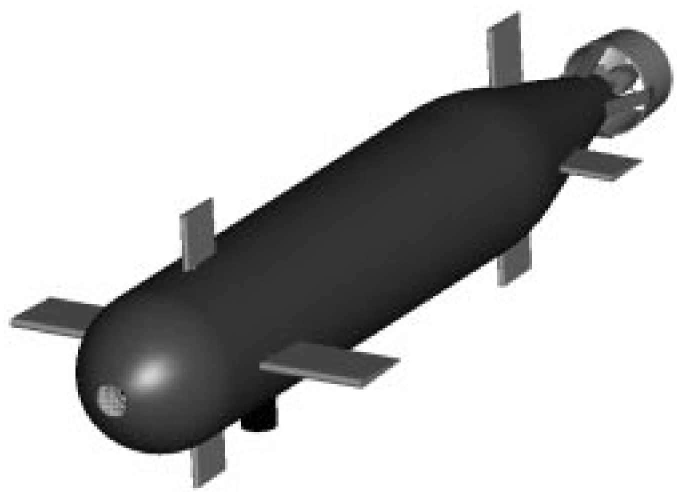
# ME 415 Group #5
# Design of an Autonomous Underwater Vehicle
# Advisor: Professor S. Thangam
# December 10, 1998



*"We pledge our honor that we have abided by the Stevens Honor Code."*


David Burchell ____________________

Robert Jankowski __________________

Claudia Mayer ____________________

James Merz ______________________

Peter Verrillo _____________________

**Table of Contents**

## Project Summary

The objective of Group #5's senior design project for this semester was to successfully design all the mechanical aspects of an autonomous underwater vehicle. The AUV is being designed specifically for the 1999 International Autonomous Underwater Vehicle Competition sponsored by the Association of Unmanned Vehicles System International, in Panama City, Florida.[11] The design of the hull, control surfaces and propulsion system will be performed by the mechanical design team and the autonomous guidance system will be handled by a team of electrical and computer engineers working concurrently with the mechanical team. The deliverables to be turned in for the autonomous underwater vehicle include parametric solid models, structural analysis, flow analysis, a complete set of detailed drawings, a governing motion equations, a bill of materials, and final purchasing requirements.

## Introduction

The ocean depths are the last terrestrial frontiers that remain a mystery, due to the extreme conditions that exist there. There are many places that humans cannot safely reach with the current submarine technology, and the Autonomous Underwater Vehicle (AUV) is one possible solution. AUVs are small, low cost, and long range vehicles that have a myriad of applications in naval and marine research. Autonomous Underwater Vehicles can gather information about vegetation, pipelines, communications lines, and a number of other structures. AUVs are unmanned and therefore have to be able to detect location and positioning of objects surrounding them. They must also have precision designed sensory, navigation, and control hardware systems.

Our senior design group has designed, and will be building, and testing an AUV to be judged in the International Autonomous Underwater Vehicle Competition at the Naval Coastal Systems Station in Panama City, Florida. Another senior design group composed of computer and electrical engineers has been designing the autonomous guidance system. At the competition the AUV will undergo performance tests as well as static and design judging. The AUV must be able to dive and travel at least 5 meters completely autonomously, pass through a maximum of 6 gates, surface inside the Target Zone, mark the maximum depth inside the Target Zone, and finish the mission within 20 minutes. The AUV will also be judged on elegance and innovation of design, presentation, safety, craftsmanship, and team uniformity [11].

Currently, there are many different varieties of Autonomous Underwater Vehicles. State of the art technology for AUVs falls into three general categories. The REMUS (Remote Environmental Monitoring Units) is an example of classic submarine technology. This type of AUV is typically used for ocean topography. The second common type of AUV is the full thruster class, of which the OTTER (Ocean Technology Testbed for Engineering Research) is a prime example [11]. This variety of AUV is used for any application that requires a high amount of maneuverability. The last common type of AUV is the hybrid of the classic submarine and the full thruster technologies. The Phoenix [12] is an example of this class of AUV, which are normally utilized in mine detection and similar applications. The AUV that has been designed and is presented in this paper is most similar to, though not exactly, to the REMUS because it employs the use of control surfaces. (See Appendix C for photos of Phoenix, REMUS, and OTTER)

A literature search was essential as a beginning point for the AUV project. Many sources were looked into and the majority of the information was found on the Internet and in the library. Information on state of the art AUVs such as the REMUS, OTTER and Phoenix was all found via the World Wide Web. Also through the Web, information on model submarine construction books was found. The book purchased through this effort was Model Submarine Technology [4] by Norbert Bruggen. Another important book that helped out with a basic understanding of model submarine technology was Concepts in Submarine Design [5]. There were several books in the Stevens library that will be very helpful as well. These include Principles of Naval Architecture [7] by Edward V. Lewis and Hydrodynamics of Ship Propellers [3] by John Breslin.

There is a great need for the development of autonomous underwater vehicle technology in several arenas. Improved AUVs would be a great asset in the laying of underwater fiber optic cable across the ocean floors. If faster, more durable, cheaper

AUVs could be used, the cost of installing the cable would be reduced. Also, improved AUV technology would be of great help in mine detection. Currently, navy specialists need to manually disarm or detonate live mines. Obviously, this is a very hazardous procedure that could be eliminated if AUV technology was improved and consequently, human life would not be put in jeopardy. Also, AUVs would lower the cost of such procedures because such intense training would not be needed. Another example of an application in which improved AUVs would be beneficial is marine research. As the cost of AUVs falls, the number that research groups could purchase and utilize would increase. This would allow for more data to be obtained about ocean floor ecology and topology.

## **Design Methodology**

A characteristic that almost all successful design projects include is a lucid methodology showing the approach to be undertaken to help the team stay focused on the problem at hand. After choosing a project, a plan was devised that would allow the team to start the design process. (See Appendix B for Flow Chart.)

The first step in the chosen methodology was a literature search, which was split into two phases. Phase one included research that familiarized the team with the design practices and standards of naval architecture as well as the state-of-the-art of similar designs described in the introduction. The result of this research provided the team with four possible design alternatives for a conceptual design based on the main hull and control systems schemes.

The main hull could either be a wet hull design where the outer hull is free-flooded with water while the submarine payload, consisting of the electronic hardware, is encased in an internal pressure hull. The main purpose for the outer hull is to aid in the hydrodynamics of the submarine, reduce the weight of the AUV, and to protect the internal components. The alternative to the wet hull is a dry hull approach, which is more popular and in synch with classical submarine design. The entire submarine is a pressurized shell and no water is allowed to enter, which allows for better optimization of space for the payload. The disadvantages of this type of system are that the weight of the AUV and the pressure differential across the outer hull is increased considerably from a wet hull AUV.

The two standard designs for a submarine's control system are a classic control surface design that incorporates a given number of hydroplanes controlled by servomechanisms, while the other approach would be a thruster design. The classical approach utilizes control surfaces where the yaw of the AUV is controlled with a rudder while the depth and pitch are controlled by horizontally mounted “dive planes” and “pitch planes” in the bow and the stern of the submarine respectively. The depth and pitch of a thruster design would be controlled by two opposing thrusters mounted vertically while two opposing thrusters mounted horizontally would control the yaw. The advantages of control surfaces to thrusters are better hydrodynamic stability and low battery usage due to the smaller servomechanisms rather than large motors that thrusters require. The advantage of thrusters to control surfaces is a higher degree of maneuverability.

From these design factors, four possible design alternatives can be created. Both a wet and dry hull approach can incorporate either a control surface or thruster control scheme, thus deciding which to choose for the conceptual design. This lead to the next required step in the design process, which would be to choose the best possible alternative as a conceptual design.

To focus the team on the right design criteria, the following design specifications were gathered, based on the objectives of the competition:

- A minimum velocity of 1 m/s is essential in order to traverse the course in the allotted 20 minutes.
- A minimum turning radius of 1.5 meters is necessary to successfully complete the 90 degree turn following gate 6.
- The maximum depth the design should consider is 12 meters.
- A failsafe mechanism is required so that a malfunctioning AUV can be shut off.

- A cover for the propeller is required in order to protect the pool lining.
- The power supply must be electric.
- No more than 60 volts can be put into an open circuit.
- The AUVs weight can not exceed 100 kg.

Based on past experience as well as professional advice and several discussions, the team came to a consensus that the benefits of a dry hull design outweighed the advantages of the wet hull approach. One of the greatest arguments for a dry hull design was the advantage of not having to resort to watertight connectors, which unbelievably a majority of naval architects agree is the greatest cause of headaches in the field. This also proves to be a more economical approach since the recommended watertight connectors start at a hundred and thirty dollars each. The weight disadvantage of the dry hull is alleviated by the fact that the freedom in payload placement would allow for a more optimized layout and the overall size of the AUV would be decreased. From a manufacturing perceptive, both approaches have similar manufacturing requirements that are feasible so this did not factor into the decision.

An assessment of the advantages and disadvantages of the two types of control systems resulted in favoring the classical control surface approach. A thruster design would have to incorporate several sensors to constantly monitor the position of the AUV so that it did not veer off course due to the erratic behavior of thrusters. A review of the specifications showed that the high degree of maneuverability that a thruster design delivers is in excess to what is required. The hydrodynamic stability and economical advantages that a control surface design offers is more suited to the requirements of the contest.

To verify the design, a qualitative analysis of the design alternatives was made based on the competition specifications. This included an assessment of the following design criteria with an assignment of a rating system of numbers one through five (five being the most important):

| Design Criteria | Importance Rating |
|---|---|
| • Maneuverability | 4 |
| • Controllability | 5 |
| • Dynamic Stability | 5 |
| • Sealing | 3 |
| • Overall Durability | 4 |

This allowed the following table to be created, tabulating the four design alternatives, their performance, and their weighted average based on the design criteria and importance rating:

| | Maneuverability | Controlability | Stability | Sealing | Durability | Weighted Average |
|---|---|---|---|---|---|---|
| **Control Surfaces with Wet Hull** | 3 | 4.5 | 5 | 4 | 3 | 3.98 |
| **Control Surfaces with Dry Hull** | 3 | 4.5 | 5 | 3 | 5 | 4.21 |
| **Thrusters with Dry Hull** | 5 | 3 | 3 | 4 | 5 | 3.90 |
| **Thrusters with Wet Hull** | 5 | 3 | 3 | 3 | 4 | 3.57 |

The qualitative result confirmed the team's initial decision and enabled the team to begin the detailed design process, which was broken down into four concurrent sub-processes that include the design of the hull, the control scheme, propulsion system, and the sensor team's guidance system. Each process for the detailed design was evaluated for feasibility.

## Static Design

The dry hull design sub-process was initiated by researching the type of shape it was to take on based on the hydrodynamic performance and manufactorability. Technical research and professional advice convinced the design team that a torpedo shape was an ideal shape for AUV. This shape would result in the lowest resistance force in all planes of motion and it would have the best overall hydrodynamic stability. This valid assumption allowed the team to bypass the tedious job of testing several different design configurations and initiated a good start on the unforeseen problems in the flow analysis using ANSYS software. Based on the electronic internals provided by the sensor team, a solid-model of the outer hull was created as to begin analysis. After confirmation of the approximate dimensions of the sensor team's electronic package, the size of the hull was adjusted accordingly and detailing was initiated (Detailed drawings included in Appendix D). The solid model was also updated in the analysis using ANSYS so as to confirm the chosen hull design and to get an analytical model of the AUV.

Following the agreement on the overall size of the AUV, the dry hull was considered from a manufacturing point of view. The design would have to allow for an easy access to the payload to ease any adjustments that would have to be made without sacrificing water-tightness. As a result, an absolutely dry hull design would not be feasible. Instead, a hybrid dry hull was proposed where the nose and tail portions of the

AUV would be free-flooded as in a wet hull design. This compact and modular design not only eases manufacturing, it allows for an excellent o-ring configuration while utilizing a portion of the weight saving advantages of a wet hull design. The mid-section of the boat will be made out of PVC pipe while the nose and tail sections will be made out of fiberglass. The fiberglass sections where sized with the stereo-lithography constraints of DMI so that a mold could be made for the fiberglass.

The mid-section of the boat will be connected to two other smaller sections by a series of step-down caps using an o-ring seal in between each connection. Research conducted by the team showed that o-rings where the best type of sealing arrangement. With an internal pressure of 10 psi and a pressure differential of approximately 5 psi, o-ring seals are more reliable than gaskets in that they are more resistant to warping.

The control scheme consists of control surfaces that will be driven by servomechanisms. The advantage of using servos is that they do not require a microcontroller, unlike stepper motors, and can easily be adopted to be controlled by a RF remote control for testing purposes. The proper torque output requirement for the servos was calculated by taking moment calculations on the control surfaces. The following formula yielded the required torque[2]:

$$M = C_m * 0.5 * \rho * V^2 * A_p$$

where M is the torque, $C_m$ is the coefficient of moment, $\rho$ is the density of the fluid, V is the velocity of the fluid and $A_p$ is the planform area of the fin. Below, figure 1 shows the torque rating for each hydroplane.

| | Velocity m/s | Moment in*oz |
|---|---|---|
| Bow Planes (Min.) | 1 | 51.14 |
| Bow Planes (Max.) | 1.5 | 115.06 |
| Pitch Planes (Min.) | 1 | 34.09 |
| Pitch Planes (Max.) | 1.5 | 76.71 |
| Rudder (Min.) | 1 | 10.39 |
| Rudder (Max.) | 1.5 | 23.37 |

Figure 1

**Propulsion System**

The propulsion system will consist of a properly sized motor to drive a propeller. A good approximation for the propeller sizing was difficult to find in that they are usually sized using past experience and experimental data. By sizing the propeller, based on the number of blades, and the number of revolutions per minute at which it will rotate, the pitch and diameter of the propeller can be solved. This, in turn, will provide the power necessary to propel the AUV at 1 m/s and thus provide an approximation for the size of the driving motor needed. Based on a Taylor's approximations for a propeller design [2], a propeller diameter of 3.5inches with a pitch of 1.75 radians was calculated. A motor of 75watts and an rpm of 1500 is needed. See Appendix A for detailed calculations of propeller size.

## AUV Dynamics

The contest specifications and the sensors picked by the Sensor and Control Team place limitations on how our AUV can perform dynamically. The contest requirements state that the submarine must complete the course in a minimum of 20minutes and that points will be awarded for a faster completion of the course. The layout of the gates also forces the AUV to have a minimum turning radius of 1.5meters. The sensor and control team will be using sonar and a pressure sensor so the AUV can know it's position in the pool. Both of these sensors require that the submersible stay flat and not pitch forward or back and it cannot roll side to side even when diving or surfacing. Any pitching or rolling would cause inaccurate readings from each of the sensors.

## System Analysis

The system equations were set up in TKSolver to get a model of the submarine motion. The model would be able to show how long it took to accelerate to maximum velocity and predict slow down when turning and similar dynamics. Unfortunately, the file got corrupted and an older version was recovered but we were not able to get it back to it's latest revision and therefor time dependant nature of the submarine could be given in the report. We are working to get it back in time for the final presentation on the 14th.

## Motion in the Vertical Direction

The ability to dive is of primary importance in submarine design. In order for a submarine to dive, the net force acting on it must be downward. The converse is true for surfacing. Typically, three forces act on the submarine during vertical motion, a buoyant force, a lifting force, and a drag force. These forces make up the governing equation for vertical motion[5]:

$$\Sigma F_z = ma_z = B + L + D$$

where B is the buoyant force, L is the total lift force, and D is the total drag force. The location of these forces are shown below in figure 2.

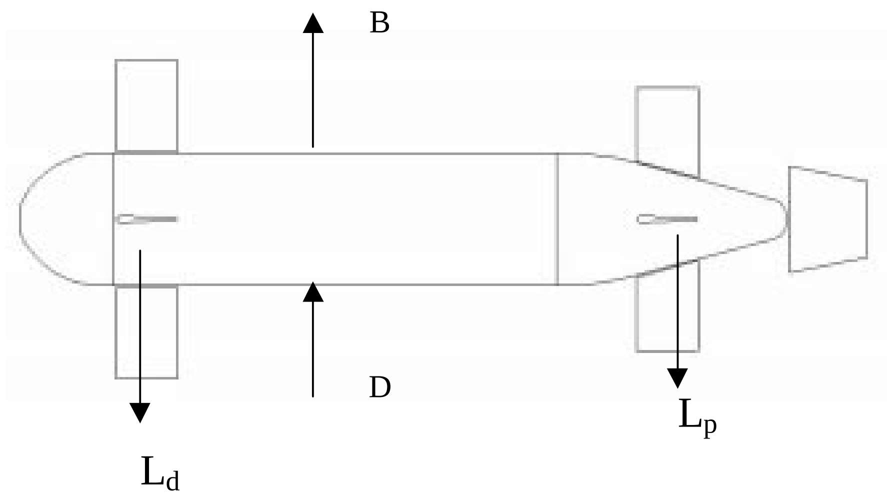



Figure 2

The buoyant force is equal to the difference between the dry weight of the submarine and the weight of and equal volume of water. Typically, it is up to the designer to pick a buoyant force for the submarine. For the AUV, a buoyant force of 1lb upward was chosen. The direction was chosen so that if there is a problem generating lift, the AUV with naturally rise to the surface for easy retrieval. The value was chosen to be small, since the buoyant force must be overcome by the lift in order to dive. The larger this force, the larger a lifting force the AUV will need to dive.

**Lifting Theory**

The total lifting force ,L, is the force that will mainly control the vertical motion of the AUV. For this AUV, it will be generated mainly by pair of forward horizontal hydroplanes. Lift will also be generated by a second pair of horizontal hydroplanes placed after the center of gravity. The lifting force generated by a pair of hydroplanes are determined by two main equations. First, the dimensionless lifting coefficient, $C_l$, must be determined. Once this is found, the lifting force can be calculated.

The coefficient of lift is mainly a function of the angle of attack of the fin. As the angle increases, $C_l$ increases. This trend continues until the stall angle is reached at which the effectiveness of the fin decreases greatly and the lift drops sharply. The coefficient of lift is defined[9] by :

$$C_l = 2 * \Pi * \sin(\alpha)/(1+(2/AR_{eff}))$$

where $\alpha$ is the angle of attack of the fin and $AR_{eff}$ is the aspect ratio of the fin. This equation suggests that one way to get a higher stall angle at the same $C_l$ is to decrease the aspect ratio of the fin.

As aspect ratio decreases, the stall angle increases. Figure 2 shows this relationship. The graph shows that at an AR of approximately .5, a coefficient of lift of .8 is generated at 25 degrees while at an aspect ratio of 2.5 the same $C_l$ is generated at 14 degrees. While it then seems logical to have very low aspect ratio wings, the drag on the fin greatly increases as the aspect ratio increases. This is due to an increase in the Reynolds's number of the wing which causes the flow over the wing to become more turbulent. For this reason, a aspect ratio of 1.5 was chosen for the forward hydroplanes and 1 for the after planes.

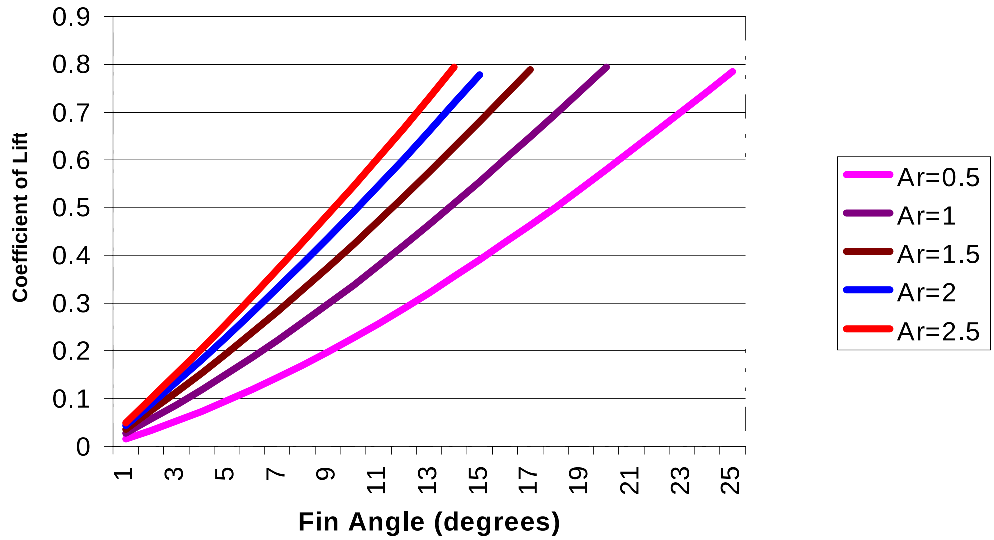



Figure 3

The lifting force generated is related directly to the coefficient of lift. The amount of lifting force is also dependent on the planform area of the wing and velocity of the submarine. The lifting force is determined by the following equation[9]:

$$L = C_l * .5 * \rho * V^2 * A$$

where $\rho$ is the density of the fluid, V is the velocity of the body or the fluid, and A is the planform area or area parallel to the flow over the wing.

**Drag Theory**

The total drag force is the sum of the individual drag on each component of the body. For the submarine, it is composed of the drag on the hull and then the individual drag on each control surface.

The drag on the hull is determined by the coefficient of drag for the hull shape, the area perpendicular to the flow and the velocity of the fluid or body. For the torpedo shape chosen for the AUV, Hoerner's book Fluid Dynamic Drag [6] recommends a coefficient of drag, $C_d$, of 0.75 for a flow perpendicular to the length of the hull. . This coefficient and the other variables relate to the drag force, $D_h$, similarly to the coefficient of lift [9]:

$$D = C_d * 0.5 * V^2 * A * \rho$$

where in this case A is the area perpendicular to the flow. These concepts also apply to the drag on each control surface except that the $C_d$ varies with the $C_l$.

The drag on each control surface is dependant on the same variables as the lift. The coefficient of drag on a fin is determined by it's coefficient of lift. The two are related by [9]:

$$C_d = C_{do} + (C_l^2/A*AR)$$

where $C_{do}$ is the two dimensional drag coefficient and A is the planform area. This equation also shows that as the aspect ratio decreases, the coefficient of drag increases.

**Pitch Theory**

A requirement from the sensors being used is that the submarine remain level at all times, even when changing depth. For this to be met, two sets of horizontal control surfaces must be used. This allows the depth and pitch to be controlled independently. The forward horizontal planes or dive planes typically generate the bulk of the diving force and a second set of horizontal planes or pitch planes control the overall pitch of the submarine. Figure 3 shows a free body diagram showing all the moments on the sub around the y-axis( coming out of the page) and acting at the center of gravity.

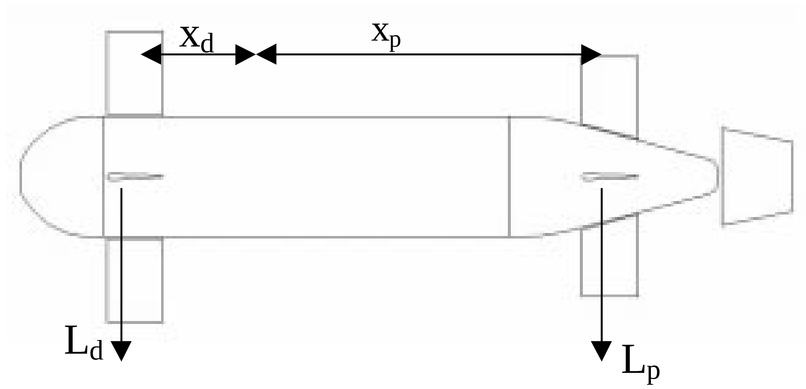



Figure 4

The pitch will also be generating a lift to offset the moment of the dive planes. This must be taken into account so that the submarine can exactly counter act the buoyant force and hover at a specified depth ($\Sigma F_y=0$). If it isn't, the submarine will oscillate as it tries to maintain a constant depth and zero pitch angle at the same time.

**Horizontal Plane Size and Placement**

The placement of the dive and pitch planes will determine how well the submarine can be controlled. The dive planes should be placed in forward one third of the submarine length for the best efficiency. Ideally, the planes should be placed exactly over the center of gravity so that there is no moment generated on the hull. This would reduce the size of the pitch planes and then the dive planes could negate all of the buoyant force. However, this would also mean that the should the pitch planes fail, there would be no way to control the pitch of the AUV. Also, if the submarine was to go into a steep dive and the pitch planes could not straighten the hull, the sub could not generate a force on the nose to help pull it out of the dive. In the case of this AUV, a distance of 8 inches forward of the center of gravity was chosen. This was decided upon to keep the dive planes in the first one third of the hull and close enough to the COG so as to keep their moment small.

The pitch planes must be placed carefully as well. It seems logical to just place them as far back as possible, but this is not the case. The part of the buoyant force that the pitch planes are going to negate must be chosen. Then the moment generated by the planes must counter act the moment created by the dive planes when they are negating the rest of the buoyant force. This is how the pitch planes location is calculated.

The size of the dive planes was determined only after certain design criteria were decided. It was picked that the dive planes would offset 75% of the buoyant force when the submarine remains at a constant depth. However, this 0.75lb force must be generated at a low angle of attack so that they can create a larger lifting force to allow the submarine to dive. Using the equations described above for the lift and empirical data from Davidson Labs and books on wing design [8] other design criteria were determined. From test data from Davidson labs for fins operating near the Reynolds's number of the AUV fins (approximately 45,000) it was found that the maximum coefficient of lift before stall would be approximately 0.75. "Theory of Wing Sections" contained experimental data which showed that an expected stall angle for our wing would be around 15 degrees [1]. The NACA0012 wing shape was chosen based upon correspondence with the navy's AUV project team. With this information and using the lifting theory described above, the dive planes were sized. At an aspect ratio of 1.5 and a maximum coefficient of lift of .75, the bow planes are sized at a 6" span by a 4" chord each, giving the bow planes a total span of 20"(8" for the hull). The maximum angle before stall was 12 degrees, and at an angle of 4 degrees, a lift of .775 lbs is generated. The maximum lift is approximately 2.5 lbs total for the forward hydroplanes.

The pitch planes were sized using the same theory as the dive planes(see appendix 2b for sample calculations). The final size of the stern planes is 4" by 4" giving an aspect ratio of 1 and their maximum angle of attack is 15 degrees. The maximum lift generated by these planes is 1.81 lbs. They will generate the remainder of the buoyant force, .225lbs, at 2.5 degrees. They were placed approximately 27" behind the center of gravity so that the moment due to their .225lbs of lift is equal to the moment due to the .775lbs of force from the dive planes. A safety factor of two was used, meaning that the pitch planes should be able to counter double maximum moment generated by the dive planes.

The maximum dive rate was checked to see if it would suitable. At the maximum dive rate, the drag force on the hull in the vertical direction would equal the lift generated minus the buoyant force. Since the maximum lift down was equal to the sum of the dive and pitch lifts, from the drag theory the maximum velocity down was found to be 1.17 ft/sec. The design was iterated several times until this result was found.

## Horizontal Motion

Analysis of the horizontal motion is simpler than that of the vertical motion. There are only two forces acting in the horizontal motion when the AUV is moving in a straight line, the thrust from the propeller and the drag on the hull. They are related by the equation [5]:

$$\Sigma F_x = ma_x = T - D_{tx}$$

where T is the thrust and $D_{tx}$ is equal to the total drag in the horizontal direction. Figure 5 shows the freebody diagram for the thrust and the components of the total drag. As can be seen from the figure, the total drag is composed of the drag on each control surface and the hull itself. The hull shape has a $C_d$ of 0.1 [6] and the drag on each control surface was found using the drag theory described above. The minimum velocity of the AUV was chosen to be 3.25ft/sec so that it can complete the coarse with time to spare. This velocity is the terminal velocity at maximum drag in the horizontal direction. The maximum total drag in the horizontal direction was found to be approximately 3.13 lbs. This is the drag when all control surfaces are turned to a 12 degree angle of attack. This means that the thrust must be equal to that drag for the AUV to move at 3.25ft/sec. The minimum drag on the submarine was found to be approximately .5lbs. This is when all control surfaces are at zero angle of attack.

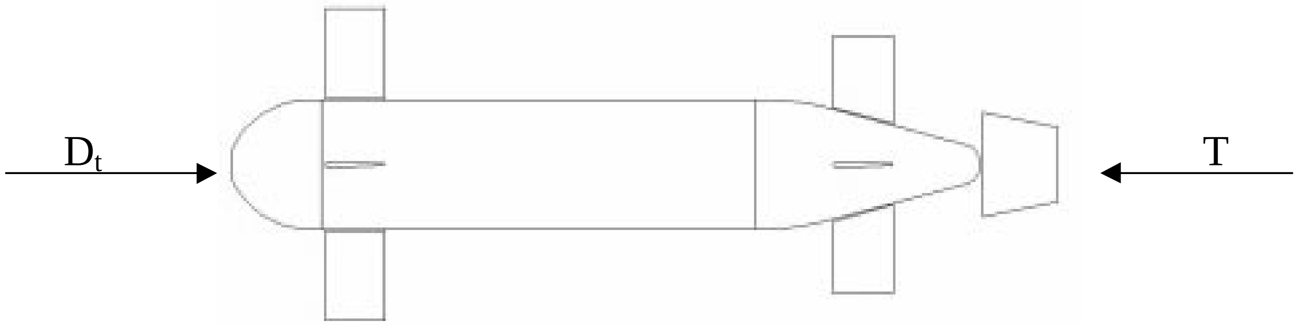



Figure 5

**Turning Analysis**

The forces necessary to turn the AUV will be generated by  pair of vertical hydroplanes that will create lift in the horizontal direction causing a net moment around the z-axis and acting on the COG of the AUV. At the suggestion of the Principals of Naval Architecture [7], there is no straight forward method of calculating the proper size of a rudder to generate a given turning radius. Instead, they recommend using previous design experience and experimentation to properly size the rudder on a given hull. However, these references had some information for the placement of the rudders. Rudder efficiency is improved greatly by placing the rudder directly behind the propeller wash. In this AUV, the stern rudder will be placed forward of the propeller so it can be controlled directly by a servo. It is recommended that the rudder area be increased 30% when not placed directly behind the propeller. Using the design experience from the summer, a rudder size was calculated. Since the size and shape of the two submarines are very similar, a rudder size was calculated. The old area was $7.5in^2$ with a span of 2.5in and a chord of 3in. The new area is $9.75in^2$ with a chord of 2 and a span of 4.875. The change in span and chord was due to information from the Principles of Naval Architecture which stated that rudder effectiveness depends more on span than chord.

The old submarine could turn within 1.5 times it's boat length which is as tight as a boat can achieve with a single rudder. It is expected that the new boat will achieve the same results. However, a forward rudder of the same size was added so that the submarine can make a tighter turn when it needs to enter the target zone after gate six or at any other time during the course. This forward rudder will also add stability to the submarine when it is not in use. Next semester, the effectiveness of the rudders will be determined experimentally as suggested by the Handbook of Naval Architecture by tow tank testing performed at Davidson Laboratories.

**Note: See Appendix A for governing equations and sample calculations.**

**Computational Fluid Dynamics**

The analysis part of this design project was a very challenging and difficult task for our team. Initially it was thought that there would be a dynamic modeling of the control surfaces and hull based upon the data from ANSYS/Flotran. This proved to be an incorrect evaluation of the software and the ability of our group to operate it. There were many obstacles hindering progress with our dynamic simulation. Initially it was the inability to properly input data and boundary conditions into Flotran. We received help to establish an understanding of some of the quirks of the program and how to analyze certain 2-dimensional situations. After a respectable understanding of the example problems was attained the three-dimensional case was set to evaluate.

During the semester it was realized that the true design of our parts was coming from the dynamic system equation that was being calculated concurrently, the CFD would merely be verification of their design. If there had been previous instruction of the software in the classroom, the original intention of using Flotran for design would have been possible. Due to the steep learning curve of the software, only one three-dimensional flow model was analyzed. This model was first developed in Pro/Engineer using the design that our team had modeled. The parametric solid models are very robust with many relations and features which required a “stripping” of the models down to very basic features, to facilitate the finite element modeling procedure.

The model that was analyzed was constructed from the models and assemblies excluding the propeller and shaft. This model was then assembled with a “box” that simulated the flow that would be needed to analyze the problem. The size of the domain was basically one chord length surrounding the circular section of the hull. There was one chord length in front of the submarine and approximately 6 chord lengths behind too fully realize the wake. This domain was determined from the expert advice of Professor Thangam. The model of the submarine was incorporated into one singular part and then "Cut out" of the fluid domain part in Assembly Mode. This yielded one part that could be loaded and meshed using Pro/FEM.

For such a complicated model, there was no other choice then to use Pro/Engineer's sophisticated finite element modeling software. The only problem with using the software was that it did not output the desired boundary conditions needed to analyze our simulation. Pro/Engineer will only output a thermal, structural or modal analysis and the simulation required a fluid analysis. This required a significant amount of tweaking to derive correct loading conditions. With the aid of Professor Pochiraju a better understanding on how to utilize the .ANS file that Pro/Engineer output was developed. These .ANS files are merely a text file with a string of ANSYS commands assigning nodes, elements and their boundary conditions. A line of the .ANS file is similar to typing in the text command in the ANSYS Dialog Box, or using the Graphical User Interface (GUI) selecting commands through menus.

The Pro/FEM mesh delivered 37,601 elements and 7658 nodes. Assigning a Boundary Condition to the nodes that derived the shell of the submarine was a very difficult process. Using a methodology that Professor Pochiraju had suggested, pressure was applied to each surface of the submarine from within Pro/FEM. When the .ANS file was output, the text was imported into Microsoft Excel delimited by commas. From this output a list of all the nodes, elements and loading conditions for the model are obtained. The loading conditions are defined by elements that can be traced back to which nodes are being loaded as well because each element has four nodes.  By sorting and organizing this data, all the nodes that define the outer shell of the submarine were defined.  This data was used to place the specific loading command for each node into Excel and exported the file as a .CSV (delimited by commas).  The only other “tweak” that was required was changing the Element Type in the .ANS from the 3-D structural number (72) to the Flotran 3-D number (142).

The following is an example of ANSYS Commands that were used:

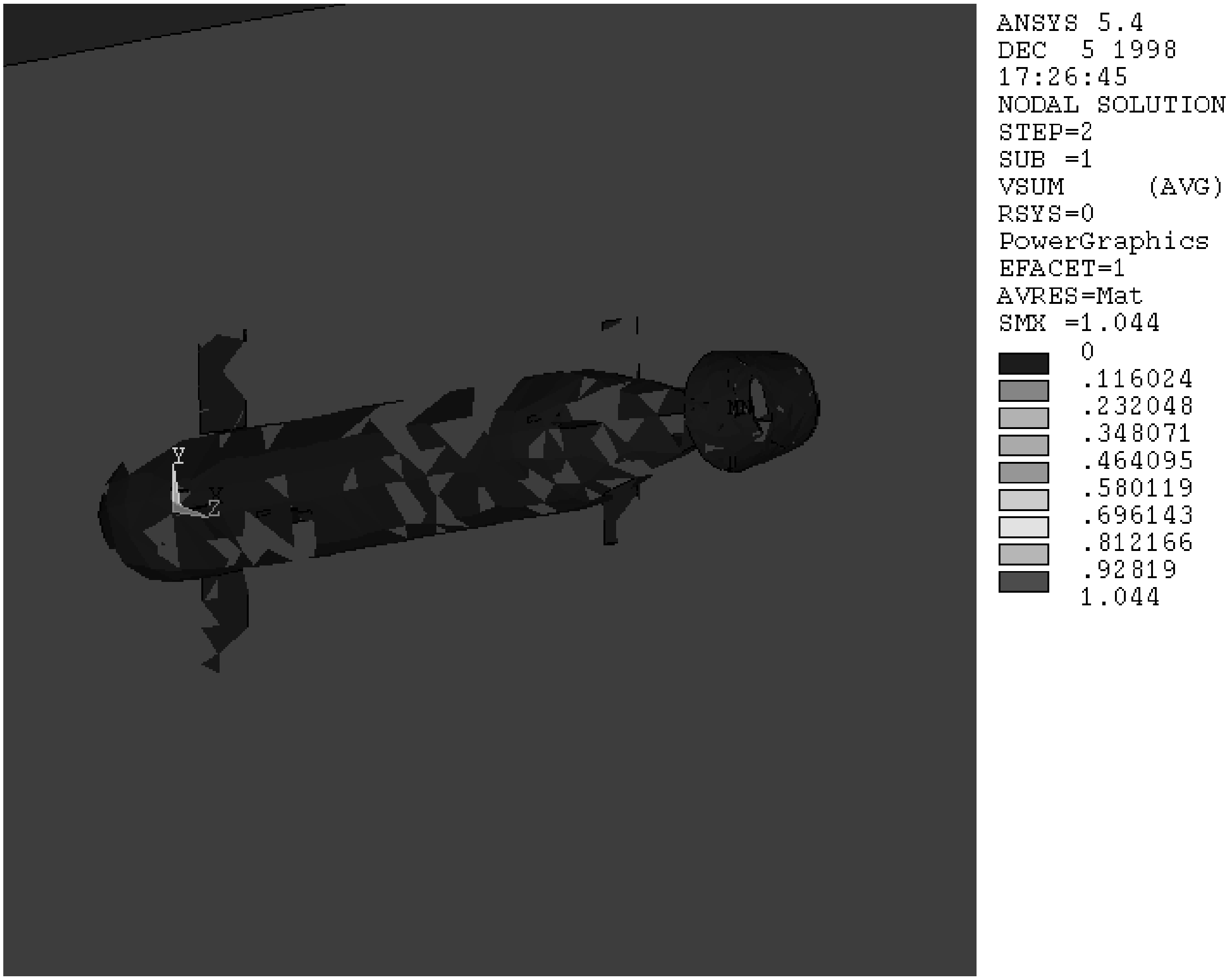
| | |
|---|---|
| ET,1,142 | >> Defining Element Type 142 |
| N,1,2,1,1 | >> Node, #1, x = 2, y = 1, z = 1 |
| EN,7,85,33,68,77 | >> Element #7, Nodes 85,33,68,77 |
| D,1,VX,0,0,129,1,VY,0,0,VZ,0,0 | >> Assigns VX,VY,VZ = 0 from Node 1-129 |

The AUV will travel at approximately 1 meter per second in a man made lake. To model this with Flotran there is a need to exact the submarine at zero velocity, while the water travels past it at the speed of the sub (1 m/s).  This requires the model to exhibit free flow stream over the boundary of the “stationary” submarine.  After the .ANS file was brought into ANSYS/Flotran, boundary conditions were set.  By selecting the all the nodes at the entrance from a front view, the Velocity in the X direction was set to 1 (m/s). The submarine had constraints place from the imported file, basically for each node on the surface of the sub Vx, Vy and Vz = 0.  The fluid can be modeled with a boundary for the length of the domain with Vx=1 m/s and set as a moving wall.  This is done for all four walls surrounding the length of the vessel.  Finally, the nodes at the end of the fluid domain are selected and set to zero pressure.  Through the analysis, SI units are kept similar to obtain the charts in a consistent way. As for the Fluid Properties, the pressure was approximated at 115,000 kPa for the depth of the vehicle. The density = 998 $kg/m^3$ and viscosity = $1.003x10^{-3}$ kg/ms were taken at standard temperature.

The analysis was run for over 100 iterations but the solution converged after about 50 steps. The analysis took over 45 minutes to analyze and there was several times where the limited amount of disk quota hindered the analysis process unknowingly. The suggested technique to run this analysis is to set the interactive swap space to 196 MB and the database to 32 MB. This allows ANSYS to use the hard disk for any memory need past the amount of RAM. The other necessity to run such complex three-dimensional case analyses involves using the /scratch/ directory for increased hard memory requirements during the solution.

The following pages show the graphs of our flow analysis. Basic flow and pressure characteristics around the vessel can be determined from these graphs.

## Validation of Boundary Conditions




## Validation of Boundary Conditions

**Explanations:**

1. The Blue areas depict the boundary of the submarine.

2. The Red areas depict the fluid that is flowing around the submarine vessel.

3. The Velocity of the Red elements is designated by the chart on the side to be

1.044 m/s.

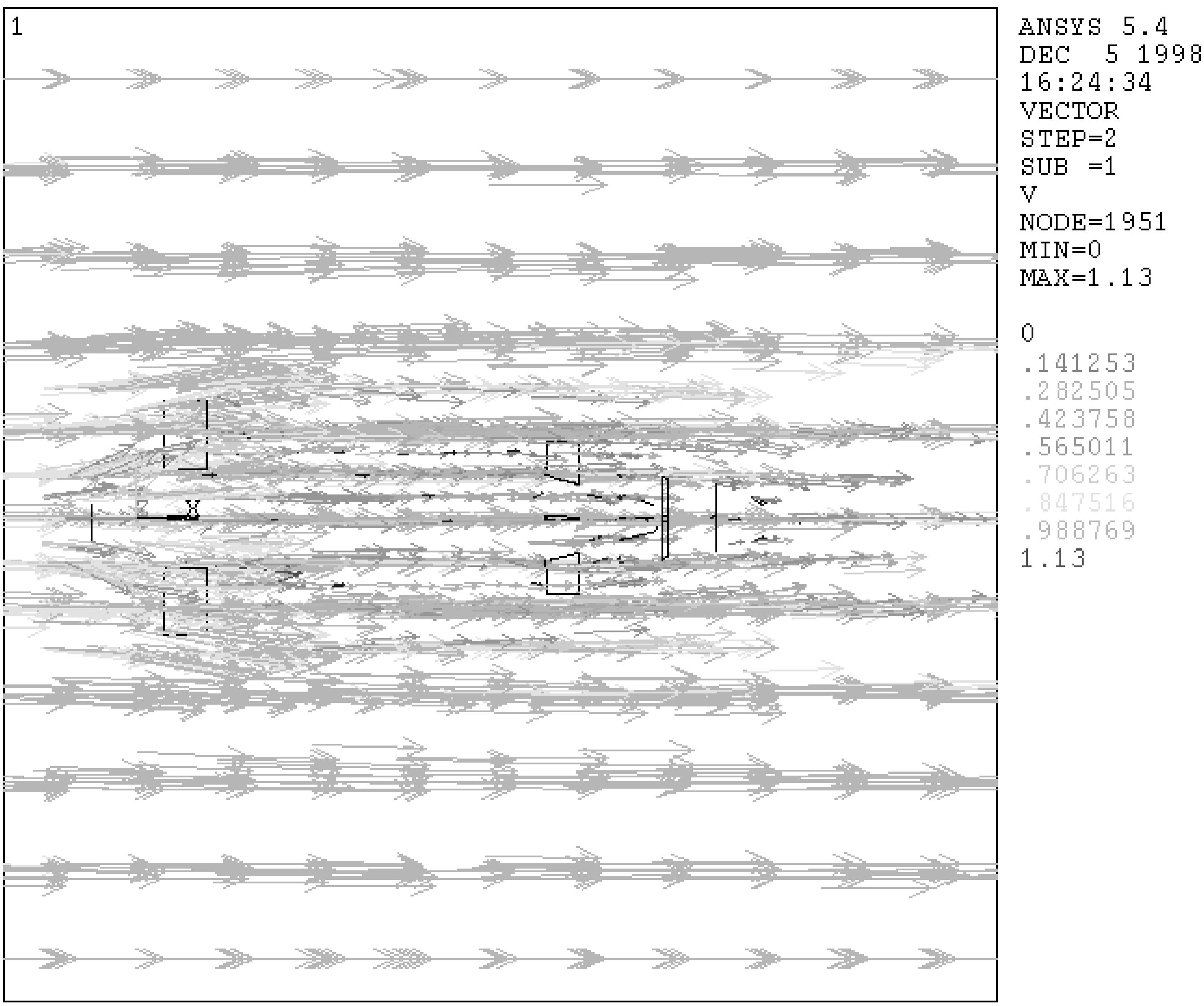
4. The Velocity of the Blue elements is defined as zero m/s.

**Considerations:**

1. The reason for the inconsistency of the element color is most likely due to the complexity of the graphics involved.

2. This graph is merely a validation of the boundary conditions that were specified in the report and in the Flotran Analysis.

**Velocity Flow around Submarine (Side View)**



## Velocity Flow around Submarine (Side View)

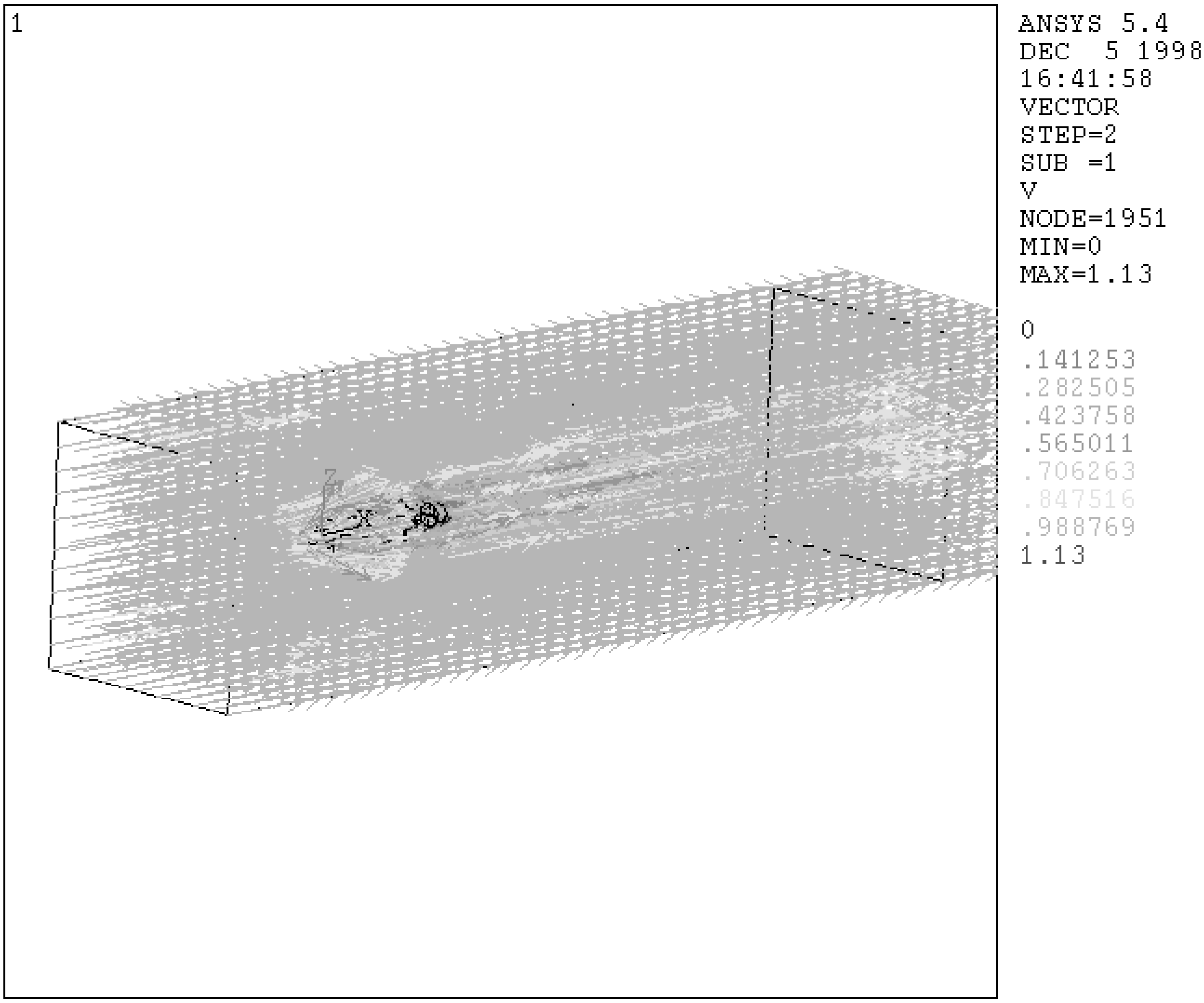
**Explanations:**

1. The Orange vectors currently exhibit a velocity of .9887 m/s, this is close to the inlet stream set at 1 m/s.
2. The Blue vectors exhibit the physical boundary of the submarine vessel with a velocity of 0 m/s.
3. The nose of the vessel diverts the flow well without too much hindrance; most of the velocities at the front are close to 1 m/s.
4. The control surfaces do not significantly affect the flow around the submarine.

**Considerations:**

1. The single Red Vector on the plot may be caused by an improperly assigned/deformed element.
2. The plot is very busy due to the depth of the plot due to the 3-Dimensional analysis.
3. The real-life situation would have opposite speeds on the boundaries.
4. The sub would have a velocity of 1 m/s surrounding it and the water at the top of the domain would be at rest.

## Velocity Flow for Fluid Domain




**Velocity Flow for Fluid Domain**

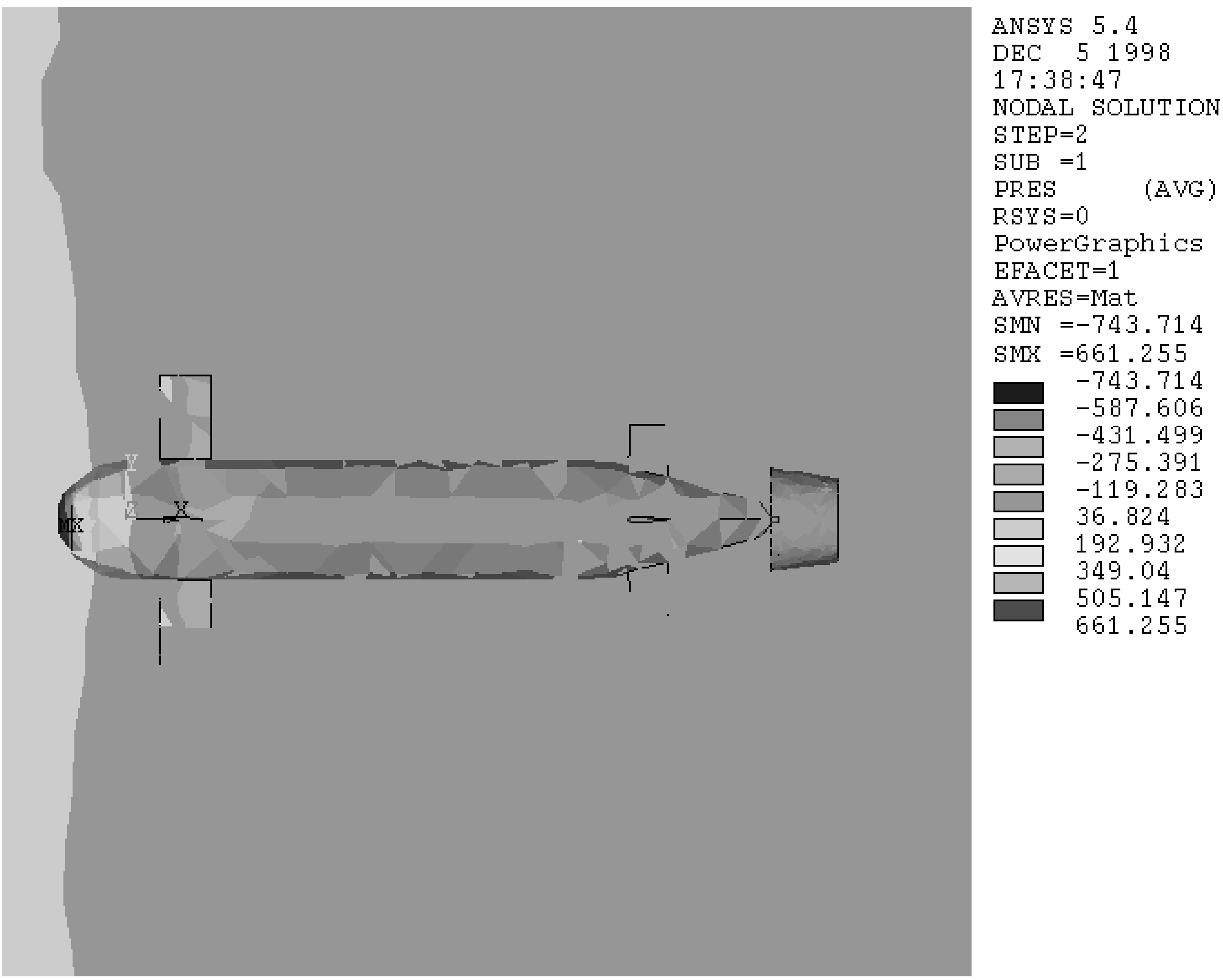
**Explanations:**

1. This is merely the same profile as before, but looking from the front top right corner down as an isometric.

**Considerations:**

1. This vector flow of the fluid domain shows the effect the vessel has on the fluid around it and gives a better definition of the wake of the sub.

## Pressure Distribution over Submarine



## Pressure Distribution over Submarine

**Explanations:**

1. The Red pressure zone is showing a pressure of 661 kPa on the front of the sub.

2. The Yellow pressure zone soon follows with a pressure listing of 349 kPa.

3. The Green pressure zone is displaying the pressure reverting back to standard atmospheric pressure.

**Considerations:**

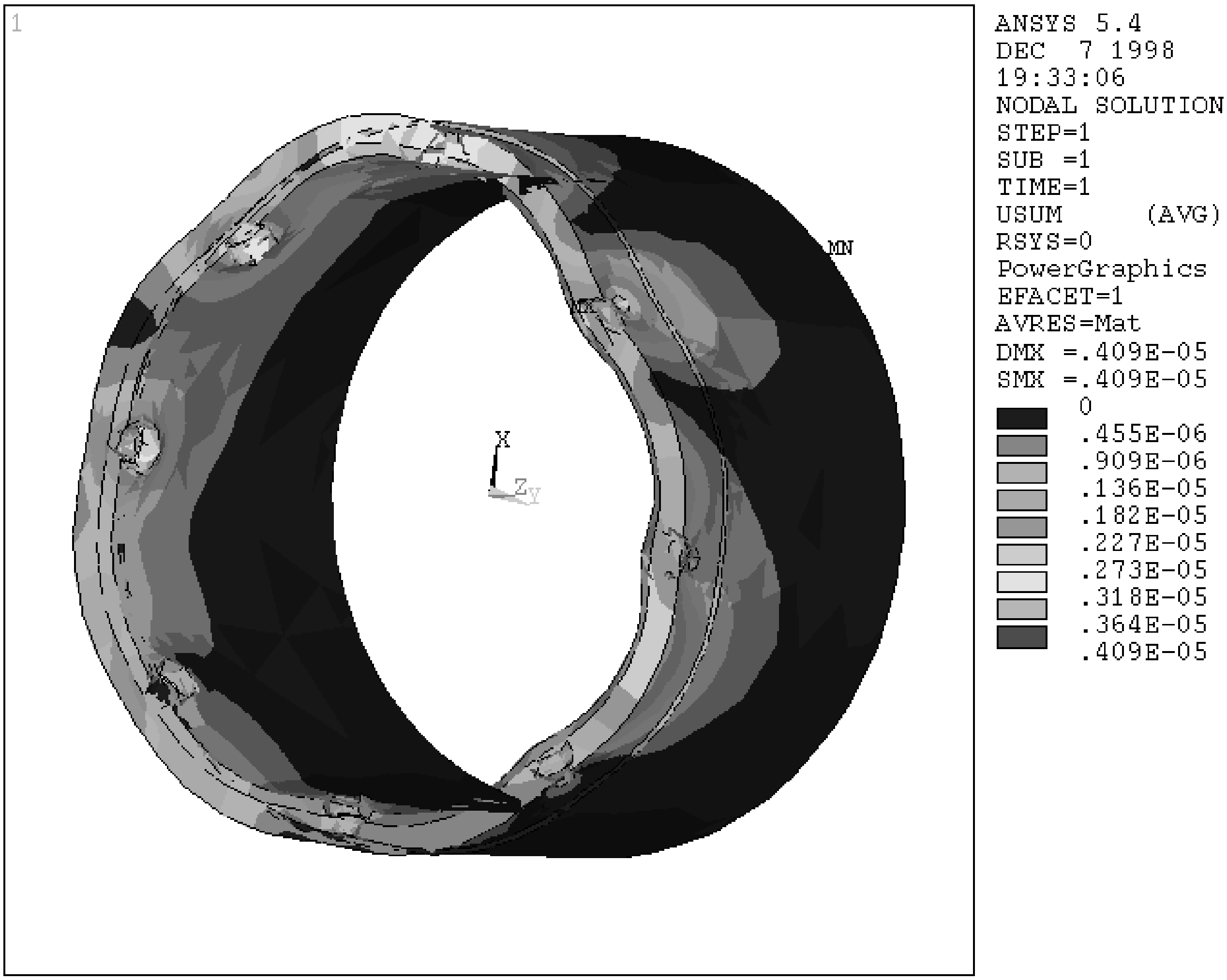
1. From the pressure distribution, the drag force for the model can be evaluated. By integrating the pressures across the boundary surface, forces in the X, Y, and Z can be determined. The force in the X direction is the drag.

## Structural Finite Element Analysis

The structural analysis was much simpler due to its integration into Pro/Engineer's finite element modeling package. Due to the very comprehensive static design of our submarine there is only one case that presented a worry to the stress of the structure. The Flotran analysis showed high pressures on the front of the submarine but this is negligible due to the strength of the PVC pipe and Polypropylene. The area that was determined in need of inspection involved the forces placed on the PVC from the internal pressure of 10 psi when the boat is sealed and not in motion. The thickness of the pipe is 0.4675" and .37" away from the edge of the pipe, a thin enough amount of material to require analysis for this pressure load. There were three cases involved, the 8" with 8 screws, 4" with 6 screws and the 3" with 6 screws as well. The worst case was determined to be the 8" situation due to a larger surface area of pressure per screw hole.

The loading conditions for the finite element analysis is as follows. There is a pressure of 10 psi on the screw hole of the PVC pipe and there was a displacement of zero for X, Y, Z on the other end of the part to act as a ground. The material was defined according to properties gathered from the Internet. The simplicity of this analysis is that ANSYS will solve the model without any other work. The following pages include several snapshots from the ANSYS Graphics Window that quells the argument of this designs structural integrity.

## Deformed Edge of the 8” PVC Pipe




**Deformed Edge of the 8” PVC Pipe**

**Explanations:**

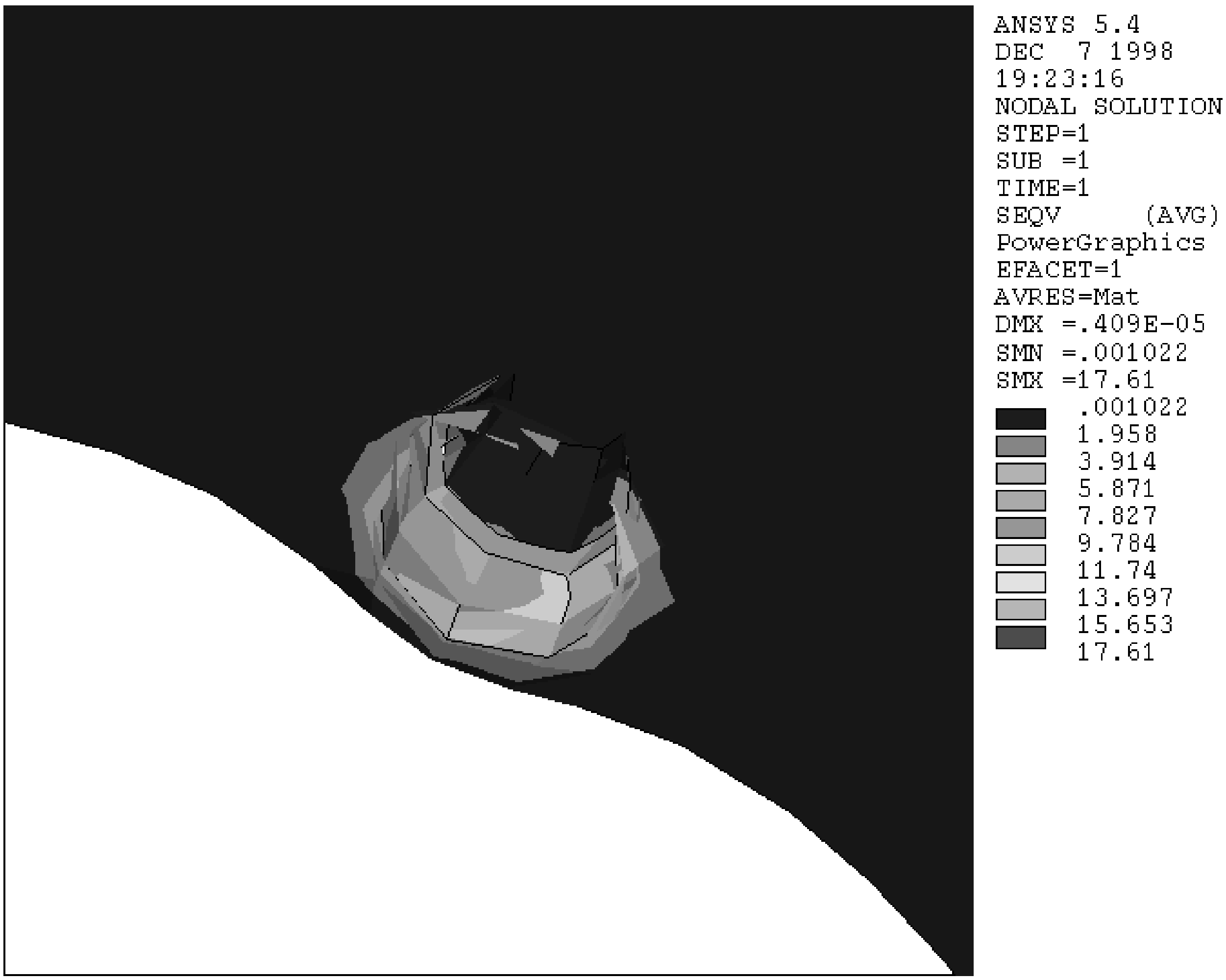
1. The graph is showing the displacement of the PVC pipe.
2. The edge is deformed because ANSYS exaggerates the displacement to show the direction of the material.

**Considerations:**

1.The Maximum Displacement is in the Red regions of the plot.

2. There are very few Red regions, even so, the displacement at those points is only .00000409 inches, thus resulting in negligible displacement.

## <u>A Closer Look at the Stresses</u>




**A Closer Look a the Stresses**

**Explanations:**

1. The surfaces that were loaded are currently Green regions, these regions were loaded at 10 psi, which is shown by the graph.

2. These pressures did not cause a pressure to go above 17.61 psi (minimal).

3. Once again, the edge is deformed due to the exaggeration mentioned previously

**Conclusion:**

1. There is no worry of failure or crack propagation from the design pressures.

## Bill of Materials

| Description | Quantity |
|---|---|
| Polypropylene 1.5" Thick | 2' X 2' |
| Polypropylene 2.5" Thick | 2' X 4' |
| PVC 80 8" Diameter | 5' |
| PVC 80 3" Diameter | 5' |
| PVC 40 4" Diameter | 5' |
| Hitec/RCD Servos | |
| HS-77BB | 4 |
| HS-605MG | 2 |
| HS-615MG | 2 |
| 4in Diameter Propeller | 1 |
| 540 28 turn | 1 |
| O-Ring(s) | 12 |
| Miscellaneous Hardware | - |

* Refer to Appendix C: Detailed Drawing for specific dimensions.

## Budget & Resources

Labor Costs

| Hours x 5 Jr. Engineers | Rate | |
|---|---|---|
| 700 | $20/hr | $14,000 |

Overhead (100% Labor)

| | | |
|---|---|---|
| | | $14,000 |

Materials and Supplies

| | | |
|---|---|---|
| Literature (Research) | $100 | |
| Supplies | $100 | |
| Materials: | | |
| Motors/actuators | $500 | |
| Propeller | $100 | |
| PVC Pipe | $95 | |
| Polypropylen | $355 | |
| O-ring seals | $50 | |
| Linkages | $100 | |
| Misc. | $100 | |
| | | $1,500 |

Machine Shop

| Hours | Rate | |
|---|---|---|
| 30 | $35/hr | $1,050 |

Computer Resources

**Lease of:**

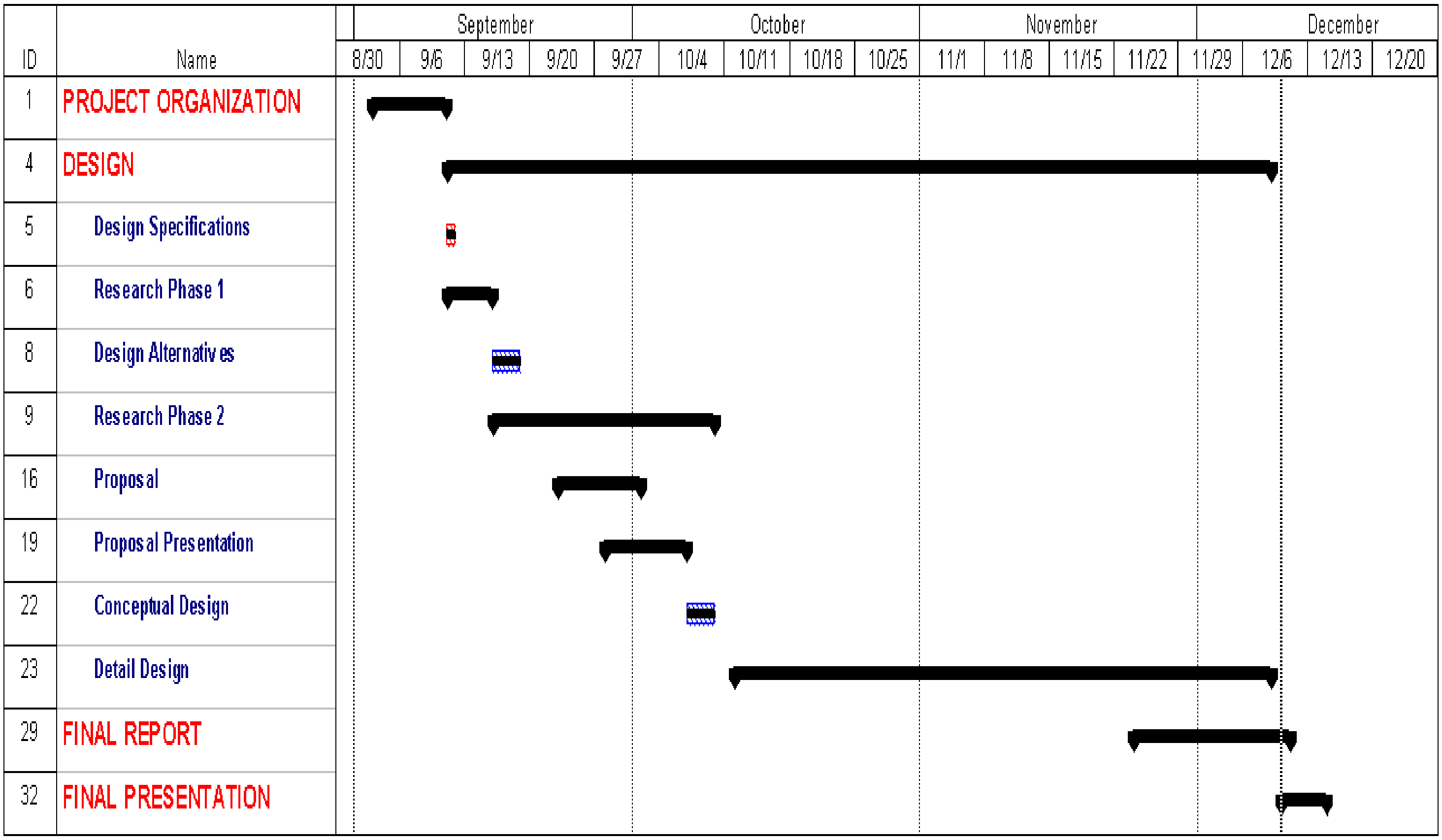
**3 DEC Alpha**

| Weeks | Rate | |
|---|---|---|
| 15 | $150/wk | $2,250 |

| Software: | Hours | Rate | |
|---|---|---|---|
| Flotran | 50 | $8/hr | $400 |
| Ansys | 50 | $10/hr | $500 |
| Pro/Engineer | 100 | $8/hr | $800 |

TOTAL **$34,500**

## Project Management




## Discussion and Conclusions

The project team has been successful in completing all of the objectives specified earlier and feels that the proposed entry will be a very competitive one. The design methodology was followed very closely and proved to be an excellent plan in that it broke down a complicated design into various stages. This allowed for a more scrupulous attention to detail in the design processes of the four key components systems. The Gantt Chart let the team know how far along or behind schedule they were and gave a good estimate in how much time had to be put into each specific task. The team feels that the research they have acquired has set a good foundation for a successful AUV even though several problems can be expected.

The biggest of these problems is going to be “trimming” the boat, which is making sure that the AUV is trimmed to 1lb of buoyant force in the water.  The other major problem will be the placement of the servos for the control surfaces.  Due to funding constraints, each control surface might not get its own servo but instead linkages would have to be incorporated.  The manufacturablity was considered in the detailed design so the team has already eliminated several foreseen problems, which will allow the team to concentrate on building rather than redesigning in the upcoming semester.

## Acknowledgements

Our senior design group would like to thank our Faculty Advisor Professor Thangam, for giving us the opportunity to work on the Design Autonomous Underwater Vehicle Project, and for his insight and guidance into helping us complete the project.  Also, we would like to thank Professor Pochiraju for his help in understanding ANSYS/Flotran.  Thank-you to Doctor Paul Spense for his insight into Naval Architecture.  Thank-you to Tony Healey, Director of the Phoenix Project.  Thank-you to Dean Gallois for his continued support in the AUV Competition.

## Appendix A: Governing Equations and Sample Calculations

| Variable | Description |
|---|---|
| $C_d$ | Coefficient of Drag |
| $C_{do}$ | Coefficient of Drag ( infinite span) |
| $C_l$ | Coefficient of Lift |
| AR | Aspect ratio |
| $AR_{eff}$ | Effective Aspect Ratio |
| c | Chord Length |
| s | Fin Span |
| $A_p$ | Planform Area |
| $A_f$ | Area Perpendicular to Flow |
| L | Lift |
| $D_{tx,y}$ | Total Drag in x or y direction |
| $D_{hx}$ | Drag on Hull in horizontal direction |
| $D_{hy}$ | Drag on Hull in vertical direction |
| $D_f$ | Drag on Forward Rudder |
| $D_s$ | Drag on stern Rudder |
| $D_d$ | Drag on Dive Planes |
| $D_p$ | Drag on Pitch Planes |
| T | Thrust |
| $a_x$ | Acceleration in the horizontal direction |

| | |
|---|---|
| $a_y$ | Acceleration in the vertical direction |
| m | Mass of AUV |
| B | Buoyant Force |
| $\alpha$ | Angle of Attack of hydroplane |
| x | Distance from Center of Gravity of Sub |
| $M_{z,COG}$ | Moment about Z-axis on Center of Gravity |
| $\rho$ | Density of Water |

**Motion Equations**

$\Sigma F_x = T - D_t = ma_x$

$\Sigma F_y = B + L + D = ma_y$

$\Sigma M_{z,COG} = L_d x_d - L_p x_p - M_{misc} = 0$

Drag Equations

$D_t = D_h + D_f + D_s + D_d + D_p$

$C_d = C_{do} + (C_l^2/A_f*AR)$

$D = C_d * 0.5 * V^2 * A_f * \rho$

Lift Equations

$C_l = 2 * \Pi * \sin(\alpha)/(1+(2/AR))$

$L = C_l * 0.5 * V^2 * A_p * \rho$

**Moment Calculation on a Servo**

$M = C_m *0.5 * \rho * V^2 * A_p$

**Lift on Dive Planes**

$AR = s/c = 6/4 = 1.5$

$AR_{eff} = 2 * AR = 3$

$C_l = 2 * \Pi * \sin(\alpha)/(1+(2/AR_{eff})) = 2 * \Pi * \sin(12) / (1+(2/3) = 0.720313$

$L = C_l * 0.5 * V^2 * A_p * \rho = 0.720313 * 0.5 * (3.25)^2 * 24 * 2 * 1.97 = 2.5\text{lbs}$

**Drag on Dive Planes**

$C_d = C_{do} + (C_l^2/A_p*AR_{eff}) = 0.1 + (0.720313^2/ ((24*2/144) * 3)) = .6188$

$D = C_d * 0.5 * V^2 * A_p * \rho = .6188 * .5 * (3.25^2) * (24*2 /144) * 1.97 = 2.14\text{lbs}$

Location of Pitch Planes

$\Sigma M_{z,COG} = L_d x_d - L_p x_p - M_{misc} = 0 = .775(8) - .225(x_p) - 0 = 0$

$x_p = 27.556\text{in}$

**Propeller Calculations**

3-Blade
Pd = delivered power
Ps = power to the shaft
Va = speed of advance
P/D = Pitch ratio
D = Propeller diameter (m)
n = rev. per min
Cb = block coefficient
Bp = Taylor's propeller coefficient
d = Taylor's advance coefficient
ho = open-water propeller efficiency

1 knot = 0.515 m/s
Pd = 0.98 Ps
Bp = (1.158 * n * (Pd0.5)) / (Va2.5)
d = (3.281 * n * D) / Va

Input

| | | | | | |
|---|---|---|---|---|---|
| Ps = | 70 | W | | 0.07 | kW |
| Pd = | 68.6 | W | | 0.0686 | kW |
| V = | 1 | m/s | = | | |
| Va = | 0.8 | m/s | = | 1.5534 | knots |
| n = | 1500 | rpm | | | |

Calulations
Bp = 151.27

From Typical Taylor Propeller Characteristic Curves (pg. 11-60, Mark's Handbook)

| | | | |
|---|---|---|---|
| ho = | 0.5 | | |
| P/D = | 0.5 | | |
| d = | 290 | | |
| D = | 0.0915 m | 3.6037 | in |

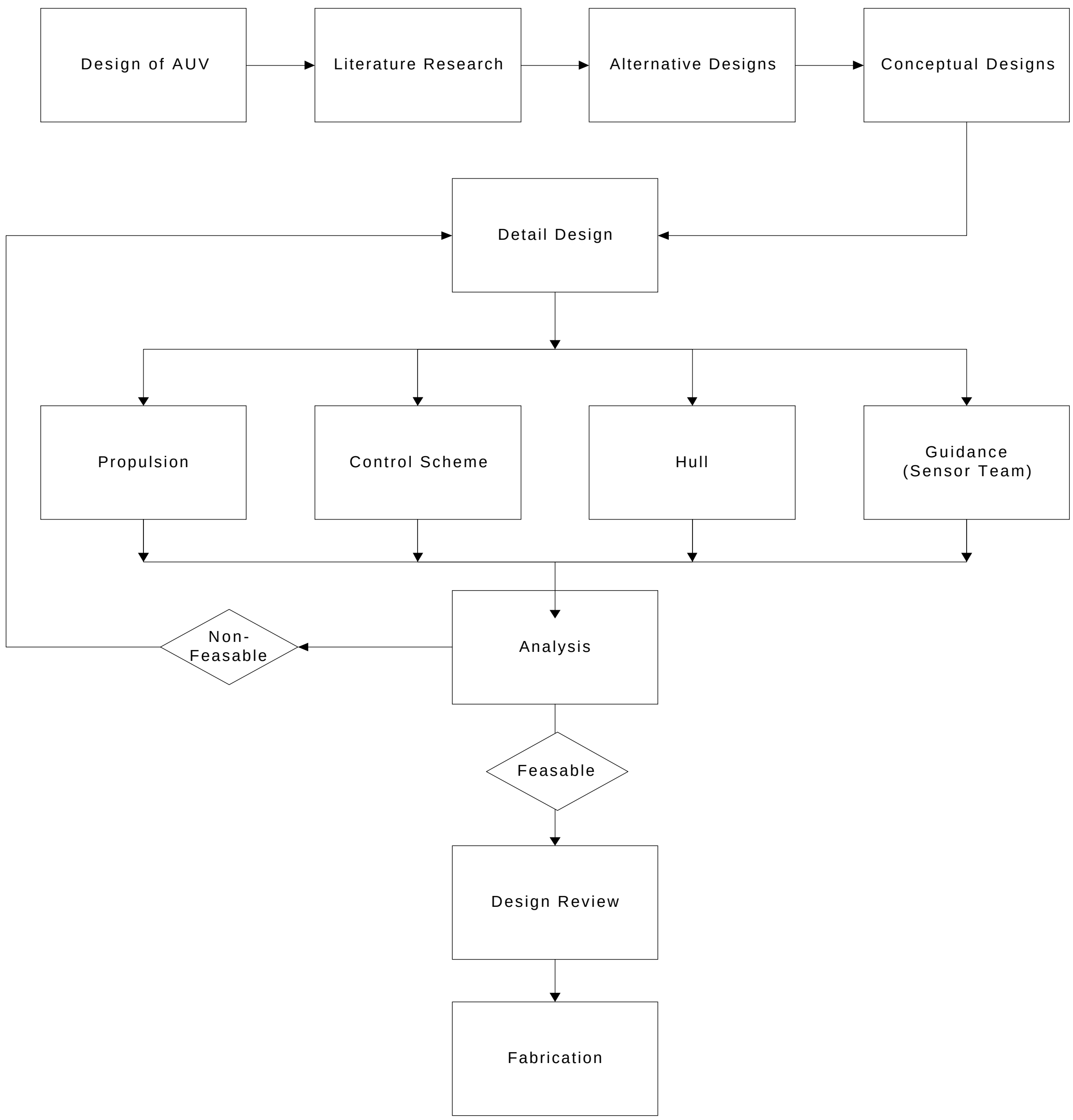
4 Blade Prop

| | | | |
|---|---|---|---|
| P/D | 0.5 | | |
| D4Blade = | 0.97*D3-Blade | | |
| | 0.0888 m | 3.4956 | in |

### Calculation of Moment of Control Surface

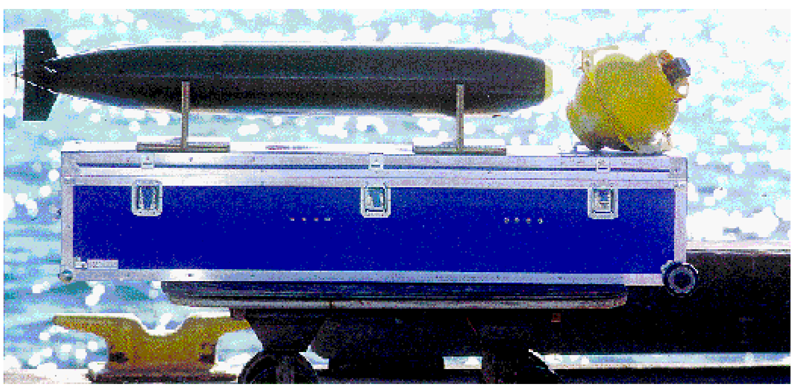
| | | |
|---|---|---|
| Cm | 0.23 | |
| density | 998 | kg/m^3 |
| Velocity | 1 | m/s |
| Chord | 4 | in |
| Span | 6 | in |
| Plane Area | 24 | in^2 |
| Factor of Safety | 2 | |
| Moment | 51.14 | in*oz |

## Appendix B: Flow Chart

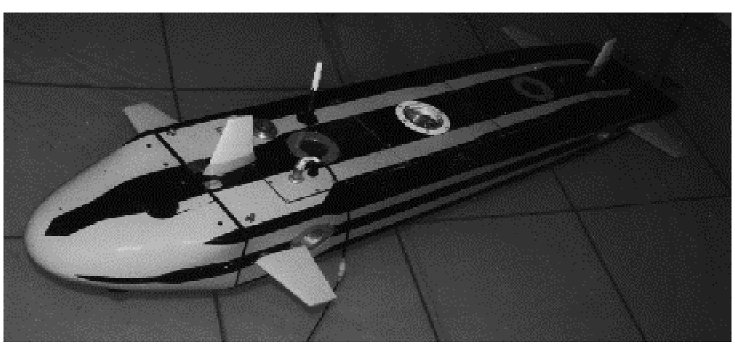


**Appendix C: State of the Art**


OTTER

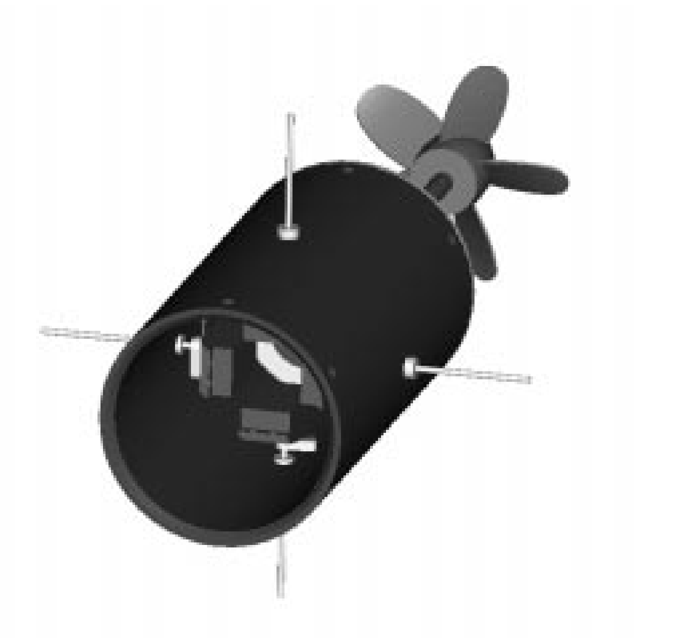


REMUS

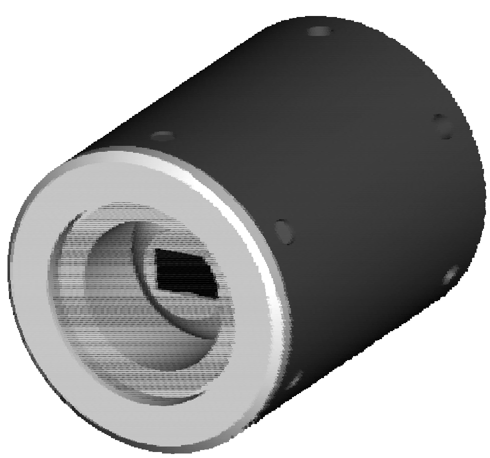


Phoenix

<u>**Appendix D: Renderings and Drawings**</u>



## Rear View of Tail Pressure Hull



## Front Pressure Hull

## References


1. Abbott, Ira H., Theory of Wing Sections. Dover Publications, Inc., New York,1959.
2. Avallone, Eugene A. Mark's Standard Handbook for Mechanical Engineers. McGraw-Hill, New York, 1978.
3. Breslin, John P., Hydrodynamics of Ship Propellers. Cambridge University Press, New York, 1994.
4. Bruggen, Norbert., Model Submarine Technology.
5. Burcher, Roy, Concepts in Submarine Design. Canmbridge University Press, New York, 1991.
6. Hoerner, Sighard F., Fluid-Dynamic Drag. Hoerner, New Jersey, 1965.
7. Lewis, Edward V., Principles in Naval Architecture. Society of Naval Architects and Marine Engineers, Jersey City, 1998.
8. Tara, Abdullah S., The Lift of Symmetrical Hydrofoils, of Effective Aspect Ratio 3, at Reynolds Numbers from $1.14x10^4$ to $4.25x10^4$ . Master's Thesis, Steven's Institute of Technology, 1959.
9. White, Frank M. Fluid Mechanics. McGraw-Hill, New York,1994.
10. www.soc.soton.ac.uk/OTD/asub/labels.html
11. www.erols.com/auvsicc/auvcomp.htm
12. www.acim.usl.edu/ACIM/auv.html