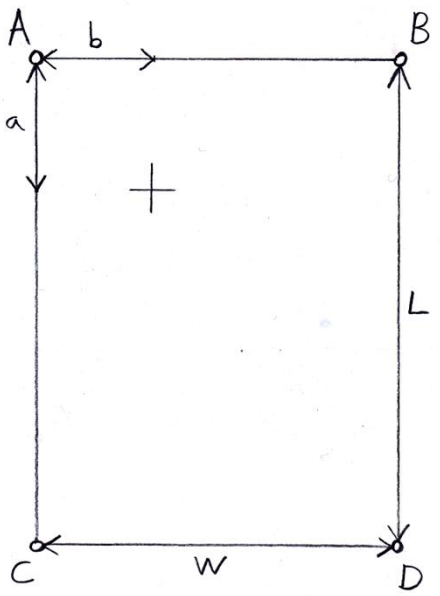


Figure 7.5 – Layout of main hull actuators, A,B,C and D, showing centre of mass at coordinates (a,b)

$$(F_A + F_B)(a) = (F_C + F_D)(L - a)$$

$$(F_A + F_c)(b) = (F_B + F_D)(W - b)$$

$$\therefore a = \frac{(F_C + F_D)L}{F_A + F_B + F_C + F_D}$$

$$b = \frac{(F_B + F_D)W}{F_A + F_B + F_C + F_D}$$

Equations 7.3

These equations (Equations 7.3) give the coordinates of the centre of mass taking actuator A as the origin. These were mapped to the global coordinate system for integration with the rest of the simulation. The vertical component for centre of mass is irrelevant as the users weight always acts vertically.

It is also important to note that we want the centre of mass of the user relative to the boat. Not the centre of mass of the whole system including the boats weight. The forces used (Equations 7.3) are therefore calibrated values equal to the force fed back from the load cell minus the force due to the self weight of the stationary simulator on each actuator.

8. Hydrodynamic Drag

8.1. Governing Physics

For a flat plate of length l and breadth B, dimensional analysis leads to the following result (Equation 8.1).

$$Drag = \frac{1}{2} C_d(Re, \frac{B}{l}) \rho S U^2$$

Equation 8.1

where the drag coefficient (Cd) is a function of Reynolds number and the dimensions of the plate. Here ρ is water density, S is wetted surface area and U is the velocity of the flow.

For flow around a bodie the drag coefficient can be split into two parts (Equation 8.2); a frictional drag coefficient that is a function of Reynolds number and a 'pressure drag' coefficient. This pressure drag arises due to turbulent flow and separation at the surface of the body.

$$C_d(Re) = C_f(Re) + C_p$$

Equation 8.2

For a boat's hull another term arises in the drag coefficient called 'wave resistance'. This is not resistance due to waves on the surface of the water but arises because the boat generates its own waves, known as a wake, as it pushes through the water. This expends energy and therefore appears as a term in the total drag coefficient, working against the forces that propel the boat forwards. This wave drag is a function of Froude number and when combined with the pressure drag is known as residual drag. This gives an expression for the total drag coefficient as a function of Reynolds and Froude number (Equation 8.3).

$$C_d(Re, Fr) = C_f(Re) + C_r(Fr)$$

Equation 8.3

$$Re = \frac{Ul}{\nu} \qquad Fr = U/\sqrt{gl}$$

Equation 8.4

where v is the dynamic viscosity of water and l is a characteristic length usually defined as the 'length at water line' (LWL). Note that this is an approximation to assume residual drag is only a function of Froude number. It is strictly a function of both Reynolds and Froude number.

8.2. Finding the Drag Coefficient

The ITTC (International Towing Tank Conferences) have a standard for calculating the frictional drag coefficient (Figure 8.1). Although these standards are for much larger hulls than the laser, they provide a good approximation as they scale with Reynolds number. With these larger hulls, a roughness allowance is added due to wielding seams, rivet heads etc. For the smooth monocoque hull of the laser this is not necessary. The ITTC line is described by the following equation (Equation 8.5).

$$C_f(Re) = \frac{0.075}{(logRe-2)^2}$$

Equation 8.5

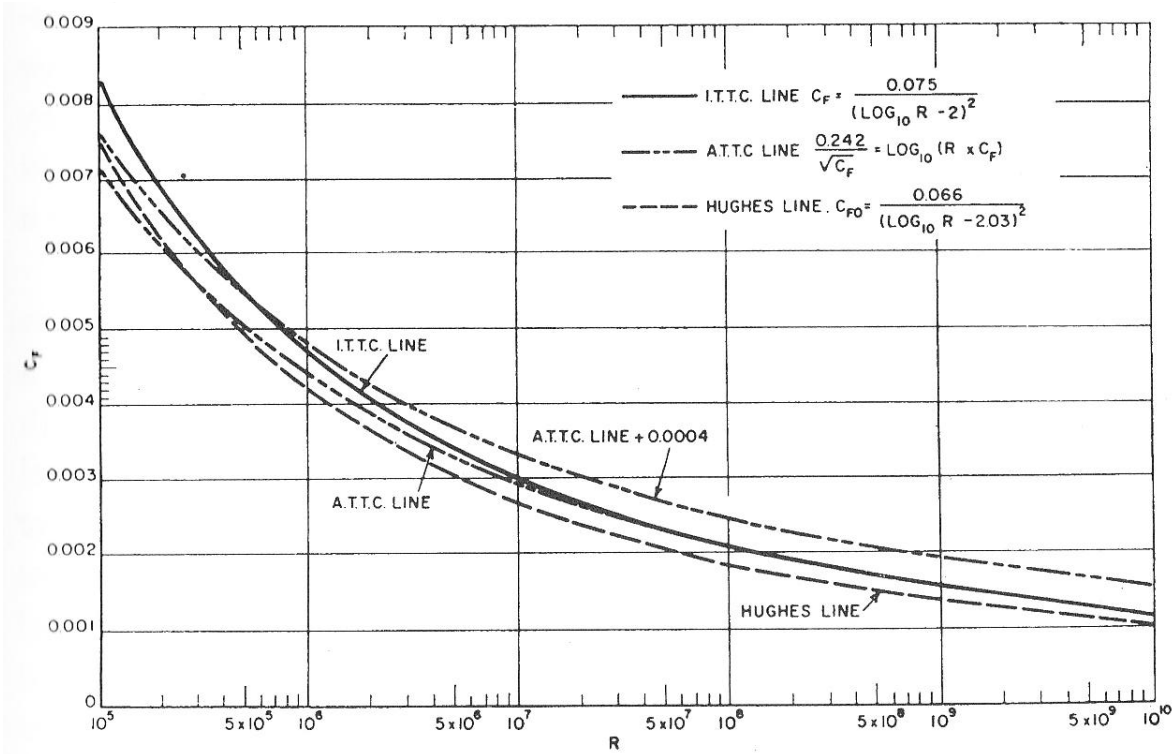


Figure 8.1 – Frictional drag coefficient as a function of Reynolds number

(J.N.Newman, Marine Hydrodynamics, The MIT Press, London, 1977)

With the information about frictional coefficients from the above graph, model testing can be done to determine the residual drag coefficient. A model test, scaled at constant Froude number would allow the residual drag coefficient to be found for the model and hence the total drag coefficient for the actual hull (Equation 8.6).

$$(C_d)_{boat} = (C_d)_{model} - (C_f)_{model} + (C_f)_{boat}$$

Equation 8.6

Fortunately a lot of tests have been done with model hulls of various shapes and sizes allowing the residual drag coefficient for a given Froude number to be read from a graph (Figure 8.2 - Graff, Kracht and Weinblum 1964). For the laser hull the 'measured shallow-draft' curve was used.

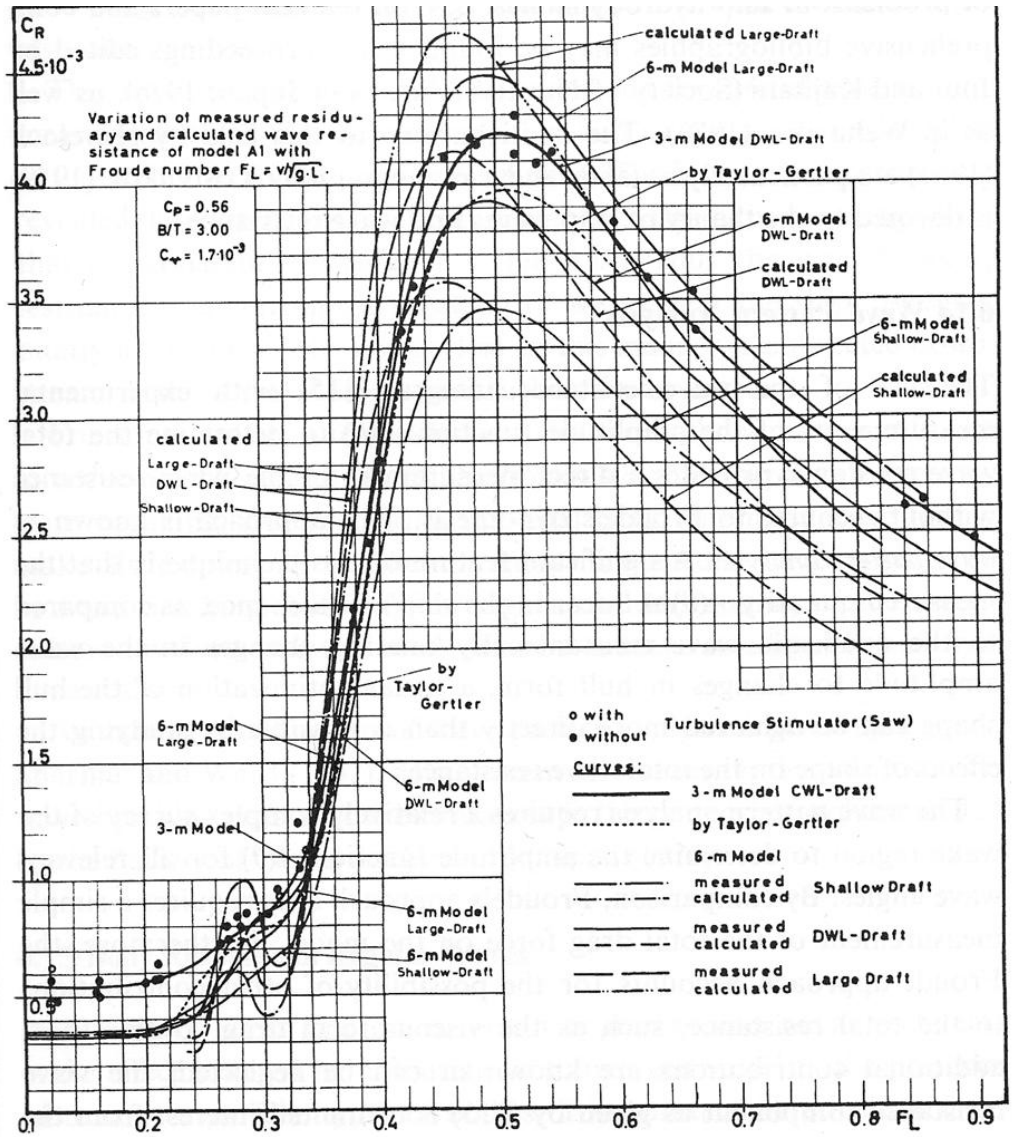


Figure 8.2 – Residula drag coefficient as a function of Froude number

(Graff, Kracht and Weinblum, 1964)

A Rather crude process was used to calculate the hydrodynamic drag resisting the sideways drift of the hull. The hull was assumed to be equivalent to a flat plate moving normal to the flow with an area equal to half of the wetted surface area. The coefficient of friction can be assumed to be assumed constant at about 1.28 allowing the sideways drag to be calculated (Equation 8.7).

$$D_{drift} = \frac{1}{2}\rho U_{drift}\left(\frac{1}{2}S\right)C_d$$

Equation 8.7

8.3. Implementation

The residual drag curve is not described by a function, it is an experimental result, so a set of 44 points on the curve (Figure 8.2) were used to recreate it (Figure 8.3). It was also extended to reach a Froude number of zero. There was an attempt to fit a 10th order polynomial to the curve so that drag coefficients could be calculated from equations only and the simulator would not have to store data points. However, the fit was inadequate, particularly at low speeds where the drag coefficient should remain constant (Figure 8.3). As an alternative the simulator stores the data points and interpolates between them for any given Froude number. Obviously you cannot interpolate outside of the data range but with data up to a Froude number of about 0.8, corresponding to a speed of about 5 m/s (about 10 Knots) this will not be an issue.

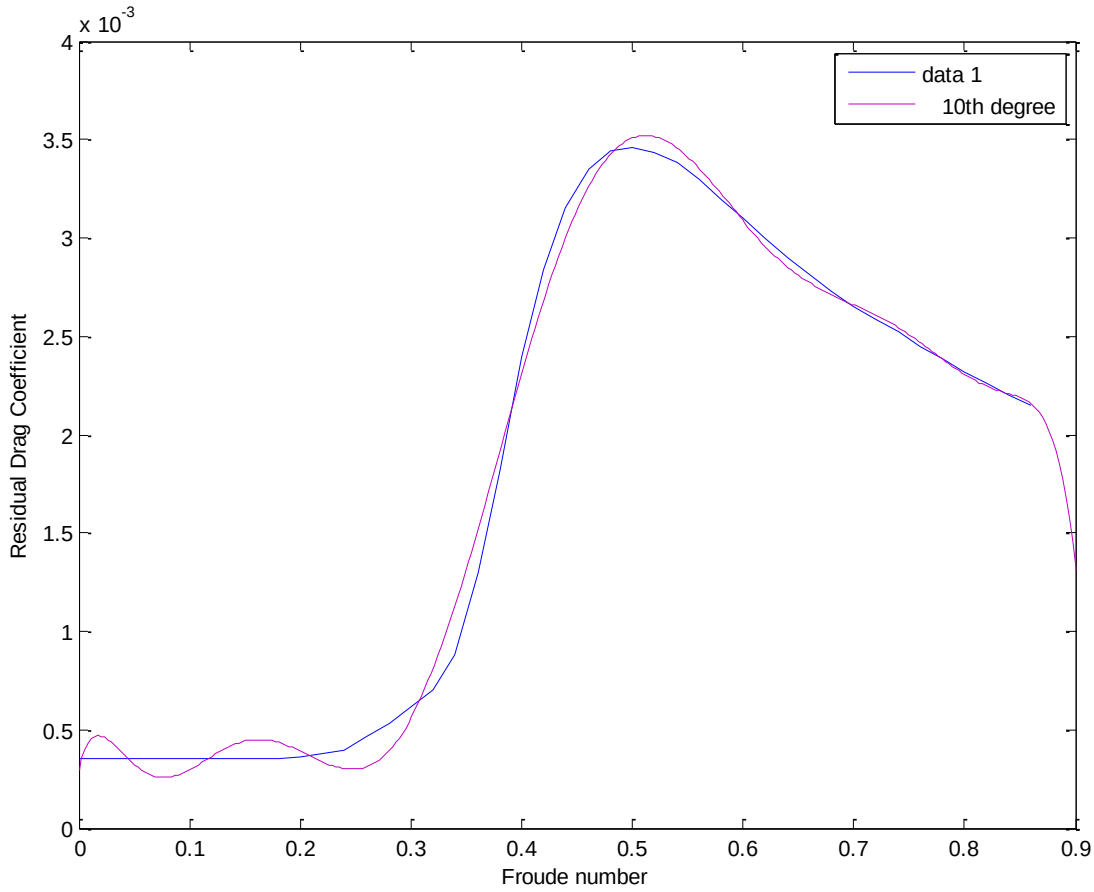


Figure 8.3 – A 10th degree polynomial estimate for residual drag coefficient, compared to experimental values

The process for calculating hydrodynamic drag forces within the simulation is therefore;

- Take the current values for the Velocity, Drift Velocity and Wetted Surface Area variables.
- Calculate Froude number and Reynolds number (Equation 8.4)
- Calculate the frictional drag coefficient (Equation 8.5) and interpolate to obtain the residual drag coefficient (Figure 8.2). Then add the two to obtain the total drag coefficient (Equation 8.3).
- Finally Calculate the total drag forces for forward motion (Equation 8.1) and for drift (Equation 8.7).
- These forces are fed into the rest of the simulation giving updated values for the Velocity, Drift Velocity and Wetted Surface Area variables.

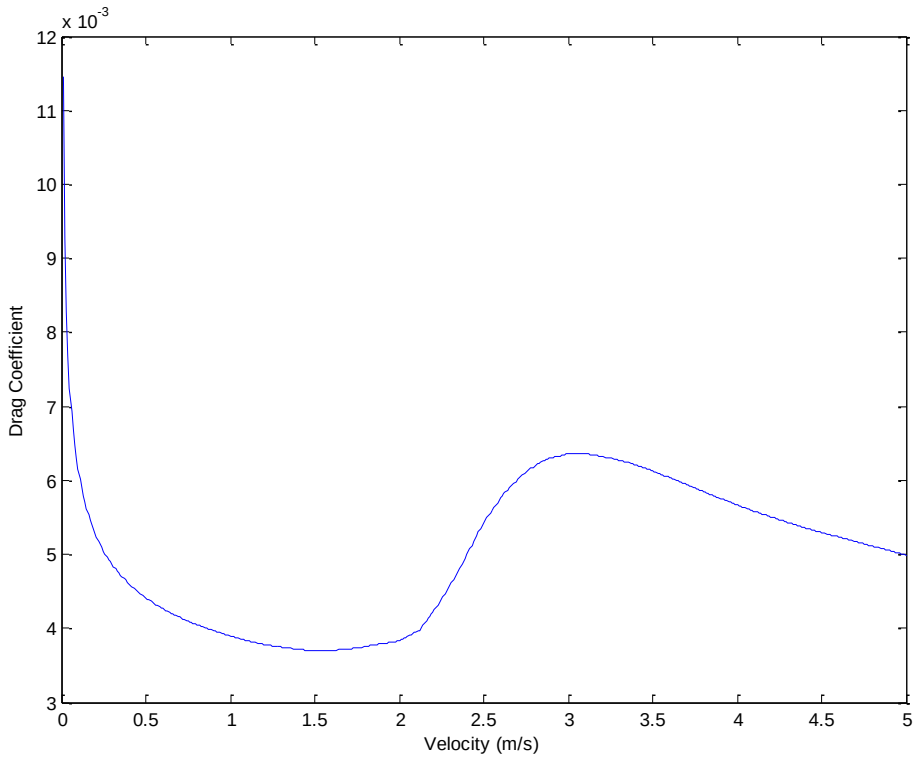


Figure 8.5 – Calculated drag coefficient as a function of velocity for a wetted surface area of 3m^2

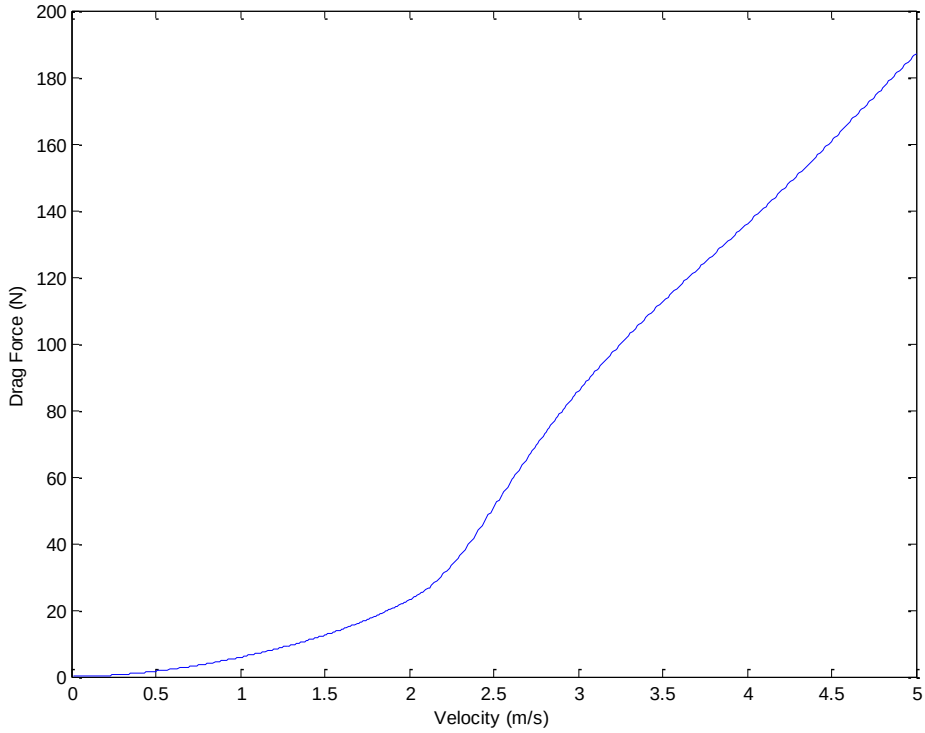


Figure 8.4 - Calculated drag force as a function of velocity for a wetted surface area of 3m^2

From these graphs you can clearly see the point at which the increasing residual drag coefficient begins to dominate over the decreasing frictional drag coefficient, suddenly increasing the total drag force at about 2 m/s (about 4 knots).

9. Sail, Rudder and Centreboard

9.1. Underlying Physics + Equations

The sail, rudder and centreboard all share the same underlying principles as they are all aerofoils; the centreboard has it's angle of attack varied by apparent water angle, the rudder by the sum of apparent water angle and rudder angle and the sail by the sum of apparent wind angle and sail angle. (Figure 9.1) shows how the apparent wind (and water) vectors are calculated. Alongside this there is a static pressure generated on the side of the rudder and centreboard due to the water pressure, represented as a function of roll and apparent water angle. (Equation 9.1) shows how the standard equation for force on a submerged flat plate is multiplied with an area projection to take into account the roll and angle of the surface with respect to the water. A dynamic pressure component would also act on the side of all the components when the boat is rolling, however as this is small it was neglected to favour a quicker running code; in keeping with the initial design ethos. This underlying similarity means that the sail, rudder and centreboard follow the same two driving equations (see Equation 9.2).

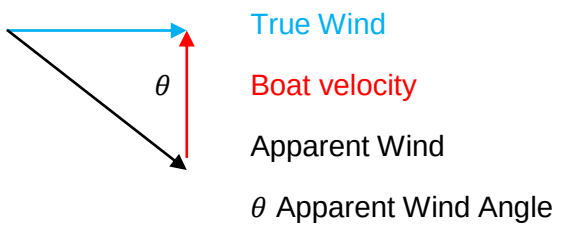


Figure 9.1: Apparent wind vector

$$Force_{90} = \rho g h_{avg} A$$

$$Force = Force_{90} \times \sin\theta_{roll} \times \cos\theta_{app\ water}$$

Equation 9.1: Static hydraulic force

$$Drag = C_d \frac{\rho v^2}{2} A \qquad Lift = C_L \frac{\rho v^2}{2} A$$

Equation 9.2: Surface general equations

It is obvious that the lift acts perpendicular to the aerofoil, with the drag acting parallel to the chord. To calculate torques, a point of action was required for the forces and this was taken as the centre of pressure for lift and hydraulic pressure, and the trailing edge of the surface for drag.

At higher levels of sailing varying sail and centreboard profile is used to keep the boat in trim and in balance. This will be briefly discussed here as a scope for future work however it was not included

in the simulation as it would have required the inclusion of a full height mast and centreboard which would have in turn made the simulator much larger and caused numerous problems with assembly, disassembly and transport.

The centreboard is usually left at its natural fully submerged position, causing a hydrodynamic resistive moment to the heeling moment generated by the sail thus helping to keep the boat upright and in turn preventing air from “spilling” out of the top of the sail. It also prevents the boat from sliding sideways in the water, allowing the force generated by the sail to translate into useful forward drive. However, when on a reach, the centreboard is raised to reduce drag on the hull as the heeling moment generated by the sail is minimal. When running the centreboard can be raised even more for the above reasons. Counter intuitively, raising the centreboard slightly (10-15cm) can also be used to stop a boat heeling under strong winds or when beating; this is a trade off however as it works by increasing the boats leeway hence loosing forward velocity to horizontal drift.

Reefing the sail (reducing its area) is used to reduce a boats heel. This is done by reducing the sail area and lowering the centre of pressure, in doing so reducing the overall heeling moment generated by the sail. The side effect of this is a reduction in sail profile efficiency and hence a loss in forward drive. There are three main methods of doing this; slab reefing (lowering by around one quarter of the luff and tightening at the bottom of the sail), in-mast reefing (vertically rolling the sail into the mast) and in-boom reefing (horizontally rolling the sail into the boom).

A third adjustment comes in the form of raking the mast; tilting it slightly towards the boat aft resulting in greater stability. This is because the centre of pressure moves rearward pushing the boat into the wind; similar to a weather vane. This action is called “weather helm”. This is also a safety feature as it means the boat naturally points into the wind and waves in heavy gusts of wind preventing capsize. Too much rake however exaggerates this effect meaning rudder needs to be constantly applied to keep the boat on course which greatly increases drag thus greatly decreasing boat speed.

9.2. Control

The drag and lift coefficients generate the link between angle of attack and force. Graphs of coefficient against angle of attack were taken from "Development Of A More Realistic Sailing Simulator"[12] (see Figure 9.2), inserted into MATLAB and then equations reproduced by using an adjusted and scaled (to stop poor conditioning for single values) 10th degree polynomial. This gave a good fit except from the extremes of the polynomial (around 20 and 160 degrees), were there was an error in the region of 0.05 (see Figure 9.3). Due to the nature of the fitting, negative values gave very poor results so negative angles of attack in the code were converted to positive for the calculations, and then the final coefficient reverted back to the correct negative value; essentially mirroring the graphs in the x axis.

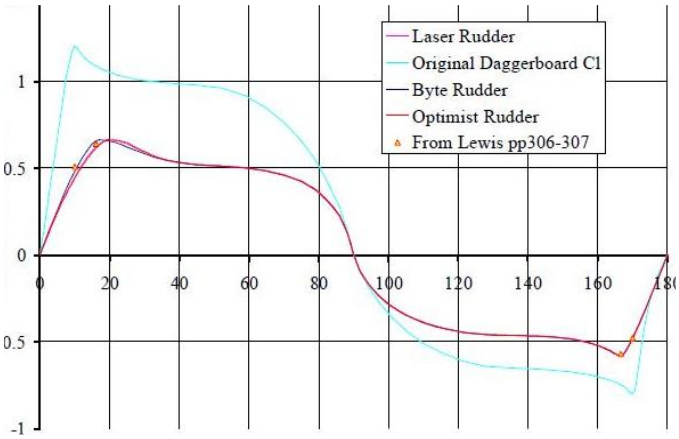


Figure 9.2: Binns et al – rudder lift coefficient vs angle

As all the surfaces, in particular the sail, are non-symmetric shapes code was written which broke the surfaces down into small 1mm thick rectangular strips which were then treated as symmetric. This also enables the code to be easily transferred to a wide variety of sail shapes whilst giving an accurate centre of pressure location. To speed up the discretization process the sail was initially broken down into 4 horizontal strips comprising of a rectangle and a triangle each (see Figure 9.4). This then allowed a looped code to further break these down into the desired 1mm strips quickly and efficiently by adding a constant length incrimination to the previous length of 1mm strip each loop due to the constant slope of the triangle

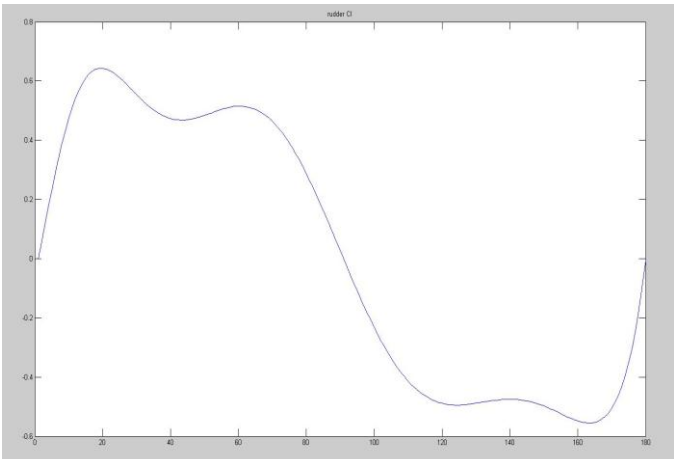


Figure 9.3: MATLAB rudder lift coefficient vs angle

[12] J.R. Binns, F.W. Bethwaite & N.R. Saunders

hypotenuse. Within each loop the force and resulting torque generated by each strip was calculated and these were then continually summed so that only two values had to be stored at one time; the previous total and the new additional value (see Code 9.1). The centre of pressure was taken as ¼ of the way down the chord length behind the leading edge (see Figure 9.4). The above driving equations were applied, and the forces and torques then summed to give a total force and torque for each surface. Up till this point the code treated each surface in its own co-ordinate system with the exception of the centreboard which is fixed in the global (boat hull) system (x = port, y = up mast, z = towards aft).

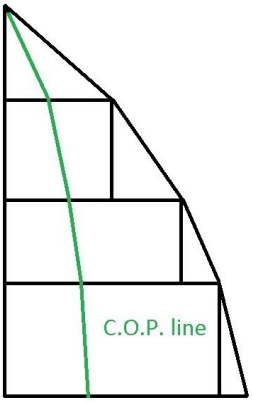


Figure 9.4: C.O.P. line

```
for i=1:h3;

   area=ch2+(i*1*(ex2/h3));                    %calculate strip area
   lift=Cl*(0.5*ro_air*(v_wind^2))*area;       %lift F on strip
   torque_y=lift*(0.25*(ch2+(i*(ex2/h3))));    %associated torque
   torque_z=lift*((hT-h2-h1-i)+sail_height);   %associated torque

   drag=Cd*ro_air*((v_wind^2)/2)*area;         %drag F on strip
   torque_x=drag*((hT-h2-h1-i)+sail_height);


   section3_lift=lift_old+lift;                %add up total lift
   section3_torque_y=torque_y_old+torque_y;    %add up total torque
   section3_drag=drag_old+drag;                %add up total drag
   section3_torque_z=torque_z_old+torque_z;    %add up total torque
   section3_torque_x=torque_x_old+torque_x;

   torque_y_old=section3_torque_y;
   lift_old=section3_lift;
   torque_z_old=section3_torque_z;
   drag_old=section3_drag;
   torque_x_old=section3_torque_x;

   area_3=area;
end
```

Code 9.1: Code for calculating F and T on sail quarter strip

$$R_{xyz} = \begin{bmatrix} \cos\theta & 0 & -\sin\theta \\ 0 & 1 & 0 \\ \sin\theta & 0 & \cos\theta \end{bmatrix}$$

Equation 9.3: Rotation matrix

Code was then written to rotate each of the force and torque vectors into the global co-ordinate system using the Euler angles approach, taking the rotation angle as the rudder and sail angle respectively (see Equation 9.3).

The code was adjusted so that it took into account the offset of the surfaces from the boats centre of rotation for each axis, and then all the toque and force vectors were summed to give a global vector acting on the boat hull.

An intuitive layout at the start of the code was used to make entry of boat dimensions easy and this, combined with the discretization of the surfaces into strips led to the code being very versatile and can be easily transferred to any boat and sail/rudder/centreboard shape for single sail yachts and dinghies. (Figure 9.5) shows the inputs required for the code and the outputs generated.

Inputs	**Outputs**
Boat dimensions	Global torque vector
Sail, rudder and centreboard dimension	Global force vector
Location of boat centre of rotation	Sail, rudder local torque vector
Apparent wind vector	Sail, rudder local force vector
Apparent water vector	
Roll angle	
Sail angle	
Rudder angle	

Figure 9.5: Table of code inputs and outputs

Finally, a test code was written which varied the sail and rudder angles whilst keeping all other variables constant. The results were then plotted to ensure the code and driving equations reproduced accurate responses. (Figure 9.6) shows the plot of the y-torque when the rudder angle was varied between 0 and 180 degrees with the sail, water and wind angles set at 0. The green graph represents rudder rotation to the starboard (right) side with red representing rotation to port (left). This is the expected response as it tracks the lift coefficient variation with angle, slightly distorted by the inclusion of rudder drag. It should be noted that in reality the rudder can only be varied +/- 90 degrees however the full 180 degree was tested to ensure the code was robust. Due to the nature of the fitting and the real, non-ideal data, some of the surfaces generated very minor

torques (<1Nm) at 0 and 180 degree positions where it would be expected there would be 0. However the offset is so small it could not be noticed by the user and it was decided not to address the problem further and keep the code as slim as possible and adhere to the original “speed over accuracy” design ethos.

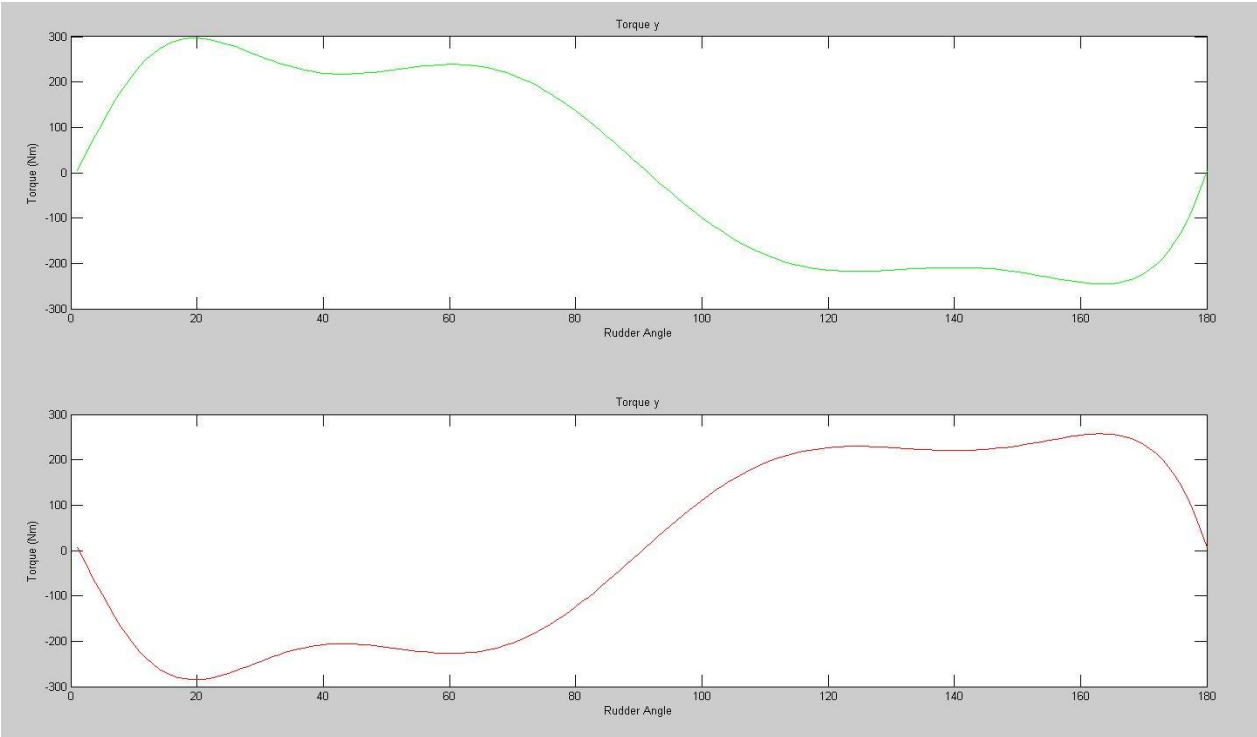


Figure 9.6: Torque around boat y-axis with varying rudder angle

10. Waves and Wave Effects

An important part of the hydrodynamics relating to a sailing simulator is the waves. Visually they are needed to represent a level of realism associated with the environment. Physically, the forces from waves on a boat may be subtle, especially if you consider the sailing environment being a calm lake, but can still affect the realism of the simulation. The wave effects are also important for use in medical research. The wave effects and forces were therefore explored.

10.1. Specification

It was specified that the simulation will include waves which will impact on the boat.

10.2. Waves

Theory

The types of waves we want to represent are free surface waves, where the fluid is assumed ideal (i.e. neglecting viscosity) and the motions are sufficiently small to be linearised. Nonlinear effects such as the breaking of waves will be ignored for the purpose of this simulator.

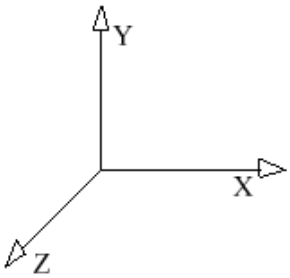


Figure 10.1 – Coordinate system.

The fluid velocity $\boldsymbol{V}$ is expressed by the gradient of a velocity potential $\emptyset$ such that $\boldsymbol{V} = \boldsymbol{\nabla}\emptyset$. The free surface condition gives rise to a single boundary condition for the velocity potential, $\frac{\partial^2 \emptyset}{\partial^2 t} + g\frac{\partial \emptyset}{\partial y} = 0$, on $y = 0$, which is the plane of the free surface and the y-axis positive upwards. g is the acceleration due to gravity. The vertical elevation of any point on the free surface is defined by the function $y = \sigma(x, z, t)$.

The simplest, and most important, solution to the boundary condition above is the plane progressive wave. In 2D this can be written in the general form $\sigma(x, t) = A\cos(kx - \omega t)$. A is the amplitude of the wave, ω is the radian frequency and k is the wavenumber. Important relationships are that $k = \omega/V_p = \omega^2/g$ (from the boundary condition) where V_p is the phase velocity, and

$k = 2\pi/\lambda$ where the wavelength λ is the distance between successive points on the wave with the same phase.

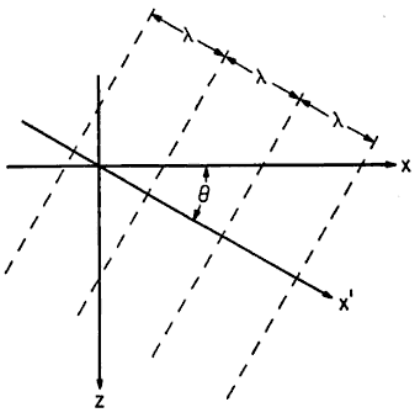


Figure 10.2 – View from above a plane progressive wave system. Wave crests are dashed lines.

In three dimensions, a wave can be represented as moving in an arbitrary direction, at a polar angle θ in the horizontal x-z plane (see Figure 10.2). The wave is moving along the x' axis, and since $x' = x\cos\theta + z\sin\theta$, the plane progressive wave in 3D can be generalised as $\sigma = A\cos(kx\cos\theta + kz\sin\theta - \omega t)$.

In an environment such as a lake or ocean, there is not just a single propagating wave, but rather a collection of individual waves propagating at different speeds, in different directions, and with different wavelengths. These different waves arise from a range of different sources, for example, wind, reflection and moving objects in the water. While there may be a dominant wave (dominant in terms of power), it would be inadequate to only represent the surface of the water as one plane progressive wave. The water profile can therefore be written as a sum, or superposition, of individual waves, $\sigma = \sum_n^N A_n \cos(k_n x\cos\theta_n + k_n z\sin\theta_n - \omega_n t)$ for N sources.

In the case of a large lake or ocean, the dominant source of waves is the wind. Other sources such as bodies in the water and Coriolis acceleration exist, but can be neglected as an approximation. A level of realism is desired such that the water surface looks realistic.

The wind interacts with the surface of the water in a number of ways. Turbulence produces random pressure fluctuations and subsequently small waves on the surface (wavelengths of a few centimetres). Wind acts on smaller waves, creating pressure differences along the wave profile which causes it to grow. Finally, waves interact with each other and transfer energy, creating longer and faster waves. These processes are far too complex to physically model, so an empirical approach can be taken. From data collected, the frequency spectrum

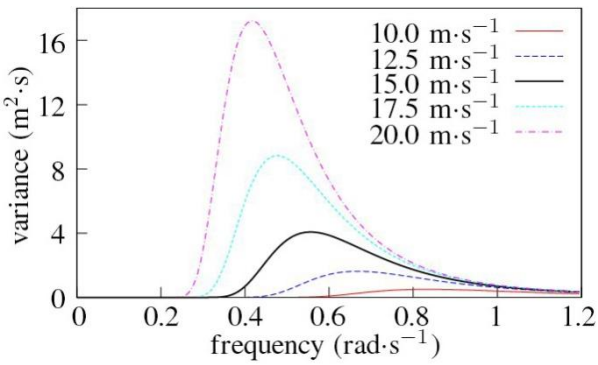


Figure 10.3 – Pierson-Moskowitz Spectrum.

of fully developed waves caused by a steady wind which has blown for a long time over a large area can be described by $S(\omega) = \frac{\alpha g^2}{\omega^5} e^{(-\beta(g/U\omega)^4)}$ where $\alpha = 8.1 \times 10^{-3}$ and $\beta = 0.74$. This is the Pierson-Moskowitz spectrum, and produces a spectrum as shown in Figure 10.3. There are other spectrums[13], such as JONSWAP spectrum, but these are very similar to the Pierson-Moskowitz spectrum which can therefore be used to a satisfactory level of accuracy.

From this spectrum, amplitudes and wavenumbers can be calculated, which gives us a realistic profile of the waves in the water.

Computational Implementation

The implementation of such a wave model is fairly straightforward. While the final simulator will be built in C++, Matlab provides a suitable environment for prototyping during the design phase.

The frequency spectrum must be discretised and the discrete Pierson-Moskowitz spectrum, amplitudes and wavenumbers computed. This can all be pre-computed for different wind speeds, which reduces real time computation.

```
% Discretize the frequency spectrum
dw = 0.1;
w = dw:dw:2;
% Set wind speed and direction
speed = 20; %wind speed @19.5m above surface
dir = 0;
% Compute frequency spectrum from
% Pierson-Moskowitz formula
Sw = (8.1e-3*world.g^2./(w.^5)).*exp(-0.74*(world.g*((speed*w).^-1)).^4);
% Compute amplitudes of wave spectra
% (http://en.wikipedia.org/wiki/Sea_states)
A = 0.5*sqrt(2*Sw*dw);
% Compute wave numbers
k = w.^2/world.g;
```

Code 10.1 – Pre-computation of Pierson-Moskowitz spectrum and wave characteristics.

The different frequency components can then simply be summed up with the function in Code 10.2 overleaf.

[13] For more information, http://www.wikiwaves.org/index.php/Ocean-Wave_Spectra provides a good resource.

```
function Z = generateWaves(w, Z, X, Y, A, k, dir, ti)
% w: frequency spectrum, Z: previous height profile
% X,Y: horizontal axis meshgrid, A: amplitude vector
% k: wavenumber vector, dir: direction of waves
% ti: current time value

for n=1:length(w)
    Z = Z + A(n)*cos(k(n)*(cosd(dir).*X + sind(dir).*Y) - w(n)*ti);
end
```

Code 10.2 – Superposition of waves function.

While this code is written to run in real time at the current time step, `ti`, it could be pre-computed to reduce real time processor load.

This computation produces the result as shown in Figure 10.4, which has shading and lighting to enhance the effect. Clearly, the profile is good but lacks realism as the waves are regular and all in a single direction. A real water surface consists of waves from multiple wind sources, as well as random fluctuations. Multiple random strengths of wind can be added to the simulation with directions from 0° to 90° in discrete steps. These secondary sources of wind can have random amplitudes. A primary wind can also be defined with a specific direction to represent a dominant local wave producing wind. Finally, some random noise is added, and smoothed over time and space by a smoothing filter, to increase the effect of randomness.

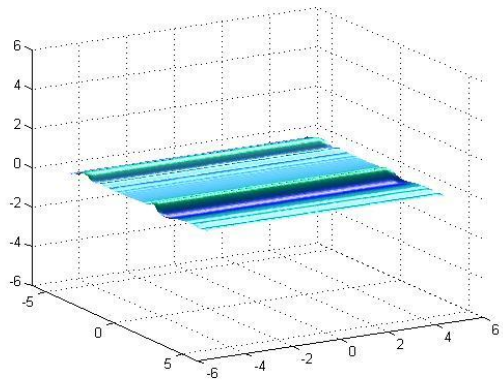


Figure 10.4 – Wave profile for a single wind source.

For a 5° discretisation in secondary wind directions, with the primary wind at 40 knots at 0°, the result is a realistic water surface, as shown in Figure 10.5.

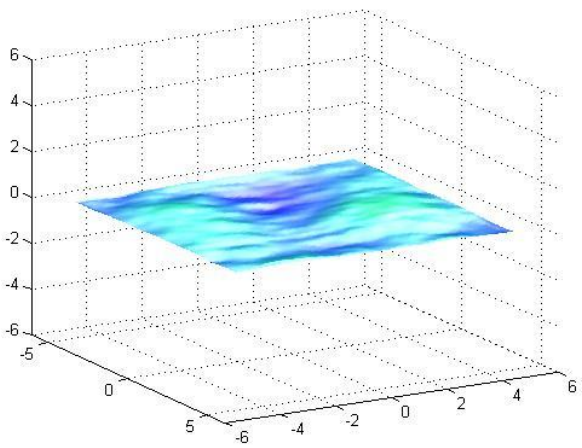


Figure 10.5 – Wave profile for multi source random wind.

Finally, a texture is introduced on to the mesh, which is similar to the technique that will be employed during the graphical rendering in the finished

simulator. This produces the result in Figure 10.6, which gives the illusion of higher frequency detail, with only the computation for low frequency detail needed.

This illustrates that a realistic wave system can be built for the simulation.

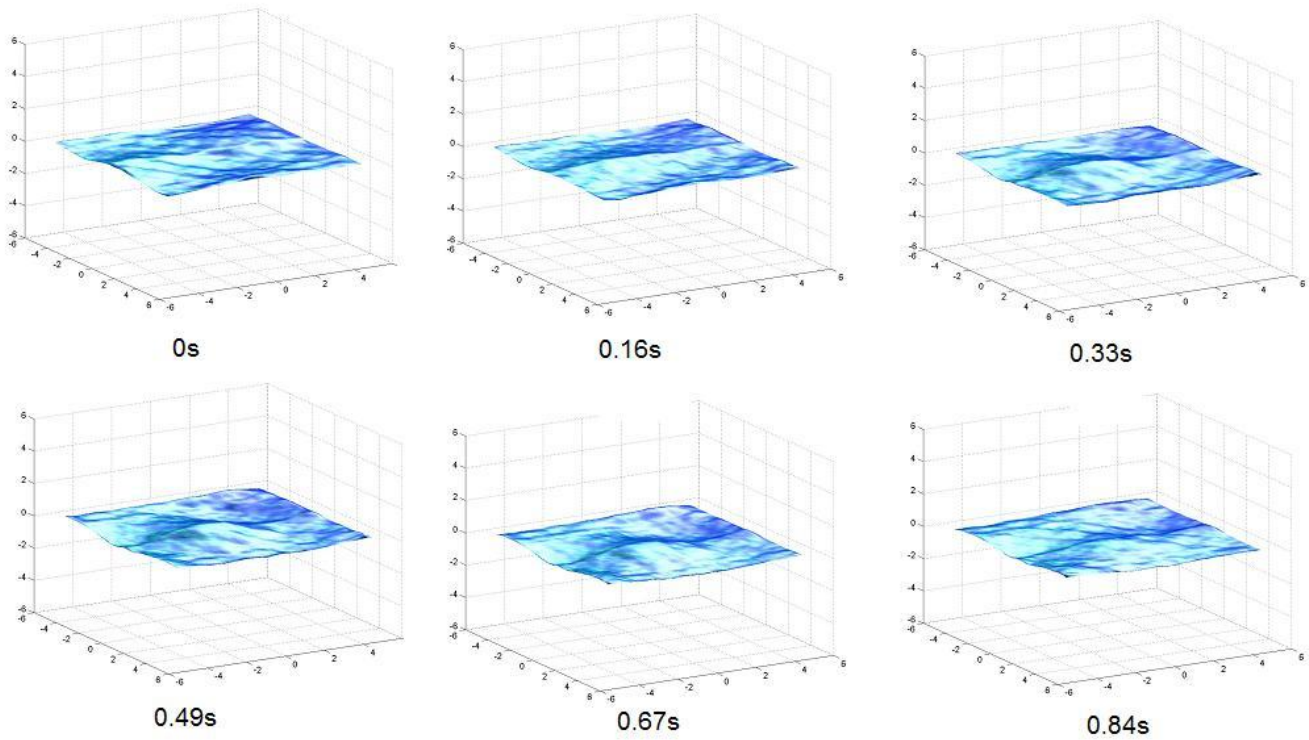


Figure 10.6 – Wave profile for multi source random wind with texture mapping. Frames taken at simulation times labelled.

10.3. Wave Forces

A system of plane progressive waves impacting on a sailing boat will create a dynamic response. The forces of the waves can be modelled by a body in regular, plane progressive, waves. Actual ocean waves are of course irregular, however this approach is still of significance, since irregular ocean waves can be thought of as a superposition of regular sine waves (for example as described by the Pierson-Moskowitz spectrum). In reality, the ocean environment is most realistically described by a random process, which in turn causes a random response of the body. Initially, randomness will be ignored and the focus will be on the response from regular waves, as this will be dominant.

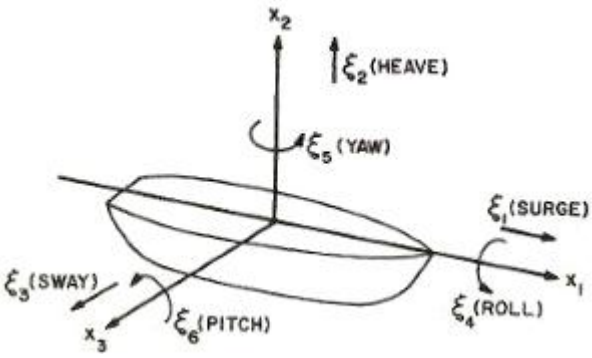


Figure 10.7 – The body motions in six degrees of freedom

An oscillatory wave will cause an oscillatory response to a body in six degrees of freedom (see Figure 10.7). The waves are assumed plane progressive with amplitude A and direction θ. The corresponding velocities $U_j(t)$ will be sinusoidal in time with the same frequency as the incident waves, therefore $U_j(t) = \mathrm{Re}\left(i\omega\xi_j e^{i\omega t}\right)$, $j = 1,2,\ldots,6$. Coupling the velocity potential from this with Bernoulli's equation yields an expression for the wave exciting sinusoidal force,

$$\boldsymbol{F} = -\rho \mathrm{Re}\left\{ i\omega A e^{i\omega t} \iint_{S_B} (\emptyset_0 + \emptyset_\tau) \begin{pmatrix} \boldsymbol{n} \\ \boldsymbol{r} \times \boldsymbol{n} \end{pmatrix} dS \right\}.$$

Here $S_{\boldsymbol{B}}$ is the submerged area of the boat, $\boldsymbol{n}$ is the unit normal to the boat surface, $\boldsymbol{r}$ is the position vector on the boat surface, $\emptyset_0$ is the incident wave velocity potential and $\emptyset_\tau$ is the scattering potential. Using boundary conditions [14], this can be simplified to

$$X_j = -\rho \iint_{S_B} (\emptyset_0 + \emptyset_\tau) \frac{\partial \emptyset_j}{\partial n} dS, \qquad j = 1,2,\ldots 6$$

where X_i is the complex amplitude of the exciting force of moment such that $F_j = \mathrm{Re}\left\{A e^{i\omega t} X_j\right\}$. The incoming wave potential $\emptyset_0$ is simply that derived from the plane progressive wave, i.e. $\emptyset_0 = \frac{gA}{\omega} e^{ky} \sin(kx - \omega t)$ in 2D, and the scattering potential (due to the reflection of waves from the body in water) can be ignored since its effect is negligible. In the x direction, the result is the following for the surge exciting force, $X_1 = 2\rho g \iint \frac{\partial \xi}{\partial x} e^{ky - ikx\cos\theta} dxdy$. Similar expressions can be derived for the other dimensions.

Computational Implementation

As before, the prototype implementation is in Matlab, while the completed simulator will be written in C++.

[14] For full analysis see Chapter 6, Marine Hydrodynamics by J N Newman.

The approach outlined in the previous section, of modelling the frequency response to a body in regular waves, was implemented computationally in two dimensions initially to reduce complexity. The code to do so is outlined below for the surge force. Heave and pitch are neglected as the effects of waves are very small compared to the buoyancy forces in these directions.

```
% outline the domain of the boat (from boat polygon)
minBoatValues = min(boat.hull.worldVertex);
maxBoatValues = max(boat.hull.worldVertex);
bottomOfBoat = minBoatValues(2);
topOfBoat = maxBoatValues(2);
leftOfBoat = minBoatValues(1);
rightOfBoat = maxBoatValues(1);
xVec = leftOfBoat:world.dx:rightOfBoat;
yVec = linspace(bottomOfBoat,topOfBoat,length(xVec));

% for all the frequencies in the spectrum, compute the wave
% exciting response
for n = 1:1:length(world.water.w)
    % exp to first order.
    % approximate thin boat shape to a rectange
    % therefore d$/dz = constant, d$/dx = constant
    ytoint = ones(size(yVec)) + world.water.k(n)*yVec;
    xtoint = ones(size(xVec)) - 1j*world.water.k(n)*xVec*cosd(world.wind.dir(1));
    % split integration up
    xInt = trapz(xVec,xtoint);
    yInt = trapz(yVec,ytoint);
    X1(n) = 2*world.rho*world.g*0.5*(topOfBoat-bottomOfBoat)*xInt*yInt;
end
% sum the frequency components
F = [sum(real(world.water.A.*X1.*exp(1j*world.water.w*world.t(frame)))) 0 0]';
```

Code 10.3 – Wave exciting force implementation, as proposed by Marine Hydrodynamics (J N Newman).

This approach works well, and produces a realistic set of results as shown in Figure 10.8. However there is a major flaw with this method – it is designed to be used to look at the steady state response of a body in steady state waves, not the real time response of a body in random waves. Therefore it would not be suited to a real time simulator, as it would not respond convincingly to the real time waves that the body would encounter in the simulation.

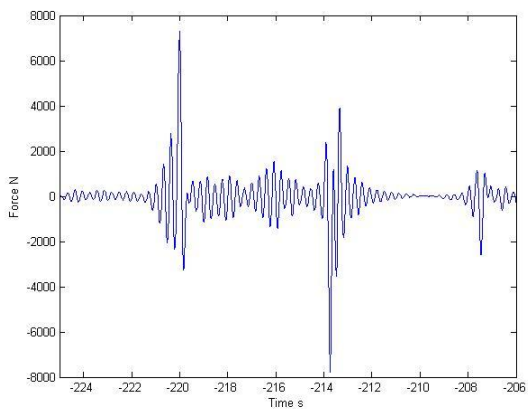


Figure 10.8 – Computational results of steady state wave exciting force analysis on model boat. Force shown is the surge force.

A solution to the problem of how to compute the wave exciting force is to go for a more observational approach. That is, the surge force is due to the impact of an incoming wave on the area of the boat as seen by the wave, the effective area. This method is employed in 2D by simply looking at the submerged area either side of the minimum vertex of the boat, see Figure 10.9. This produces very realistic results in 2D.

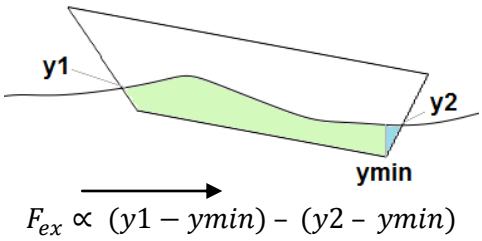


Figure 10.9 – Boat profile in wave

As the wave exciting force is of minimal significance, further approximations could be made. One effective one is to say the horizontal forces from the waves are negatively proportional to the moments, $\begin{pmatrix} F_x \\ F_y \end{pmatrix} = -\begin{pmatrix} 0 & a_1 \\ a_2 & 0 \end{pmatrix}\begin{pmatrix} M_x \\ M_y \end{pmatrix}$. This is very similar to the method in 2D proposed above, but rather than evaluating around the minimum vertex (which would be ideal), the submerged area is evaluated around the centre of gravity. The advantage is that no extra computation is required, other than the multiplication of the arbitrary constants a_1 and a_2, as the moments are previously computed in the simulation from the buoyancy calculations. This gives us a wave exciting force with almost no computational overhead. While not completely physically accurate, it gives a visually realistic 3D response which is all that is required. The constants a_1 and a_2 would need to be selected to give the most realistic response.

10.4. Conclusion

It has been shown that a realistic wave model can be computed. There is a fairly significant computational cost, however the bulk of this computation can be done before the real time simulation, making it feasible to implement in the final simulator without significant impact on frame rate. The wave exciting force is very subtle and a case could be made to neglect it from the simulation. However, a 3D wave force model has been devised which produces visually realistic results requiring negligible processing. Though physically inaccurate, the minor processing required means it can be implemented to enhance user experience.

11. Computer Simulation

11.1. Specification

The simulator must realistically simulate the sailing of a boat, react to user inputs, and output to the display adapter, sound adapter and actuator control system.

11.2. Simulator Structure

The simulator program must be designed in such a way that allows the various components of the computation to fit together, as well as being able to accept user inputs and output to a graphics engine, sound engine, and also the actuator engine, as shown in Figure 11.1.

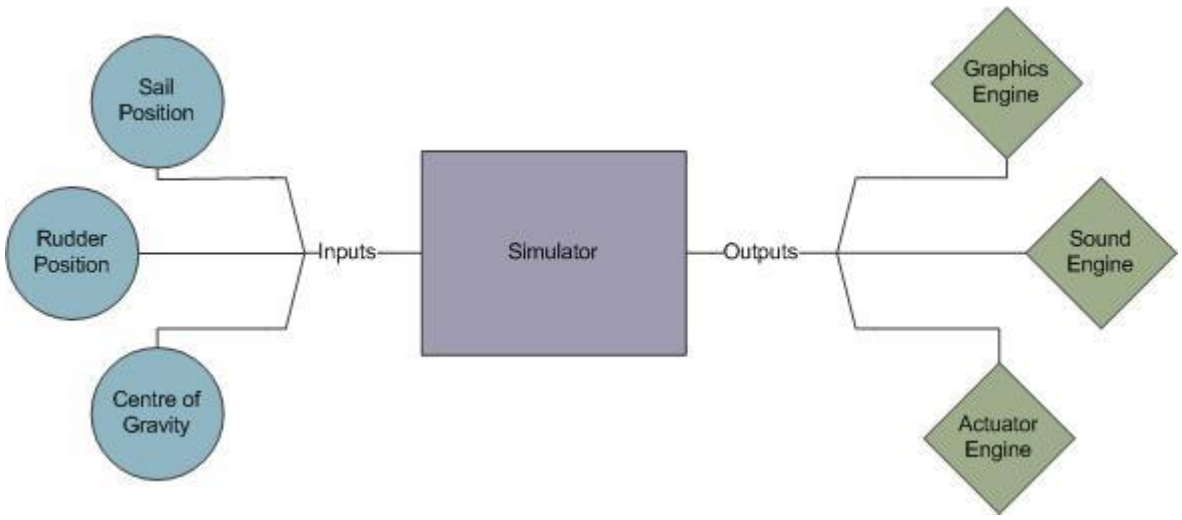


Figure 11.1– Simulator overview.

The simulator comprises of two stages – the preparation stage and the execution stage. The preparation stage occurs once the user has selected the desired configuration for the simulator. Configuration includes locale, wind speed, weather, and any other parameters which define the game world. Once these have been selected, the pre-processing can begin. This involves setting up the environmental variables and computing wind and water characteristics. Also the textures, display data and sounds must be loaded from disk and be ready to be used, the actuators must be reset into their start positions, and the user input interface must be initialized. Once this is complete, the real time simulation is ready to run.

The execution phase now follows. This takes the form of a continuous loop which occurs until the simulation is finished. Contained within the loop are the processes required for the real time simulation. They are:

- accepting user inputs and process them,
- calculating the forces on the boat, sail and rudder,
- updating the boat state,
- updating the world state,
- outputting boat and world state to user.

Time can then be advanced and the process repeated until the user exits, or an end condition has been met (e.g. capsize, exit). The simulation must then shutdown – hold actuators in an appropriate position, and reset the display to prompt the user for a new simulation. The flow diagram of the process is shown in Figure 11.2.

User inputs

The three user inputs are the force due to the mass of the boat and user (i.e. the changing centre of gravity of the system), the sail velocity and the rudder velocity.

The forces due to the changing centre of gravity of the boat are calculated from the forces exerted on each actuator. Each linear actuator supporting the cradle has a load sensor. The load measured is the combination of the static mass and the dynamic force due the acceleration, $F = m(\ddot{x} + g)$. As the acceleration is known from the velocity sensors on the linear actuators, the static mass can be calculated at each corner of the cradle. The four static masses can then be resolved around the boat's centre of mass to find the forces and moments, F_z, M_x and M_y, exerted on the boat hull by the position of the user.

The sail and rudder actuator velocities will be outputted and fed into the simulator program. The angular position of the rudder and sail in relation to the boat hull will be tracked by the velocity and used to determine the positions in the simulation.

Boat Forces

All the individual forces acting on the boat must be added together. The computation for each type of force has been dealt with in proceeding chapters. The simulator will make a call to each function and add the results in the form shown in Code 11.1. Each force will have its own function which is a wrapper to the computation of that force, taking in the current `boat` and `world` objects which represent the current states of the boat and the world, and returning the 6 element force vector.

```
%% Sum all the forces on the boat
    boat.F = [0;0;0;0;0;0];

    % Gravity
    boat.F = boat.F + gravityForce(boat, world);
    % Buoyancy
    boat.F = boat.F + buoyancyForce(boat, world);
    % Drag
    boat.F = boat.F + dragForce(boat, world);
    % Wave exciting
    boat.F = boat.F + waveExcitingForce(boat, world);
    % From sail and rudder
    boat.F = boat.F + forceFromSailRudder(boat, world);

    %% Sail force
    boat.sail.F = forceOnSail(boat, world);

    %% Rudder force
    boat.rudder.F = forceOnRudder(boat, world);
```

Code 11.1 – Summing of forces in the simulation.

The computation of each force is done using the methods outlined in the previous sections of this chapter.

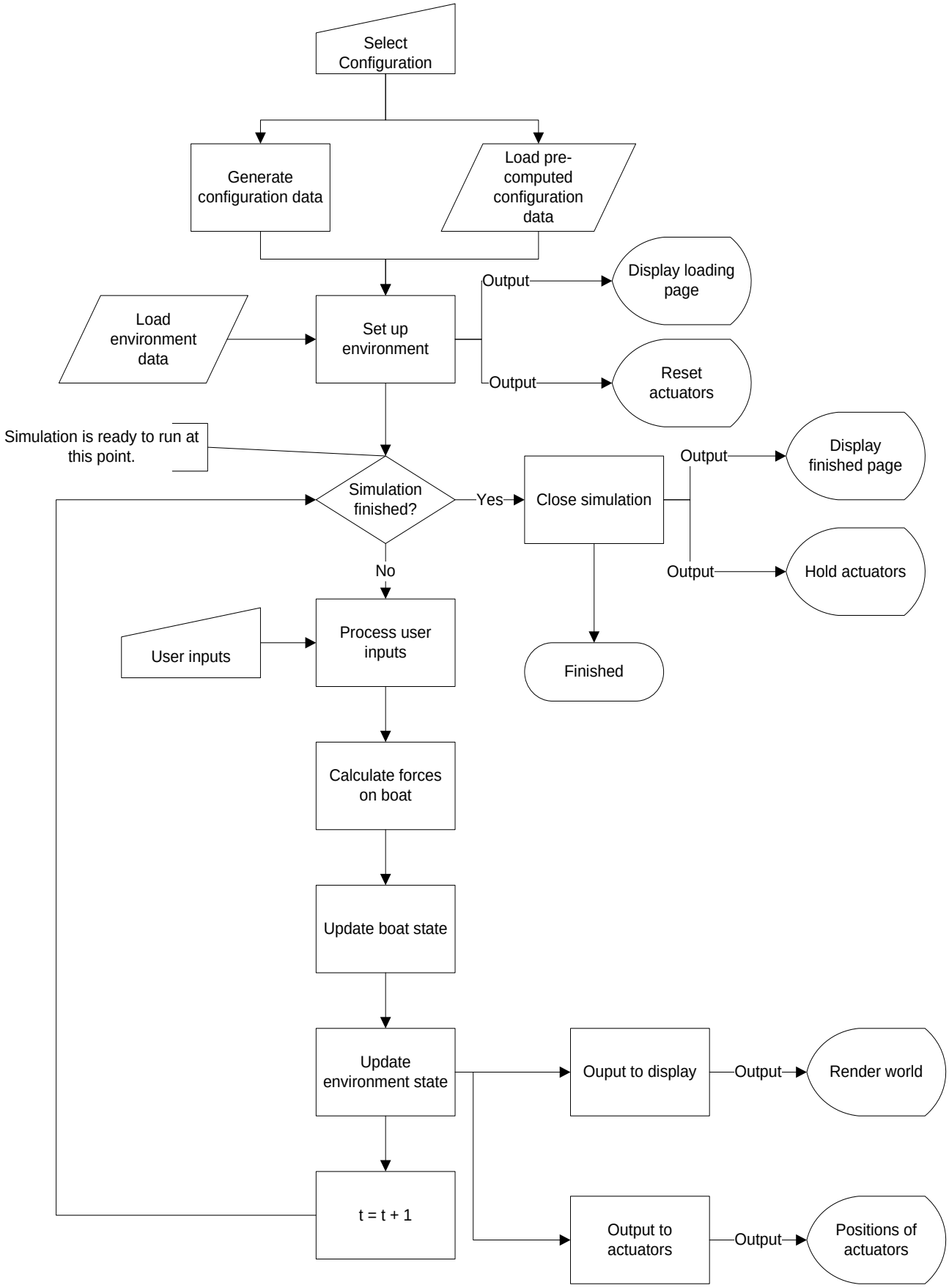


Figure 11.2 – Flow diagram for simulator program.

Boat State

The state of the boat must be updated for each time step. This is done simply using the Euler method[15]. For each dynamic component of the boat – the boom, rudder and hull – the current acceleration, velocity and position vectors are computed. Basic dynamics gives that $\boldsymbol{a}_k = \boldsymbol{M}_k{}^{-1}\boldsymbol{F}_k$ where $\boldsymbol{M}_k$ is the mass matrix at time step k and $\boldsymbol{F}_k$ is the force vector at time step k. It then follows that $\boldsymbol{v}_k = \boldsymbol{v}_{k-1} + \delta t \boldsymbol{a}_k$ and $\boldsymbol{x}_k = \boldsymbol{x}_{k-1} + \delta t \boldsymbol{v}_k$, where δt is the time elapsed since the last time step, i.e. $\delta t = t_k - t_{k-1}$. This now gives the current state of the boat.

World State

Subsequently, the world state must be updated. This involves updating the wind vectors and water state. The wind for the simulation is represented as an array of wind vectors at each time step. As the simulation is of unknown length and the array must be of finite length, the wind array will represent a repeating pattern of wind behaviour. An example implementation is given in Code 11.2.

```
% Called from the simulation loop to update the world state
function world = updateWorldState(world)
    % update the wind state at the current time step, k
    world.wind = updateWindState(world.wind, world.k);
    % update the water state at the current time step
    world.water = updateWaterState(world, world.k);
end

    % Updates the wind state
    function wind = updateWindState(wind, k)
        % array of wind pattern
        windPattern = wind.windPattern;
        % set current wind to the current point in the windPattern array
        wind.currentWind = windPattern(mod(k, length(windPattern)));
    end

    % Updates the water state
    function water = updateWaterState(world, k)
        water = world.water;
        Z = zeros(size(water.Z));   % initialize new water state
        % generate current wave profile
        Z = generateWaves(water.w, Z, world.X, world.Y, water.A,...
                          water.k, world.t(k));
        water.Z = Z;
    end
```

Code 11.2 – Example implementation of updating the world state

The wind patterns are computed and loaded in the preparation stage of the simulation and

[15] See http://en.wikipedia.org/wiki/Euler_method for more information.

correspond to the wind configuration which was set for the simulation, e.g. “constant”, “gusty”, “swirling”, etc.

Output

Once the current states of the simulation have been computed, they must be outputted to the display and sound adapter for presentation to the user, and to the actuator engine to update the actuator positions. The functions which handle these actions will be called asynchronously, in different threads or as different processes. This avoids the bottlenecking of performance by these functions which do not require synchronicity with the computer simulation, as calling a function asynchronously starts the function as a new process, in a separate thread. The simulator can immediately begin the computation for the next time step while the display rendering and actuator control is processed in parallel.

11.3. Prototype Implementation

A prototype implementation was developed in Matlab to outline the structure of the simulator and test the feasibility of the computation.

The boat is represented as a collection of coloured polygon objects built up using a rendering package[16]. Each frame requires the components of the boat to be rotated and translated to the position of the boat as defined by the current state. This process is shown in Code 11.3. The function `transformObject` translates the object back to the origin, applies the new rotation, then applies the new translation, which transforms the object to the desired state.

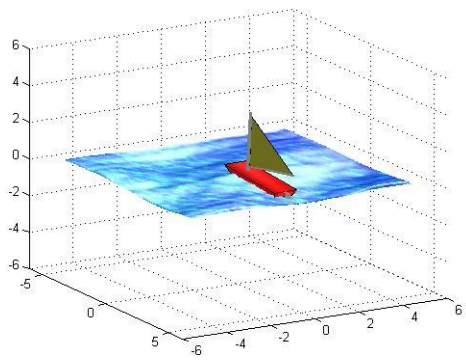


Figure 11.3 – Screenshot of the prototype implementation.

[16] See http://www.nbb.cornell.edu/neurobio/land/projects/hierarchy/ for an outline of the graphics modeller used.

```
%% draw frame
    function drawFrame(boat, world, frame)
        figure(1); clf; % clear last frame from figure

        % render water
        b = double(world.texture);
        b = b(1:size(world.water.Z,1),1:size(world.water.Z,2));
        m                                   =                                   ...
mesh(world.X,world.Y,world.water.Z,b,'FaceColor','texturemap','EdgeColor','none');
        set(m,'FaceLighting','phong','FaceColor','interp','AmbientStrength',0.5)
        colormap winter
        shading interp

        % transform the rudder and sail objects
        rudder = transformObject(boat.model.rudder, boat.rudder.x);
        sail = transformObject(boat.model.sail,boat.sail.x);
        % combine the rudder and sail with the hull
        fullboat.model = combine(boat.model.hull.model,rudder.model,sail.model);
        fullboat.transVector = boat.model.hull.transVector;
        % transform the complete boat
        fullboat = transformObject(fullboat,boat.x);
        % render the boat
        renderpatch(fullboat.model);

        % set lighting and camera position
        light('position',[-10,10,10])
        axis([-6 6 -6 6 -6 6]);
        view(60,20);
    end
```

Code 11.3 – The prototype function to draw a frame of the simulation.

The computation produces realistic results, and runs at an average of 5 frames per second. This is encouraging, as Matlab is not efficiently compiled, so a 6x increase in performance to 30 frames per second is believed possible when implementing in C++.

11.4. Production Implementation

The production version of the simulator will be implemented in C++ and run on Windows 7 operating system. Windows 7 is chosen for its stability, robust hardware interface, and its support for Microsoft DirectX graphics renderer. Some reasons for choosing C++ for implementation are:

- Rich object oriented language allowing for complex objects utilizing the Component Object Model (COM).
- Hardware interfacing and memory management possible with low level coding.
- Multi-threading and parallel computing supported.
- Large number of standard libraries for maximally efficient code.

- Does not rely on Microsoft .NET which increases performance.

The simulator will be developed using well structured classes and objects, allowing encapsulation and modularity of code. The COM interface will allow it to be easily plugged into the user input module and output to the user, for example, by using Microsoft DirectX 9 for video and sound rendering (covered in a later chapter). Object models will be created for logical and intuitive programming, for example the `Boat` object, which consists of `Vector` objects `x, v, a, F` as well as a `Matrix` object, `M`, and three objects of type `Component`, representing the sail, rudder and hull. Where possible the C++ standard library templates will be used, such as the `std::vector<double>` type for the vectors.

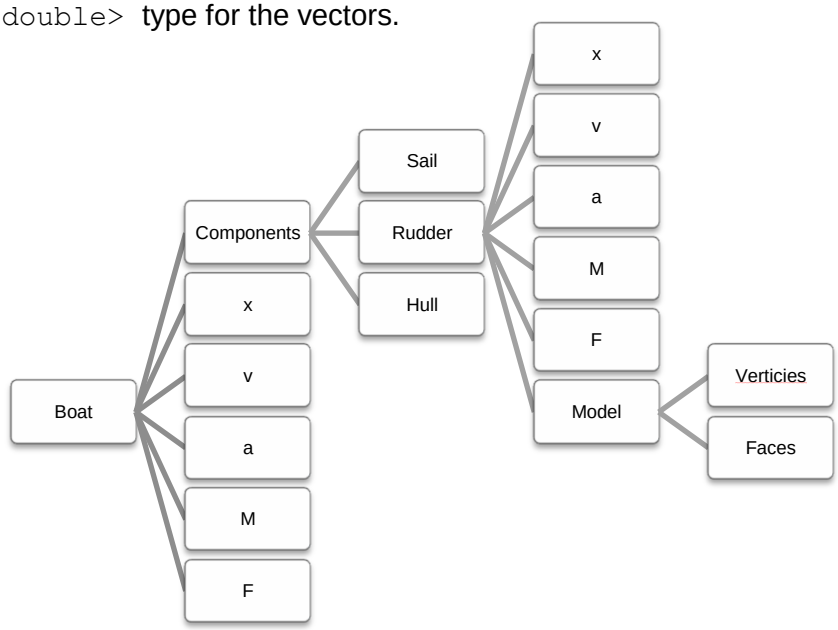


Figure 11.4 – Example structure of a boat object, consisting of its current state vectors, and the individual components it is made from.

```
class Rudder : public Component {
public:
      Rudder();
      Rudder(vec6 x, vec6 V, vec6 a, Matrix M);
      vec6 X();
      vec6 V();
      vec6 A();
      Matrix M();
      vec6 Force();
      Model Get3DModel();
private:
      vec6 x, v, a;
      Matrix m;
      Model *model;
};
```

Code 11.4 – Example Rudder class implementation in C++.

12. Boom and rudder actuators

To begin to specify actuators to give the correct torques around the sail and rudder, a maximum apparent wind and water operating speed had to be chosen. Values of 40 knot (74kph) wind and 10 knot (18.5kph) water speeds were selected to give the optimal balance between a wide range of accurately simulatable conditions whilst keeping the overall cost and weight of the actuators and hence total simulator to a minimum. This resulted in maximum torques of 1015Nm for the sail and 78.4 Nm for the rudder.

The major challenge in designing the simulation system was to create a mechanism which would provide an accurate torque at a certain position, whilst still being able to be pulled against and return to a natural position (in line with the apparent wind and water angles) when the controls were released. A standard electric motor would not be suitable as when “locked” (the user holding the motor static by holding the sail in a certain position) the coils would very quickly overheat and burn out due to the high current.

A clutch system was initially thought of as a solution however this was quickly ruled out due to it being heavy, expensive and inherently very wasteful.

A second idea was to use a linear actuator or shock with a linkage system similar to that found on high end full suspension mountain bikes. The linkage system would then manipulate the actuator rate to the desired curve. However this would require a large area for the links alongside time consuming and lengthy in -depth analysis into the linkage placements and size. On top of this the driving actuator would have to be extremely powerful whilst maintaining a very high degree of accuracy which in turn would make the overall package very costly and large.

A third idea which was explored in more detail was to use a system of worm and wheels connected to a spring. The base of the mast would be driven by a first worm and wheel system, which would quickly move the mast into its natural position and then switch off; due to the nature of the worm and wheel this would effectively lock the base in place. A second worm and wheel system would be mounted on the rotating plate and connected to a spring which would then connect to the sail

boom (see Figure 12.1). This would allow force to be wound on/off the spring hence varying the torque on the mast as required. It would also give a quick return to the natural position when released; slightly overshooting first and giving a more natural feel to the response. This would be combined with a kill switch to return the spring to its natural position once the natural position was slightly passed; stopping overshoot or the sail spinning off the required no wind angle.

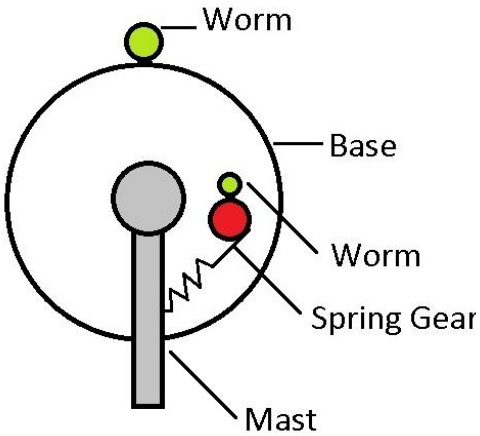


Figure 12.1: Alternative actuator design

Although this is a solution, a more elegant one was found in brushless servo motors that can generate high continuous stall torque. These types of motors are designed with the capability to run stalled for prolonged periods of time whilst not damaging the coils and circuitry. The servo motors also have in build position and velocity detection which eliminates the need for any other form of feedback loop.

12.1. Actuator Motors

Continuous stall torque motors can come in packages developing up to 150Nm + of continuous stall torque, however these can be in excess of 100kg. The best power to weight and size distribution for the sail actuator was found to be generated by a Parker Hannifin Maxplus 8” Servo Motor 1902BSG. These come in both high and low current variants, however as the required rpm is low even after gearing, the low current version was selected. The rudder actuator requires a much smaller torque, so a Parker Hannifin Maxplus 4” Servo Motor 1141BSG was selected, one again the low current variant. A further advantage of the motors is that they both require a standard line voltage of 220V with a continuous line current maximum of 12.8A so can be driven off a mains supply with no transformation required and allowing for slight voltage drop between components. The specifications for each are shown overleaf (see Figure 12.2).

To give a life like and quick response a minimum rotational velocity of the sail and rudder was chosen to be 60rpm, allowing the sail in particular to “snap” back to its natural position when released in high wind simulation. It should be noted that the boom is covered in 30mm thick foam padding to protect the users from damage induced by contact with the swinging boom.

Specification	Rudder (1141 BSG)	Sail (1902 BSG)
Continuous Stall Torque	6.1 Nm	67.5 Nm
Max Acceleration	54000 rad/s	20000 rad/s
Peak Torque	21.3 Nm	176.3 Nm
Continuous Line Current	2.7 A	12.8 A
Line Voltage	230 V	230 V
D.C. Resistance	3.3 Ω	4.4 Ω
Max Angular Velocity	1500 RPM	1250 RPM
Weight	8.8 kg	40.5 kg
Approx Size		
Required Ratio	1 : 14	1 : 20
Geared Continuous Stall T	107 RPM	62.5 RPM
Geared Max Angular Velocity	114 x 218 x 114	190x 397 x 190

Figure 12.2: Motor specifications

12.2. Actuator Gearing

Gearing was required to step the motor torques up to the required torques. For the rudder a 1:14 ratio was selected, giving an overall maximum stall torque of 85.4Nm. This in turn took the angular velocity for maximum torque down to 107rpm, well above the desired 60rpm. To keep the overall actuator dimensions to a minimum, the gearing ratio was obtained in two steps; using a 1:2 ratio with a further 1:7 ratio driven by the second primary gear. The 1:7 ratio had to connect to the rudder, as using the 1:2 ratio gears as the secondary caused mechanical interference between the rudder and the motor housing. The gear module was set at 2 for the rudder gearing system. This was the minimum module and hence gear size, which allowed a 10t gear to be mounted to the motor shaft. The module was kept constant throughout the gearing to maintain the same torque ratio as gear ratio and to give an optimum mesh and hence power transfer.

For the sail a 1:20 ratio was selected which in turn resulted in a stall torque of 1356Nm and an angular velocity of 62.5rpm. This was still above the originally desired speed, strengthened by the fact that the continuous stall torque is well below the maximum torque capacity of the motor so on release the motor will accelerate at close to its maximum potential. Once again the gearing ratio was achieved in two stages to keep the actuator compact; a 1:4 gear ratio drove a 1:5 gear ratio connected to the mast. The gear module for the sail gearing was set at 4 due to the same reasons as for the rudder; the minimum diameter that allowed insertion of a 10t gear onto the motor axle. (Figure 12.3) shows the gear layout for both drive systems, with green representing the driving gear and red representing the driven gear connecting to the mast and rudder respectively.

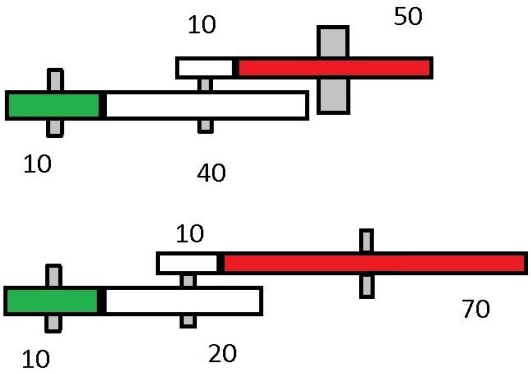


Figure 12.3: Gearing layout

12.3. Actuator Mounting

The next stage was to design mounting kits so that the motors could be securely mounted on the hull, alongside holding the gearing in position to give an optimum mesh and hence the best power transfer whilst reducing long term wear. For both mounting plates it was decided to use instrument ball bearings press fitted into the top and bottom of the plate to securely hold the link axle in place. The link axle diameter was chosen as 10mm for the rudder, and a thicker 25mm for the sail to deal with the higher torques generated by the sail and gearing ratio. It can be seen from the calculations (see Equation 12.1) that mild steel would be suitable for both axles providing they are solid and this contributes towards the aim of keeping material cost to a minimum.

$T = \frac{Driven\ torque}{link\ axle\ ratio}$ $\sigma = \frac{Tr}{I}$

For a solid cylindrical bar: $T_{max} = \frac{\pi}{16}\sigma_{max}d^3$

Von Misses for pure torsion: $\sigma_{max} = \frac{\sigma_y}{\sqrt{3}}$

For rudder: $T = \frac{78.4}{7} = 11.2Nm$ For mast: $T = \frac{1015}{5} = 203Nm$

∴ Rudder: $\sigma_y = \frac{22.4 \times 16}{\pi \times (10 \times 10^{-3})^3} \times \sqrt{3} = 197.6MPa$ (applying x2 factor of safety to torque)

∴ Mast: $\sigma_y = \frac{406 \times 16}{\pi \times (25 \times 10^{-3})^3} \times \sqrt{3} = 229MPa$

σ_y for mild steel = $240MPa$

Equation 12.1: Link axle material calculations

Two small grooves were placed 1mm from either end of the axle, allowing circlips to snap on and prevent the spacers and gears from moving in the axial direction. A groove ran the length of the axle to mate with the gear lugs and enable the torque transmission from gear to axle and visa versa. With the inner diameter for the ball bearings set, the bearing size could be determined; a 100-15 was used for the rudder and a 250-32 was used for the sail.

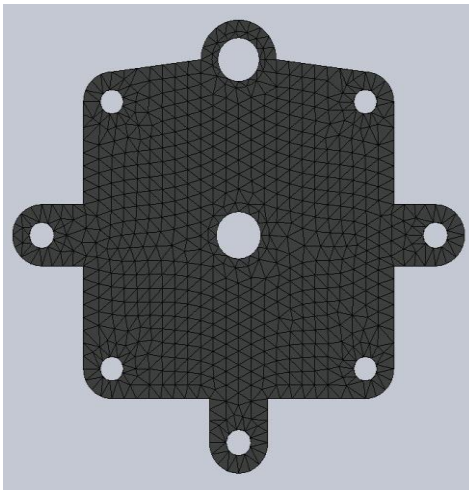


Figure 12.4: FEA mesh

With the bearing sizes set, the overall thickness of the mounting plate could be determined. As the rudder bearings were 4.5mm thick, a mounting plate thickness of 19mm was chosen to give a 10mm thick spacing between the bearing sets. Similarly, for the sail a thickness of 20mm was chosen as the bearings each had a depth of 5mm. Drawings for both mounts can be found in the appendix. Finite element analysis was then run in

Solidworks to determine the strength of the mounts under loading from the motor and to establish if any reinforcement was required. Both the rudder and sail mountings are well within the yield strength of mild carbon steel with the major stress concentrations located around the bolt holes as expected. A finite element analysis simulation for bearing lock in the link axle combined with the motor load was also run (see Figure 12.4 and Figure 12.5), once again showing that the maximum Von Mises stress was well below the mild steel yield stress of 240MPa. (Figure 12.6) shows the

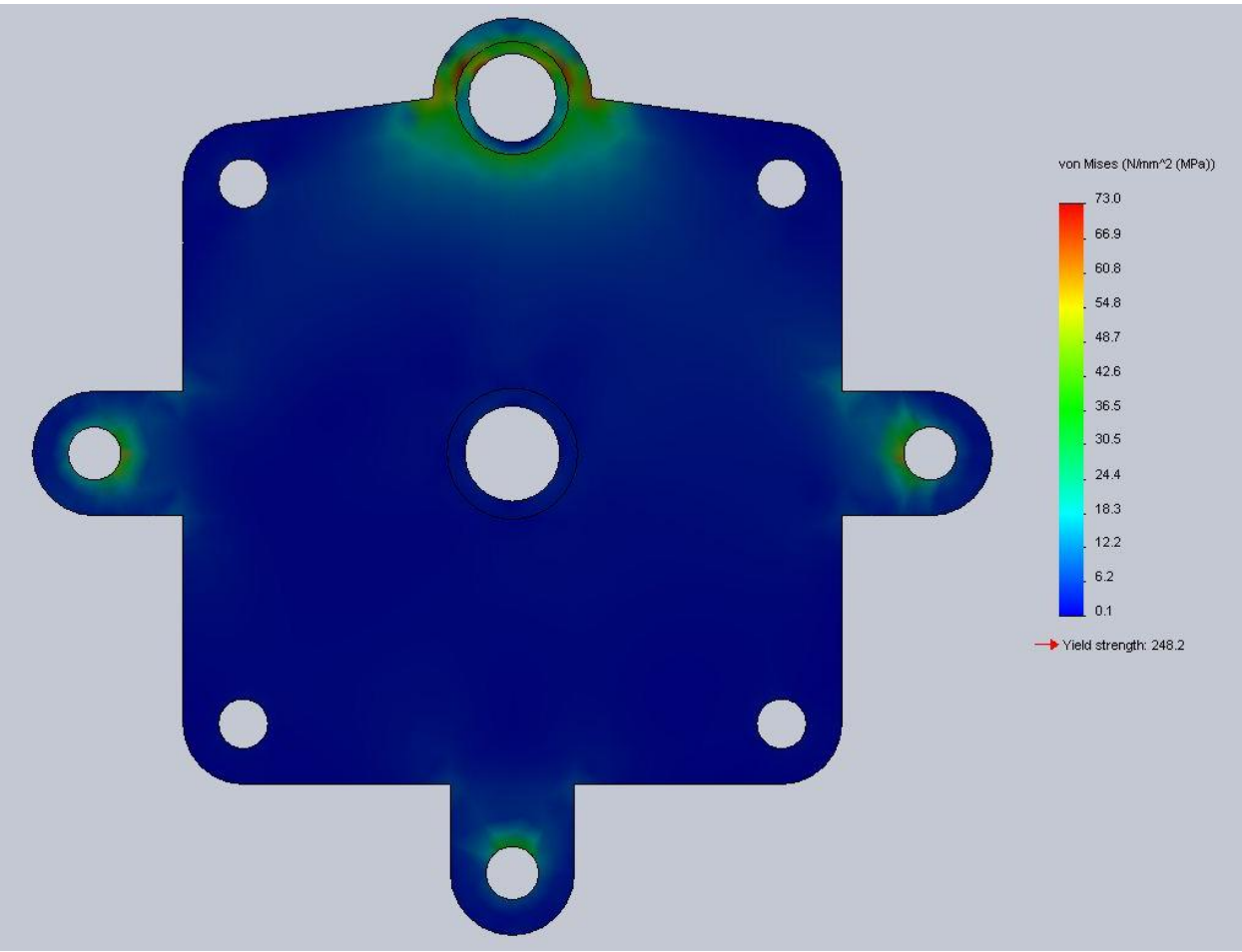


Figure 12.5: FEA with combined torque and motor load for sail motor mounting

results for all the tested scenarios. As the width of the mounts are fixed due to the requirement of at least 5mm thickness between bearings to stop them puncturing through, weight could be saved by drilling holes at certain locations throughout the plate. However this would increase machining costs dramatically with relatively minimal weight reductions so it was decided to remain with the basic original plate design.

	Max stress motor load	Max stress combined load
Rudder Mounting Plate	6.9 MPa	12 MPa
Sail Mounting Plate	14.1 MPa	73 MPa

Figure 12.6: FEA results

A series of aluminium spacers and circlips were then used to mount the gears and axle onto the plate in the correct position (see Figure 12.7). Finally a second mounting plate attached to the end of the axle with the mounting bolts connecting both plates together, encasing the entire gearing and giving better tensional rigidity.

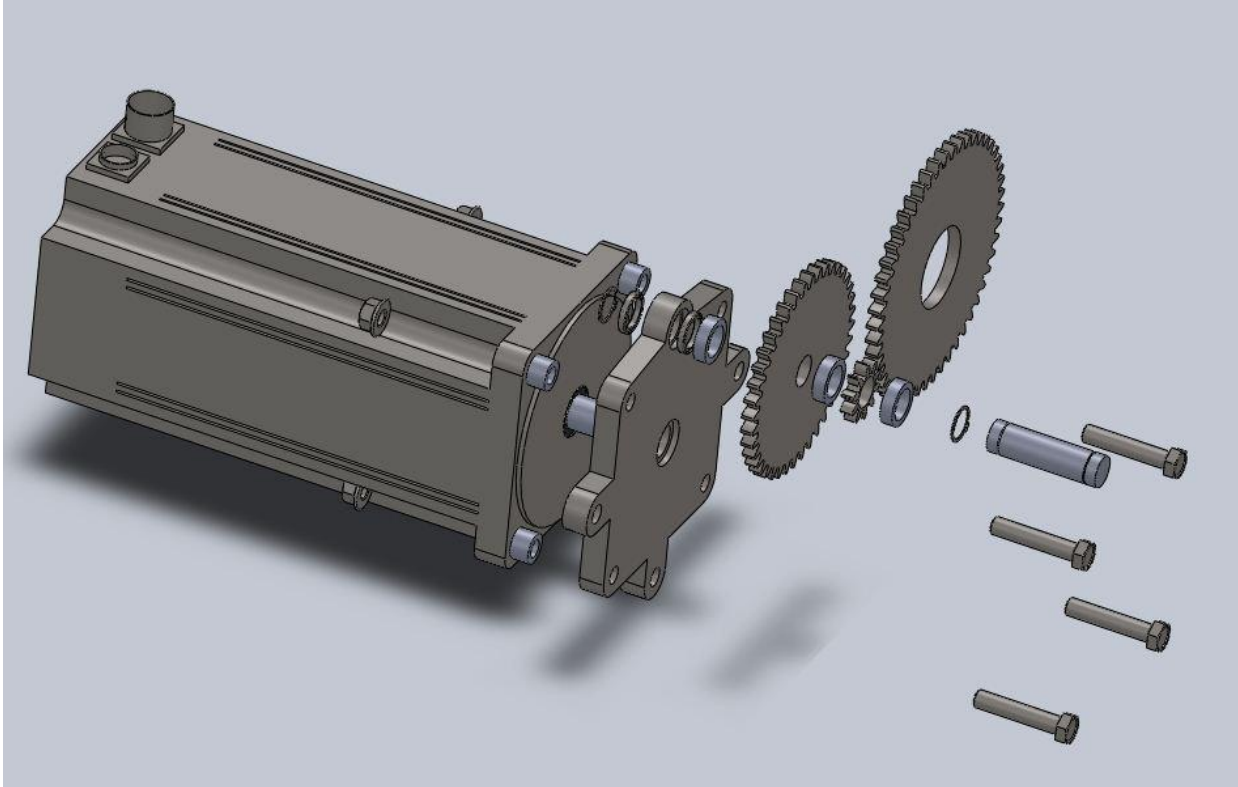


Figure 12.7: Exploded diagram of sail motor minus bottom mounting plate

The rudder mounting kit was attached to the motor using four 3/8” x 2” hex head bolts (sizing determined by motor) with M10 bolts used to mount the whole package to the hull. The sail mounting kit attached to the motor using four M14 x 40 hex head bolts sizing determined by motor) with M15 bolts securely mounting the package to the hull.

Theoretically the bolt is taking no shear load; the axles spin freely in their bearings and when mounted vertically the mounting plate takes the mass of the motor. However, in the rare occasion

that the bearings fully seize and transfer all torque to the plate, the bolts should be able to cope with the shear stress.

Due to the gearing ratio bearing seize in the link axle would generate the most torque and hence the highest shear on the bolts. Using the calculations shown (see Equation 12.2) it can be seen that M10 and M15 bolts of high yield steel exceed the respective design strengths by a factor of over two, which means the system is still safe even if poorer quality bolts with a lower ultimate yield strength were used in error, e.g. mild steel. Alongside this it also gives a good margin to take into account the assumption that the shear force is split evenly between the bolts. (Figure 12.8 and Figure 12.9) shows the fully assembled motors complete with both housings.

$F = \frac{T}{3r}$ for 3 bolts holding housing

Rudder = $\frac{11.2}{2\times(78.49\times10^{-3})+(197.53\times10^{-3})} = 31.42N$ each bolt *(assuming force split evenly between bolts)*

Sail = $\frac{203}{3\times(154.69\times10^{-3})} = 437.43N$ each bolt

$M_{max} = \frac{F}{2} d_{mount}$ $M_{max} = \frac{My}{I} < \sigma_y$ for high yield steel $\sigma_y = 400MPa$

$400 > \frac{4\times M_{max}}{\pi\times r^3}$

Sail: $r > 0.96mm$ M2 bolts minimum

Rudder: $r > 3.03mm$ M7 bolts minimum

Equation 12.2: Motor housing bolt calculations

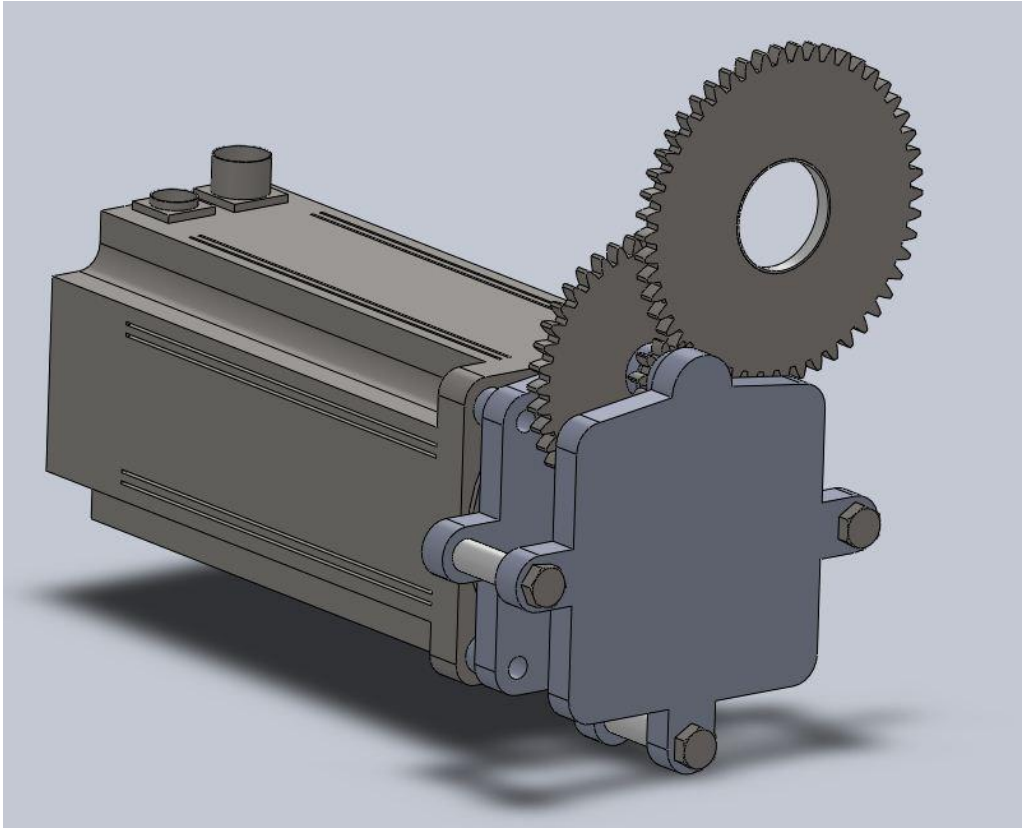


Figure 12.8: Complete sail actuator assembly

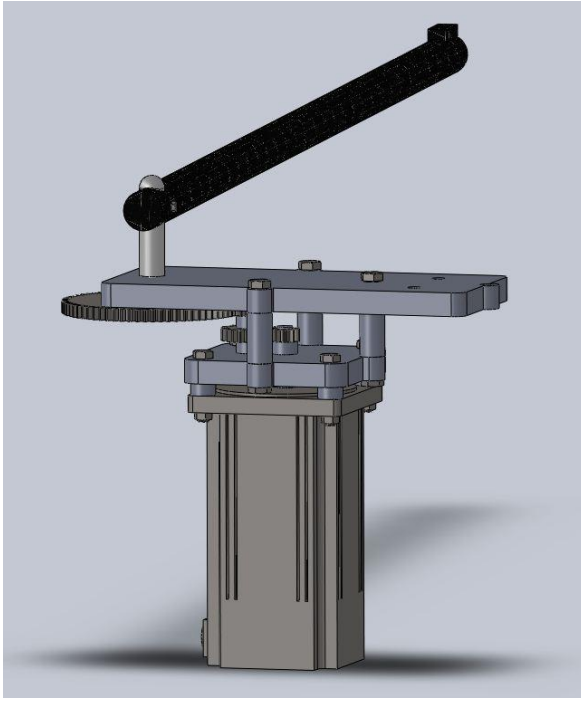


Figure 12.9: Complete rudder actuator assembly with tiller handle

13. Cradle and Hull

13.1. Hull

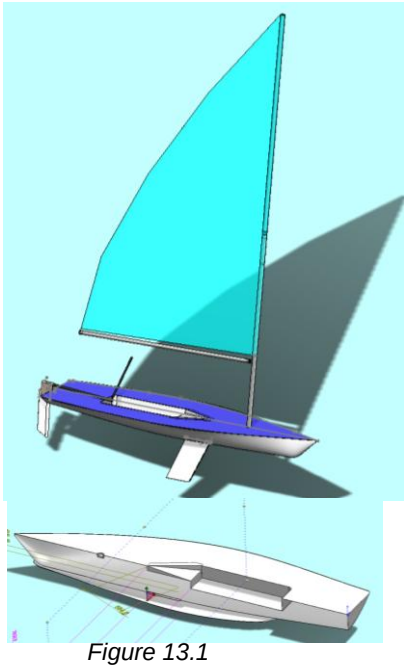


Figure 13.1

The first task to enable the equations of motion to be completed, and the cradle for the simulator to be designed, was to draw a realistic hull of the laser sail boat. This was then able to be used to find the centre of gravity for equations of motion.

This proved to be quite difficult due to minimal information about laser hull design being available. The hull was created from sight and from the dimensions given in the Laser Class Rule Interpretations[17]. The materials were taken from 'The Complete Book of Laser Sailing'[18]. The final mass was compared to the actual mass of a laser hull (59kg), and shell thickness was slightly altered until these were the same (see logbook pg24).

Cradle – choosing actuator's/pneumatics

The specification for the cradle is to be able to produce a three degree of freedom simulation that can carry the weight of one or two people in the hull. The difficulty in producing a simulator with three degrees of freedom is that there will not be a pivotal point in the cradle, the actuators need to be able to take the weight of the hull and move this against gravity. All of the calculations for cradle creation will be made for the weight of two people for worst case scenario.

To reduce the weight, and so force needed to be provided by the actuators, we will cut a section away from the hull from after the mast point to the tip of the front of the hull. This reduces the weight by 20kg from 59kg to 39kg.

The design of the cradle and resulting actuators is taken from five factors: overall force that needs to be provided, max angle of tilt before capsize, acceleration due to gravity, maximum speed, and distance the hull will move vertically.

[17] http://www.laserinternational.org/sites/default/files/Rules_2012.pdf
[18] Richard Tillman (2005)

Angle of Tilt - it was decided that the hull should have a maximum angle of tilt of approximately 30° before capsizing. This angle was calculated by the buoyancy forces on the boat showing where water overflow and capsizing might occur.

Vertical movement of hull – Maximum width of the hull is 1.44m. Therefore at an angle of tilt of 30°, vertical distance = $1.446 \times tan30$ = 0.84m. This will be the minimum stroke length of each actuator.

Acceleration – The acceleration desired is $3ms^{-2}$ against gravity, which will provide 1.327G.

Max speed – Maximum speed calculated by constant acceleration equation, such that $v^2 = 2as$, where v is velocity, a is acceleration, and s is distance travelled. Therefore with an acceleration of $3ms^{-2}$, and s = 0.8384m, maximum velocity is $2.23ms^{-1}$.

Force – by assuming that each person weigh's a maximum of 95kg, and using the hull weight of 39kg, we can calculate the force that the actuators need to provide. Against gravity, an upwards force of (39 + (2 x 95))x9.81 = 2246.49N is needed just to ensure the actuators can take the weight. This is added to the force needed for acceleration (39 + (2 x 95)) x 3 = 697N, resulting in a total of 2933.49N needed to be provided by the four actuators.

Cradle Design

To enable three degrees of freedom, we would need to use three or more actuators.

Figure 13.3

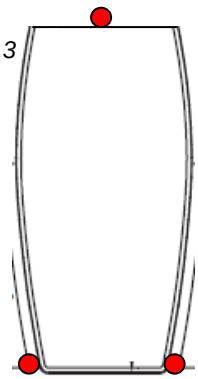
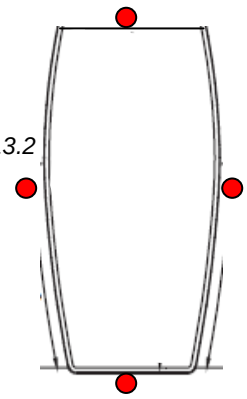


Figure 13.2



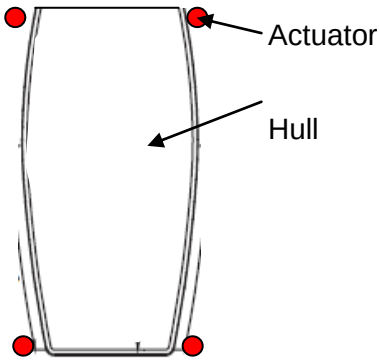


Figure 13.4

This resulted in choosing one of three configurations above. The decision on the number of actuators needed and their positioning is due to the force that needs to be provided. The maximum

force the actuators need to provide against gravity will not be distributed evenly. This is because the gravitational centre is not at the centre of the boat, and due the occupants being able to move around. However the off centre of boat gravity will be small compared to the positioning of the people inside the boat, and so will be rendered negligible.

With actuator system in Figure 13.3, it is necessary to have one actuator that can withstand at least half of the total force. This actuator needs to provide $\frac{2933.5\text{N}}{2} = 1466.75$. To ensure this does not fail, multiply by 1.4 as may have up to 40% more weight on this actuator (if the occupants are situated near to the top of the boat). Total force able to supply: $1466.75 \times 1.4 = 2053N$.

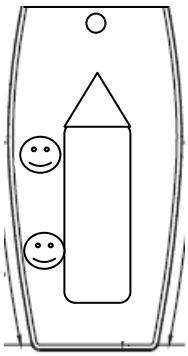


Figure 13.5

Figure 13.2 provides a pivotal system, in which one of the actuators needs to provide force for half of the weight carried, particularly if two people are sitting on the same side of the boat (Figure 13.5).

Therefore each actuator will need to provide force of $2053N$.

For the actuators in system Figure 13.4 the actuators will always be taking the weight in pairs in worst case scenarios (Figure 13.6). Therefore the maximum force needed to be provided by these actuators is $\frac{2933.49\text{N}}{4} = 733.37N$.

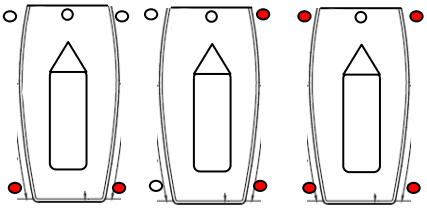


Figure 13.6: Pitch, roll and heave

Multiply this by the previous factor of 1.4, results in a maximum force required to be 1026N. We made the decision to use linear actuators rather than pneumatics. This is because pneumatics are more expensive, noisy and are hard to control the motion precisely due to the fact that the gas used is a compressible substance. Then found the strongest linear actuators available that are able to provide the maximum speed required of 2.4ms^{-1}.

13.2. Actuator choice

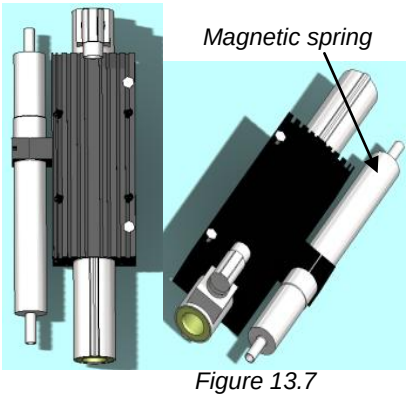


Figure 13.7

This resulted in finding the 1024N LinMot[19] linear motor, which can provide maximum speeds up to $2.4ms^{-1}$, with a stroke length of up to 1.5m. Unfortunately, there are new linear motors that will be brought out at the end of this year, capable of providing up to 2500N of force at the same speeds. However they have not been properly tested, and there are no confirmed prices. Therefore we are unable to produce configurations show in Figure 13.2 and Figure 13.3, and so will produce a cradle with actuators shown in Figure 13.4.

The actuator stators are encased in an aluminium flange, so that they can be mounted using four M10 bolts.

13.3. Magnetic Spring

In order for the actuators to provide equal forces on the way up and down, as they are mounted vertically, magnetic springs can be used as counterbalances.

The magnetic spring also prevents the linear motor from falling to the lower end stop if the linear motor is turned off or the power is lost. LinMot produce magnetic springs that correspond to the actuators used (see Figure 13.7).

13.4. Creating the cradle

The cradle needs to be able withstand the forces provided by the actuators, and also able to rotate. This rotation can be achieved by using high misalignment spherical rod ends, which have capabilities of providing rotation in both axes. These can be then simply attached to the cradle by using a cylindrical rod, such that the rod ends can rotate about them.

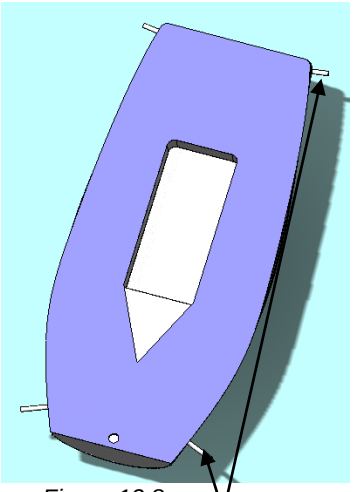


Figure 13.8

[19] http://www.linmot.com/index.php?id=15

When the simulator moves up and down, there needs to be an extension in the direction of travel.

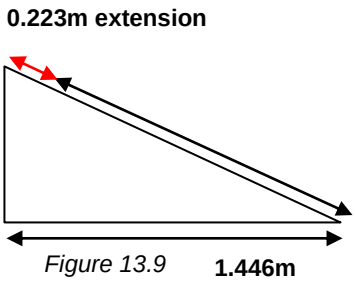


Figure 13.9

This can simply be provided in the port-starboard direction by extending the length of the cylindrical rod. This extension is calculated to be (at widest point of the hull) $\frac{1.446}{cos30} - 1.446 = 0.223$m.

As the rods will be attached at different widths of the hull (is not the same width the whole way), these cylinders will vary in length.

This extension is much more difficult to achieve in the bow-stern direction, and a slider bearing system will have to be used instead. This slider system will either be created on the rods, or at the actuators. It would have been preferable to create the system shown in Figure 13.2 as rods could have been used to make the extension on both sides, however the current actuators were not capable of the force that would have needed to be provided (this is further investigated in section 13.8 on pg79).

13.5. Rod end

The spherical rod ends are providing the pivotal movement, and need to have a misalignment of 30° and a large bore width, capable of holding a large cylindrical rod. Looking online, the largest bore width available for a rod end with high misalignment 30° is 31.75mm [20]. This will be the corresponding diameter of the cylindrical rod.

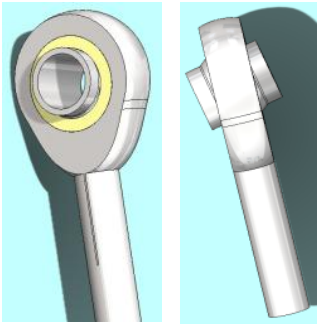


Figure 13.10

13.6. Bracket

The rods can be attached to the hull using a bracket, which will be like a belt encircling the outer perimeter of the hull, and bolted onto the sides. The rods will be bolted through the bracket to the inside of the hull. The material for this bracket needs to be strong enough to withstand the forces due to the actuators (needs high tensile strength), whilst also being ductile enough to be shaped around the hull of the boat.

Calculate tensile strength: Assume that the bracket will have a small width, and be relatively thin,so that it is as light as possible as not to interfere with actuators. The bracket needs to have a

[20] http://www.rbcbearings.com/repb-aerospace/index.htm

much higher yield stress than the stress generated by the force of the actuators, so that it can be bent around the perimeter of the hull, but does not yield due to actuator force. I applied a safety factor of 5 to all calculations, in order to ensure yield stress is much higher than actuator stress.

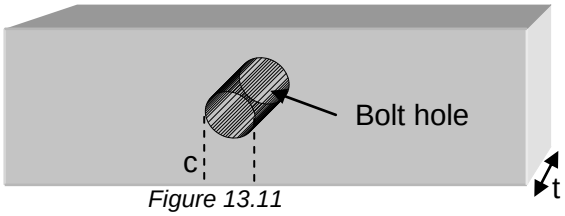


Figure 13.11

The bracket stress state will be in pure shear due to actuators. According to Von Mises, shear strength = $\frac{yield\ strength}{\sqrt{3}}$. The bracket shear stress needs to be less than the shear strength. If we assume thickness of bracket *t* is approximately between 2mm and 10mm (to have a snug fit with the cradle). Also assume any bolt holes will be centred across the bracket. Finally assume that the bracket width is no greater than 75mm, and no less than 35mm (such that cylindrical rod can be large diameter). Material shear stress > $\frac{F \times sf}{2 \times c \times t}$.

So maximum (worst case scenario) material shear strength > $\frac{5 \times 1024}{2 \times 17.5 \times 10^{-3} \times 2 \times 10^{-3}} > 73.14 MPa.$

Therefore material yield strength > 73.14 x $\sqrt{3}$, >126.7MPa

Calculating elongation: The material needs to be able to be bent around the outside of the hull without fracturing. The force needs to 5120N or greater. So need to calculate the size of elongation necessary for the bracket. If we approximate the hull to a cylinder, this is easily done (see Figure 13.14). Firstly the maximum radius of approximated circle was calculated, and so the curved arc length. Then the difference in length of the bent section compared to the initial straight section. This change in length is divided by the initial length to find the percent strain (or percent elongation).

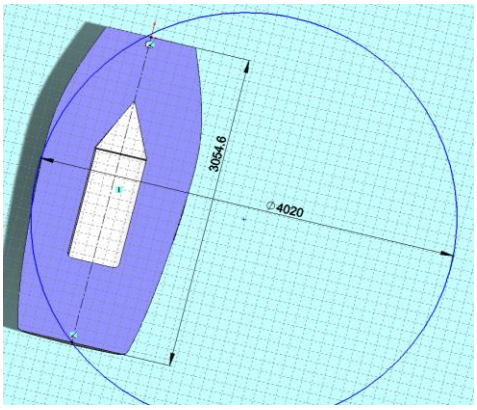


Figure 13.13

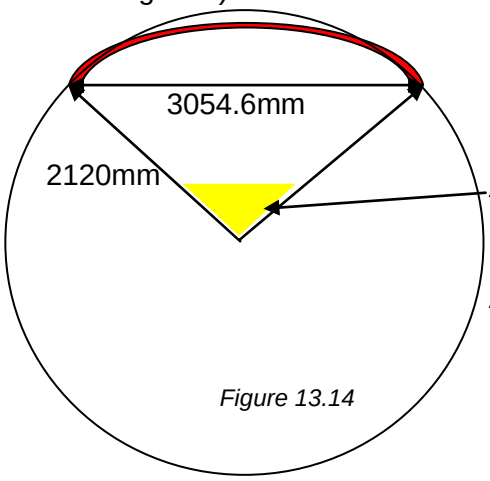


Figure 13.14

Length of hull section = 3054.6mm

Maximum radius of approximated circle = 2120mm

Angle of arc = $2 \times cos^{-1}(\frac{1527}{2120})$ = 87.8°

Arc length = $\pi \times 2120 \times 2 \times (\frac{360}{87.844})^{-1}$ = 3250.325mm

Percent Elongation (Strain) = $\frac{3250.325-3054.6}{3054.6} \times 100$ = 6.41%

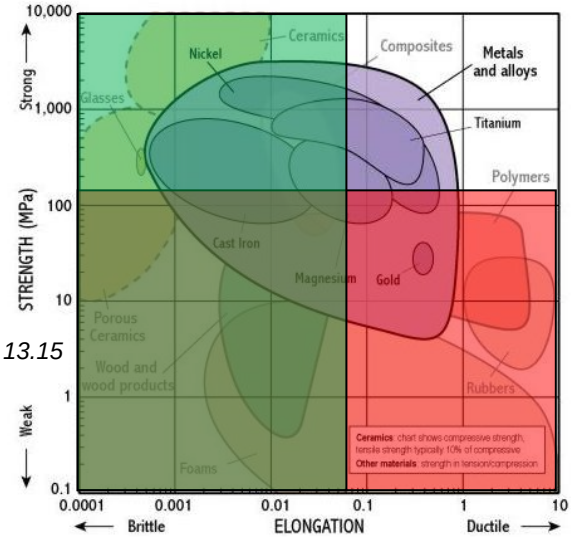


Figure 13.15

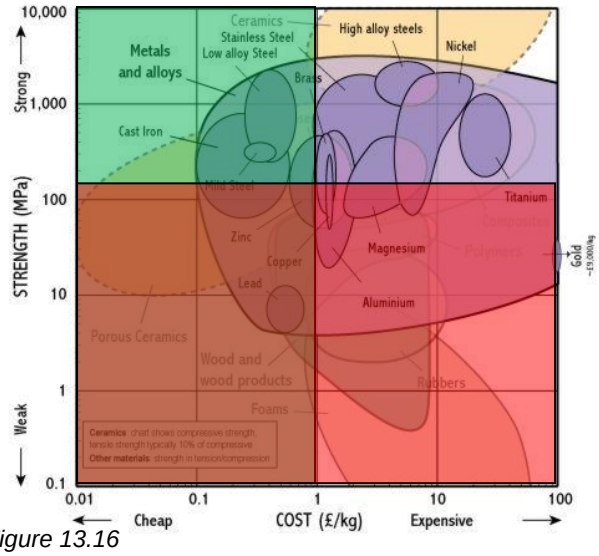


Figure 13.16

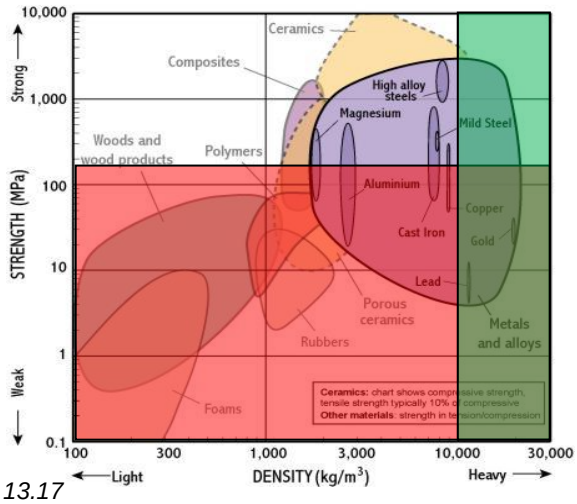


Figure 13.17

By using a material selection chart (Figure 13.15) to section off area's that fall within requirements, it is obvious that the only adequate material are metals, such as steels, titanium or magnesium

The second point of material choice is density. The material needs to be quite light so it will not affect the capabilities of the actuators. The volume of the bracket is 0.00096m^3 and we want a total weight of less than 10kg, so density must be less than 10,416kgm^{-3} (Figure 13.17).

Finally the total weight will affect the cost, ideally a cost of less than £1/kg is ideal (Figure 13.16).

Chosen material: Hot rolled mild steel A36, with yield strength of 250MPa, and elongation of 23%. It is also relatively cheap, at £0.75/kg, and density of 7850 kgm^{-3}. This is readily available in the engineering department in 50mm x 3mm lengths, which shall be used.

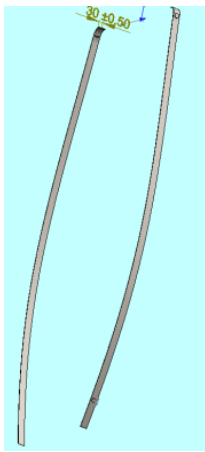


Figure 13.18

Calculating number of bolt holes needed to attach bracket to the boat

In order to not split the hull, we want to use as few bolt holes as possible with a small diameter. Assume that bolt material used is high-yield structural steel. Need to ensure that the average bearing stress < yield stress of bolt.

1. $\frac{Force\ on\ bolt \times safety\ factor}{number\ of\ bolts \times diamter\ of\ bolt \times thickness\ of\ bracket\ and\ hull} < \sigma_y$. On each side of the bracket there will be a force on the bolts given by two actuators, so the force on the bolts will be 2048N on each bracket side.

2. $\frac{2048 \times 3}{x \times d \times 0.005}$ < 400MPa, so 3 x 10^{-3} < x.d

This means that there needs to be at least one bolt hole, of at least 3mm diameter on either side of the bracket. If we put in 2 bolts spaced out along the bracket, each 3mm in diameter, the bracket will not fail, and will not pivot about bolt holes.

13.7. Connecting cylindrical rods to bracket and hull

Due to the fact that the bracket will be bent around the hull, the rods will protruding out of the boat at different angles at the front and back of the hull. However, as we will be using three piece spherical rod ends, they will account for any misalignment up to 30°. Used Solidworks to calculate the angles needed to place the cylinders on the bracket, at 15° and 21°. Therefore this misalignment will be accounted for using the rod ends.

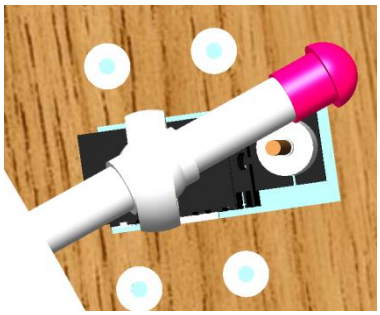


Figure 13.19

Bolt material and diameter

To determine the diameter of the bolt and the material needed, I will have to ensure that the bolt will not fail under two load states, static loading of 1024N, and then cyclic loading due to wave movement of 1024N (the force provided by each actuator).

Static Loading

Need to be able to calculate the minimum bolt diameter connecting the rod to the boat, and the material required, so that the bolts do not yield. There will be two static stress situations applied to the bolt, one created by the force of the actuators, and one by the tightening preload that needs to be applied to ensure that the bolt doesn't slip from contact with the hull in shear.

A) We consider the stress caused by the actuators. The largest stresses occur at the maximum length of cylindrical rod, as it will create the largest moment. The maximum length of the cylindrical

rod is found by calculating the extension needed at the narrowest hull width. This width is at 1.269m, so that extension = $\frac{1.446}{cos30}$ - 1.269 = 400mm.

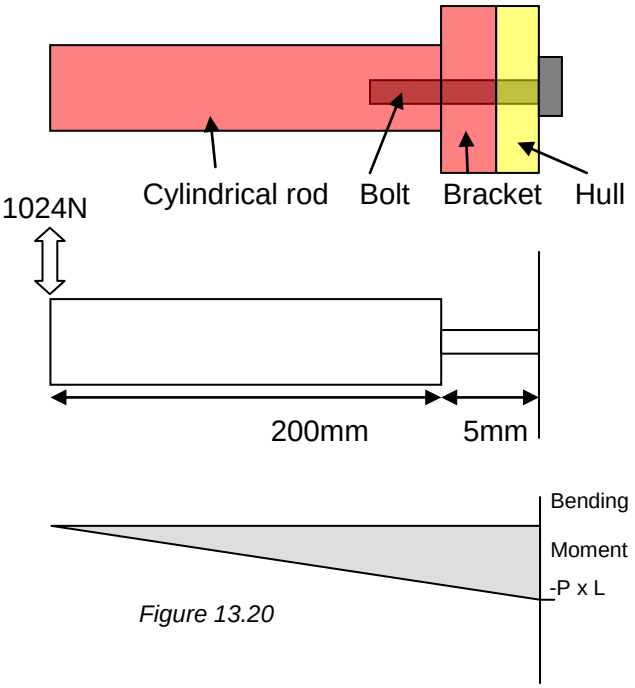


Figure 13.20

Therefore each rod has to be 200mm (shorter rod length at wider part of the hull at 1.4457m, so rod length = 112m.) This cylindrical rod will act as a cantilever (see Figure 13.20).

There will then be a total bending moment of (1024 x (0.2 + 0.05)) = 209.92Nm. If we apply a safety factor of 3, then the maximum bending moment is found to be 629.76Nm.

Maximum stress in the bolt due to actuators is found using the formula:

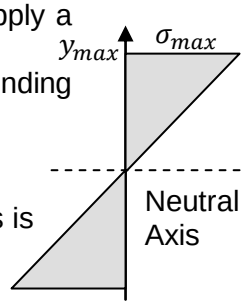


Figure 13.21

3. $\sigma_{max} = \frac{M_{max} \times y_{max}}{I}$ (Figure 13.21)

4. $\sigma_{max} = \frac{629.76 \times r \times 4}{\pi r^4} = \frac{629.76 \times 4}{\pi r^3}$ (where r is radius of the bolt).

B) There is a secondary stress coming from tightening the bolt itself, called a preload. This is proportional to the tightening torque:

5 . $N = \frac{T}{k \times d}$, where N = Tension force, T = tightening torque, k = nut constant, d = diameter.

We want the tightening bolt stress to be greater than the stress applied by the actuators, so that it is always in compression on the outside of the hull, so that it doesn't slip in shear.

6. $\sigma_{max} < \frac{N}{A_B}$, where A_B is the bolt area.

The bolts should then not yield due to the combined stress states:

7. $\sigma_{max} + \frac{N}{A_B} < \sigma_y$.

If we rearrange and combine equations 6 and 7 to eliminate N, such that:

8. $2\sigma_{max} < \sigma_y$,

9. $\sqrt[3]{\frac{2 \times 629.76 \times 4}{\pi \times \sigma_y}} < r$.

If we assume σ_y is high yield structural steel, with tensile strength = 400MPa, then the diameter of the bolt has to be greater than 31.76mm. However, the maximum bore of the spherical rod end is 31.75mm, which means that the diameter of the bolt necessary not to fail using this material is too wide.

This means that the material of the bolt needs to be changed, to ultra high strength steel H11[21]. According to matweb[22], H11 steel σ_y = 1650MPa. By re-performing previous calculations, the diameter of the bolt reduces to 20mm, which should fit easily into the cylinder.

10. σ_{max} (using equation 4) = 802MPa.

The desired preload on the bolt also has to be greater than the shear force created by the actuators, so that it does not slip and lose contact with the hull. Assume that the washer material will be metal, such that coefficient of friction[23] $\mu = 0.1$. Then Nμ > actuator force, N>$\frac{3072}{0.1}$, N>30.7kN.

N is also affected by the stress generated by the actuators force so that $\sigma_{max} < \frac{N}{A_B}$, as such N>251kN. Therefore can find the tightening torque using equation 5, with nut factor of material taken as 0.2 for plain finished bolts. T>1004Nm, so can assume T=1010.

Property Class	8.8		10.9		12.9 Socket Head Cap Screw	
Minimum Tensile Strength MPa	M6 - M16: 800 M20 - M30: 830		1040		1220	
Nominal Size and Thread Pitch	Bolt Torque Specs in Newton Meters - Nm					
	Dry	Lubed	Dry	Lubed	Dry	Lubed
M10 x 1.50	50	38	72	54	84	64
M12 x 1.75	88	66	126	94	146	110
M14 x 2.00	141	106	201	150	235	176
M16 x 2.00	218	164	312	233	365	274
M18 x 2.50	301	226	431	323	504	378
M20 x 2.50	426	319	609	457	712	534

Table 13.1

Using generic metric bolt torque table provided by CNC machines[24] (Table 13.1), can see that an M20 bolt of highest grade 12.9 steel has a bolt torque specification that is quite similar to that calculated for the torque, in fact T = 1010Nm is quite generous. Using equation 5, N = 253kN.

[21] http://www.vegasfastener.com/raw-materials.php#steel
[22] http://www.matweb.com/search/DataSheet.aspx?MatGUID=2285b7f0dd2f440abc57dea68458bb22&ckck=1
[23] http://physics.info/friction/
[24] http://www.cncexpo.com/MetricBoltTorqueNm.aspx

Cyclical Loading

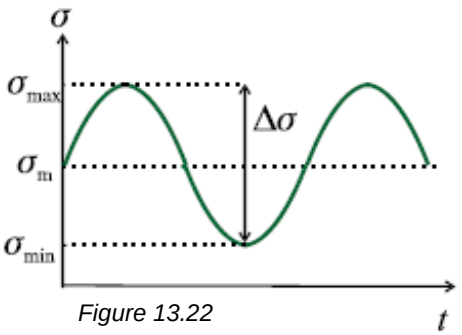


Figure 13.22

Due to the nature of the waves, the bolt will have to endure cyclic loading. The loading endured will be a maximum of 1024N upwards and 1024N downwards (as magnetic spring accounts or gravitational forces) Figure 13.22.

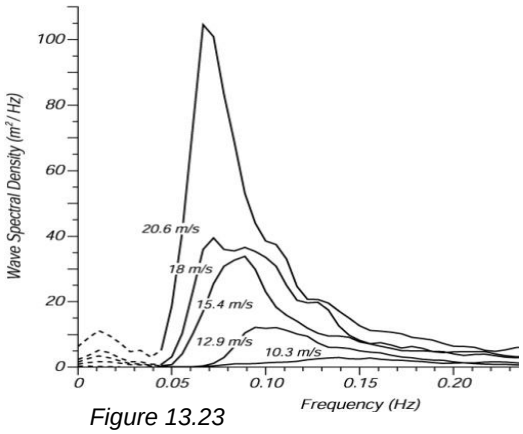


Figure 13.23

Therefore σ_{max} = 802MPa, σ_{min} = -802MPa, and $\Delta\sigma$ = 1604MPa, about σ_m = 0N (generated from actuator force 1024N equation 10). We would like to ensure the life of the cradle, at approximately 5 years. Using the Pierson-Moskowitz spectra[25] (Figure 13.23) on a fully developed sea, average wave frequency is between 0.07-0.12 waves per second. This means that number of cycles to failure:

11. N_f = s.f x seconds in 5 years x wave frequency
12. $N_f = 5 \times 60 \times 60 \times 24 \times 365.25 \times 0.12 = 1.89 \times 10^7$

As this is high cycle fatigue, we need to ensure that $\sigma_{max} < \sigma_y$. However, from previous calculations we have already ensured that $2\sigma_{max} < \sigma_y$ (equation 8). Therefore we know that the bolt will not yield under cyclic loading until a high number of loads.

The endurance limit (S_e) for steel is the stress below which cyclic loading never causes failure, at $\sigma_{max} < S_e$.

S_e = 690MPa for steel with tensile strength > 1400MPa[26]. Therefore the cyclic loading will eventually cause failure, as 802MPa $\nless$ 690MPa. However it will be after several years, as $\sigma_{max} < \sigma_y$, so $N_f > 10^5$. This estimate has also been made for the boat to be in continual use. However this is not the case, and so the lifetime of the simulator will probably be much longer.

Using fatigue sample tests, life cycles to failure for H11 steel can be found using the following Basquin equation, where $C = \Delta\sigma\ N_f^a$, and a and C are experimental constants.

[25] http://oceanworld.tamu.edu/resources/ocng_textbook/chapter16/chapter16_04.htm
[26] http://www.roymech.co.uk/Useful_Tables/Fatigue/Fatigue.html

Calculating washer size

There is going to be a large stress state around the small bolt head area generated by the actuator force. Therefore a washer is likely to be needed to be used, due to the low yield stress of the fibreglass hull material. For an M20 bolt, the diameter of the washer head is 34.6mm[27]. Therefore total stress area over which the stress state acts due to preload tension is: 13.A_{hex} = (1.5 x 34.6 x 19.97) – (π x 0.01²) = 7.22 x 10^{-4}m².

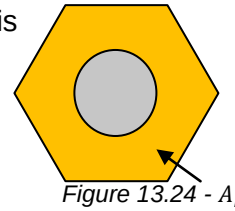


Figure 13.24 - A_{hex}

We can approximate area of which σ_{max} acts as a circle of diameter 34.6mm, and so σ_{max} occurs where $y_{max} = 17.33mm$ (equation 3). Yield strength of chopped strand mat laminate fibreglass (hull material) is 100MPa[28]. To prevent yield in the fibreglass hull:

14. $\frac{N}{A_{hex}} + \sigma_{max} < 100MPa.$

15. $\frac{253 \times 10^3}{7.22 \text{ x } 10^{-4}} + \frac{629.76 \times 4}{(17.33 \times 10^{-3})^3 \pi}$ = 504MPa $\nless$ 100MPa

Therefore the hull will yield under this stress. This means a large washer will be needed, with a material yield strength > 504MPa. This washer radius needs to be such that

16. $\frac{253 \times 10^3}{(\pi r^2 - \pi \text{ x } 0.01^2)} + \frac{629.76 \times 4}{r^3 \pi}$ <100MPa.

Therefore r>0.0283m (full calculations logbook page 50). Therefore diameter of washer needs to be 0.057m, using titanium grade 5 material, of yield strength 880MPa.

Due to the gradient of the boat, the washer will need to be angled. There is a difference in gradients at either end, so requires two washers that need an angle of 10° and 20°. These will have to be machined.

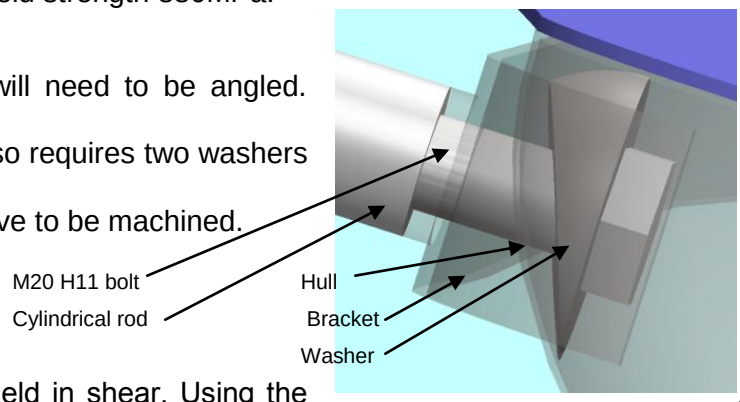


Figure 13.25 – angled washer at 20°

Hull

Want to ensure that the hull material doesn't yield in shear. Using the fact that material shear strength > $\frac{F \times sf}{2 \times c \times t}$, can ensure that the hull won't yield.

Shear strength = $\frac{yield\ stress}{\sqrt{3}}$, and hull material yield strength = 100MPa, so shear strength = 58MPa.

[27] http://www.roymech.co.uk/Useful_Tables/Screws/Hex_Screws.htm
[28] http://en.wikipedia.org/wiki/Glass-reinforced_plastic

Minimum c is from the top of the hull to the bolt = 25mm (assume the bracket is placed directly at the top of the hull, and it is 50mm wide). Hull thickness = 2mm.

So $\frac{F \times sf}{2 \times c \times t} = \frac{1024 \times 3}{2 \times 0.025 \times 0.002}$ = 27.4MPa. 27.4MPa <58MPa, so the hull won't yield.

Cylindrical rod

As the diameter of the rod is determined by the maximum bore width of the rod end of 31.75mm, and maximum cantilever length of 200mm, σ_{max} can be found to be (from equation 1)

17. $\frac{614.4 \times 4}{\pi(0.015875)^3} < \sigma_y$. Therefore $195.53MPa < \sigma_y$.

We also want to ensure that the failure will be of high cycle fatigue.

The endurance limit, $S_e = \frac{tensile\ strength}{2}$ for materials with tensile strength < 1400MPa[29]. Therefore $195.53 < \frac{\sigma_f}{2}$. We can then use a cheap steel, such as a mild steel, with σ_y = 240MPa and σ_f = 500MPa, satisfying both static and cyclic restraints[30]. This means that the rod should never fail under static or high cyclic loading.

13.8. Mounting cradle

Due to the large stroke length of the actuators, the total distance at maximum stroke length from the floor will be 2m. This will be dangerous for those using the simulator, as to perform hiking positions the sailor will be positioned quite far out of the sides of the boat.

According to health and safety for working at heights, a fall over from 2m or over can cause serious injuries[31]. Therefore we will use a wooden plywood stage[32] to "sink" the actuators into, such that the maximum distance to fall is reduced to only 82cm. For added safety, crash mats will be put either side of the hull.

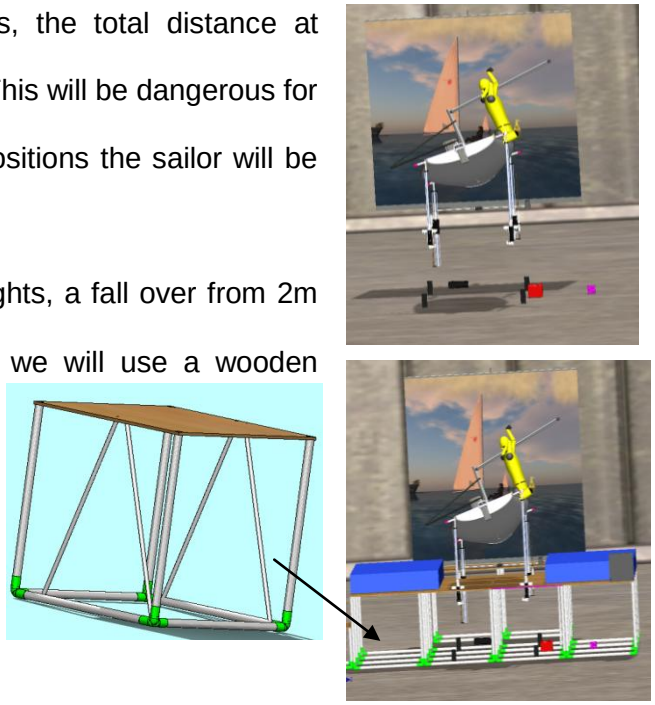


Figure 13.26

[29] http://www.roymech.co.uk/Useful_Tables/Fatigue/Fatigue.html
[30] http://www.metals4u.co.uk/detail.asp?cat_id=134&prd_id=2536
[31] www.hse.gov.uk
[32] http://www.alibaba.com/product-gs/376288197/display_stage_wooden_stage_show_stage.html

The staging gave an unplanned benefit, all of the electronics; controllers, power packs, sound system, projector etc can be easily put underneath. This added bonus is particularly beneficial in a teaching environment with children around.

Slider

To produce the bow-stern movement, either the actuators of the end of the cylindrical rods need to be able to move forwards and backwards in their relative housing. Due to the fragility and thin shell of the hull, I think it is more sensible put the slider at the actuators as they are mounted on thick plywood staging.

The vertical extension is provided by the maximum stroke length of the actuators, of 0.84m, which provides an angular title in the bow stern direction of $tan^{-1}\left(\frac{0.8348}{3.0546}\right) = 15.3°$. Due to extension needed to provide angular tilt of 15.3°, actuators also need to move forwards and backwards by a distance of $\frac{3054}{\cos(15.3)} - 3054 = 112$mm.

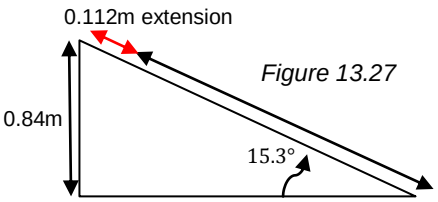


Figure 13.27

We can use a simple slider bearing to produce this movement, and so cut these holes of this size in the staging so that the actuators can be moved forwards and back. As the maximum speed of the actuators is 2.4ms^{-1}, need to use a slider that is capable of such fast movement. This can be achieved using the Speed Demon[33] linear rails, capable of taking speeds up to 10ms^{-1}.

The rail is mounted onto the staging by six M10 bolts[34], and so calculations are made to ensure that the plywood stage will not yield due to stress generated by the bolt. Firstly we have to ensure that the bolts will not yield, and so decide on material to be used.

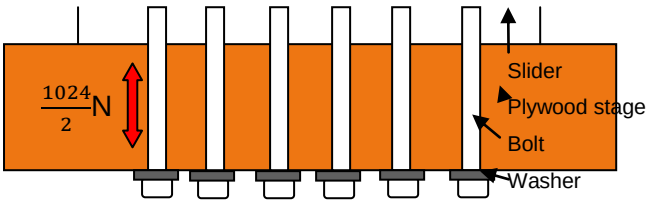


Figure 13.28

18. $\frac{force\ on\ bolt \times safety\ factor}{no\ bolts \times bolt\ shaft\ area} < \sigma_y$

Assume bolts are high tensile steel 8.8 grade,

$\sigma_y = 400MPa.$

19. $\frac{1024 \times 3}{6 \times 2\ \times \pi \times 0.005^2} < \sigma_y$, 3.3MPa < 400MPa so they won't yield.

[33] http://www.drivelines.co.uk/product1.aspx?id=64
[34] http://www.screwfix.com/p/bolt-bzp-high-tensile-steel-m10-x-70mm-pack-of-50/13827

To check for cyclic failure, $\frac{force\ on\ bolt \times safety\ factor}{no\ bolts \times bolt\ shaft\ area} < \frac{\sigma_f}{2}$, 3.3MPa < 300MPa so won't yield ever cyclically.

Now had to see if the combined stress states of bolt preload and the actuator force stress won't cause yielding in the plywood, if not then a washer will need to be used.

20. $\frac{N}{A_{hex}} + \frac{actuator\ force \times s.f}{no.bolts \times A_{hex}} < \sigma_y$

N can easily be calculated using the generic torque Table 13.1, by assuming that T = 50, k = 0.22 and d = 0.01 so from equation 5, N = 22.7kN. A_{hex} is calculated as before for M20 bolt (equation 13) so = 2.54 x 10^{-4}m^2. σ_y plywood = 13.8MPa[35].

21. $\frac{22.7 \times 10^3}{2.54 \times 10^{-4}} + \frac{1024 \times 3}{6 \times 2.54\ 10^{-4}} < 13.8$, 91.4MPa $\nless$ 13.8MPa

Therefore washers will be needed for the plywood.

Bolt shaft area = $\pi 0.005^2$ = 7.85 x 10^{-5}m^2.

22. $\frac{22.7 \times 10^3}{(\pi x^2 - 7.85 \times 10^{-5})} + \frac{1024 \times 3}{6\ (\pi x^2 - 7.85 \times 10^{-5})} < 13.8$MPa

x > 0.02367.

So need to use washers of diameter > 47mm, and mild steel material.

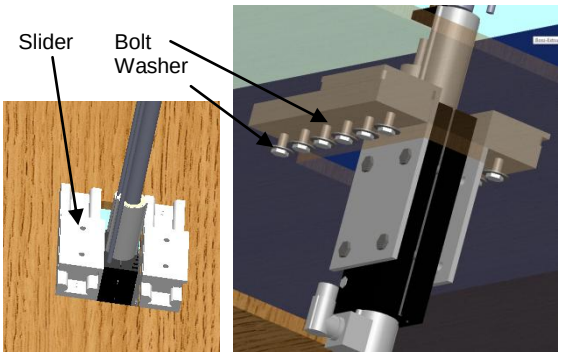


Figure 13.29

Actuator bracket

The actuator needs to be supported vertically on this slider, and in a stationary position at the other end of the hull (Figure 13.31 and Figure 13.30). The easiest way to do this is to attach an L-shaped piece of aluminium to the flange and corresponding slider or plywood staging. There will be two stress states that have to ensure does not yield. A) stress due to actuators and preload on the top of the L shape, and B) a shear force stress state and preload on the side of the L shape (again both static and cyclic loading conditions).

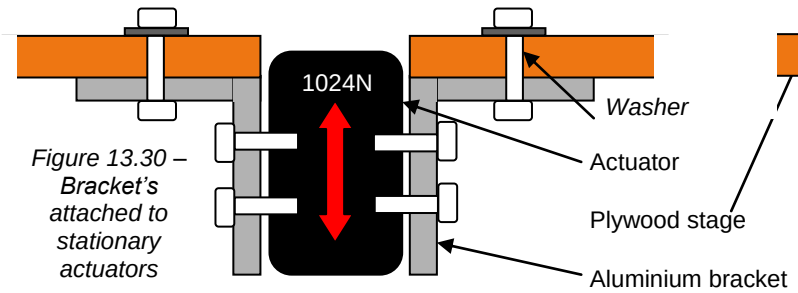


Figure 13.30 – Bracket's attached to stationary actuators

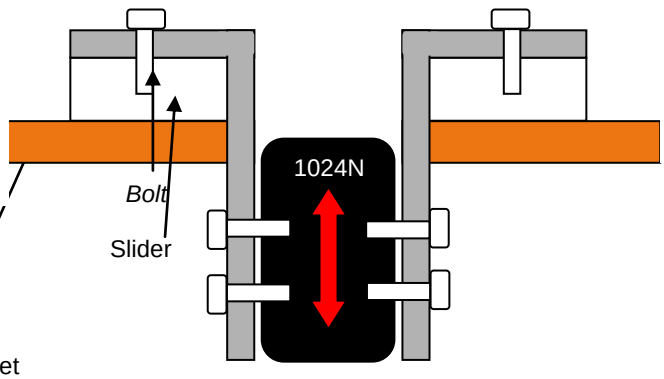


Figure 13.31– Bracket's attached to sliding actuators

[35] http://robots.freehostia.com/Materials/MaterialsBody.html

A) The dimensions of the top of the L shape are given by the dimensions of the slider, to ensure correct force distribution they should be exactly equal. The bolt sizes are specified already by the slider as two M10 bolts. We need to ensure that this will take the forces, such that we do not have to drill any more holes. In order to not yield in static loading:

17. $\frac{force\ on\ bolt \times safety\ factor}{number\ of\ bolts \times bolt\ area} < \sigma_y$ and assume σ_y =400MPa for high-yield structural steel.

18. $\frac{1024 \times 3}{2 \times 0.005^2 \pi} < 400MPa$, so 19.95MPa < 400MPa, so they will not yield.

For cyclic loading, $\frac{force\ on\ bolt \times safety\ factor}{number\ of\ bolts \times bolt\ area} < \frac{\sigma_f}{2}$, 19.95MPa < 300MPa, so will not yield cyclically.

We will need to use washers so that the plywood doesn't yield, as there are fewer bolt holes than the slider calculations (equation 20), but equal stresses generated. The aluminium bracket will not yield, due to the higher yield strength, so no washers are needed.

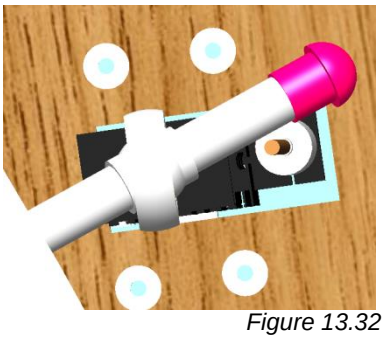


Figure 13.32

Using the same calculation as equation 20, but with (2 x 2) bolts instead, washer diameter > 48mm (see logbook pg79 for calculations). So we can use the same 50mm diameter mild steel washers for the slider and the actuator bracket[36].

B) The side plate of the aluminium bracket is loaded in shear by the actuator. Need to ensure that the bolts don't yield, there is no washer required, the normal force x friction is greater than shear force, and the bracket won't yield.

To calculate the stress in the bolts, the situation can be approximated to a cantilever similar to that for the cylindrical rod bolts, however the cantilever distance is much smaller. Use equation 7 of combined stress states, and equation 3 to generate σ_{max}, with the length creating the moment = 16.35mm.

Therefore σ_{max} = 64MPa, and $\frac{N}{A_B}$ = 289MPA, $\frac{N}{A_B} + \sigma_{max}$ = 353MPa < 400MPa. This means bolts won't yield.

Check for washers: as A_B is smaller then A_{hex}, and 353MPa < 395MPa, then we know that no washers are required (logbook page 84).

[36] http://shop4fasteners.co.uk/acatalog/Repair_Penny_Mudguard_Washers_Zinc_Plated_BZP.asp

Need to ensure $number\ of\ bolts\ \times N\ \times \mu > \frac{F \times sf}{2}$, 4 x 22.7 x 10^3 x 0.1 > 1536N

9080N > 1536N, so we know that the bolts won't slip in shear.

For cyclic loading, σ_{max} < σ_y, and to ensure yielding never occurs, $\sigma_{max} < \frac{\sigma_f}{2}$.

As 64MPa < 300MPa, they will never yield.

To ensure the bracket doesn't fail, use same equation material shear strength > $\frac{F \times sf}{2 \times c \times t}$.

C = 13.5mm (see logbook pg82), giving worst case shear stress= 1.1MPa. So yield strength must be > 1.1 x $\sqrt{3}$ = 1.93MPa. Therefore the bracket won't yield, σ_y = 395MPa.

Staging sections

Holes were needed to be cut into the staging to allow for the slider movement and the stationary actuators to be mounted.

13.9. Human

Finally, I drew a life size human with fully rotational joints that could be placed on the simulator. This was to be able to see how they moved in the boat, and to tell how large the screen had to be and where it should be placed. It was easy to see how a person would interact with the moving boat, and control the rudder and tiller.

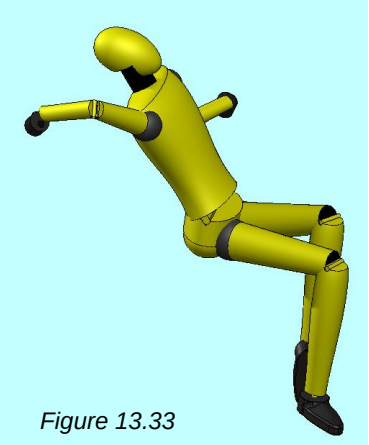
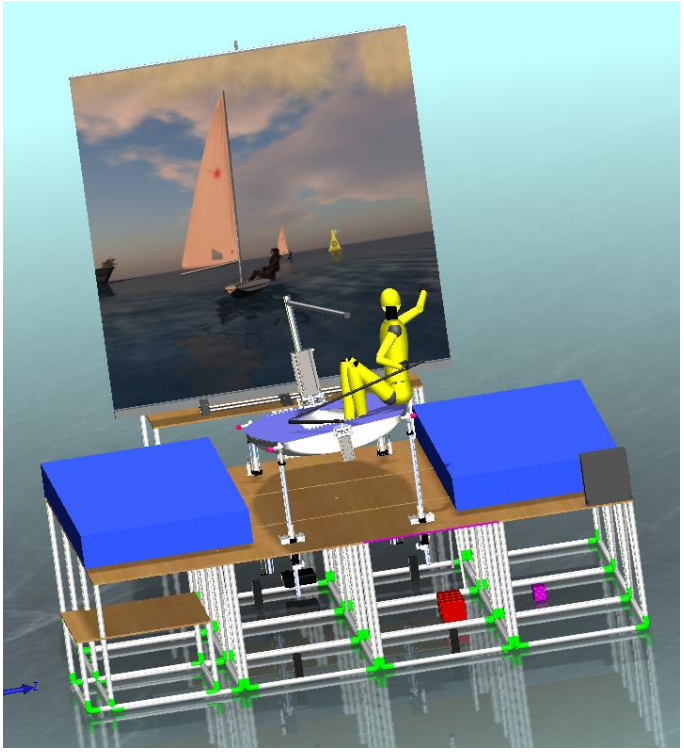


Figure 13.33



Hinge joints and restrictive mates to certain angles and distances were used, so that the human made sensible movements to the requirements. This now simulated how the human model reacted to hiking.

All cradle components were then drawn together to give the finished product.

Figure 13.34- finished cradle

Figure 13.35

14. Display and sound

14.1. General Specification

We are aiming to produce a sailing simulator such that the user is fully immersed in the simulation, and is as real and life like as possible. We will do this by using a HD screen and projector, with the aim of progressing to 3D capabilities in the future. A surround sound system will also be used to generate wind sounds, such that the sailor can tell by listening where the wind is coming from. This will provide a much more realistic simulation experience, and the novice sailor will learn skills that are more transferable to the water.

14.2. Display

Projector

Choosing a projector is dependent on many factors, such as room lighting (lumens required), size of screen projecting onto, throw distance (how far away it needs to be from projected area), capabilities (HD, 3D, wireless etc), and lens shift. After going through each of these factors, a list of potential projectors was created. The most cost efficient of these was chosen. This then led to the screen choice.

Room Lighting - It can be assumed that room lighting can be an issue for most sailing schools, with little control over light. Therefore a projector with a high number of lumens is desired. The average projector range is around 1000-6000 lumens. To overcome high levels of ambient light, a projector with over 2000 lumens or higher is desired.

Screen Size – To make the viewer feel like they are in a “real world” environment, a large screen size is very important. The end of the hull is to be removed, such that on screen the viewer can see the hull as if it is in front of them. It would be preferable to generate an image that is close to the same size as the tip of the hull. However, at around 1m wide, it is unrealistic to generate such a large image. It would be more preferable for the viewer to see wider either side of the hull, which is more realistic and enables a wider view of any upcoming obstructions. If we assume that the hull is going to be around half real size (0.5m), and we want a view of at least 2 metres either size real

time (1m on the screen), we need a screen that is bigger than 2.5m wide. Therefore we need a projector that is able to project onto such a large size.

Throw Distance – such that the simulator is widely marketable, we want to choose a projector that has a very short throw distance. This is because for ease of use, we want to be able to put the projector underneath the staging area such that it doesn't get broken. It will be difficult to implement ceiling projectors due to varying ceiling heights, and they also have to be placed at a level higher than the mast at its highest point for uninterrupted viewing. Therefore the projector will be placed at the front of the stage area (so that the stage doesn't interrupt viewing). To be commercially viable we need to be able to produce a viewing area in small rooms. This results in choosing a projector with a very short throw distance.

Lens shift – The screen will most probably be placed on a wall, or on some form of staging. As the projector will be placed on the floor, and not level with the bottom of the screen, projectors with lens shift capabilities need to be used. If we assume that the minimum acceptable distance from the projector to the screen is 2m, and the maximum height off the ground of the bottom of the screen is 1.2m, then we need a projector with a lens shift of $tan^{-1}\left(\frac{1.2}{2}\right) = 30.9°$. There is an option to use keystone correction (which cheaper projectors use), however using lens shift to correct the simulation will be a visual improvement. Using digital keystone correction, the image undergoes a compression and conversion that recreates edge pixels, sometimes causing noticeable artefacts along the corrected portion of the image. Keystone correction works well with still images and video in a temporary setting. In a permanent installation, using lens shift will produce the best images. Therefore it is ideal to use lens shift as opposed to keystone correction.

Capabilities – the minimum requirement for the projector is to be able to display in HD. An added benefit is to be able to display 3D images, which will give sailors a fully immersive experience. We are not in this project, aiming to create 3D graphics. However, this is an area of further investigation, and would be easier to have existing 3D projectors for future 3D capabilities. It would be ideal to have a fully wireless component, so that the projector can be easily moved and do not have to worry about trailing wires.

Using the BenQ and Projector People throw distance calculator[37], the throw distance to provide an image of 2.5m wide was found, see Figure 14.1.

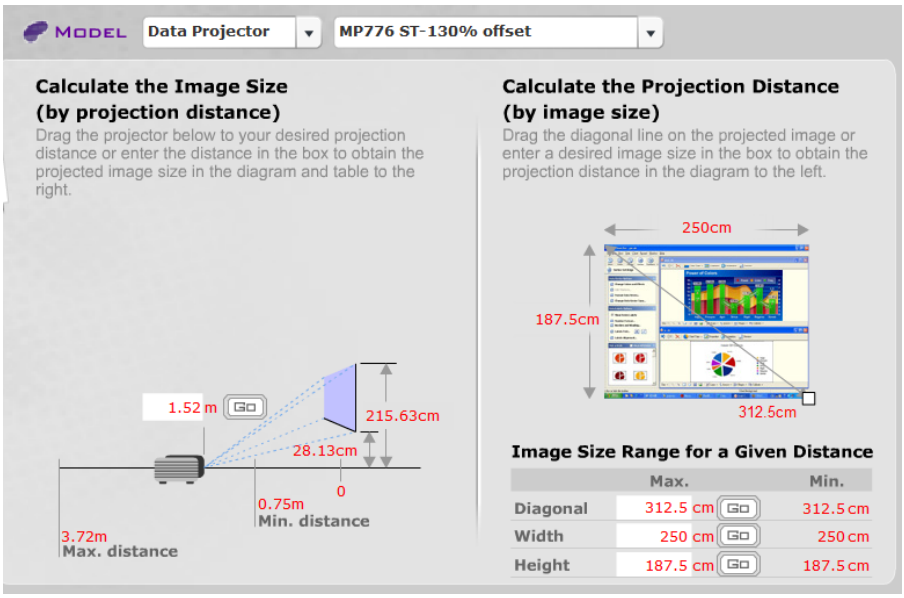


Figure 14.1

Possible Projectors

Projector	Lumens	Image Size	Throw Distance at image size 2.5m	Lens Shift/Keystone	Capabilities	Price
BenQ W6000	2500	Up to 7.62m	4.32m	Keystone +/- 30° Lens Shift +/- 30°	HD	£1029.47
BenQ MP776 ST	3500	Up to 7.62m	1.52m	Auto Keystone Lens shift +/- 30°	HD, 3D	£807.00
Epson Powerlite 4300	5200	Up to 7.62m	4.24	Keystone +/- 40° Lens shift +/- 50°	HD, wireless	£1345.00

Table 14.1

Result

The high price tag for the Epson Powerlite 4300 did not justify the slightly better specifications. The higher keystone and lens shift is unnecessary in the situation, and the wireless capability is less of an improvement than 3D ready. The BenQ MP776 ST[38] is the best solution to the problem, with high lumens, adequate lens shift and

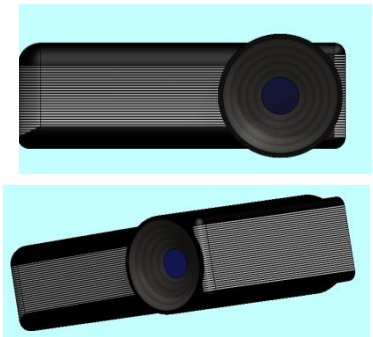


Figure 14.2

[37] http://www.benq.com/product/projector/calculator/
[38] http://www.projectorcentral.com/BenQ-MP776_ST.htm

keystone correction, and a very short throw distance. It is also 3D ready, for when 3D graphics are designed.

Screen

Now that the projector has been chosen, the screen decision can be made. This is dependant also on many factors, projection method, ambient light conditions, display brightness, room configuration size, projected image make up, projector placement, and screen maintenance.

Projection Method **–** The BenQ MP776 uses DLP or 3D projection technology. The screen manufacturer will ideally be able to have a choice of screen fabric, but in this case we will only focus on the DLP projection system.

Ambient Light Conditions - Ideally have a screen that is able to cope in any ambient light conditions, however we will assume that most ambient light can be controlled, and the brightness of the projector will make up for any short falls with the screen capabilities.

Display Brightness – If the projector is not producing bright images on the required screen size, then a screen that is optimised for low display brightness is necessary. This occurs if the display brightness is less than 25 foot lamberts. This can be calculated using formula for foot lamberts $\frac{LUMENS\ of\ projector}{Square\ footage\ of\ screen} = foot\ lamberts.$ Using measurements for a screen of approx 64 square foot (around 2.5x2.5 m²), and using the projector lumens of 3500, a brightness of 54 foot-lamberts was calculated.

Room configuration size **–** If the viewing angle of the audience is greater than 30°, then a screen fabric providing a greater viewing angle is required. The width of the screen will be larger than the width of the boat, so that at any point the viewer will be viewing the screen at an angle of less than 0°. The instructor will be standing at the widest point on the staging at 1.83m from the centre of the screen, so around 0.5m outside of the screen, and 5.66m away from the screen. Therefore, the viewing angle is $90 - tan^{-1}\left(\frac{5.66}{1.83}\right) = 18°$. This is much less than the 30° angle which needs a screen fabric providing a greater viewing angle.

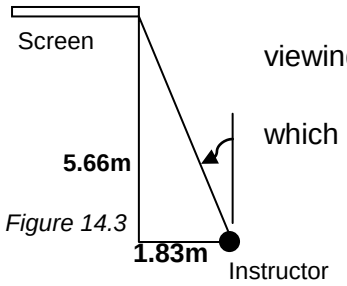


Figure 14.3

Projected Image make up **–** Due to the DLP projection image by the BenQ MP776, the image make up is digital or pixelized data with fine detail.

Projector placement - this is dependent on where the audience and projector are compared to the screen normal. In this case, the audience and the screen will be on opposite sides of the screen normal, a reflective screen is required (see Figure 14.4).

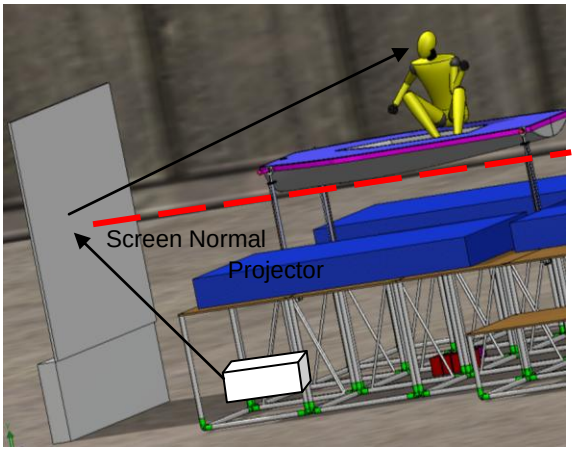


Figure 14.4

Maintenance **-** As the simulator will be used in an educational setting with children around, the screen needs to be hardy and wipe proof.

Resulting Screen Material

Matte White[39] **–** This is of the most versatile screen surfaces, and also the cheapest. Used in situations with projection type is DLP, display and the display is bright. It is hardy and can be wiped clean, which is good in an educational situation.

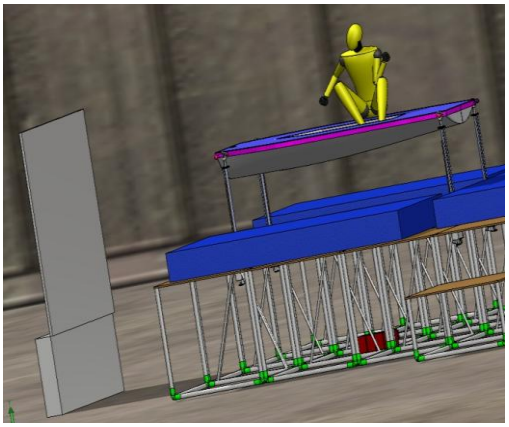
Screen Size

As stated before, the screen size needs to be at least 2.5m wide. The aspect ratio can range from 1:1, 4:3 or 16:9, but 1:1 or 4:3 is more desirable. This means that it needs to be at least 2.5m tall also. The screen also needs to have the option of being wall mounted or mounted on the floor. The screen also needs to be made out of matte white material. It should also be portable, for use when the simulator is being rented.

I decided on several portable screens that met these specifications, and then created basic drawing of them in Solidworks. I then input these drawings into my finished assembly of the boat and cradle, with the boat raised to the highest position possible. I was then able to see whether the screen was appropriate for the person sitting in the boat.

[39] http://www.da-lite.com/education/guide.php?page=3

Screen 1 - Picture King 96" x 96" £281.94



This screen is tripod based, and when raised to the highest setting off the floor, the viewer is above the top of the screen. The screen is just under specified width at 2.41m, and is simply not large enough.

Figure 14.5

Screen 2 – Model C 12' x 12' £619.00

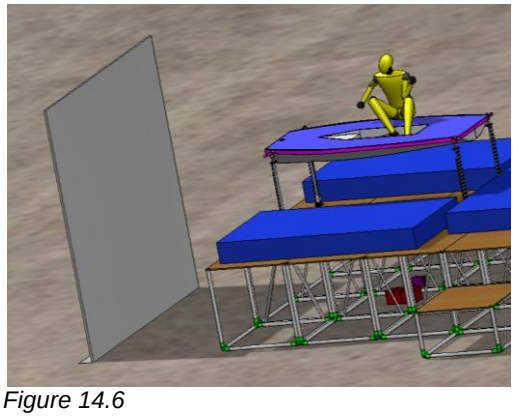


Figure 14.6

This screen rolls up from the base, and so there is no support from underneath. The bottom of the screen would be difficult to see from the top position of the boat. It is at a good height, but could be higher for better viewing. It is also expensive.

Screen 3 - Model C 10' x 10' £515.00

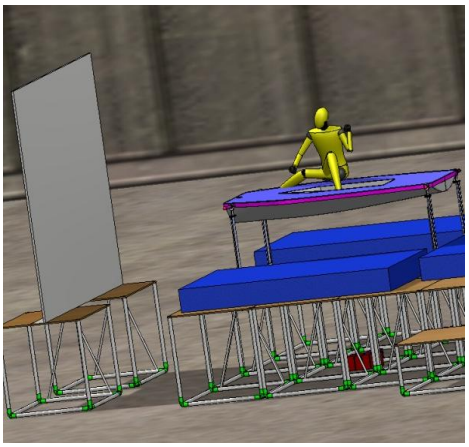


Figure 14.7

This screen is smaller, so cheaper than previous screen. The screen can be raised by placing on a table/staging. The screen is larger than desired, so will give an even wider view of surroundings. It can also be optionally wall mounted.

Figure 14.8 – chosen screen

Result

The chosen screen is the Model C 10'x10', with White Matte screen material[40].

14.3. Sound

We are aiming to produce a real life experience with the simulator, and so want to create a surround sound system that can create directional wind noises. The wind direction will be pre programmed at the start of the simulation, and so then will be calculated relative to the direction of the boat at short time intervals. This will be found as an angle, and each speaker will be assigned to a range of angles. The speed of the wind is programmed at the start, which will be calibrated to producing a certain volume level from the speakers.

Wind Direction

The angle calculated relative to the boat will depend on the original wind angle. In the example given below, the angle range relative to the boat is calculated for an input direction of a northerly wind. These angles are recalculated and the speakers are assigned a different angle range dependant on the initial wind direction input.

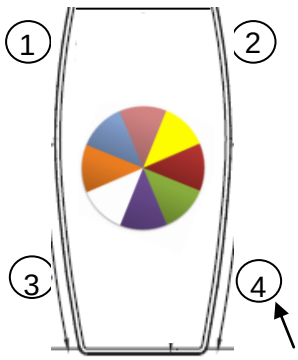


Figure 14.9

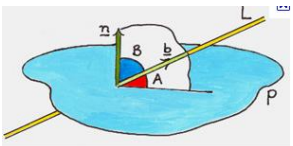
Wind Direction	Base wind angle	Angle range relative to boat	Speakers
N	0°	338°- 22°	1 and 2
NE	45°	23°- 67°	2
E	90°	68°-112°	2 and 4
SE	135°	113° - 157°	4
S	180°	158° - 202	3 and 4
SW	225°	203° - 247°	3
W	270°	248° - 292°	1 and 3
NW	315°	293° - 337°	1

Table 14.2

[40] http://www.dreamav.co.uk/dalite_model_c_screen.html

Matlab Code

Coding will be written so that the angle of the boat relative to the wind plane can be calculated.



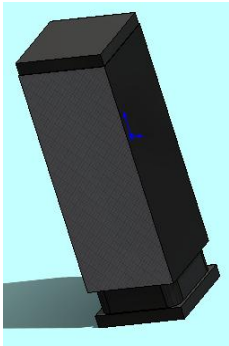
The wind plane will be assigned according to original wind direction.

The boat can be assigned a vector, generated by two points on the hull, at the bow and stern. This vector can be extended until it reaches the plane.

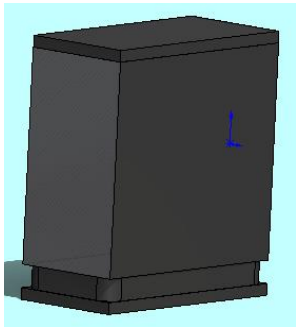
If b = boat vector, n = normal to the plane, can perform dot product between normal and boat vector: $cosB = \frac{n \cdot b}{|n||b|}$. Then cosB =90 – A = sinA. Therefore A = $\sin^{-1}\left[\frac{n \cdot b}{|n||b|}\right]$.

This can then be input as a function into matlab or C++ to generate the wind angle at 0.5 second time intervals. The same code will work on screen to show the wind generated visually, as in Code 11.3.

Speaker Choice



The specification for the speakers is limited, as the noise level of the actuators are minimal as they are electrically driven. They should be wireless for portability, but there is no minimum level of sound that is needed. However a system with over 100W/140dB of sound power is desirable for adequate range of noise level. Preferably the cheapest wireless speakers that are quite durable, with an optional sub woofer. The device should produce HD and also ideally 3D surround sound.



The cheapest speaker over 140dB, and with wireless connection is the Sony HT-AS5[41], with four front speakers and one subwoofer.

[41] http://www.sony.co.uk/product/hcs-home-cinema-speaker/ht-as5/tab/technicalspecs

15. Graphics and Sound Rendering

15.1. Specification

The simulator is required to display photorealistic graphics at a reasonable frame rate (no less than 30 frames per second). Sounds should also be rendered in sync with the simulation for auditory realism.

15.2. User Interface

An important aspect of the graphics is the user interface – how information is displayed to the user. There are two seperate components of the user interface, the menu system and the “in game” heads up display.

The menu interface will be designed to be very simple to navigate. It will be navigated by a mouse and provide a simple interface for setting up the configuration for the simulation. The settings and advanced configurations will be able to be accessed by advanced users, but there will be simple pre-set configuration options for beginners to begin a simulation quickly with minimal setup time. For example, a user will be able to select a location, e.g. “English Channel”, with weather settings as “Sunny and calm”. This would load a pre-set advanced configuration of a specific lattitude and longitude for the starting position, a wind speed of 10 knots with direction 20° and calm wave settings. An advanced user would be able to edit these settings if desired. In this way, a beginner can very quickly enter a simulation and start sailing, while an advanced user can set specific properties for a specific simulation.

The “in game” heads up display must show the extra information displayed on top of the 3D rendered world. The simulation is focussed on realism, and in reality you have very little information available when sailing a Laser. For this reason, the heads up display will be minimal, only showing a compass for direction, as a sailor may well have a compass on their person. Another crucial piece of information which needs to be conveyed to the user of the simulator is the wind direction and wind speed. This will be done by subtle arrows pointing in the direction of wind flow, with their opacity and length denoting the speed of the wind.

Extra display information, such as boat speed and a map will be incorporated as add on modules to the interface, if the user selects them.

15.3. Three Dimensional Graphics

The simulation will be displayed with three dimensional graphics[42], which will be rendered in real time. The graphics must be photorealistic and dynamicly dependent on the weather conditions. There are four main aspects of the 3D graphics – the boat rendering, water rendering, sky rendering, and background environment.

The boat model will be based on the CAD model produced for the design. This will be animated and ported in to a format which can be used within the game engine, using Blender[43], a powerful open source 3D modelling suite. The model will be texture mapped with high quality images to portray the materials. If supported by the graphics engine, bump or parrallax mapping will be also used to futher enhance the realism of the boat materials. There will also be the option of incorporating a custom image onto the textures to allow the boat model to have sponsoring on it.

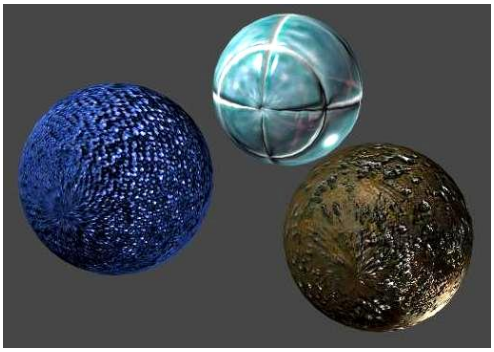


Figure 15.1 – Three identical sphere models using different texture and bump maps to drastically change the material look.

The water shall be rendered as a mesh surface, with a high quality repeating texture map, and also a bump map if supported. There will be a number of different texture and bump maps, depending on the environment and wind strength. For each environment there will be three different texture and bump maps – ligth wind, medium wind and heavy wind. This allows the high frequency detail and colour of the water surface to change depending on the intensity of the wind. There will also be reflection off the water surface. Due to the complexity of the rendering of the water surface, it is highly unfeasible to render the entire game world's water surface at such high quality. To overcome this limitation while keeping visual fidelity, we will implement a draw distance. The draw distance is a distance from the camera, beyond which the number of polygons rendered dramatically decreases. There will be a high quality water mesh and texture for within the draw

[42] 3D graphics denotes a 2D representation of a 3D object on a 2D display.
[43] See http://www.blender.org/ for more information.

distance, beyond which a low quality mesh and texture is used, then at a further distance still, no mesh and just a colour are rendered. This will dramatically incease the performance. The draw distances will have to be set once the performance of the system is known.

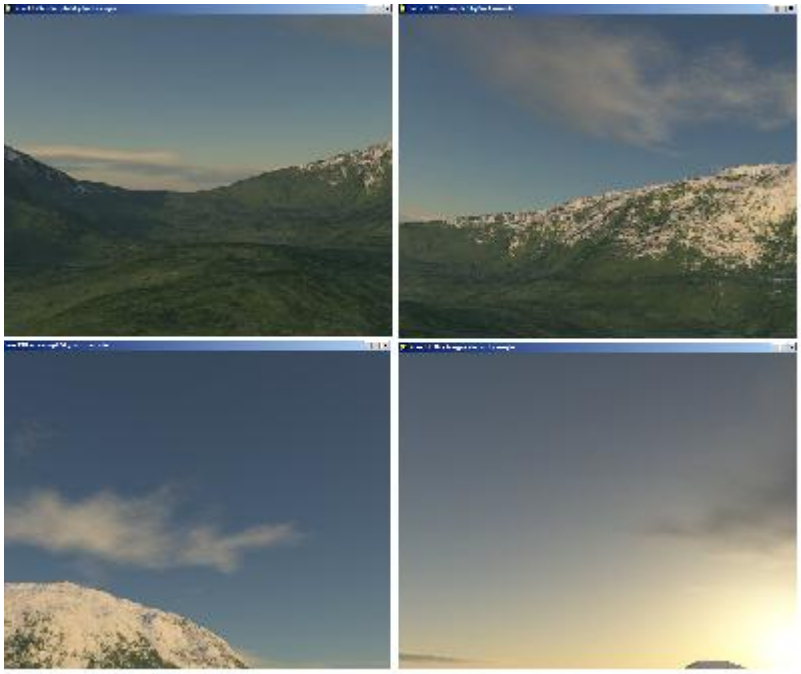


Figure 15.2 – Prototype implementation of a sky box rendered using DirectX 9, implemented in C++.

The sky will be rendered using a sky box. A sky box is a cube which surrounds the camera in the 3D world. Each face inside the cube has a different image mapped on to it. The image is a cube map, which maps a panoramic view on to the faces of a cube. From the perspective of the camera inside the cube, the result is a 3D, seamless world. The cube moves with the camera, giving the illusion that the scene on the sky box is infinitely far away. This creates an immersive background environment for the simulation. There will be a number of different cube maps for the sky box, one for each weather type for each location. If there are no clouds in the chosen weather system, the sun will be represented by a point light source at the edge of the sky box, which can move depending on the time of day, and the lighting of the scene will be a combination of the point light source and global illumination[44]. However, if a cloudy weather system is selected for the simulation, there will be no point light source, instead a different global illumination configuration to simulate a cloudy day.

The sky box technique can also be used for the background environment, an example is shown in Figure 15.2, and a sample of the code shown in Code 15.1.

[44] http://www.thepolygoners.com/tutorials/GIIntro/GIIntro.htm provides a good introduction to the global illumination concept.

```
        // ---------
        // Main Loop
        // ---------
        while( uMsg.message != WM_QUIT )
        {
              if( PeekMessage( &uMsg, NULL, 0, 0, PM_REMOVE ) )
              {
                    TranslateMessage( &uMsg );
                    DispatchMessage( &uMsg );
              }
          else
              {
                    // Timer
                    g_dCurTime      = timeGetTime();
                    fElpasedTime = (float)((g_dCurTime - g_dLastTime) * 0.001);
                    g_dLastTime     = g_dCurTime;

                    // Process the keyboard presses
                    ProcessInput( fElpasedTime );

                    // Render the current frame
                  RenderFrame();
              }
        }
```

Code 15.1 – The main program loop for the graphics rendering using DirectX9, creating the scene shown in Figure 15.2.

The result of using these 3D graphics rendering techniques will be a photorealistic, emmersive environment which will allow the user to believe they are sailing. Pulling together all the parts of the graphics will produce a product similar to Figure 15.3. The exact rendering engine and implementation of these techniques is explored in the next section.

Figure 15.3 – Prototype graphical representation of the simulation.

15.4. Graphics Engine

Three dimensional graphics are required to be rendered, and then outputted to the projector. To do this, the graphics will be rendered using a graphics card – a special type of I/O card in the computer with processors and memory designed for efficient processing of display data. The graphics card our system will use is a NVIDIA GeForce GTX 550 Ti, which features a 1.8GHz processor and 1GB of memory. This graphics card also has a DVI and HDMI output, either of which can be used to connect the output to the projector.

To take advantage of the graphics card, an API (application programming interface) should be used to interface with the display adapter. Using an API reduces the complexity of utilising the graphics card, and allows the same code to work with different graphics cards and on different systems. This is important for the robustness of the simulator, allowing the computer hardware to change due to new developments of client needs. The two feasible APIs which could be used are OpenGL and Microsoft DirectX.

OpenGL (Open Graphics Library) is a cross platform API which produes 3D grpahics. It is released under the GPL license so requires no licensing for our application. DirectX is a Windows only platform, requiring no licensing fees, which provides an interface to not only the display adapter, but also the sound adapter and input devices. DirectX would therefore be preferable to use as it would provide the interface for all inputs and outputs. However, either option would require a significant programming effort to create a 3D world with the level of realism required by the simulator. An alternative is to use a ready made rendering or game engine.

Figure 15.4 – A high resolution ocean scene rendered with OGRE at over 200 frames per second by using a low resolution mesh.

We will use OGRE (Object-Oriented Graphics Rendering Engine) utilising DirectX 9 for this. While there are many game engines available, OGRE is most desirable for a number of reasons. Firstly, OGRE is only a rendering engine, which allows the physics, input, and output to be handled by the proprietry code developed by us. Its object oriented design and native C++ interface allows for perfect incorporation with the simulator program, and it supports a number of 3D rendering effects, including bump mapping, for rich, photorealistic graphics. Finally, OGRE is released under the MIT license which means there is no cost for using it, in comparison to other game engines such as the Unreal 3 engine which can cost up to $700,000 for a commercial license.

OGRE will therefore be used for the graphics rendering, implementing the techniques outlined in the previous section. OGRE will use DirectX 9 for the graphics card interface, so the sound of the simulation will also be played using DirectX 9, which supports 3D sound. DirectX 9 handles the sound processing on the sound card of the computer, which is outputted to the speakers. The expanded simulator schematic is shown in Figure 15.5.

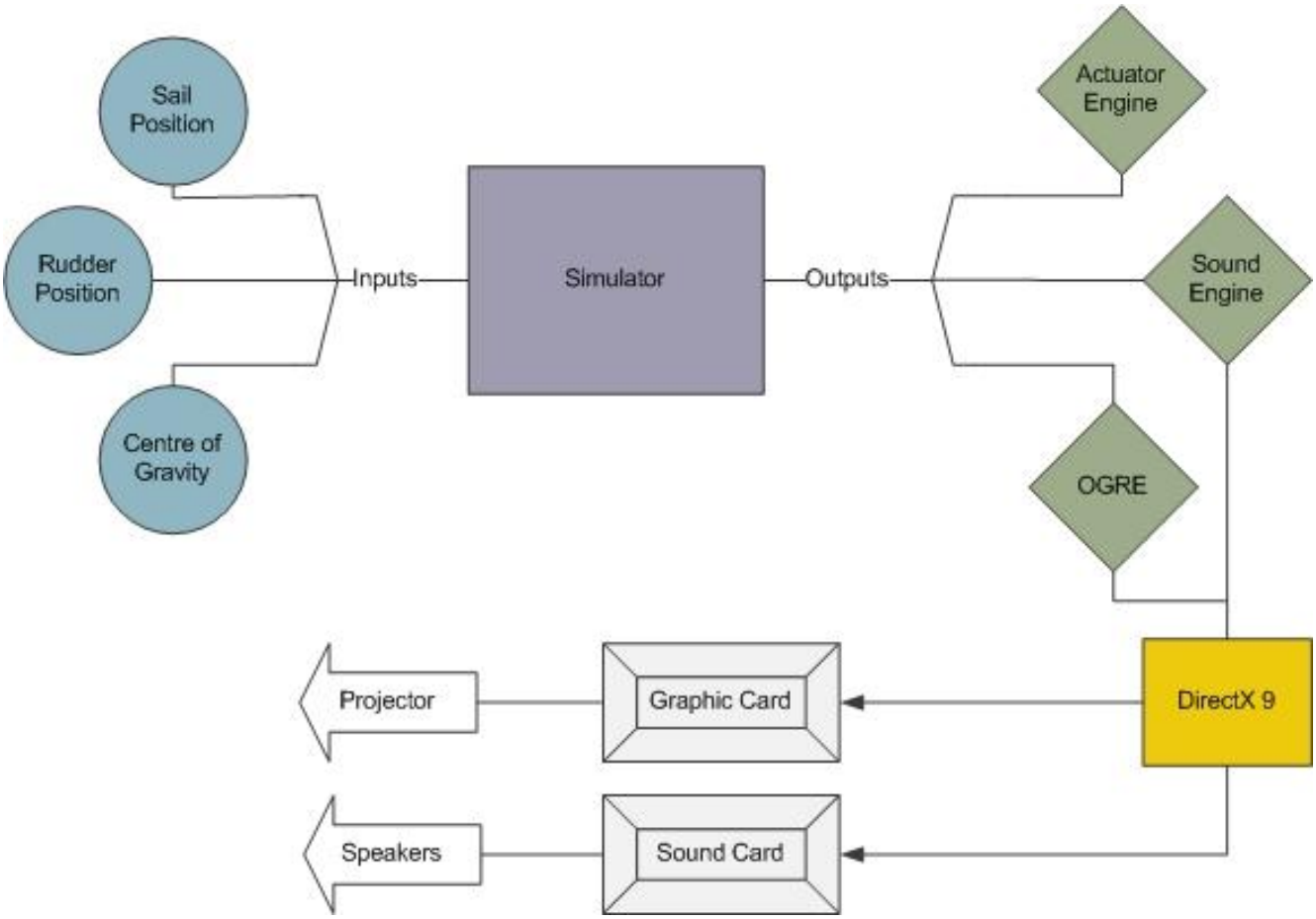


Figure 15.5 – Simulator program showing graphics and sound rendering engines.

16. **Electronics**

Main Hull Controllers

Each of the main hull actuators (LinMot P01-48x360F) is controlled through a LinMot E1250-IP-UC Servo Controller. The controller takes position and load information from the actuator and outputs required position to the actuator using the control lines provided. The controller receives and feeds back the relevant information to the computer using an Ethernet connection. Each actuator has its own controller and so the four Ethernet lines are connected to the computer via a four way router.

Each of the controllers is powered from a mains supply via its own LinMot S01-72/500 Transformer power supply. These supplies were chosen for their ability to provide the 32Amp peak at 72VDC demanded by the actuators.

Sail and Rudder Controllers

The sail actuator (MaxPlus 1902BSG Servo motor) is controlled by a MaxPlus 230 series MPA-35 Servo amplifier (required as the motor demands a peak current of 61.0 Amps), which is wired straight into the mains supply.

The rudder actuator (MaxPlus 1141 BSG Servo motor) is controlled by a MaxPlus 230 series MPA-06 Servo amplifier (required as the motor demands a peak current of 10.8 Amps), which is again wired straight into the mains supply.

Both controllers/amplifiers are interfaced with the computer via a PIC18F4550 microchip using a USB connection. The micro chip outputs 0 to ±5V but the controllers take command inputs of 0 to ±10V so two small transformers are required between the microchip and the controllers. The output voltage from the controllers, providing velocity and torque monitoring, can be handled directly by the microchip and sent to the computer via the USB connection.

Computer, Projector and Sound System

The computer, projector and sound system all plug straight into the mains supply at 230V, 50Hz

Power Requirements

It is vital that the total power requirement for the simulator is known so that venues for the simulator can ensure that their mains supply can take the demand.

Equipment	Number Required	Peak Power Consumption
LinMot S01-72/500 power supply	4	750W
MaxPlus 230 series MPA-35 Servo amplifier	1	5000W
MaxPlus 230 series MPA-06 Servo amplifier	1	880W
Sony HT-AS5 5.1ch Satellite Speaker System	1	110W
PC (specified in the 'system integration')	1	600W
Benq MP776 Projector	1	425W

It can be seen that this gives a total power requirement of 10.015KW that needs to be drawn from the mains supply at any location, requiring at least four contact points from which to draw power. An alternative option would be to generate the power required using a portable petrol generator. If this option was specified by the consumer then a Loncin LC12000DDC Silent Petrol Generator could be supplied with the simulator to fulfil its power requirements allowing the simulator to be completely portable. This would obviously be at an extra cost to the consumer.

17. Control

17.1. Specification

The actuators must be controlled so that the user experiences a range of motions in the simulator similar to that in the computer simulation.

17.2. Cradle Control

The user sits in a Laser hull, which is mounted on a cradle. The cradle is supported on each corner by an actuator; four actuators in total (see Figure 17.1). Four actuators were chosen over three to meet the force requirements. This arrangement allows three degrees of freedom - roll, pitch and heave - with four variables, the four actuator lengths. With four variables and three degrees of freedom it is implied there is one unconstrained variable.

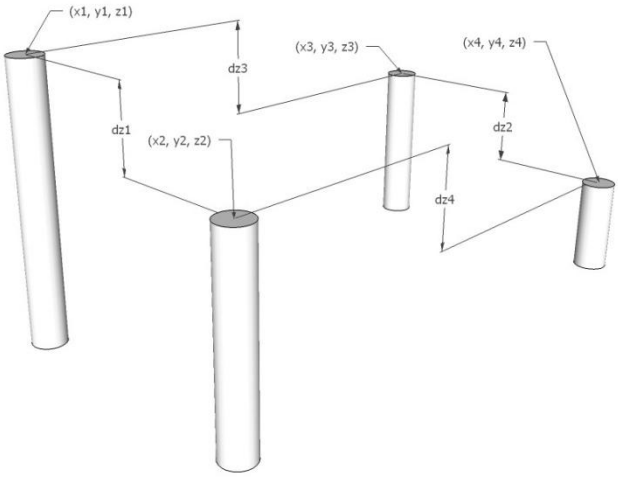


Figure 17.1 – Layout of the cradle actuators. Constraint requires dx1=dx2 and dx3=dx4.

However, the rigidity of the cradle (and the desire not to warp the cradle) imposes a further constraint which eliminates the unconstrained variable. This constraint is that the lengths of the actuators must be such that the ends of the actuators all lie on the same plane. A plane is defined by $\boldsymbol{n} \cdot (\boldsymbol{r} - \boldsymbol{p}_1) = \mathbf{0}$, where $\boldsymbol{n} = (\boldsymbol{p}_2 - \boldsymbol{p}_1) \times (\boldsymbol{p}_3 - \boldsymbol{p}_1)$ and $\boldsymbol{p}_n = x_n \boldsymbol{i} + \mathrm{y_n}\mathbf{j} + \mathrm{z_n}\mathbf{k}$, see Figure 17.1. Therefore, three actuator positions can be chosen and the fourth must be chosen to lie on the plane.

The goal of the physical cradle simulation is to provide the most convincing perception of the computer simulation. The linear actuators are of finite length and only have a finite range of motion, which provides a constraint on the domain of the three dimensions which are simulated. The problem is therefore a contstrained optimisation problem: minimise the difference between the cradle hull orientation and the computer simulated hull orientation within the physical bounds of the actuators, given the actuator ends lie on a common plane. A final condition is that the roll and pitch

of the boat should be priorotised over the heave. This is because humans are sensitive to roll and pitch positions, veloctiy and acceleration, but in the vertical direction only sensitive to the velocity and acceleration.

This optimisation problem can be solved in Matlab. The `fminsearch` function can be employed to solve this, which uses the simplex algorithm to find a minimum to a cost function. The parameters to be optimised are the three vector parameters describing the position of the cradle plane $\boldsymbol{p} = (z\ \theta\ \emptyset)^T$ where z is the vertical position of the centre of the plane, and θ and $\emptyset$ are the roll and pitch of the cradle plane. The cost function is threrefore the sum of the square difference, $c(\boldsymbol{x}, \boldsymbol{p}) = \sum_1^3 (x_i - p_i)^2$, where $\boldsymbol{x}$ is the vector of height, roll and pitch for the computer simulated boat. However, the contstraints of the finite actuator extension must be introduced. This is effectively a barrier function which increases the cost to infinity if the actuator length is above or below the bounds of the actuators, i.e. $cost = c(\boldsymbol{x}, \boldsymbol{p}) + \sum_1^4 I(z_i - h_{max}) + \sum_1^4 I(h_{min} - z_i)$, where $I(u) = \begin{cases} 0, & u \leq 0 \\ \infty, & u > 0 \end{cases}$. This function is problematic as it is not differentiable and so would not be suitable for an optimisation algorithm. To overcome this, a logorithmic barrier function is used approximate the barrier function, $I(u) \approx -\frac{1}{t}\log(-u)$, Figure 17.2.

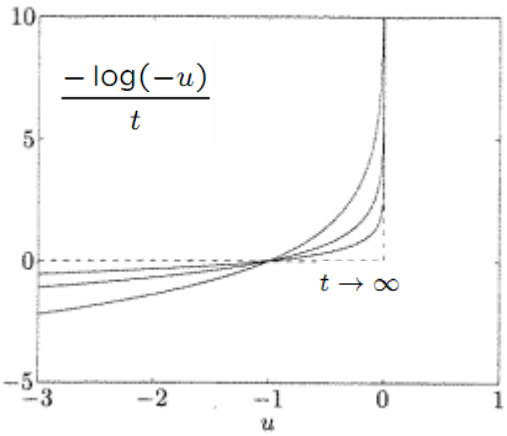


Figure 17.2 – The logarithmic barrier function with increasing values of t

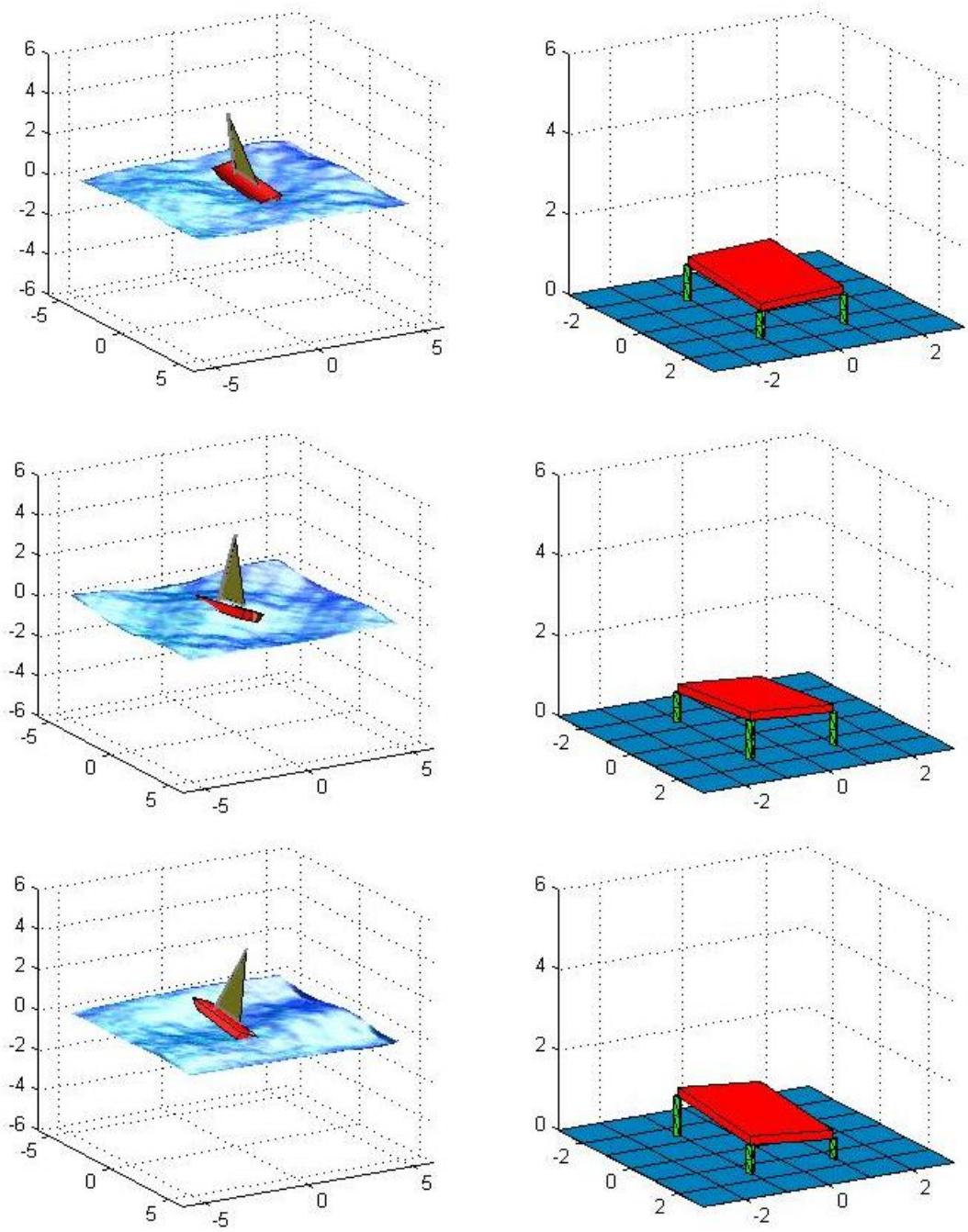


Figure 17.3 – Prototype implementation of mapping simulated boat to actuator positions. The figures on the right show the optimal position of the cradle based on the simulation (left). Boat positions artificially set for this demonstration.

This entire process was implemented in Matlab and produces the real time desired positions of the actuators as the simulation is running. These desired positions can then be outputted to the control system of the acuators to affect the cradle orientation.

17.3. Actuator Interface

We will be using a control unit to control the linear actuators (LinMot P01-48x360F). The control is implemented automatically given a desired height for each actuator. The interface we will use is

Ethernet as this is easy to communicate with using C++. The C++ simulator program will send the control unit the reference height for each actuator.

The control unit is the LinMot E1250-IP-UC Servo Controller. The actuator will be controlled using the VA interpolater mode (see Figure 17.4) which must be given a reference position as well a maximum velocity and acceleration. The supplied control reference vector is $\boldsymbol{r}(k) = (x_{Goto}(k), v_{max}(k), a_{max}(k))^{\boldsymbol{T}}$ at timestep k. $x_{Goto}(k)$ is the desired extension of the actuator, set by the optimised position of the actuator (see previous section). $v_{max}(k) = x_{Goto}(k) - x_{Goto}(k-1)$ and $a_{max}(k) = v_{max}(k) - v_{max}(k-1)$, therefore $\boldsymbol{r}(k) = \begin{pmatrix} 1 & 0 & 0 \\ 1 & -1 & 0 \\ 1 & -2 & 1 \end{pmatrix} \begin{pmatrix} x_{Goto}(k) \\ x_{Goto}(k-1) \\ x_{Goto}(k-2) \end{pmatrix}$.

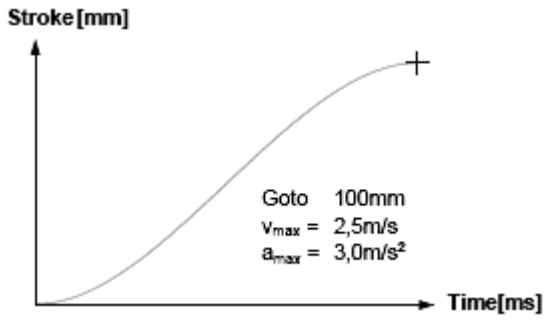


Figure 17.4 – Position profile for a Goto command to an actuator using the VA interpolater.

17.4. Sail and Rudder Control

The sail and rudder must be simulated physically. Intuitively, it is the torque on the sail and rudder which provides the most feedback to the user, as a sailor must apply a constant force on both these components to maintain a certain heading and speed. The servo amplifier controlling the motors will therefore be operated in torque mode. The simulation models the torque on the sail and rudder around their main axis. These torques will be outputted to their respective servo amplifiers and set the reference torque for the motor.

For safety the physically simulated torques will be limited.

18. Systems Integration

The simulator is comprised of various systems which must be interfaced together. Central to this will be the computer, which needs to connect with the projector, sound system, actuator control system and sensors.

18.1. Computer

The computer will be a PC running Windows 7 to provide the operating system for the simulator program. The hardware of the computer will be as follows:

- Intel Core i7 Quad Core Processor (3.20 GHz)
- ASUS P6X58D-E Motherboard
- 6GB Samsung DDR3 RAM (1333MHz)
- 1GB Nvidia GeForce GTS 450 Graphics Card
- Asus Xonar DG 5.1-Channel PCI Sound Card
- 500GB Hard Drive
- 600W Power Supply
- Windows 7
- USB mouse and keyboard

This specification will provide the necessary computing power to run the simulator program, as well as the I/O ports required to connect to the various other systems.

18.2. Display

The display system is the projector (BenQ MP776 ST). This will be connected to the graphics card via a DVI cable.

18.3. Sound

The speaker system is the Sony HT-AS5, which is a 5.1-Channel digital surround sound system. It will be connected to the computer via an HDMI cable which is connected to the PC's sound card HDMI output.

18.4. Linear Actuators

The linear actuators are each connected to an actuator servo amplifier (LinMot E1250-IP-UC Servo Controller). The controller can be assigned an IP address and interfaced with over Ethernet. Each control unit will then be connected to a router (Netgear FS108) which will also be connected via Ethernet cable to the PC. Ethernet was chosen over other connection schemes (such as serial RS232) as the router allows modular connectivity, easy referencing of each actuator via an IP, and the Ethernet interface is well defined for the controller. This will allow the PC to communicate with each actuator controller and send control signals as well as receive sensor data from the actuators to feed into the simulation.

18.5. Sail and Rudder Actuators

The motors providing actuation to the sail and rudder are the Parker Hannifin 1902 BSG and1141 BSG MaxPlus motors. The sail actuator will be connected to a MaxPlus 230 series MPA-35 Servo amplifier and the rudder actuator to a MaxPlus 230 series MPA-06 Servo amplifier. The servo amplifiers have differential 0 to ±10V command inputs and 2.8V per 1000rpm velocity output. The I/O for each amplifier will be connected to the PC via a USB interface. The USB interface will be handled by a PIC18F4550 microchip. This chip can implement the Human Interface Device (HID) protocol for its USB interface, meaning no custom drivers are needed on the PC to communicate with it. Also the HID interface is easy to manipulate in C++ allowing easy reading and writing of data to the microchip.

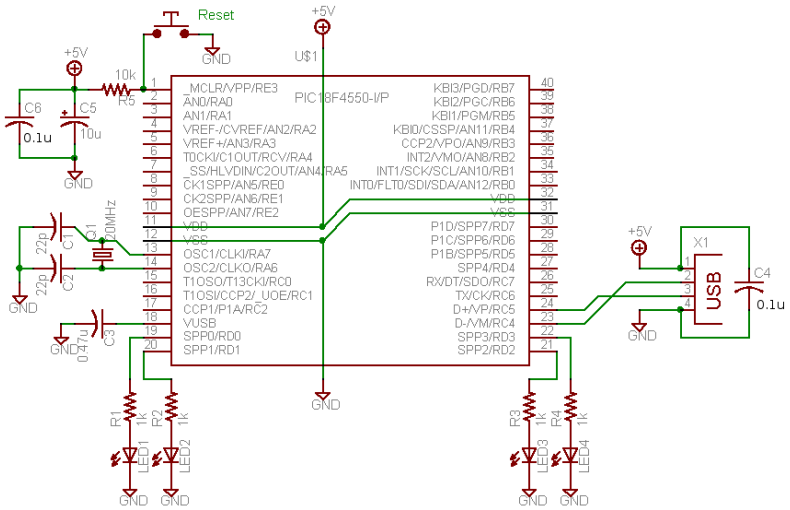


Figure 18.1 – Circuit diagram for PIC18F4550 USB interface.

Figure 18.1 shows the circuit diagram for the USB connection to the PIC. There are many free I/O pins which can then be used to connect to the two MaxPlus units, via amplifiers if required. Communication will occur across USB in a simple manner – firstly indicating which actuator is being referred to, followed by whether to set or read the actuator, then if required, a parameter, and finally a character indicating the end of the message. Code 18.1 shows an example communication between the PC and the PIC in ASCII encoding. Message 1 indicates the sail (S) speed (s) should be set to 200, ending with a stop byte (N) to indicate the message has finished. The PIC responds with an acknowledgement (line 2). Line 3 is a speed setting for the rudder motor. Line 5 is a request for the sail speed, to which the PIC responds with the message on line 6, indicating the sail speed is 198rpm.

```
1.PC->PIC: Ss200N
2.PIC->PC: aN
3.PC->PIC: Rs50N
4.PIC->PC: aN
5.PC->PIC: SrN
6.PIC->PC: Ss198N
7.PC->PIC: aN
```

Code 18.1 – Example communication between the PC and PIC.

The PIC will be programmed in C using the mikroC package[45], which includes a number of built in software libraries for the PIC18F4550, including USB communication protocols, pulse width modulation for digital to analogue conversion, and analogue to digital libraries.

In this way, the simulator program running on the PC and the PIC can communicate, reference signals can be sent to the actuators and sensor readings sent to the simulator program.

[45] See http://www.mikroe.com/eng/products/view/7/mikroc-pro-for-pic/.

18.6. System Diagram

Figure 18.2 shows the complete connected system.

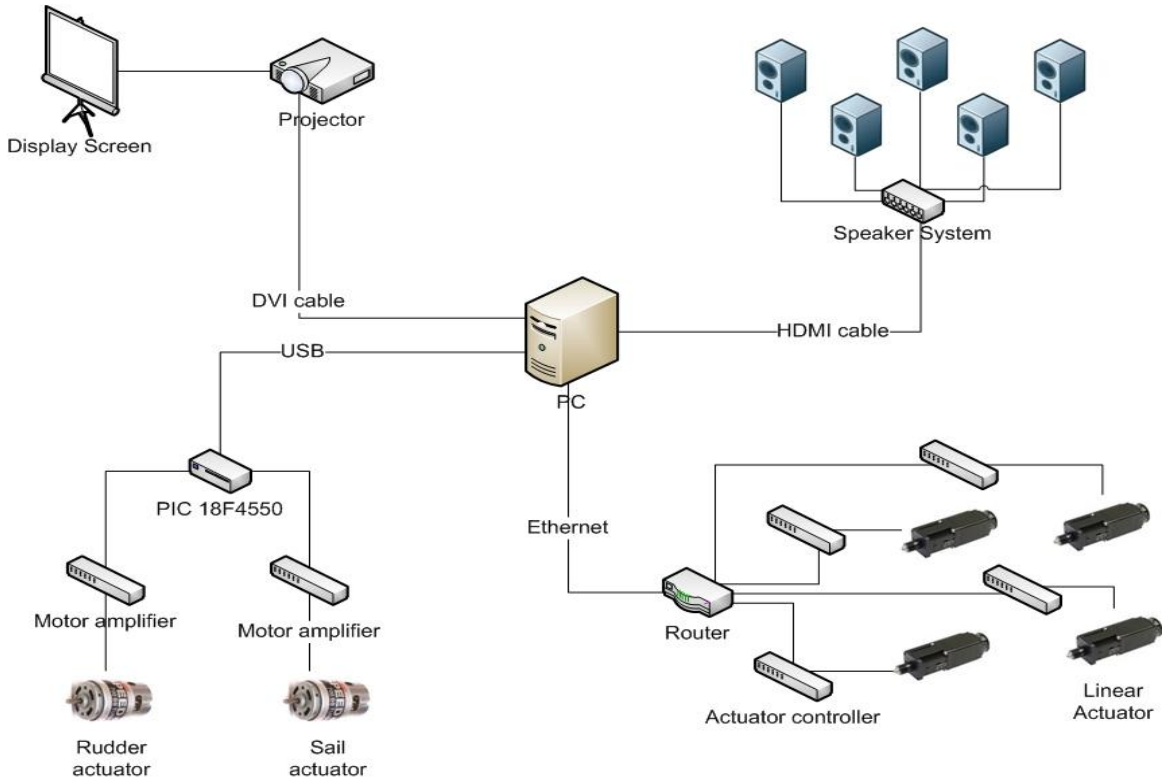


Figure 18.2 – System layout.

19. Pricing

We aim to produce simulators at a selling price higher than £16500 (current sailing simulator price), and with much improved capabilities. From market research it has shown that sailing schools are willing to pay an average of £18,500 per simulator (see logbook pg19), so the minimum sell price will be slightly under this value.

To be able to encapsulate both markets of novice sailors and professional athletes, the simulator can be tailor made to the customer needs. Sailing schools will not be willing to pay as much money for the simulator as Olympic sailing teams with much larger budgets. A sliding cost scale will be used to create packages, such that buyers can design a tailor made simulator package.

The selling price is aimed to be 20% higher than the cost price, such that the design team is able to have a wage, whilst giving investors returns on company investment.

19.1. Price list of all components (starred components are optional)

Component	Manufacturer	Part Number	Quantity	Cost per item	Delivery Cost	Total Cost	
Laser Hull	Laser		**1**	£259.60	N/A	£259.60	
Carbon Fibre Sheet	King Carbon	AUXCFS1 500250	**1**	£63.25	£2.50	£65.75	
Linear actuators	LinMot	0150-1388	**4**	£1952.97	N/A	£7811.88	
Controller	LinMot		**4**	£1110.00	N/A	£4440.00	
Flange	LinMot	0150-1975	**4**	£50.05	N/A	£200.02	
Magnetic Spring	LinMot	0250-2313	**4**	£70.97	N/A	£283.88	
Magnetic Spring Flange	LinMot	0250-2307	**4**	£10.93	N/A	£43.72	
Power supply	LinMot		**4**	£331.47	N/A	£1325.88	
Bolts M10	Screwfix	13827	**48**	£0.13	£0.20	£6.90	
Rodend	RBC bearings	MSSE 20H 01-827	**4**	£53.99	£2.00	£55.99	
Steel for bracket	Oxford Uni	-	**2**	£2.82	N/A	£5.65	
Cylindrical Rods	Metals4u	-	**700mm**	£14.33/m	N/A	£10.03	
Washer for plywood	Shop4fastners	WRZ-M1050-B	**32**	£0.01	N/A	£3.10	
Titanium angled washer	Corse meccanica	Custom	**4**	£20.39	N/A	£81.56	

Actuator bracket	Oxford Uni	-	**8**	£46.29/m		£51.85	
H11 Steel M20 bolt	Vegas Fasteners	Custom	**4**	£23.88	N/A	£95.52	
Stage	Guangzhou	Stage-108	**12**	£24.89	£20	£318.68	
Linear bearing	Speed Demon	-	**4**	£35.99	N/A	£143.96	
Bulbous Ferrules	Ross Handling	154-01	**4**	£0.39	£1.00	£2.56	
Crash mats	Mortex	10512	**2**	£29.47	N/A	£58.93	
Screen	Model C	40274	**1**	£515.00	£5	£520.00	*
Floor Stand	Model C	40959		£324.59	N/A	£324.59	
Screen stage	Guangzhou	Stage-108	**2**	£54.89	N/A	£109.78	*
Surround sound	Sony	HT-AS5	**1**	£332.00	N/A	£332.00	*
3D Projector	BenQ	MP776	**1**	£740.00	N/A	£740.00	
Computer with dedicated nvidia graphics card	Any	-	**1**	£597.00	N/A	£697.00	*
Boom motor	Parker Hannifin	1902BSG	**1**	£1447	N/A	£1447.00	
Rudder motor	Parker Hannifin	1141BSG	**1**	£576	N/A	£576.00	
Total (all goods including optional)						**£20011.83**	

Table 19.1

If we assume bulk ordering, then economies of scale will reduce these prices significantly. Assume that this will cause a reduction of 20% of price, so that total price for non-optional goods

= £17578.46*0.8 = £14063

19.2. Price of all manufacturing/service jobs

Job	**Job provided by**	**Cost per hour**	**Hours**	**Price**	
Bending steel to make cradle	Oxford University	£22	5	£110	
Cutting tubing to size	Oxford University	£22	0.2	£4.40	
Cutting aluminium brackets to size	Oxford University	£22	0.2	£4.40	
Cutting square hole in wooden staging for actuators to sit (x4)	Oxford University	£22	0.5	£11.00	
Drilling holes (x80)	Oxford University	£22	0.5	£11.00	
Assembly of fixed product	Oxford University	£22	25	£440.00	
Transport cost		£0.75/mile each way			
Assembly of transportable product		£150		£150.00	
Coaching		£150/day			*
Subsistence		£100/day			*
Total				**£840.00**	

Table 19.2

19.3. Packages (to buy)

Package number		1	2	3	4
	Screen and sound system		✓		✓
	Computer			✓	✓
Build Cost		£14,903	£15,750	£15,500	£16,250
Sell price		£18,000	£19,0 00	£20,000	£21,000

Table 19.3

19.4. Packages (to rent)

Package number		1	2	3	4	5	6	7
	Screen and sound system					✓	✓	✓
	Computer			✓	✓		✓	✓
	Coaching		✓		✓	✓		✓
	Transport Cost	✓	✓	✓	✓	✓	✓	✓
	Set Up/Breakdown	✓	✓	✓	✓	✓	✓	✓
	Subsistence		✓		✓	✓		✓
Rental Cost/day	(+ transport)	£83.16	£333.16	£83.50	£333.50	£333.63	£83.90	£333.90
Rental Price/day		£200	£550	£250	£600	£600	£350	£750

Table 19.4

N.B rental cost/day is based on a simulator that has average life of 5 years, from high frequency cyclic load calculations.

This gives overall profit of approximately 21% when buying simulator, and average profit 51% when renting.

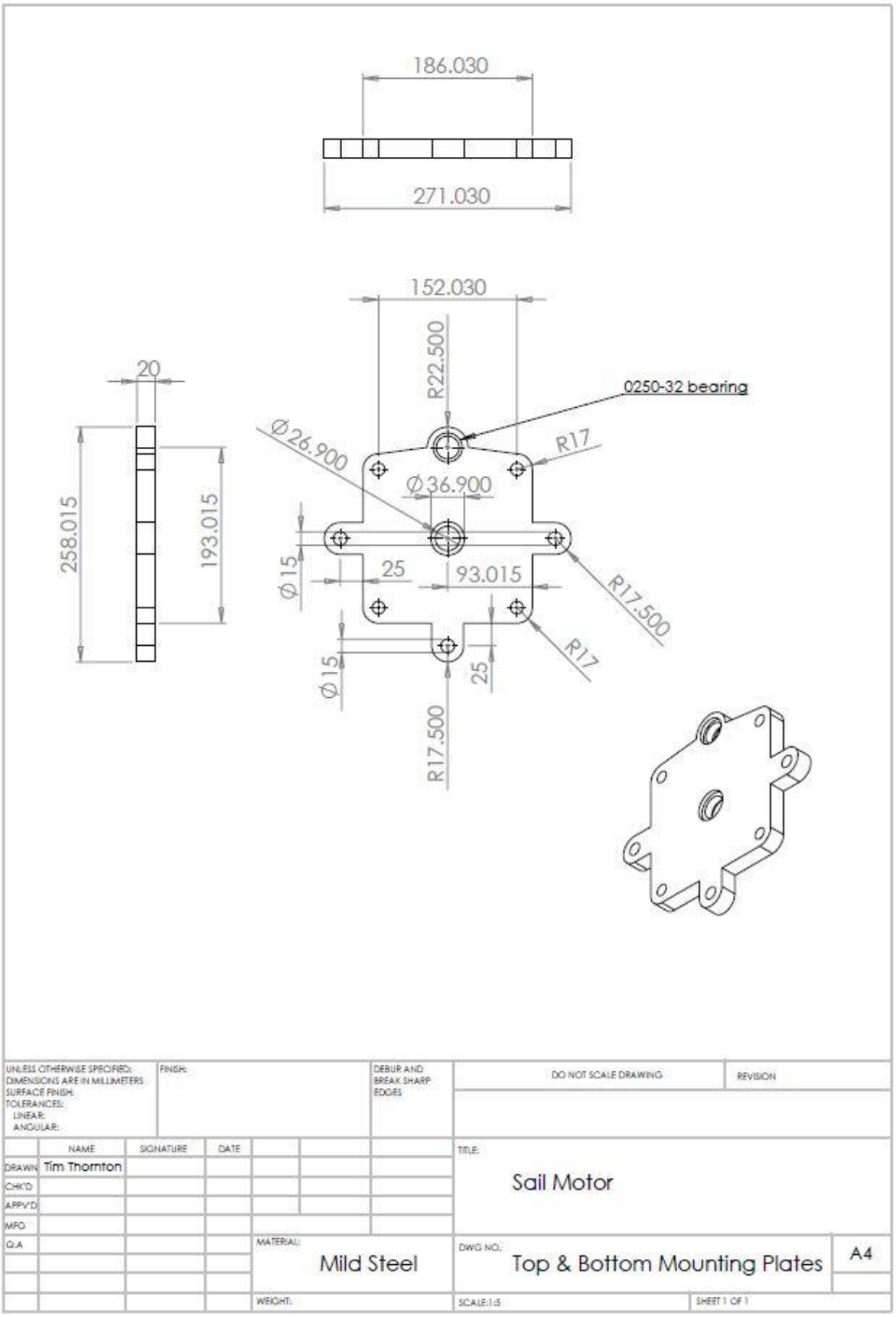
20. Summary

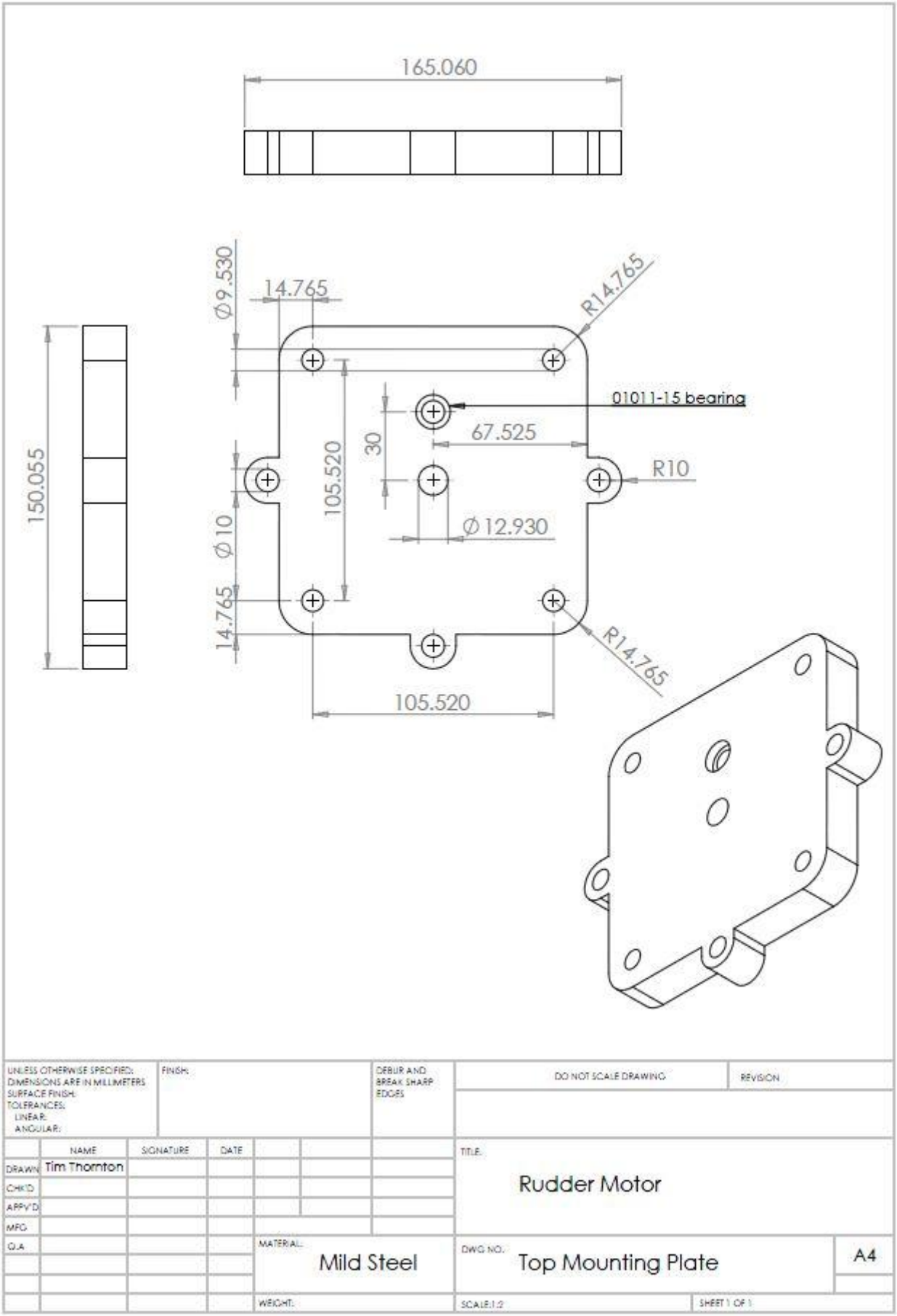
It can be seen that the new design produces a much more lifelike simulation than any of the previous attempts. This is primarily due to advanced actuation giving three degrees of freedom rather than the VS-1's one, alongside the inclusion of a fully motorised boom.

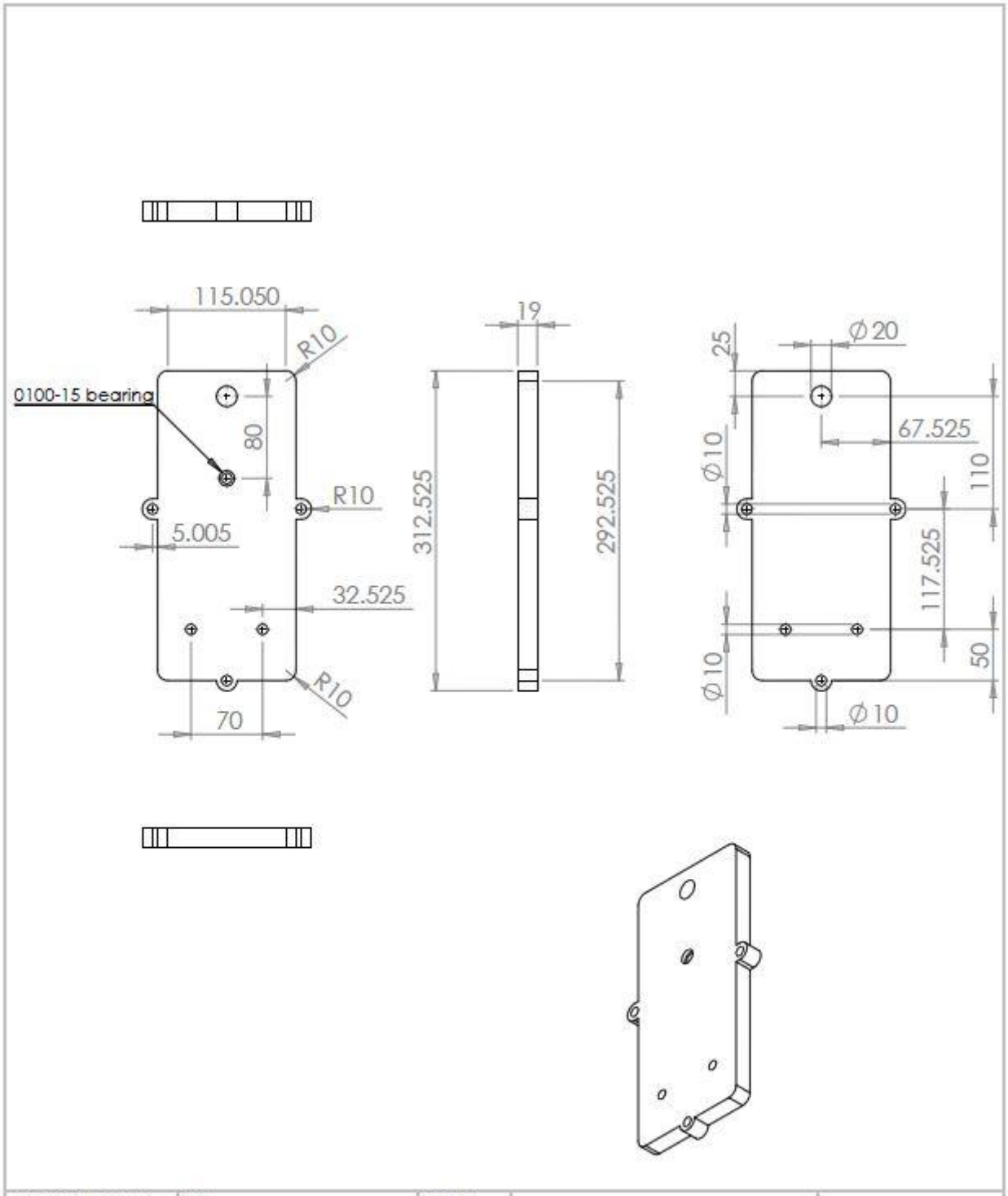
The physical model advances have been accompanied by robust, fast and efficient coding which is laid out and modularised in such a way to make boat, sail or rudder changes extremely quick; given the necessary dimensions the code can be transformed to operate on a new shape in under an hour.

With such a firm base in place, further development of the simulator will be much quicker and straightforward. Given extra time the sail motor could be mounted and geared in such a way that all the gearing can be contained within the mast, leading to the motor being virtually self contained within the boat hull. This could be done by having the inside surface of the hollow mast geared and using a planetary gear system to achieve the necessary gear and diameter ratios. Alongside this, due to the 3D capabilities of the projector a program could be written to display graphics in full 3D taking the fully immersive experience one step further. To broaden the target market a refined "basic" model could be developed aimed at home use, combined with networking capabilities so that the simulator could be used in conjunction with another to allow head to head races; for recreation of professional use. With recent advancements in virtual mapping technology such as Google Earth, it would also be relatively straightforward to fully integrate its capabilities into the current display system, allowing the user to sail on waters all around the UK and further afield if desired. This would obviously have great advantages for professional use allowing for practise runs of a course before the race date, along with providing another USP to aid sales to the general market.

21. Appendix


186.030
271.030
152.030
R22.500
0250-32 bearing
20
Ø26.900
R17
Ø36.900
258.015
193.015
Ø15
25
93.015
R17.500
R17
Ø15
25
R17.500
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBUR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
DRAWN
Tim Thornton
CHK'D
APPV'D
MFG
Q.A
TITLE:
Sail Motor
MATERIAL:
Mild Steel
DWG. NO.
Top & Bottom Mounting Plates
A4
WEIGHT:
SCALE:1:5
SHEET 1 OF 1


165.060
150.055
Ø9.530
14.765
R14.765
01011-15 bearing
67.525
30
105.520
R10
Ø10
Ø12.930
14.765
R14.765
105.520
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBUR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
DRAWN
Tim Thornton
CHK'D
APPV'D
MFG
Q.A
TITLE:
Rudder Motor
MATERIAL:
Mild Steel
DWG NO.
Top Mounting Plate
A4
WEIGHT:
SCALE:1:2
SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:			DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
	NAME	SIGNATURE	DATE		TITLE: Rudder Motor		
DRAWN	Tim Thornton						
CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL: Mild Steel	DWG NO. Bottom Mounting Plate		A4
				WEIGHT:	SCALE:1:5	SHEET 1 OF 1	

22. Bibliography

1. J. R. Binns, F.W. Bethwaite & N. R. Saunders: *"Development Of A More Realistic Sailing Simulator"* (2002)
2. T. J. Gale & J. T. Walls: *"Development Of A Sailing Dinghy Simulator"* (2000)
3. Laser International: *"The World By Law"* (2009)
4. B. Krebber & K. Hochkirch: *"Numerical Investigation On The Effects Of Trim For A Yacht Rig"* (2006)
5. P. J. Sterne *"Reinforcement Sailing"* (2004)
6. H. C. Smith: *"The Illustrated Guide To Aerodynamics"* (1992)
7. E. M. Lewandowski: *"The Dynamics of Marine Craft: Manoeuvring and Seakeeping"* (2004)
8. V. Bertram: *"Practical Ship Hydrodynamics"* (2000)
9. J. N. Newman: *"Marine Hydrodynamics"* (1977)
10. R. Tillman: *"The Complete Book of Laser Sailing"* (2000)