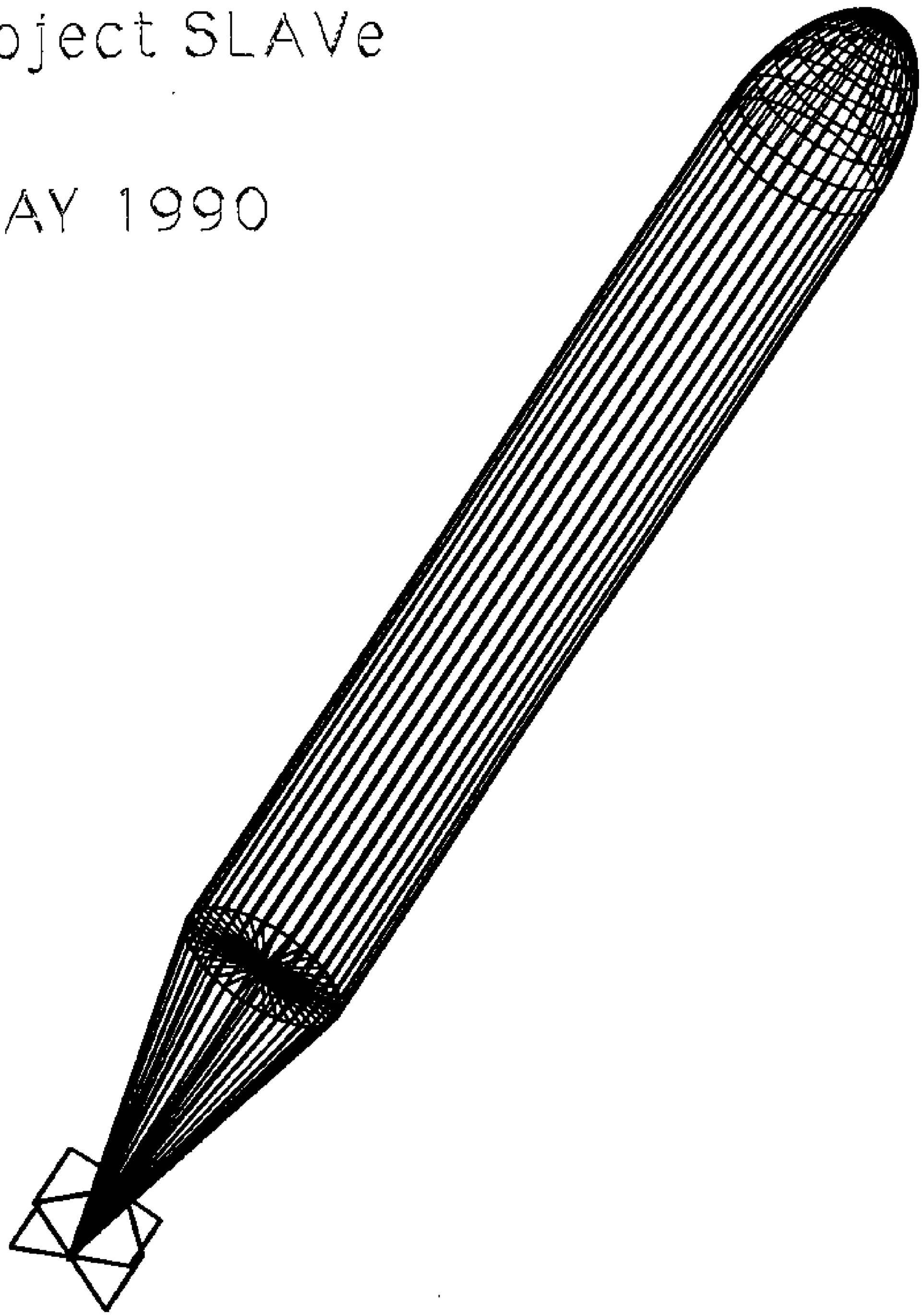
Project SLAVe

MAY 1990



University of New Hampshire
Ocean Projects Course (TECH 697)

SLAVe

SUBMERSIBLE LONG-RANGE AUTONOMOUS VEHICILE

BY

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FINAL REPORT

Submitted to University of New Hampshire
in Partial Fulfillment of
the Requirements of Undergraduate Ocean Research Program

Technology 697

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DEDICATION

The following report is dedicated to Dr. Gerald Sedor.

It was his enthusiasm and extraordinary effort

which enabled this team to survive the rough spots.

Thank you, from all the team members of project SLAVe.

ACKNOWLEDGEMENTS

Project SLAVe wishes to acknowledge the following people, corporations and businesses for their generous donations: Eptam Plastics for their donated time and effort in the construction of our pressure vessel. Motorola for their donation of computer boards. D.G. O'Brien for their donation of water tight connectors. Eagle Picher for their donation of batteries. MSEL for their donation of thrusters and transducers. Mr. Lynn Darnell for his extensive and valuable time and advice. Bob Doucette and Bob Blake for patience and advice which enabled the construction of our tail section. Mark Ragonese for his help in the wood shop. Paul Lavoie for allowing us to test the pressure vessel in the hyperbaric chamber. Jim Irish for his donation of sènsors. A special acknowledgement is make to the Sea Grant College Program for providing the necessary support that enables this course to offered at the University of New Hampshire.

PREFACE

The following is a list of personnel and their individual responsibilities who contributed to the success of project SLAVe.

Students:

Christian Antonucci (EE)	Power Generation and Distribution Command, Control, Communications
Jesse Jost (ME)	Propulsion Systems
Cheryl Kosegarten (ME)	Maneuvering Systems
Mark Morrison (ME)	Project Leader
Frederick Murdock (ME)	Hull and Fairing Structures
Pamela Wilcox (EE)	Navigation Computer Processing

Faculty:

Dr. Gerry Sedor	Advisor
Prof. Allen Drake	Course Director
Prof. Larry Harris	Assistant Course Director
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INTRODUCTION

Seventy five percent of our earth's surface is covered by water. It seems ironic that despite this vast territory, we know very little about it. Project SLAVe (Submersible Long-range Underwater Autonomous Vehicle) is our attempt to try to acquire some knowledge of how and why our underwater environment works.

For many years the United States Navy has been a major source of underwater data. They acquire this data via the use of many types of ocean platforms and vehicles, including very large and costly manned submarines. In general, the Navy acquires this data for military related purposes. The leaves scientists with only two options: 1.) To study only unclassified military related data, or 2.) To build their own data gathering platforms and vehicles and retrieve data for themselves. The draw-backs of cost and safety associated with manned submersible vehicles have led to increased interest in alternative ways of gathering undersea information.

The first alternative considered was the ROV (Remotely Operated Vehicle), which is an unmanned submarine remotely operated via a tether. The major drawbacks of a ROV are the limitations imposed by cable control and drag and the extensive cost of operating a surface ship to control the ROV.

The most recent alternative has been the development of the AUV (Autonomous Underwater Vehicle). An AUV is a programmed submersible which is untethered and needs no surface ship while in operation. AUV's are still very much in the development stage. As of 1987, there were only 27 AUV's in the world [1]. The figure has since more than doubled, but is still inadequate to satisfy demand.

The number of potential tasks such a vehicle could perform are endless. Some potential missions that an AUV could theoretically perform are: bottom mapping and survey, under-ice profiles, temperature profiles, bottom object search and identification, examination of such scientific phenomena as ocean thermal vents, ocean ridges, etc. The United States Navy would be very interested in obtaining a temperature profile of the ocean, which can be used to camouflage their submarines. Geologists would be interested in having topographical maps of ridges in the continental shelf to aid in their study of platetechtonics. Oil drilling corporations could save money by observing the ocean bottom without the use of deep sea divers.

The above factors considered, there is a definite need for AUV's. However, there is very little information on AUV's available because it is such a new field. The purpose of project SLAVe is to put a dent in the this expanding field by designing and building a protype AUV which could be adapted for extended duration missions associated with ocean data gathering.

A major part of this project involved the integration of design efforts of the individual project team members. Vehicle design is an iterative process which starts with the definition of a mission and mission requirements. Mission requirements were provided by the advisor, and lead to the development of vehicle system and subsystem requirements and the identification of alternative designs to meet the requirements. Evaluation factors were established for evaluating candidate systems and an optimization process developed to select an optimum design. For each of the vehicle's major subsystems an attempt is made in this report to describe the system design criteria established for that subsystem, the system functional requirements, alternative designs considered and a brief description of the evaluation process used to select the final design. A description of the selected system design is also provided for each system, along with supporting data and figures. Budget and schedual information is provided in the appendices.

MISSION OBJECTIVES

The AUV system under consideration is to be designed for a multiple mission capability using a modular approach. A mission module will interface with vehicle systems in such a manner as to provide all components which are unique to a selected mission. Other missions are accommodated by modifying or replacing the mission module. The generic missions for consideration include the following:

a. Ocean bottom surveys and mapping

b. Conductivity, temperature and density profiles

c. Under ice profiles

d. Ocean bottom object search and identification

e. Examination of ocean scientific phenomena

To investigate the feasibility of the AUV system to accommodate missions of this type, a prototype AUV will be designed with a specific mission module to demonstrate the mission described below:

Demonstration Mission: To obtain scientific data consisting of a temperature profile over a large area in a fresh water lake with a maximum depth of 200 feet.

The major hurdles of accomplishing the misssion objectives where that of time and money. Appendices A and B show how the team memebers of project SLAVe over came these limitations.

VEHICLE CHARACTERISTICS

Length of nose cone	16.625 inches
Length of tail cone	32.000 inches
Length overall	12.750 feet
Diameter overall	16.625 inches
Vehicle weight in air	400 Lbs
Propulsion system	2 externally-mounted electric thrusters
Energy system	8 Lead-Acid batteries (Eagle Picher)
Fairing material	Fiberglass
Pressure vessel	PVC, 1/2 inch thick, 8 ft long overall
Maneuvering	4 control surfaces, stern-mounted
Vehicle velocity	2-5 knots
Design depth	200 feet (88 psi)

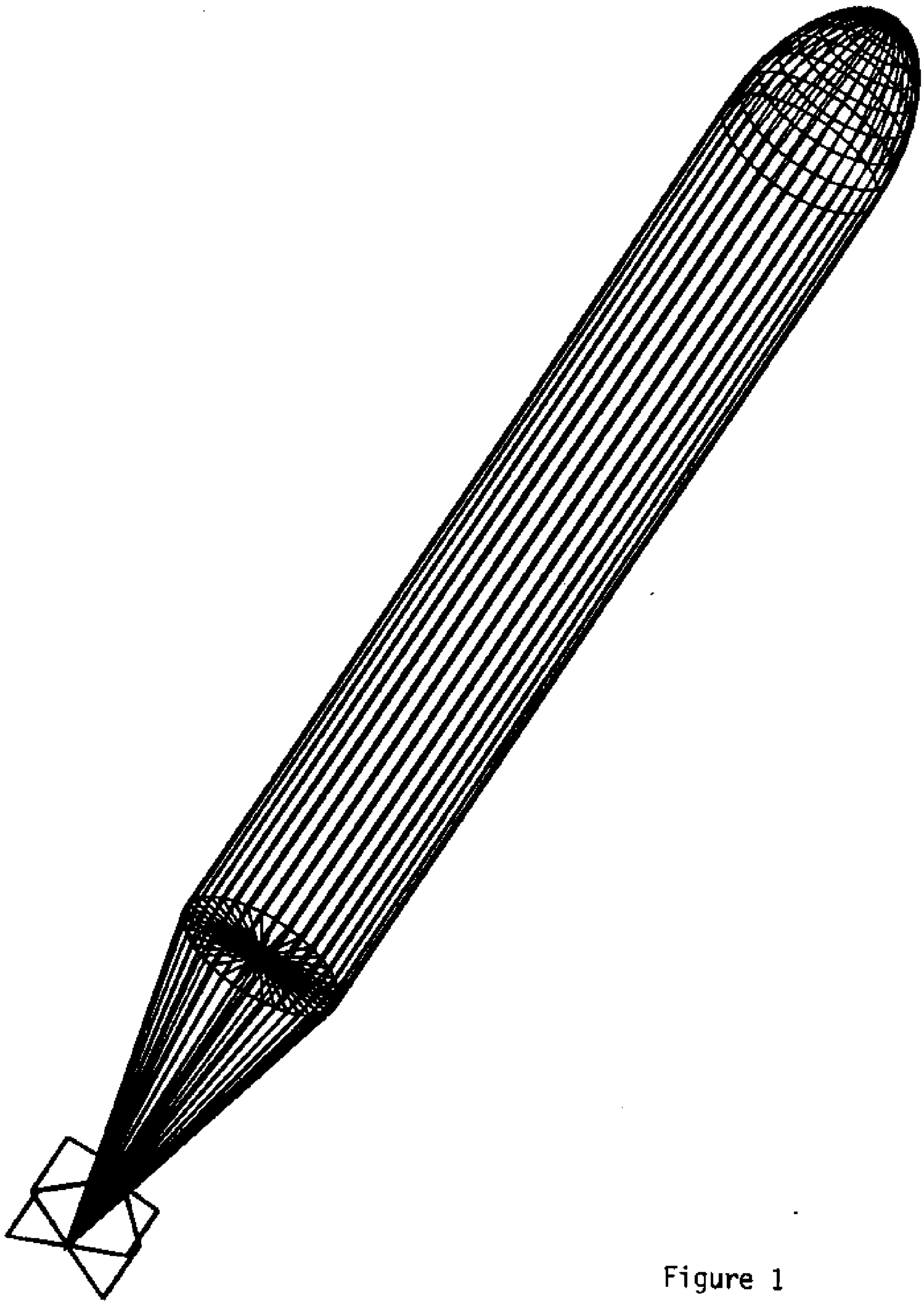


Figure 1

Assembled Fairing with Control Surfaces

Mechanical Engineering-UNH

Frederick Murdock

Scale 1:17

PROPULSION SYSTEM

I. System Design Criteria:

The criteria used in evaluating the propulsion systems include:

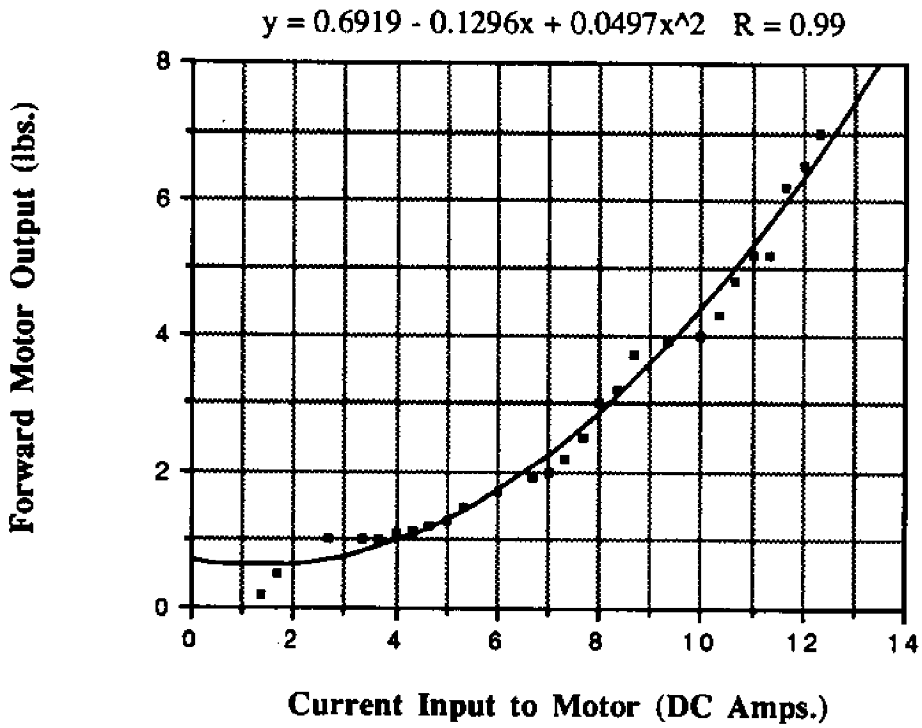
1. Efficiency
2. Power and speed requirements
3. Reliability
4. Simplicity
5. Maintainability
6. Cost

II. System Functional Requirements:

The propulsion system can be divided into three major categories: power generation, power transmission, and thrusters. Power generation is accomplished by the use of rechargeable secondary batteries which must be kept dry and at atmospheric pressure and are housed inside the main pressure hull. The energy output of the batteries must meet the criteria established in the system functional requirements. The power generating system design is described in a seperate section of this report, while power requirements for propulsion are developed in this section.

Power transmission is accomplished by electrical cables that lead from the batteries in the pressure vessel via watertight connectors to the thrusters which are attached to the control surfaces. The cables also must meet the established criteria.

The thrusters are borrowed from a currently unused EAVE vehicle developed by the Marine Systems Engineering Lab (MSEL) of the University of New Hampshire, and are manufactured by Minnesota Electric Technology Inc. of Winnebago, Minnesota. The thrusters can produce forward and reverse motion, depending on the amount of current drawn. Tests show that thrust is approximately proportional to current (see Figure 2).



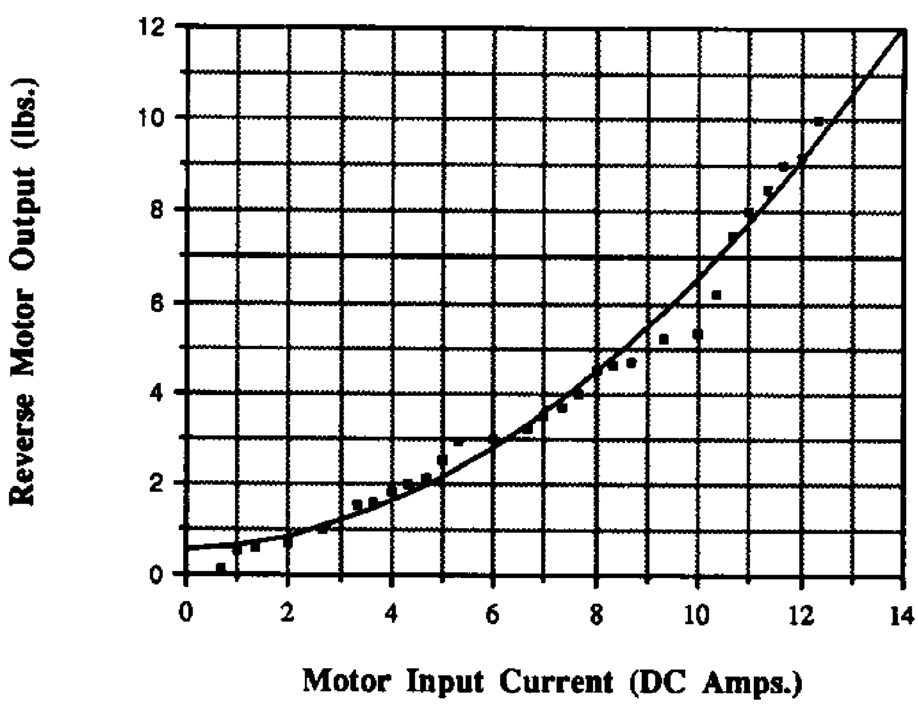


Figure 2: Thruster Current vs. Output Thrust

III. Alternative Design Concepts:

Propulsion is "... a system capable of generating a thrust force to push or propel a vehicle forward" [2]. The types of propulsors generally used on small submersibles include [3]:

1. Open propellers
2. Ducted or shrouded propellers
3. Contrarotating propellers
4. Water jets
5. Oscillating foils
6. Controllable pitch propellers
7. Stator-propeller systems
8. Vane wheel/propeller systems

The propulsor configurations listed above can be used in single or multiple arrangements, with one or more propeller blades, and can be located internally or externally. Also, some of the above categories can be combined to produce an arrangement most suitable for the system functional requirements. All of the above configurations use propellers in some form, except water jets and oscillating foils. A water jet is comparable to a propeller since it produces an output jet velocity. An oscillating foil (e.g. a fish tail) is currently being researched and has not been developed to the point where it can compete with propellers and it will not be included in this analysis [3]. In propulsor analysis, the important criteria include:

1. Efficiency
2. Reliability
3. Maneuverability

4. Complexity of fabrication and installation

5. Torque compensation requirements

Efficiency is a function of many variables, such as propeller diameter, velocity of the resulting water jet, hull efficiency, etc. The diameter of the propeller is limited by the diameter of the hull. By keeping the propeller diameter smaller than the hull diameter, the propeller is somewhat protected from damage in case the submersible travels too close to submerged objects or the bottom of the body of water. Applying shrouds, ducts, stators, or vane wheels increase the velocity of the water jet and offer a higher degree of propeller protection. Contrarotating propellers provide torque compensation advantages over single propeller configurations. A comparison of the relative efficiencies of various propeller alternatives is given in Figure 3 [3]. Advantages and disadvantages for each of the various alternatives is given in Figure 4 [3]. Maneuverability is discussed in the Maneuvering System described elsewhere in this report. Torque compensation can be provided by contrarotating thrusters or a slight skewing of the vertical control surfaces.

The results of the propulsor analysis are:

1. A screw propeller configuration is generally the optimum propulsor for most vehicle systems.

2. Contrarotating propellers and ducted propellers with stators offer the highest efficiencies of the various feasible systems.

3. Ducts and shrouds can be used to improve efficiency while providing damage protection.

4. Stators and vane wheels can be used to improve propeller efficiency [3].

A power transmission subsystem transmits energy from the power source to the thrusters. Types of power transmission systems include [3]:

1. Direct mechanical drive with reduction gear
2. Direct mechanical drive without reduction gear
3. Hydraulic pump and motor system
4. Electrical system

Mechanical systems are generally limited to large vehicles which have a prime mover, such as a steam turban or a diesel engine [3]. Between the electrical and hydraulic systems, the electrical system generally offers higher efficiency, higher reliability, lower maintenance, and lower cost. The electrical system also provides a greater number of arrangement possibilities. For a vehicle with a direct conversion energy source, such as a battery or a fuel cell, and no other demands for a hydraulic system, the selection of an electrical transmission system appears to be optimum [3].

IV. Selected Design:

An electrical propulsion system was judged optimum for this design due to its simplicity, reliability, and flexibility. With no moving parts, the electrical cables transmit energy directly from the batteries in the pressure vessel to the thrusters on the horizontal control planes. Two thrusters

RELATIVE EFFICIENCIES OF VARIOUS PROPULSOR SYSTEMS

SYSTEM	EFFICIENCY (NOTE 1)	EFFICIENCY IMPROVEMENT OVER SINGLE OPEN PROP
Single, open propeller	0.733	---
Ducted propeller	0.746	1.8 %
Open propeller with stator	0.779	6.3 %
Open propeller with vane-wheel	0.779	6.3 %
Ducted propeller with stator	0.797	8.7 %
Contrarotating propellers (2)	0.803	9.6 %
Actuator disk (theoretical upper limit)	0.874	19.2 %

NOTE: 1. All efficiencies are based on a thrust coefficient (C_T) of 0.6625, based on the diameter of a single open propeller.

Figure 3

COMPARISON OF PROPULSOR SYSTEMS

SYSTEM	ADVANTAGES	DISADVANTAGES
Open Propeller	- Simplicity - Low fabrication cost	- Low efficiency - Lower reliability than ducted or shrouded
Ducted Propeller	- Improved efficiency - Higher thrust at low speeds - Duct protects propeller	- Added complexity - Fabrication tolerances - Duct adds drag - More frequent maintenance
Water Jet	- Rotating impeller is protected - All components can be inside vehicle for ease of maintenance	- Possible loss in efficiency - Added weight & volume of pumps
Contrarotating Propellers	- Torque compensation - High efficiency	- Increase in system complexity - Fabrication: added difficulty & cost - Maintenance: increased
Controllable Pitch Propellers - Tandem - Trochoidal	- Excellent maneuvering & position keeping	- Added complexity - Fabrication: added difficulty & cost - Maintenance: increased - Control system: added complexity
Propeller with Inflow Stator	- Improved efficiency - Relatively simple to fabricate	- Added drag of stator - Slight increase in cost
Propeller with Vane – Wheel	- Improved efficiency	- Added complexity - Fabrication: increased cost & difficulty - Maintenance: increased

Figure 4

with single, open propellers were selected since they were readily available at no cost. Lowered efficiency is not a significant problem for this prototype vehicle since the thrusters effectively provide the necessary thrust.

V. Analysis of Power Requirements:

The most common method used in determining propulsion shaft horsepower, given a vehicle configuration and speed, has been developed by Sighard Hoerner and applied to the ITTC Convention [4]. Although the results are not exact, past experience indicates that it produces reasonable results. Accurate data can be produced by building a scale model and using dynamic similitude. The analytical algorithm used for the calculations follows:

1. Assume an initial vehicle diameter, D

2. Determine the vehicle's length to diameter ratio (generally between 5 and 9 for optimum drag):

$$\frac{L}{D}$$

3. Find the average Reynolds number,

$$Re = \frac{V\,L}{\nu}$$

where V is vehicle velocity, L = vehicle length and ν is kinematic viscosity of water.

4. Determine the friction drag coefficient C_f, using the ITTC convention (Internaltional Towing Tank Conference of 1957):

$$C_f = \left[\frac{0.075}{(\log_{10} Re - 2)^2}\right] + \Delta C_f$$

where ΔC_f is the roughness allowance.

5. Solve for the total drag coefficient C_T, using Hoerner's relationship [4]:

$$C_T = C_f\left[1 + 1.5\left(\frac{D}{L}\right)^{1.5} + 7\left(\frac{D}{L}\right)^{3}\right]$$

6. Determine the total resistance of the vehicle,

$$R_T = \frac{1}{2}\rho A_{ws} V^2$$

where A_{ws} is the total wetted surface area, ρ is water density, V is vehicle velocity.

7. Calculate the effective horsepower,

$$EHP = \frac{R_T V}{550}$$

8. Assume a propulsive coefficient (generally between 0.60 and 0.80 with 0.65 being the most commonly used),

$$SHP = \frac{EHP}{PC}$$

VI. Conclusions:

The propulsion system must have high efficiency, reliability, and simplicity, low cost and maintenance, while providing the necessary power and speed requirements. Power transmission for small, unmanned submersibles generally use hydraulic or electrical systems. Because of higher efficiency, higher reliability, and lower maintenance requirements, the electrical system was judged optimum. The major factors in selecting the type of propulsor was cost, and availability. While not providing the maximum efficiency and protection, the thrusters are judged satisfactory for the propulsion system requirements. Shaft horsepower required to propel a given vehicle at a given speed can be estimated with reasonable accuracy through the use of Hoerner's relationships and the ITTC convention.

POWER GENERATION SYSTEM

I. System Design Criteria:

The design criteria estblished for the selection of a power generation system for the SLAVe vehicle include the following:

1. Safety
2. Endurance to meet mission requirements
3. Sizing for installation in pressure vessel
4. Low weight relative to power or energy
5. Overall system performance versus cost
6. High operational reliability
7. Relative ease of maintenance and/or replacement

II. System Design Considerations:

This particular subsystem must store and covert energy to a usable form to move the submersible in the x-y-z plane and to power the on board diagnostic equipment. The design must provide a suitable power source to complete the mission goal of temperature data collection over a range of 200 nautical miles. The submersible must be completely autonomous, therefore the power source must be completely independent of the outside environment. The power supply must isolated from the external environment and therefore must be either placed inside of the pressure vessel or inside a pressure compensated container. The submersible, being subject to rotational forces that may invert the vehicle about any of its axes, must have a power supply that can withstand these movements. The three basic types of energy storage devices considered appropriate for this application are chemical, thermal and nuclear.

III. Design Alternatives:

After considerable time and research into power portable power sources the following systems were considered:

A. Direct Conversion Systems

1. Electrical Storage Batteries

    a. Primary (mechanically rechargeable)

    b. Secondary (Electrically rechargeable)

2. Fuel Cells

B. Thermal Conversion Systems

1. Heat Engines

    a. Steam Engines

    b. Gas Turbines

    c. Diesel Engines

    d. Stirling Engines

2. Heat Sources

    a. Thermal Energy Storage (TES)

    b. Nuclear Sources

    c. Chemical Sources

C. Mechanical Energy Storage

1. Flywheels

2. Pressurized Gas

IV. Design Selection:

The large field of choices was narrowed down after careful consideration. The Thermal Conversion Systems were ruled out due to their complexity and high cost. Since the thermal conversion systems provided some of the highest efficiencies and excellent energy-to-weight ratios, it is unfortunate that they could not be utilized. The remaining two categories considered were Direct Conversion Systems and Mechanical Energy Storage Systems. The systems then evaluated for this application were (1.) Electrical Storage Batteries, (2.) Fuel Cells, (3.) Flywheels, and (4) Pressurized Gas.

After analysis of the positive and negative aspects of each system the choices were again narrowed down. The two final canidate systems were a fuel cell which utilizes aluminum and oxygen and secondary storage batteries. Of the two systems the fuel cell was evaluated as the optimum choice. The high energy density (12.96 W Hr./cu. cm.) and the ability to be mechanically recharged in approximately thirty minutes makes this system highly attractive. The disadvantage of this system is that it is out of the budgetary capabilities of this project. The possibility of borrowing an existing fuel cell from another organization was researched.

The second choice was deep-cycle lead-acid storage batteries. Lead-Acid batteries provide an inexpensive source of power that can be easily obtained from local suppliers. They are very safe in that they are totally enclosed and will operate even if inverted. The recharging of these batteries can be done using existing equipment in UNH laboratories. The disadvantage of lead-acid batteries is their low power density(1.02 W Hr./cu. in.). The low power density would limit the running time of the mission.

V. Implementation of Design:

Research was done into the availability of the power supplies that were designated in the original research. Many different vendors were contacted and they each had their own opinions on the types of batteries that should be used. Due to the cost factors and the need for a readily available power supply that could be implemented in the designated time period a decision was made to use sealed, maintenance-free, lead-acid batteries.

Due to budget limitations for the entire project the various vendors of lead-acid batteries were queried regarding the possibility of donating the batteries. Calls were made to battery companies within the local area but a positive response could not be found. The search spread to encompass any lead-acid battery dealer in the United States. At the suggestion of various personnel at the UNH MSEL Laboratories Eagle-Picher Industries in Seneca, Missouri was contacted. Their reputation for reliable batteries and their excellent customer support network made them seem like a logical choice.

After a phone conversation with one of the Eagle-Picher sales representatives, Mr. Rex Biggs, and six weeks of waiting, Eagle-Picher donated nine lead-acid storage batteries. The batteries donated were eight CFM12V33 batteries(12 volt, 33 amp hours), and one CMF12V18 battery(12 volt, 18 amp hours). These batteries were included in the vehicle design to supply power to the thrusters, the processor boards, and the control surface actuator motors. These batteries are sealed, rechargeable, maintenance-free, lead-acid batteries. They do not need to have water added to them and they cannot be spilled unless they are subject to high impact. The charging is simple and can be done with a standard battery charger. The use of these batteries is a safe and effective way to provide energy for the various subsystems involved with the project. For further information see Appendix C.

CONTROL SYSTEM INTERFACE

I. System Design Criteria:

The design criteria established for the selection of control system interfaces include the following:

1. Provide an interface for the computer outputs to control surface actuators
2. Provide an interface for the computer outputs to thruster motors
3. Provide adequate power gain to drive motors
4. Maximize power transfer from energy source to vehicle actuators
5. Provide adequate heat sink for optimum operation
6. Size constrained for installation within pressure vessel
7. High operational reliability

II. System Design Criteria:

This particular subsystem must provide an interface between the computer and the actuators that control the movement of the submersible. The design must provide sufficient power gain to run the motors at their peak efficiency. The system must be highly efficient due to the limited energy source of an autonomous vehicle. The design must take into account the types of output that a microprocessor board can provide and it must then make up the difference between that output and the power needed to drive the motors. The system should be designed to produce a reasonable and predictable failure mode in case of component failure. The system should be versatile enough to accept a variety of motors within a limited range near the original design.

III. Design Alternatives:

After research into the various types of control systems, and discussions with members of the university faculty and staff [Professor Gordon Kraft, Mr. Dick Jennings, And Mr. David Miller (UNH MSEL Laboratories)] the basic types of control systems were defined as listed below:

A. Proportional with no feedback

B. Proportional feedback

C. Proportional-integral feedback

D. Proportional-derivative feedback

E. Proportional-integral-derivative feedback

F. Lead / lag compensation

G. Lead compensation

H. Lag compensation

IV. Design Selection:

The original control system that was designed for the submersible was a system that utilized a proportional feedback system. The simplicity of the system, the moderate speed of the submersible and the long time delay involved in the submersible's maneuvering enabled the project to implement the controller using proportional feedback. The original system controlled the movement of the submersible using DC servo motors that used feedback from three potentiometers mounted on each motor shaft within a pressure vessel. The output of the three potentiometers was averaged to provide for mechanical failure of the device. This design was never implemented due to the difficulty in manufacturing the watertight container to isolate the servomotor from the aquatic environment. The problem arose with the manufacturing of a rotating shaft seal for the motor shaft. The servo motor provided a high torque output(40 in. lbs) at 1.85 DC Amps. and at 24 volts. This design provided a highly efficient power transfer from the energy source to the control surfaces.

The system that was designed to replace the previously discussed system utilized no direct feedback but consisted of a duel acting air cylinder that is extended and contracted by means of a fluid pump. The extension and retraction of the cylinder moves a moment arm attached to the control surface shaft. The rotation of the moment arm by the piston moves the control surfaces. The position of the planes is determined by correlating the time it takes to move the control surface a the desired amount versus the time the pump should be activated. Clockwise rotation of the surfaces is accomplished by running the pump in the forward direction. Counter-clockwise rotation of the surfaces is accomplished by reversing the polarity of the voltage sent to the pump. When no power is sent to the pump the pressure is maintained in the piston and the surfaces maintain their position.

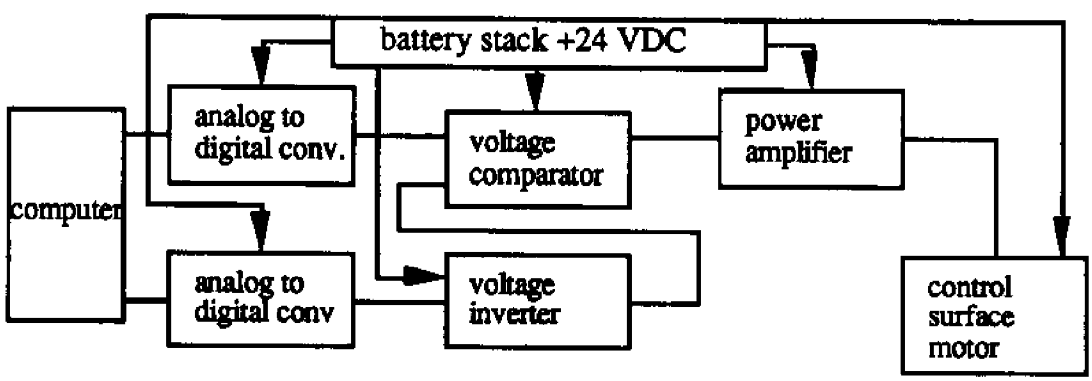


Figure 5: Block Diagram, Control Surface Motor Interface

The design of this system must take into account the fact that the thrusters need 24 DC Volts and 12 DC Amps to operate and the pumps need 12 DC Volts and 5 DC Amps. The processor boards are interfaced via a device called a digital-to-analog converter. The maximum output of a typical device is on the order of 0-5 VDC and 10-20 DC milliamps. Therefore, a current and voltage gain must be incorporated into the system to allow operation of the various motors.

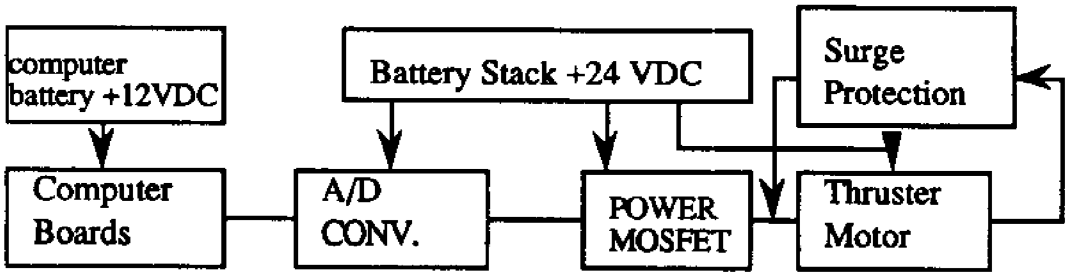


Figure 6: Block Diagram, Thruster Motor Interface

A system of control for the thrusters was suggested by Mr. Lynn Darnell, an instructor at New Hampshire Technical Institute. The possible adaption of this system to our purposes was researched and the final design incorporates concepts utilized in Mr. Darnell's original design. In this design the positive voltage side of the thrusters are connected to the positive side of the energy supply directly and the negative thruster terminals are connected to the negative supply through power MOSFET's (IRFT150). Voltage supplied to the gate of the MOSFET by the processor boards effectively connects the motor across the 24 VDC power supply and turns on the motor. If no power is supplied to the gate of the MOSFET then the motors are shut off. A capacitor and diode are used to dissipate the voltages created by the motor when power is removed from the gate of the MOSFET. With this configuration the motors are either full on or completely off. If the gate of the MOSFET is pulse- modulated the motor speed can be varied. Speed control was not used in this design.

V. Design Implementation:

The design for the thruster circuit is implemented using a Power MOSFET, a small value resistor,a heat sink, a diode and a capacitor. The circuit components were ordered through the

DIGI-KEY catalog and are standard value devices the circuit shown in Figure 7 is the circuit that is implemented in the thruster control circuit. The motor can only be run in the forward direction. This is accomplished by applying a small voltage at the base of the Power MOSFET. The motor is deactivated by setting the voltage at the gate of the MOSFET to zero voltage.

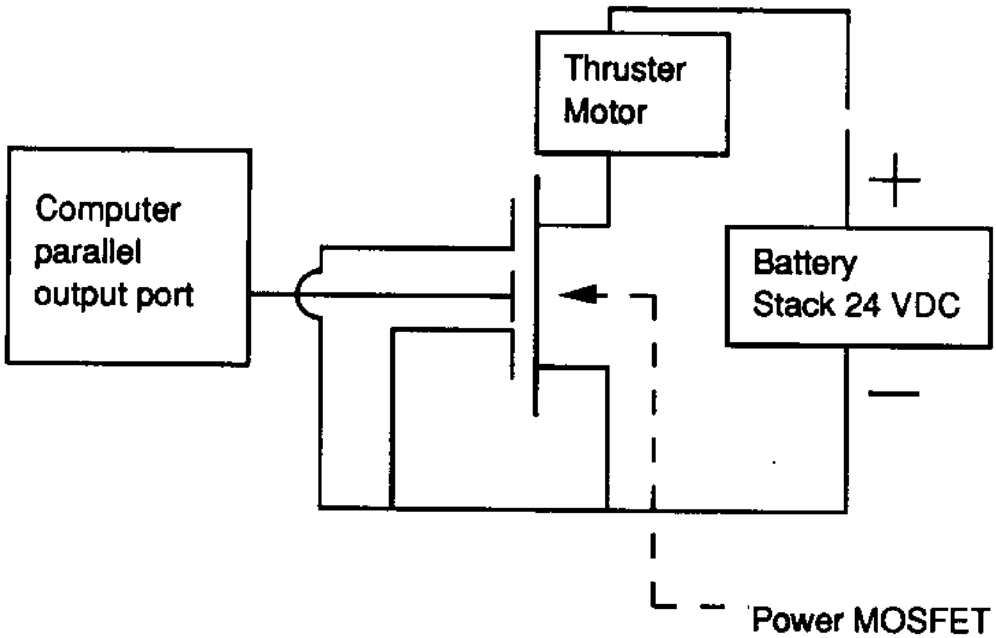


Figure 7: Block Diagram, Thruster Control Circuit

The design of the control circuit for the pump motor was implemented using four transistors set up as shown in Figure 8. The motor is turned on in the forward direction by applying a positive voltage to the forward input terminal while keeping the reverse input terminal at zero voltage. The pump motor is run in the reverse direction by applying a positive voltage signal to the reverse input terminal while keeping the forward input terminal at zero voltage. The motor will not run if both inputs are at a positive voltage or if both inputs are at zero voltage.

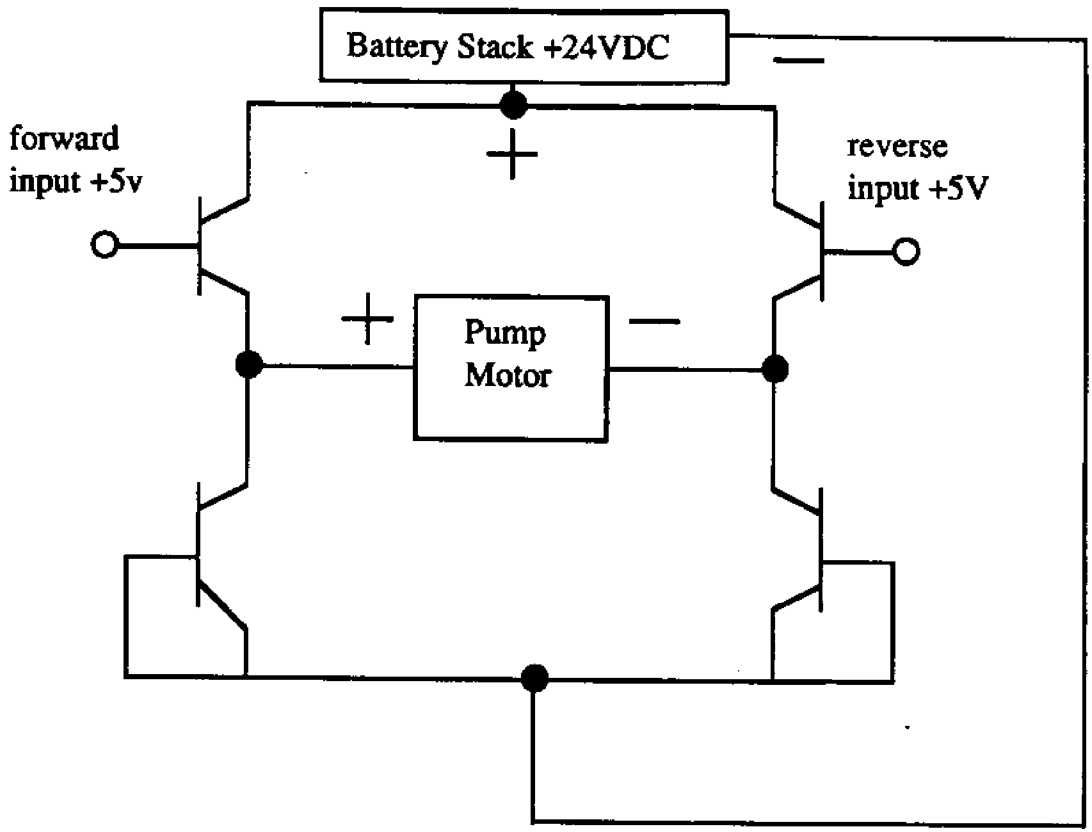


Figure 8: Control circuit for pump motor.

COMPUTER PROCESSING SYSTEM

I. System Design Criteria:

The design criteria established for the computer processing system include the following:

1. Must be able to handle navigation system processing
2. Must have adequate storage for data acquisition
3. Must be able to process control system information
4. Low cost
5. On-board A/D converters
6. Low power
7. Reliability

II. System Functional Requirements:

The computer processing boards will take in data provided by the temperature and pressure sensors, the navigation system and the control system. There is a separate board to handle each of these major areas. The data acquisition board will take temperature and pressure readings at regular intervals. The navigation board will send a pulse from the transducer and time the response from the transponder to determine distance. It will also take readings from the compass to determine bearing. It will also calculate depth from the reading of the pressure sensor. The controller board will send position signals to the controller circuit to make corrections for the mission course.

III. Design Altenatives:

There are many single-board computer processor systems with various features available. Some have more memory than others, some are easier to program than others, some have extra features which some applications need. The price range of these systems vary as much as the features found on them. Table 1 contains a comparison of the systems considered for the vehicle.

TABLE 1
COMPUTER SYSTEM SUMMARY

ALTERNATIVE FEATURE	MOTOROLA 64HC11	TATTLETALE MODEL III	GESPAC GESMPU-4B
On-board A/D Converters	8-Channel 8-Bit	8-Channel 8-Bit	None
Cost	Free	$595.00	$395.00
C-MOS Technology (Low Power Consumption)	Yes	Yes	No
Speed	2 MHz	4.9 MHz	8 MHz
RS 232 Interfaces	2	1	1
Memory Size	8K bytes	32K bytes	64K bytes
On-board Debug Capabilities	Yes	No	No
On-board EPROM Capability	Yes	Yes	Yes

IV. Selected Design Description:

The computer processor board that was chosen for the vehicle is the Motorola 64HC11 development board. It was chosen for the following reasons:

1. Motorola was conducting a promotion for the 64HC11 boards so it was much less expensive than any of the alternatives.
2. The 64HC11 boards use C-MOS technology and therefore have very low power requirements.
3. There is an on-board, 8-channel, 8-bit A/D converter.
4. There is the option of programming an EPROM and putting the program on-board.
5. An on-board debug monitor is provided.

For the above reasons and because Motorola provides excellent documentation with these boards, it was determined that they were the best choice for use in the vehicle.

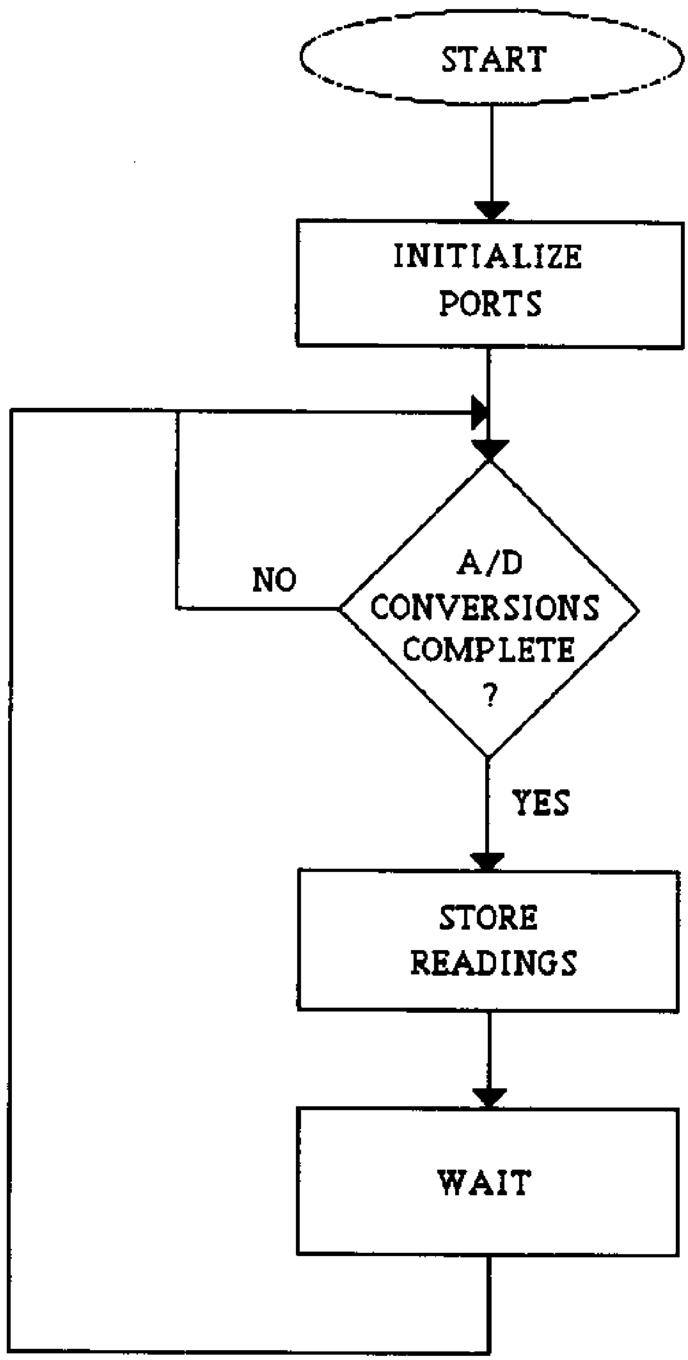
The boards need +5v, +12v and -12v which are easily obtained from the batteries on-board the vehicle using voltage regulators. A functional description of the processor boards is given below.

A. Data Acquisition Board

The data acquisition board takes readings, from the sensors, via two of the A/D ports and stores this information in memory. This information consists of a digital value proportional to the voltage read from the sensor. A temperature sensor and a pressure sensor are presently being used. These units are modular and, if it is desired, other sensors may be substituted or added to the system. The flowchart of the data acquisition program is given in Figure 9.

B. Navigation Board

The navigation board is responsible for handling several tasks. One of these is to keep track of the depth of the vehicle. This is done by determining the depth from the reading of the pressure sensor. The reading will be a digital value corresponding to the voltage reading of the sensor. It will be necessary to test and calibrate these readings to determine the actual numbers for the calculations. Depth corrections are forwarded to the controller board. Another task consists of sending a trigger pulse to the LM1812 chip. This will, in turn, fire the transducer on the vehicle. A return signal will be received from the transponder planted in the water. The time delay between the transmit signal and the receive signal will determine the distance of the vehicle from the center



DATA ACQUISITION FLOWCHART

Figure 9

of its circular course. This distance should be a constant. Corrections will be sent to the controller board. The flowchart for the navigation board is given in Figure 10.

C. Controller Board

The controller board handles sending correction signals to the control planes. It accomplishes this by taking the correction signal from the navigation board and determining the appropriate voltages to send to the control planes to turn them to the desired angle. Time constants for the control system need to be determined to calculate the exact numbers to be used. It also handles sending voltages to the thrusters. The flowchart for the controller board is given in Figure 11.

A system level block diagram of the selected system is given in Figure 12.

V. Design Problems:

One problem with the Motorola 64HC11 boards is that they have limited memory for storage space for both the data and the program. This was dealt with by writing efficient assembly code and by limiting the number of sensor readings that were recorded.

VI. Conclusion:

The Motorola 64HC11 board was the optimum choice for our vehicle. The cost of these boards was ideal. The low power requirements are an important factor for long range capabilities. It was possible to do the A/D conversions right on board. All of these factors contributed to the decision to use these processor boards.

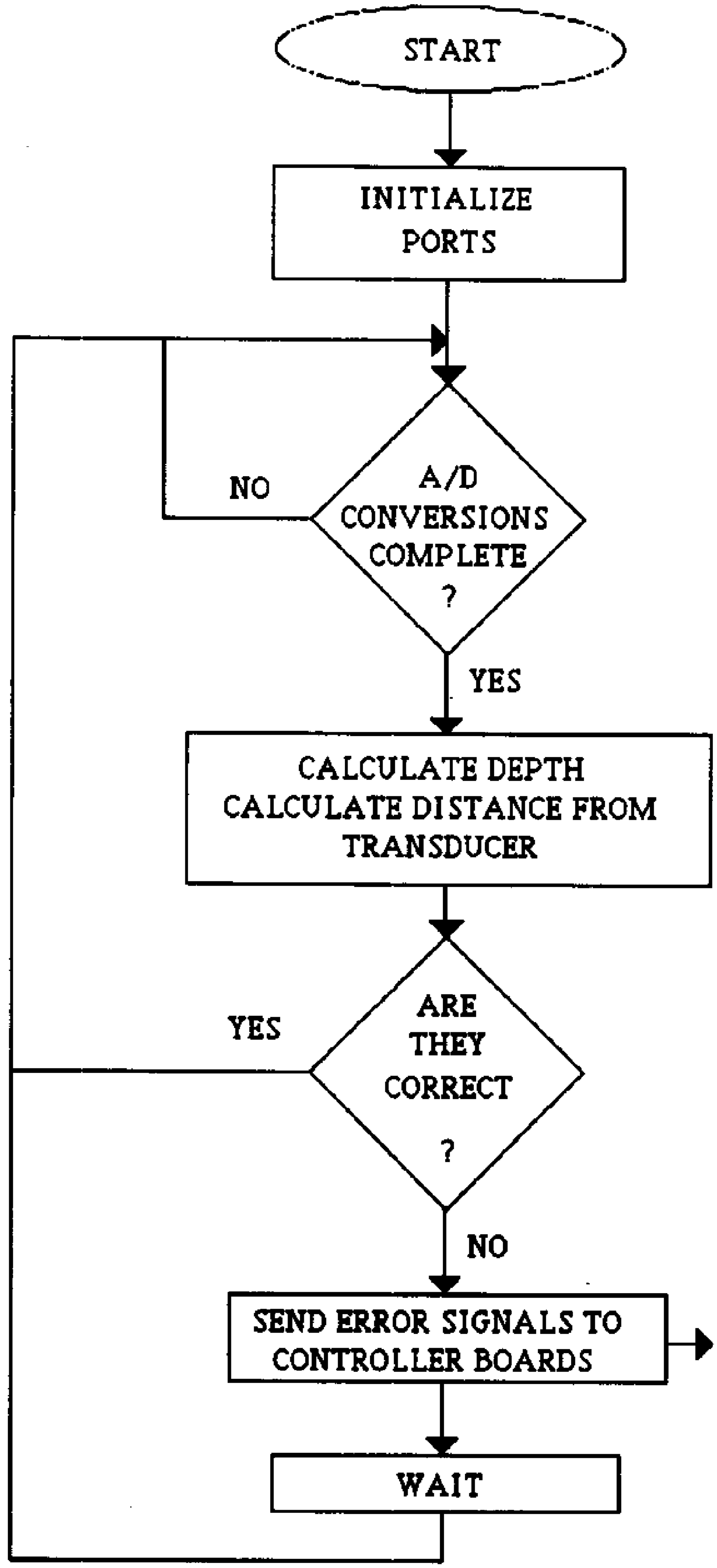


FIGURE 10: NAVIGATION BOARD FLOWCHART

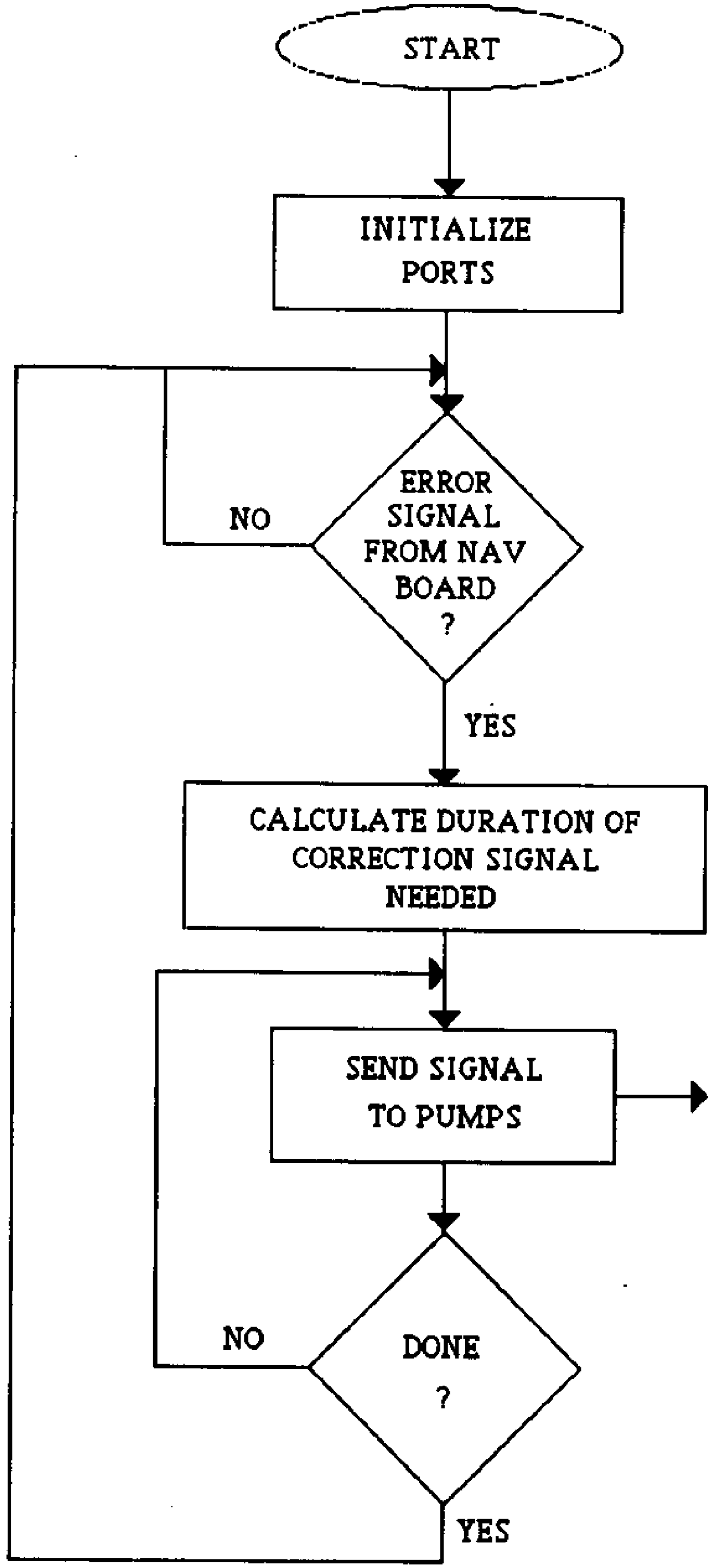
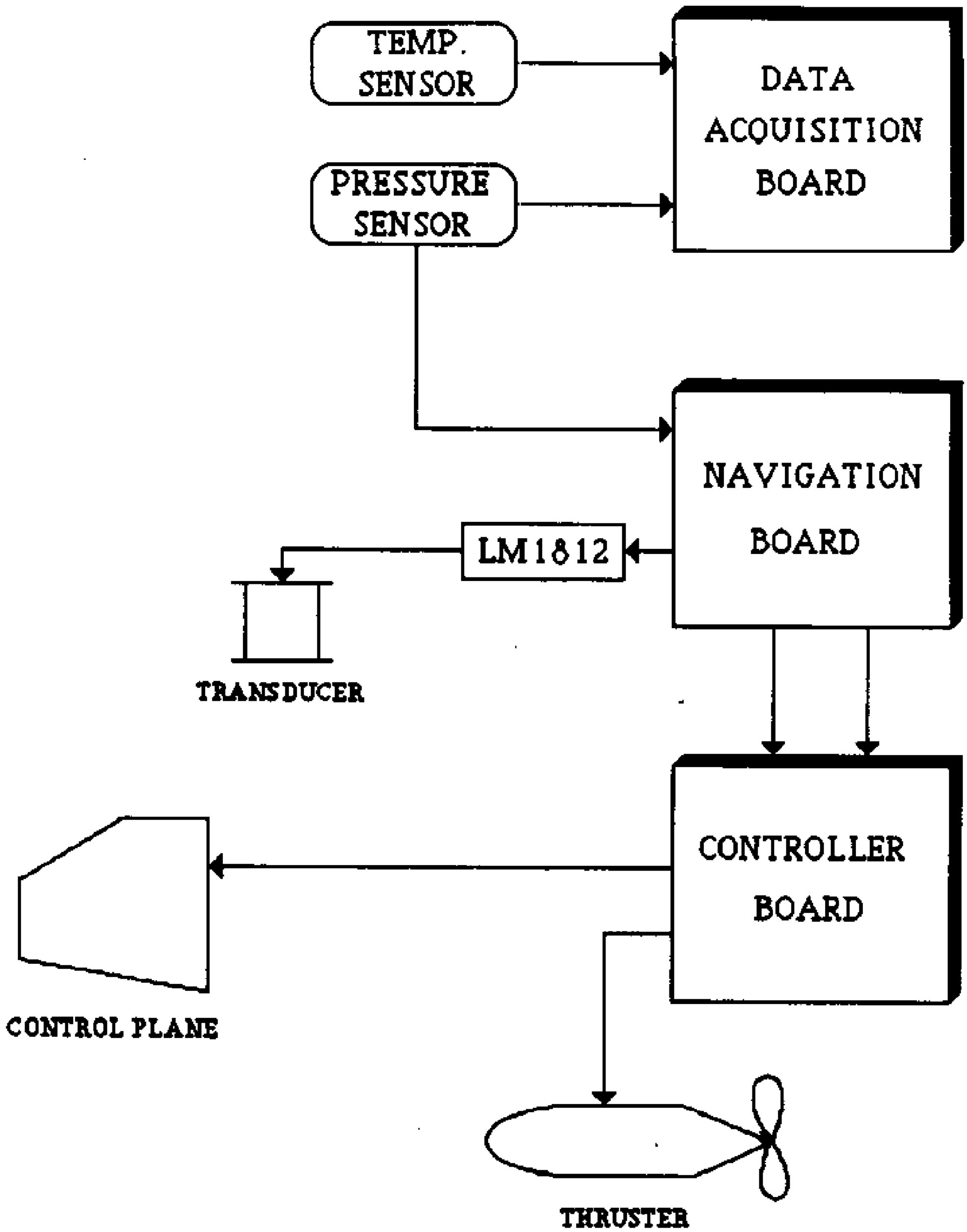


FIGURE 11: CONTROLLER BOARD FLOWCHART



SYSTEM LEVEL BLOCK DIAGRAM - COMPUTER SYSTEM

FIGURE 12

NAVIGATION SYSTEM

I. System Design Criteria:

The design criteria established for the navigation system includes the following:

1. Desired characteristics:
2. Manageable design
3. Time effective implementation
4. Ability to maintain course for length of mission
5. Low cost
6. Reliability

II. System Functional Requirements:

The navigation system monitors the position of the vehicle. Depth, orientation and distance to a reference point need to be be tracked. The navigation system is an essential system on an autonomous vehicle. The vehicle must know its current position, its desired position and be able to send this information to the vehicle controller in order to make corrections.

The navigation system must be referenced to a coordinate system. The choice of coordinate system depends on the type of navigation system chosen. Underwater vehicles have six degrees of freedom but three of these are rotational and are not a concern of the navigation system. The remaining three are translational - depth and x, y position. For accuracy, a navigation system should "make the same number of measurements as the degrees of freedom." (3N-K where N is the number of objects to be positioned in the navigational system and K, the number of constraints) [5].

The degree of accuracy required from the navigation system is dependent on its application. A vehicle which is primarily used for inspections of underwater structures needs less accuracy than one used for retrieving objects from the ocean floor. It is the ability for a vehicle to return to a predetermined position rather than to be able to calculate true position that is important for vehicle operations [5].

III. Alternative Designs:

There are several ways to approach the design of a navigation system. Most fall under two major categories - dead reckoning and acoustic navigation systems [6].

Dead reckoning systems include schemes as simple as tying a line to the vehicle and as complicated as inertial guidance systems. One is a trailing wheel or "unigator." This is a wheel mounted on a long rod which extends from the vehicle to the ocean floor. Distance is measured by an odometer that is connected to the wheel and direction is recorded by a gyrocompass. One other dead reckoning approach is a Doppler navigation system. This takes advantage of the " Doppler " shift of the frequency of a signal to determine the speed of a vehicle. Position is then calculated by combining the distance traveled with the vehicles compass heading [6].

Acoustic navigation systems use underwater acoustic markers. They may be transducers, transponders, beacons/pingers, hydrophones or combinations of these. In the case of a transducer - transponder system a signal is sent out by the transducer (on the vehicle) at one frequency and when received by the transponder (in the water), the transponder replies with one at another frequency. The time that it takes for the transducer to receive the reply is a function of the distance between the two.

Table 2 shows a comparison of some alternate navigation system designs. A graphical comparison of the accuracy of these systems is given in Figure 13. These were extracted from the MSEL AUV Search System Report [7].

IV. Selected Design Description:

The navigation system design that was selected is an acoustic navigation system incorporating transducers/transponders for measuring distance. This is a modification of an acoustic long base line (LBL) design. This type of system was selected for the following reasons:

1. There were transducers and transponders available for our use and at no cost to this project.
2. The transmission/detection circuit design parameters using an LM1812 transceiver chip are proven, reliable and available. Circuit diagrams of the LM1812, the transmitter and the receiver are given in figures 14, 15 and 16 respectively.
3. The dead reckoning systems are either too dependent on outside factors (tethers and trailing wheel) or are too complicated for the time and budget available (Doppler and inertial systems).
4. The acoustic system is a low risk, accurate and reliable system.

AUV NAVIGATION SYSTEM SUMMARY

SYSTEM TYPE	[illegible]	[illegible]	[illegible]	[illegible] RATE	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	NOTE
ACOUSTIC LBL	1 – 5	.5 – 2.5M	.3 – 1M	1s >	HIGH	SMALL	1 – 3		LOW	set,cal xponders
DOPPLER SONAR	unlimited	.25% + bias	good for short time	> 1/s	med – hi	> 2cu.ft	60 lbs		LOW	add AH sensor
INERTIAL GIMBALLED	NOT	APPLICABLE	—	—	—	—	—	—	—	—
INERTIAL LASER GYRO	unlimited	1% dist comm	good for short time	> 1/s	HIGH	.2 – 2cuft.	< 20	~100K	MED	
INERTIAL FO GYRO	unlimited	no comm	"	"	HIGH	< 1cuft.	< 20	?	HIGH	develop into com product
MULTI – SENS. INS	un;imited	.4 – .7%	good for short time	> 1/s	med	> 3cuft	> 60	?	med – hi	have to integrate

TABLE 2

AUV NAVIGATION SYSTEMS
RELATIVE ACCURACIES

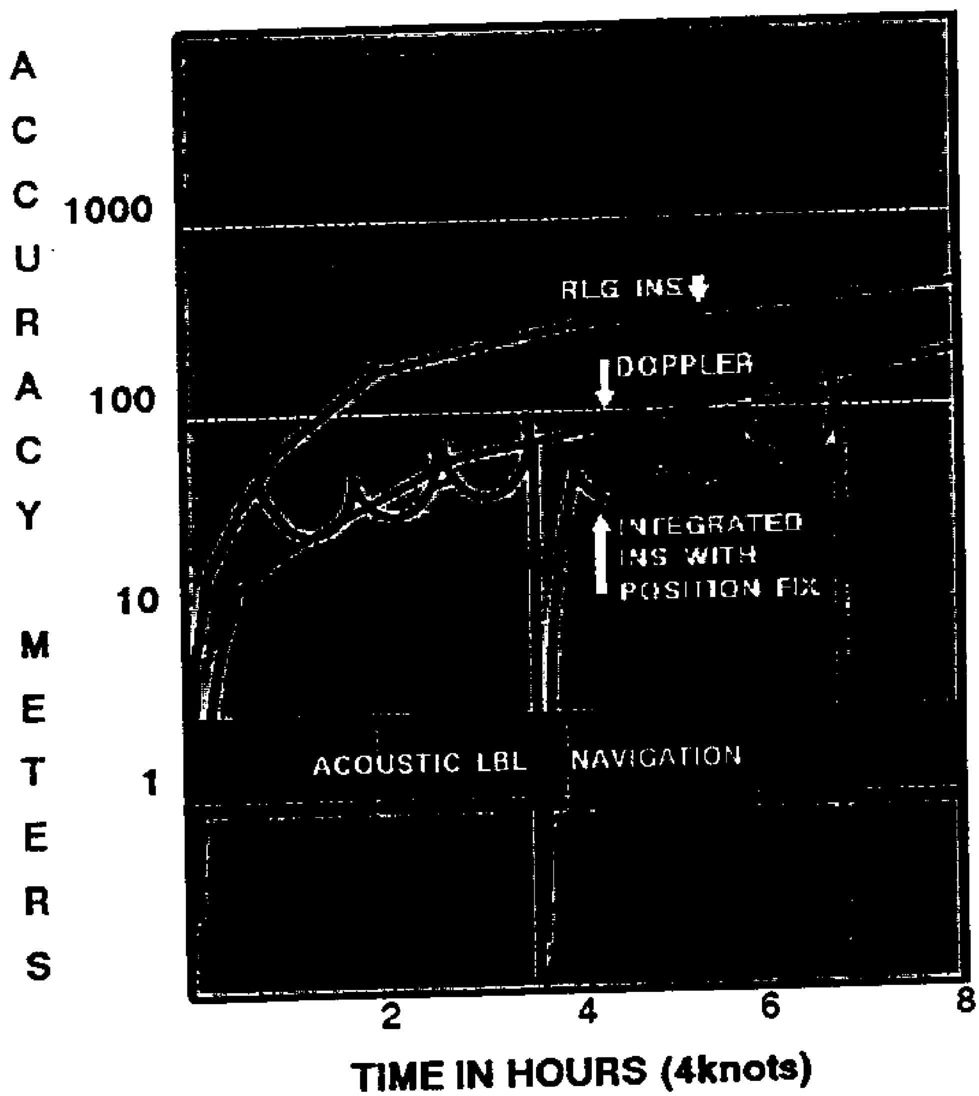


Figure 13

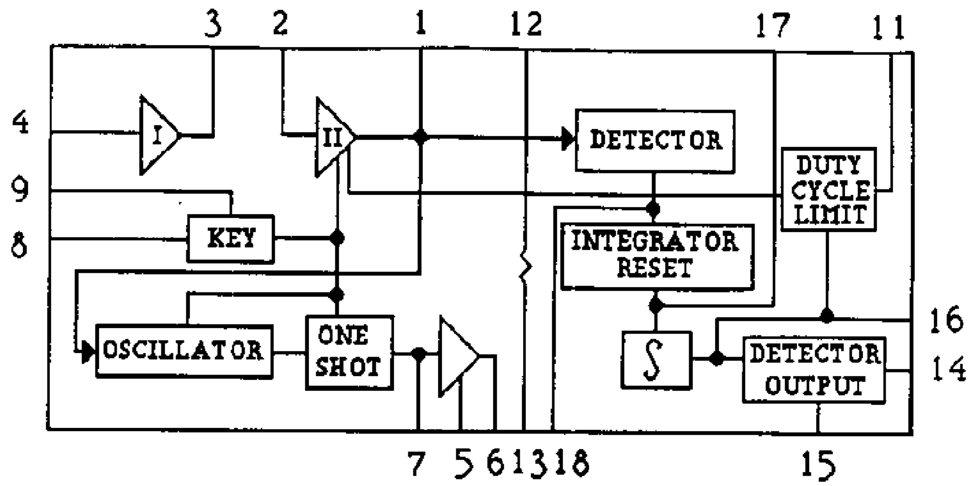


FIGURE 14: LM1812 - ULTRASONIC TRANSCEIVER CHIP

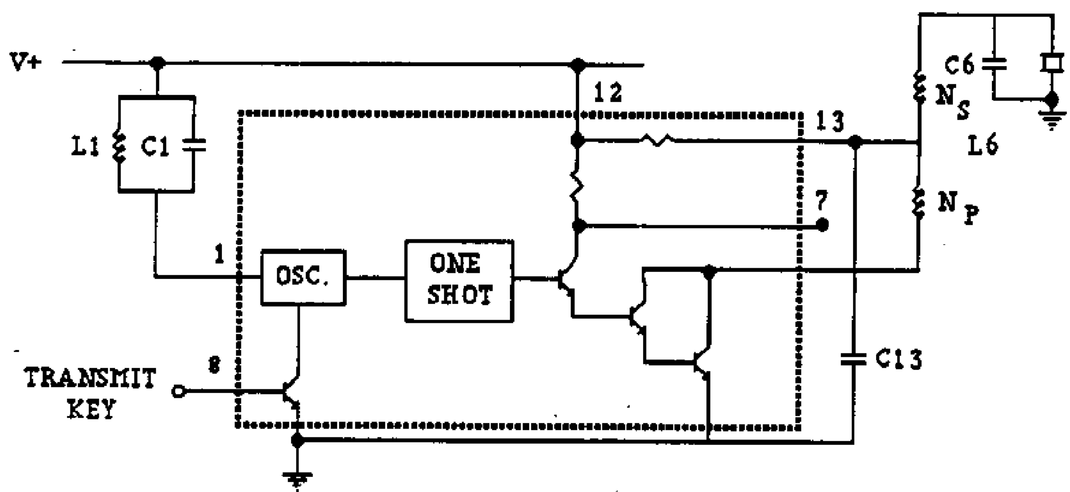


FIGURE 15: TRANSMITTER PORTION OF NAV CIRCUIT

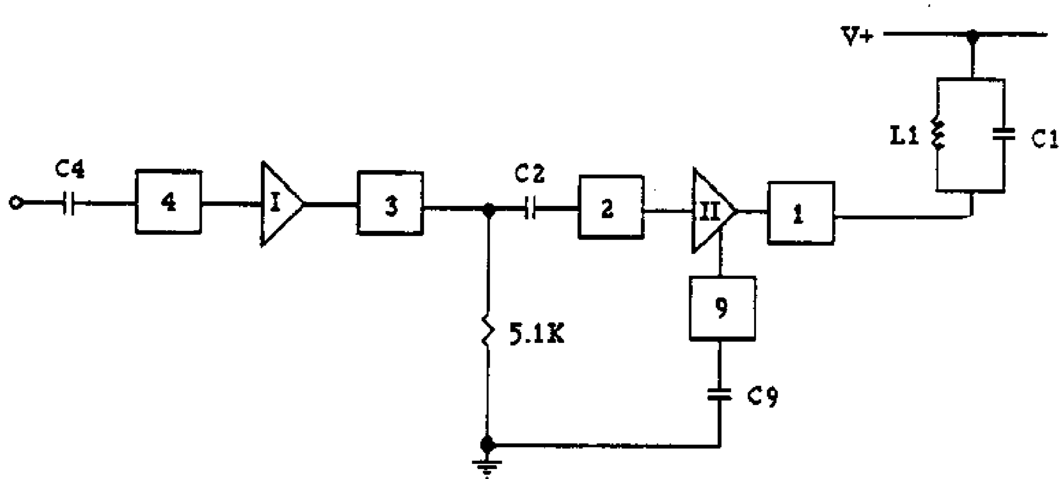


FIGURE 16 RECEIVER PORTION OF NAV CIRCUIT

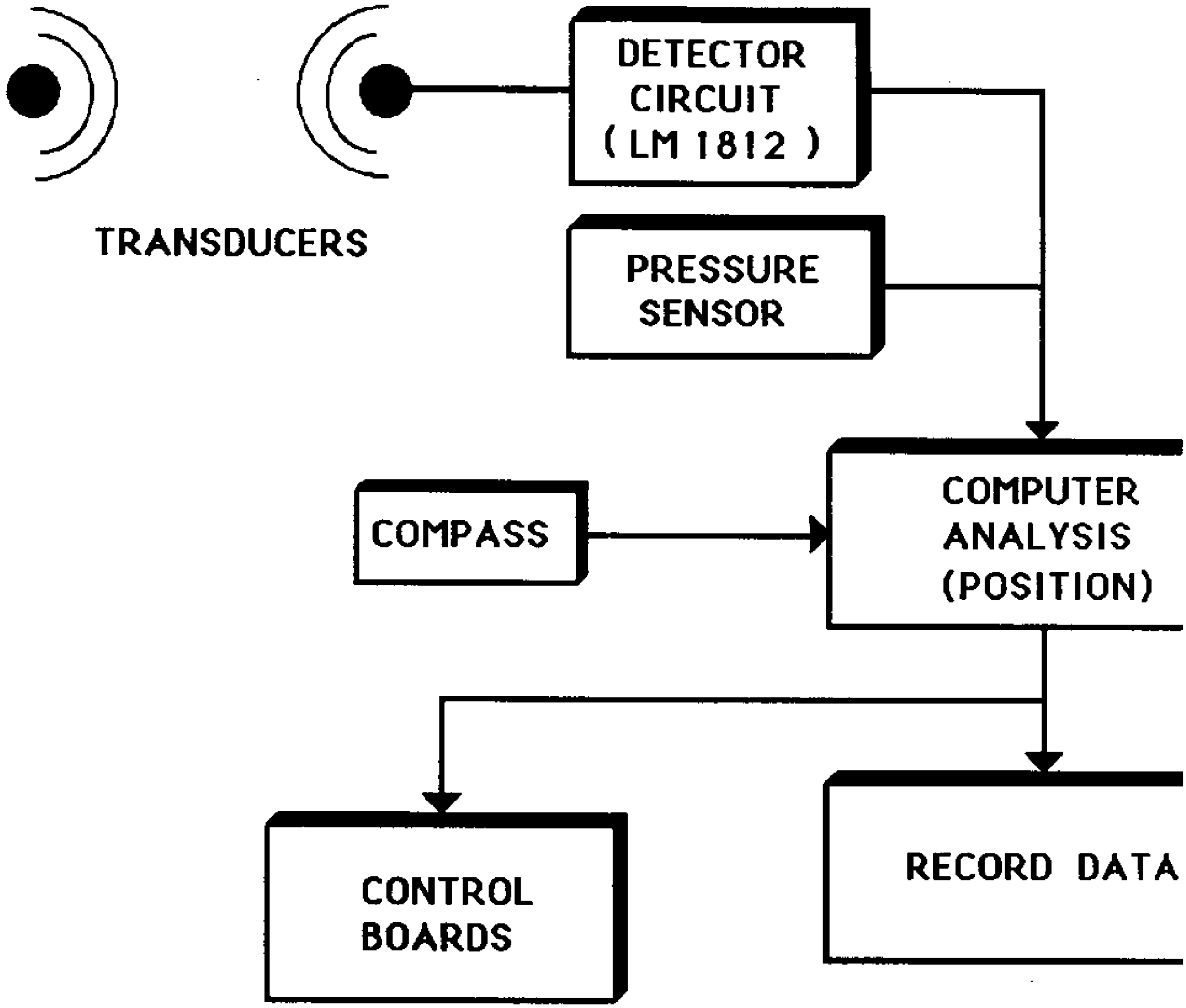
Due to the constraints on this project (money and time), it was necessary to simplify the navigational requirements wherever possible. The mission parameters were set for the vehicle to travel on a circular course at varying depths. A system level block diagram of the navigation system is given in Figure 17. In order to reduce the number of measurements that need to be taken, the polar coordinate system, with a fixed radius, was chosen as a reference. Then using 3N-K, N = 2 (one transponder and one vehicle) and K = 4 [transponder at (0,0,0) and vehicle at (R,B,D)]. Where R is the fixed radius of the mission, B is the bearing relative to the start position and D is the depth. 3N-K = 2, so two requisite measurements are needed, B and D. B can be recorded from the on-board compass and D can be determined as a function of the pressure recorded from the pressure sensor.

V. Design Problems:

A potential problem with the transducer/transponder system is that it is range limited. A sufficient number of transponders need to be planted along the vehicle's course for it to be able to track position for the entire mission. The transponders have a limited range, so the larger the course area, the number of transponders necessary to navigate effectively will increase. The transponders also have to be calibrated which is more time consuming as the number of transponders increase. The complexity of the computer software needed to control the vehicle will also increase.

VI. Conclusion:

The navigation system design chosen for the SLAVe vehicle was the optimum choice given the time and money constraints placed on the project. If this project is further developed a different navigation system may be necessary. An acoustic system could be used if it is possible to seed transponders along its course. If a limited area is being surveyed this would be a good alternative.



NAVIGATION SYSTEM BLOCK DIAGRAM

FIGURE 17

SENSOR SYSTEM

I. System Design Criteria:

The design criteria established for the sensor system include the following:

1. Modular units for installation and replacement flexibility
2. Low cost
3. Accurate measurements
4. Fast response to change in measurements
5. Reliability

II. System Functional Description:

The sensors are modular units which collect environmental parameters within the mission area. The current parameters being recorded are temperature and depth. The units were made modular so other parameters which may be of interest can be substituted or added to the system.

III. Selected Design Description:

The sensors chosen for use in the vehicle were provided by the Ocean Process Analysis Laboratory (OPAL) here at UNH. These sensors were selected based upon the design criteria and their availability for this project at no cost. Descriptions of the pressure and temperature sensors as provided by the Martek TDC Metering System manual follow below.

The depth sensor is a standard Bourdon tube potentiometric transducer. This is essentially a hollow, coil-spring type mechanism that senses pressure changes in the fluid environment that it is exposed to. It then expands or contract a certain amount depending on the type and degree of pressure change. All electronic sensors make use of the fact that sea pressure changes increase uniformly with depth level. The pressure changes are then simply translated into proportional electrical signals.

The Bourdon tube is mechanically coupled inside the sensor housing to a variable resistor (potentiometer) which converts pressure changes to an equivalent electrical resistance, and produces an output signal which is a varying voltage level. The Bourdon tube element in the Martek sensor is filled with a special oil. A flexible rubber sensing diaphragm located in a hole at the base of the housing acts as an interface between the internal transducer mechanism and the external pressure medium (environment).

This design protects and prevents corrosion of the metal Bourdon tube inside, and also ensures the instantaneous response of the sensor to pressure changes. The depth sensor has three separate, single-conductor electrical leads.

This simple design principal results in a very economical and reliable transducer for oceanographic measurements. The accuracy level of the Bourdon tube transducers (+/- 2% of full-scale measuring range) is also adequate for most measuring applications down to about 300 meters or so. Accuracy is limited in large part by static friction errors occurring at the low end of the measuring scale. This effect becomes more pronounced when trying to measure small pressure changes at depths of 15 meters or less using a wide-range sensor (such as 0 - 100 meters). Measurements of this type should be made using a narrow-range sensor (such as 0 - 100 feet or less) to obtain better sensitivity and accuracy.

The Martek temperature probe uses a factory calibrated, specially aged, glass-bead thermistor as the sensing element. The thermistor is encapsulated at the closed end of a stainless steel tube and this whole assembly is then molded in polyurethane. This is done to prevent damage from pressure effects, shock and impact. It also aided in providing electrical insulation from conductive elements. The small diameter stainless steel tube is exposed directly to the environment as the primary conductor. The entire assembly is recessed in a cylindrical protective PVC sheath bored with four flow-through holes around the exposed tube. This design provides for a rugged, reliable and highly sensitive sensor mechanism. This system has sufficient overall accuracy for temperature measurements in the surface layers of most natural bodies of water. Thermal response time of the sensor is excellent (less than 1 second).

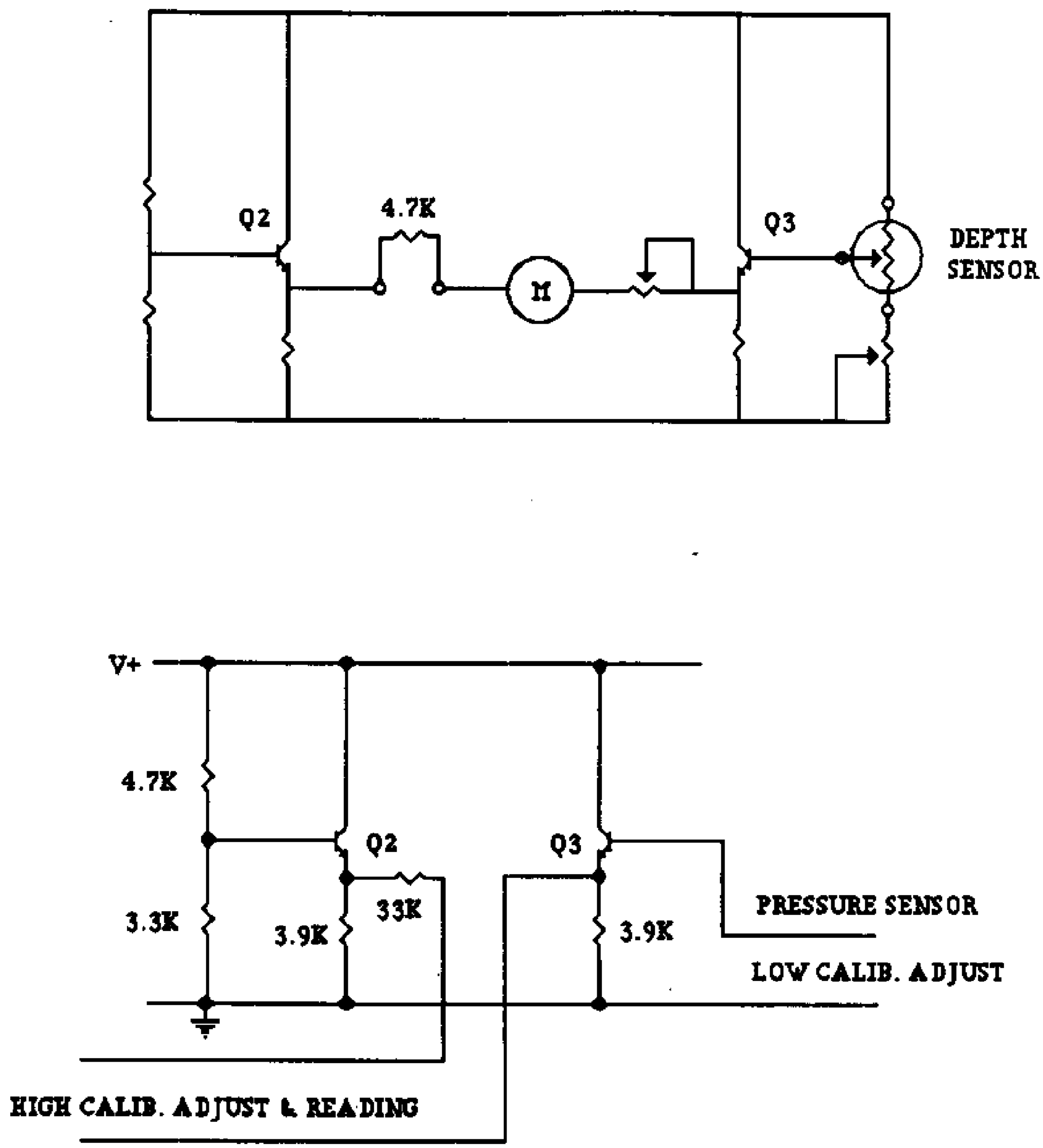
Thermistors, which are basically semiconductor resistors, may be used in electronic circuits that supply output signals which are a nearly linear function of temperature. This is especially true over the relatively small changes found in oceans, lakes and rivers. The thermistor resistance decreases with temperature. This varying electrical energy can be easily measured by converting the resistance value to a variable voltage. Thermistors are very sensitive to temperature changes, and therefore respond quickly. The Martek temperature probe has two separate, single-conductor electrical leads [8].

IV. Design Problems:

A system assembly problem was encountered in the design, since the Martek sensors were originally part of a entire metering system which included other sensors. Detector circuits had to be fabricated for this application. These are illustrated in Figures 18 and 19. It was necessary to include potentiometers for calibration purposes. These circuits were untested at the time this report was submitted.

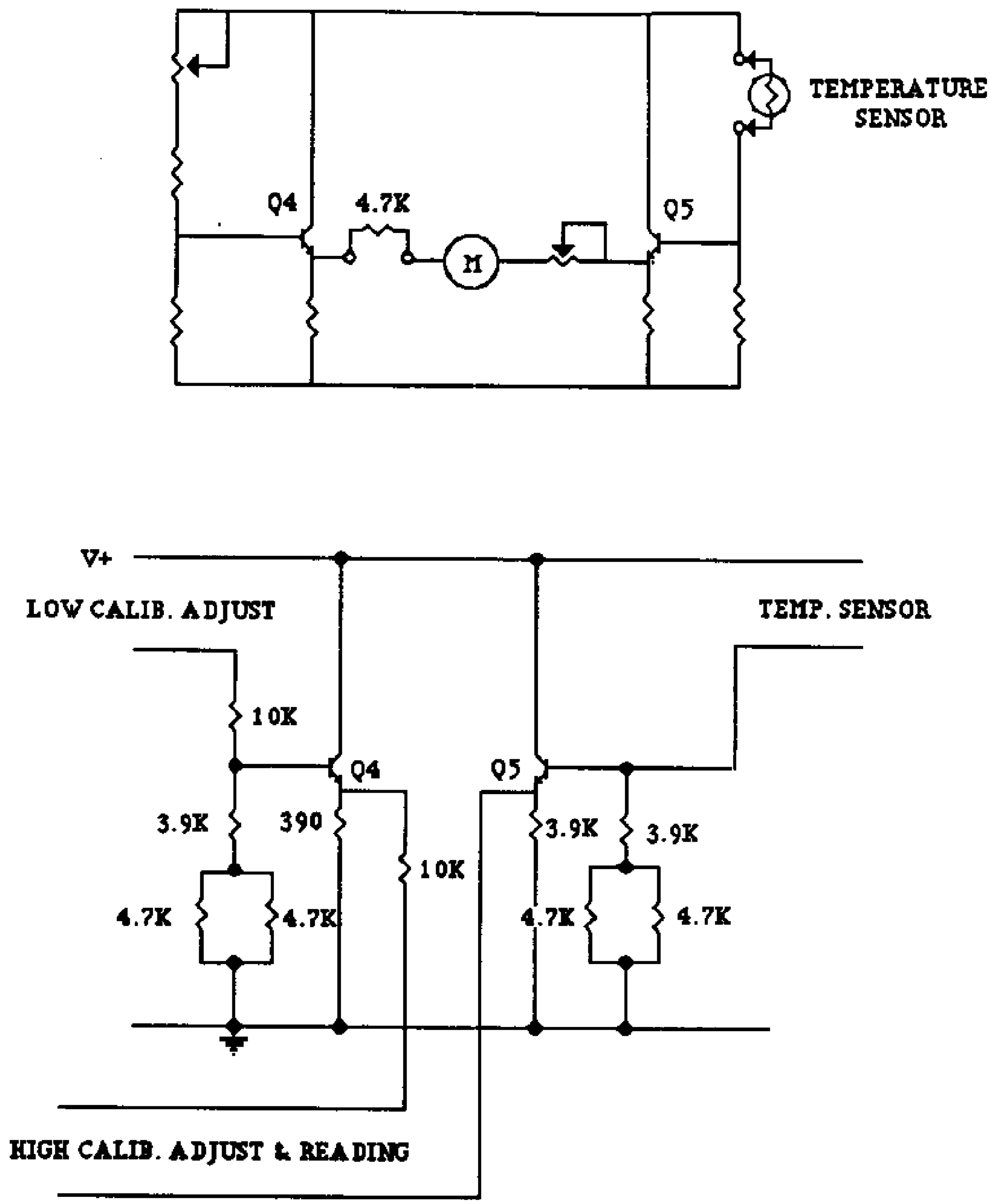
V. Conclusion:

The pressure and temperature sensor chosen for use in the vehicle were readily available at UNH, at no cost and met the system requirements.



PRESSURE SENSOR DETECTOR SCHEMATIC

FIGURE 18



TEMPERATURE SENSOR DETECTOR SCHEMATIC

FIGURE 19

VEHICLE MANEUVERING SYSTEM

I. System Design Criteria:

The criteria used to design the vehicle maneuvering surfaces include the following:

1. Minimize the hydrodynamic drag force on vehicle
2. Cost efficient
3. Ease of fabrication
4. Light weight
5. Ability to maintain control while holding a constant pitch angle, and to provide a maximum speed
6. Corrosion resistant

II. System Functional Description:

Located on the submersible vehicle, are control surfaces, which are hydrodynamic appendages whose purpose is to control the vehicle's maneuverability. Yaw control is designed to provide the vehicle with steering capability in the horizontal plane. Roll control is used to control vehicle rotation along its longitudinal axis. Finally, pitch control is used to control the pitch angle of the vehicle to help achieve and maintain a desired depth.

III. Alternative Concepts:

There are three basic ways to utilize the foil effect exhibited by the control surfaces to control a vehicles position. They are as follows: 1.) The position of the surfaces may be permanently fixed at a predesignated angle to control the vehicles path. However, this solution limits the path of the vehicle to a straight line with absolutely no flexibility. 2.) Alternative 2 is to have partially trainable portions of the fixed surface (such as the flaps on an airplane wing) to alter the lift forces on the surfaces. Experience has dictated that this method can be effective for high speed vehicles, but is of limited effectiveness at slow speeds. 3.) Alternative 3 is to utilize an all-movable control surface in which the vehicle's direction is altered by rotating the entire surface about an axis. A weighted evaluation technique (as shown in appendix D) was developed to compare the advantages of each alternative. Based on the results of this evaluation, the all-movable control surface appears to be the most effective candidate for the design.

Many different basic design configurations could be chosen which use the concept of all-movable control surfaces. One possible concept is to have two stern planes and one rudder. The stern planes would control the motion in the vertical plane, while the rudder would control the motion in the horizontal plane. Another solution would be to have two horizontal planes located towards the bow of the vehicle, which would resemble somewhat the side fins of a shark. A rudder would be also used at the stern of the vehicle in this alternative. The design selected is that of four control surfaces located at the stern of the submersible. These four planes are orthogonal to each other. Two of the planes are oriented horizontally and control the vehicle's pitch angle to aid the vehicle in changing depths either ascending or descending. The remaining two controls surfaces are oriented vertically and control the vehicle's yaw angle to aid the vehicle in controlling its heading or course.

IV. Description of Selected Design:

After the all-movable control surface was chosen, extensive analysis was performed to optimize its effectiveness. The results of this analysis can be seen in Appendix E.

The most important performance parameter of the surface is its surface profile area. The appropriate area was calculated using the following formula [9].

$$A_P = \frac{LD}{m}$$

A_P = profile area for two planes

L = overall vehicle length

D = overall vehicle diameter

m = dimensionless constant

25.6 for horizontal stern planes

30.0 for rudder

After the desired profile area was calculated, a typical stern plane shape was chosen, and its geometry altered to yield the correct area. There are many criteria to consider when choosing the appropriate geometry. The criteria often involve the less than explicit terms which describe the vehicle's control surfaces. It is therefore important at this point to discuss characteristics of control surfaces. The following terms and definitions have been extracted from Gilmore and Johnson [10].

The mean span (b) is the average of the spans of the leading and trailing edges of the control surface.

The mean chord (c) is the average fore and aft distance between the leading edge and trailing edge, or the average of the *root chord* and the *tip chord.*

The profile area (A_p) is the projected area (or planform area) of the control surface, and may be taken as the product of the mean span and the mean chord.

The aspect ratio (AR) is the ratio of the mean span to the mean chord (b/c), or the ratio of the mean span squared, divided by the profile area ($b2/A_p$).

The taper ratio is the ratio of the tip chord to the root chord.

The sweepback angle is the angle between the quarter chord line and a line perpendicular to the centerline of the ship.

The angle of attack (alpha) is the angle between the mean chord line and the direction of the free stream velocity (U).

Lift (L) is the component of the resultant force on a control surface (foil) that is perpendicular to the direction of motion (free-stream velocity).

Drag (D) is the component of the resultant force on the control surface (foil) parallel to the direction of motion.

The center of pressure (C.P.) is the point on the foil through which the resultant force may be considered to act. It is located at the quarter chord position for thin symmetrical foils of infinite aspect ratio, but varies with angle of attack for low aspect ratio foils.

With the above factors defined, the optimization process will be discussed.

Despite the fact that much experimental data has been done on control surfaces, no precise analytical method could be found in the literature which optimizes the physical characteristics of control surfaces. However, an extensive study by L. Folger Whicker, et al [11] was used which analyzed in depth, the various types of control surfaces. The shape selected for this design was modeled after the NACA 0015 section shape.

For reasons of structural reliability and minimizing the potential damage of an impact, the maximum transverse projection of a control surface is generally chosen to be less than the outer diameter of the vehicle. This criterion is based on the assumption that if an unfortunate accident should occur, (such as collision with the ocean floor or a pier), the vital control surfaces would be less likely to be damaged.

The sweepback angle was chosen somewhat arbitrarily. The Whicker study analyzed surfaces whose sweepback angle ranged between -8 and 11 degrees. Sweepback angles which lie within this range were considered to be satisfactory.

Another important control surface parameter is the location of the chord-wise and span-wise center of pressure. The steering moment is exerted about the center of pressure point and the motor which is controlling the surface must be capable of resisting this moment. Also, for stability requirements, of the center of pressure must be located aft of the drive shaft. The following equations were used to calculate the center of pressure locations [11].

$$(CP)_c = 0.25 - \frac{C_{m\frac{c}{4}}}{C_L \cos\alpha + C_D \sin\alpha}$$

$(CP)_c$ = Chordwise center of pressure

$C_{m\frac{c}{4}}$= Torque of pitching moment coefficient

C_L = Coefficient of lift

C_D = Coefficient of drag

α = angle of attack

$$(CP)_s = \frac{C_L\left(\frac{4}{3}\pi\,\frac{b}{2}\right)\cos\alpha + C_D\left(\frac{b}{2}\right)\sin\alpha}{b\,(C_L\cos\alpha + C_D\sin\alpha)}$$

$(CP)_s$ = Spanwise center of pressure

$\frac{b}{2}$= Length of semi-span

The following equations were used to calculate the resultant moment about the center of pressure point.

$$M_{shaft} = \frac{1}{2} C_{M\ shaft} \rho S V^2 c$$

M_{shaft} = moment exerted about shaft

$C_{M\ shaft}$ = moment coefficient

ρ = density of fresh water

S = Planform area

V = velocity of vehicle

c = mean geometric chord

where $C_{M\ shaft}$ is calculated from:

$$C_{M\ shaft} = \left[\frac{(l_1 - l_3)}{(l_2 - l_1)} \right] C_{M\frac{c}{4}}$$

l_1 = distance from leading edge to (CP)c-bar

l_2 = distance from leading edge to quarter-chord

l_3 = distance from leading edge to shaft

To minimize spin or roll, the vehicle was built symmetrically (mirror image to that above and below the central axis), and else the components were located along the central axis to the extent feasible.

The two thrusters were located on the distal edges of the stern planes. These planes (hydrofoils) were dimensionalized so as to minimize the drag force created by the control surfaces. The stern planes were designed to compensate for the vehicle's positive buoyancy, as well as providing the ability to change and maintain depth through control of the vehicle's pitch angle. Each plane provides enough thrust to enable the vehicle to maneuver properly. The dimensions chosen for each of the dive planes are shown in the Appendix F.

The dive planes are connected to a drive shaft which runs transversely (perpendicular to the central axis) through the stern of the submarine. A hydraulic pump/piston arrangement was chosen as the drive mechanism. These pumps were mounted in the free-flooding tail section of the vehicle. The extension and reaction of the hydraulic pistons would result in the altering of the control plane's angle of attack. A maximum of approximately ± 15 degrees is feasible for the plane angle; beyond this, stall occurs. Stall occurs when no under pressure is present on the top surface of the hydrofoil, and a drop in lift coefficient occurs. This condition is detrimental to the operation of a hydrofoil.

The vehicle course or heading is controlled by changing the angle of the rudder. The rudder is driven through a motor, hydraulic pumps and shafting in the same manner as the dive planes. Like the dive planes, the rudder is able to rotate at angles up to ± 15 degrees.

Lift and drag coefficients were obtained from an Attack Angle vs. Coefficient of Lift, and Coefficient of Lift vs. Coefficient of Drag graphs shown in figures 20 and 21. From this, forces of lift and drag were calculated by the aid of the following formulas:

$$\text{Lift force} = \frac{1}{2} C_L \rho V^2 A$$

$$\text{Drag Force} = C_D *(0.5)* p *V\text{^}2*A$$

C_L = Coefficient of Lift

C_D = Coefficient of Drag

p = Density of Water

V = Velocity of Vehicle

A = Area of Plane

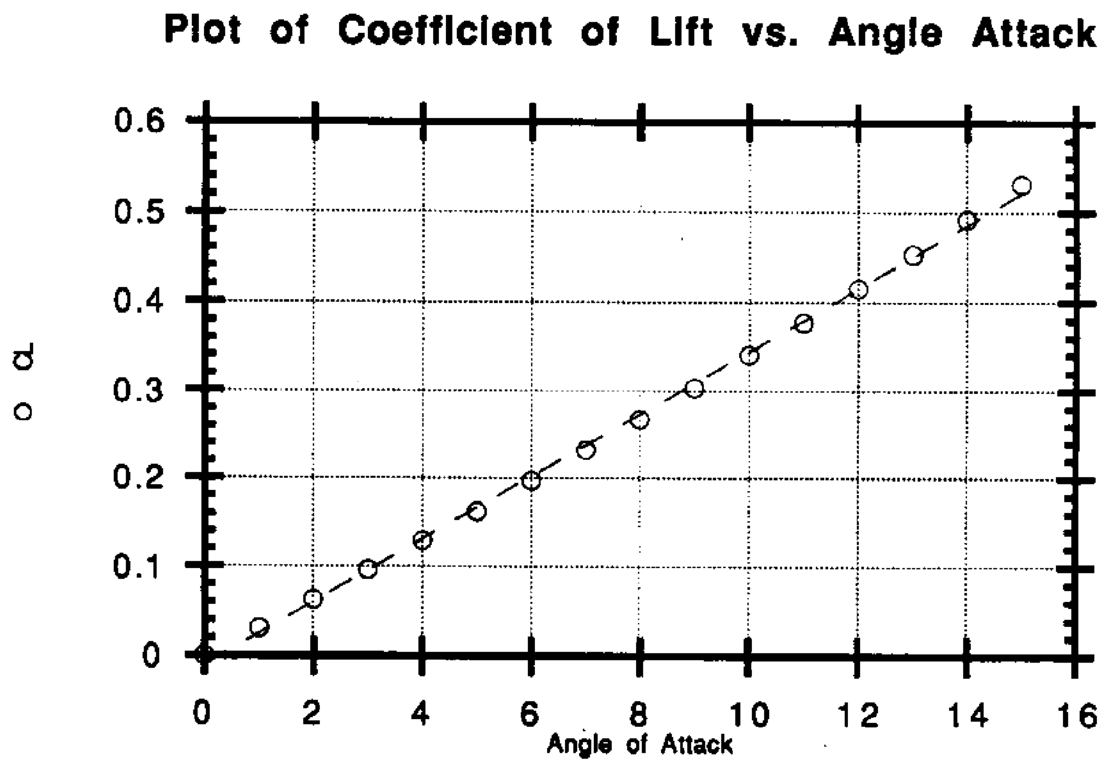


Figure 20: Coefficient of Lift vs. Angle of Attack

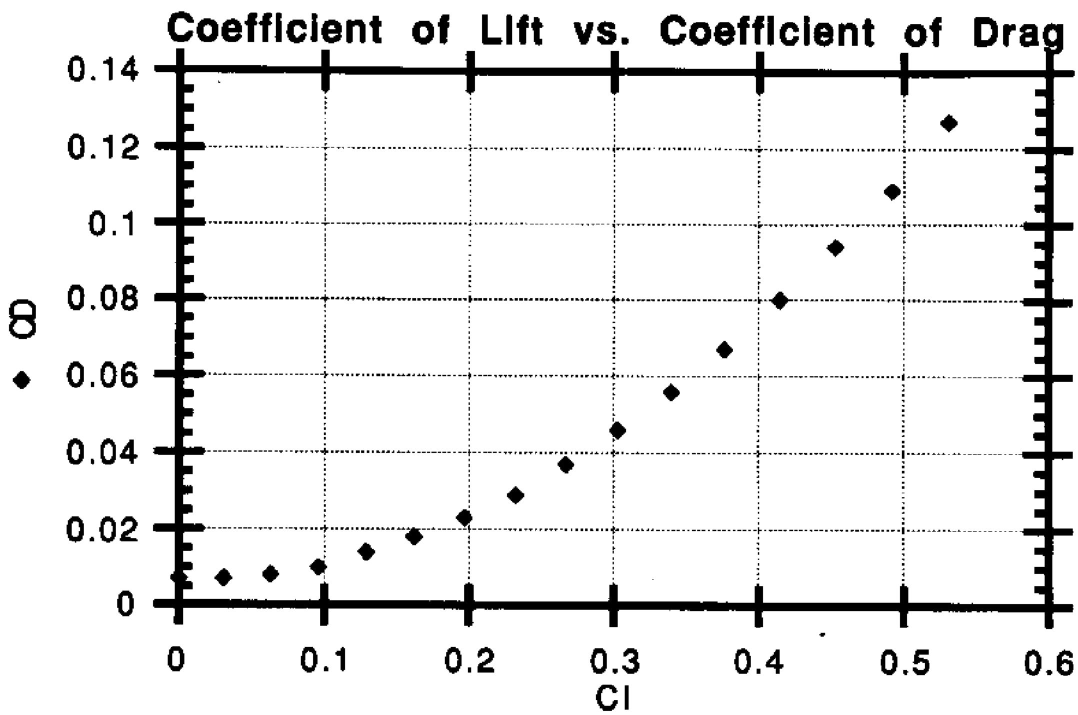


Figure 21: Coefficient of Lift vs. Coefficient of Drag

The materials that were considered for the control surfaces were plastic, fiberglass, and aluminum. These alternatives were evaluated based on thefactors of: cost, light weight, ease of fabrication and strength to withstand the change in pressure. Aluminum was the material selected which best met the design criteria for the control surfaces.

V. Design Problems:

One major concern with respect to the vehicle's stability is that of existing moments. If a moment is present, and is unbalanced (i.e. there is not an equal and opposite moment counteracting), then the vehicle will rotate onto it's longitudinal axis. This can be accounted for by balancing the moments, or else placing the components along the vehicle's central axis.

Another problem which could arise, is increasing the angle of attack beyond ±15 degrees. This would set the vehicle into stall. To avoid this, ±15 degrees was decided upon as a limitation. This included a slight factor of safety. Finally, a problem which arises concerning the pitch control deals with propulsion and steering. By moving in an ascending and descending manner, speed is lost in the horizontal direction. This is compensated for by having the pitch control (dive planes with thrusters attached) provide enough thrust to overcome the loss.

VI. Conclusion:

The chosen vehicle's maneuvering system design is one that takes advantage of the thrust and power supplied by the thrusters to provide excellent control at all vehicle speeds. The vehicle will operate in a safe manner and be able to achieve and maintain a given orientation with respect to the x, y, and z axes. This was accomplished in a cost efficient manner without jeopardizing the purpose of the controls.

HULL STRUCTURE SYSTEM

I. Pressure Vessel:

Pressure Vessel Design Criteria:

1. Lightweight and as small as possible
2. Withstand the Pressure of 200 ffw
3. House a 10 inch diameter Aluminum-Air battery and computers
4. Hydrodynamically Acceptable/ Fit within fairing
5. Ease of manufacture

II. System Functional Description:

The pressure vessel was designed with the intent that it should house an aluminum-air battery and all the necessary computer components to operate the vehicle autonomously at a depth of no more than 200 ft in fresh water. The aluminum-air battery requires a minimum tube diameter of at least 10 inches. Although our vehicle will use lead-acid batteries, which are slightly smaller, the original design criteria were still kept.

III. Design Alternatives:

Material Selection

Three materials were initially considered. The three materials were steel, aluminum and PolyVinyl Chloride (PVC). The use of steel was immediately dismissed due to weight considerations. Aluminum was later dismissed for reasons of cost. This left PVC.

Pressure Vessel Thickness

Calculations initially made for a cylinder made of PVC showed that, to withstand a pressure of 90 psi, a wall thickness of 0.48 inches was required. A wall thickness of 0.5 inches (schedule 80) was selected as it was readily available from the manufacturer. This wall thickness gave a wall deflection of only 0.016 inches (in tension) at 90 psig. It must be noted that these calculations were based on constants found in Ashby & Jones [13]. Later, during the construction phase, we learned that there are no universally accepted constants (yield strength, Young's Modulus and Poisson's Ratio) for PVC. The material qualities of PVC vary greatly from manufacturer to manufacturer. No tests of our material were made at this time.

Interior Racks

The racks were constructed in the UNH wood shop using plywood. The use of a non conductive material was essential to avoid possible electrical shorts in the circuit boards and batteries. These racks were to be mounted between the two halves of a longitudinally split 2 inch diameter PVC tube. This tubing was to be welded to the inside of the pressure vessel wall in such a manner as to prevent the batteries and computer components from shifting during vehicle operation.

End Cap Design

Pressure vessel end caps ideally are hemispherical in shape. However, once again due to budgetary and time constraints, we were forced to settle for flat caps. Using beam analysis and what sorts of flat material were readily available, a thickness of 1.625 inches was selected. This thickness of PVC was found to have an acceptable deflection of 0.016 inches at loads in excess of 90 psig. The design and specifications of the end caps may be seen in figure 22. The caps, along with the rest of the pressure vessel, were fabricated at Eptam plastics.

End Machining

When it came time to machine out the concentric circles for the end caps to fit into, a slight problem was encountered. No one that we had access to could turn out an 11.625 inner diameter

Two Section Pressure Vessel Assembly

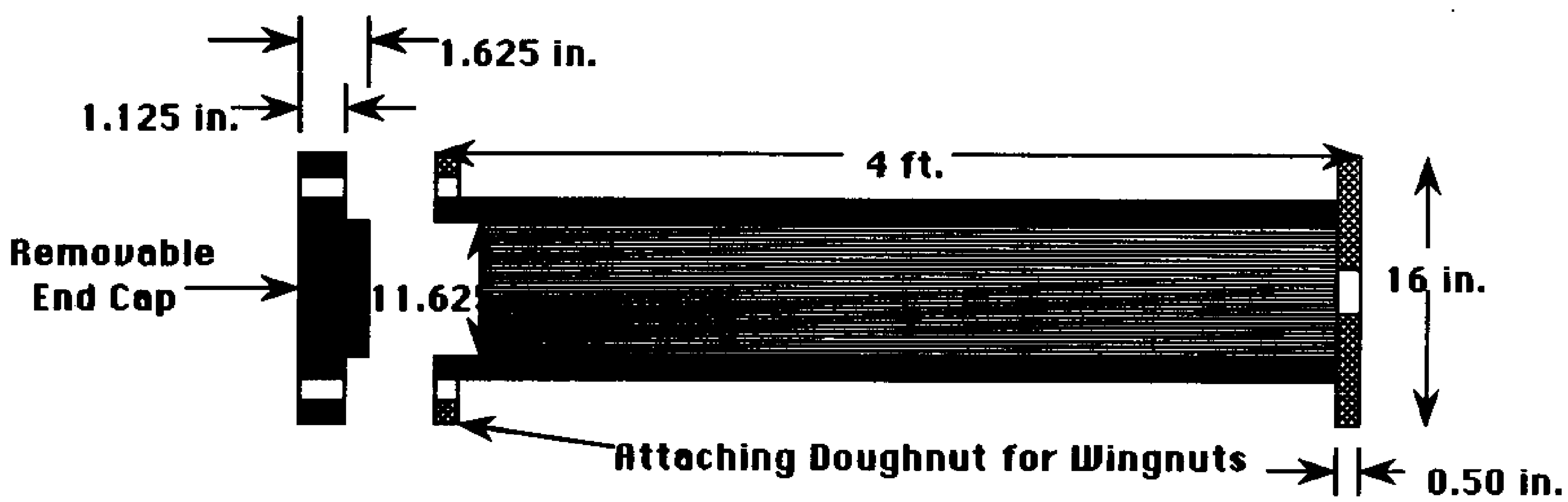


Figure 22

tube that was 8 feet long. In order that the tube end could be machined to the proper tolerance so that the end cap would fit properly, the pressure vessel had to be split into two 4 foot sections.

"O" Ring Placement

For the sake of redundancy it was deemed that the end cap should use primary and secondary "O" ring seals (see figure 23). This was done so that if the main "O" ring failed due to a deflection due to pressure or a torque caused while operating the vehicle, the pressure vessel would not be immediately flooded through the end caps. (Unfortunately, it is the placement of this secondary "O" ring which is one of the prime candidates in the failure analysis of the pressure vessel. Refer to the failure analysis section). In the actual manufacture of the pressure vessel the secondary "O" ring was not placed as specified in figure 23, which is how it was submitted to Eptam plastics. Rather it was placed midway along the lip of the turned concentric circle.

Two Section Design

As explained earlier, the pressure vessel was split into two symmetric 4 foot sections for the purpose of machining. In order to join these two pieces into a single unit, a scheme was devised where two 16 inch diameter 0.5 inch thick circular flanges were welded to both ends of the 4 foot sections opposite the end caps. A 4 inch diameter circle was then cut in the center of each flange so that the two chambers would be connected when the flanges were joined (see figure 24). This design not only facilitated manufacturing, but should have increased the pressure vessels strength by shortening the overall length between rigid supports.

Joining

The two sections were to be joined in much the same way as the end caps were to be fixed to the cylinder. Eight 0.375 inch holes were drilled in a symmetric pattern around the perimeter of each flange. This small bolt size was selected mainly for convenience. The only loads they were

End Cap Assembly

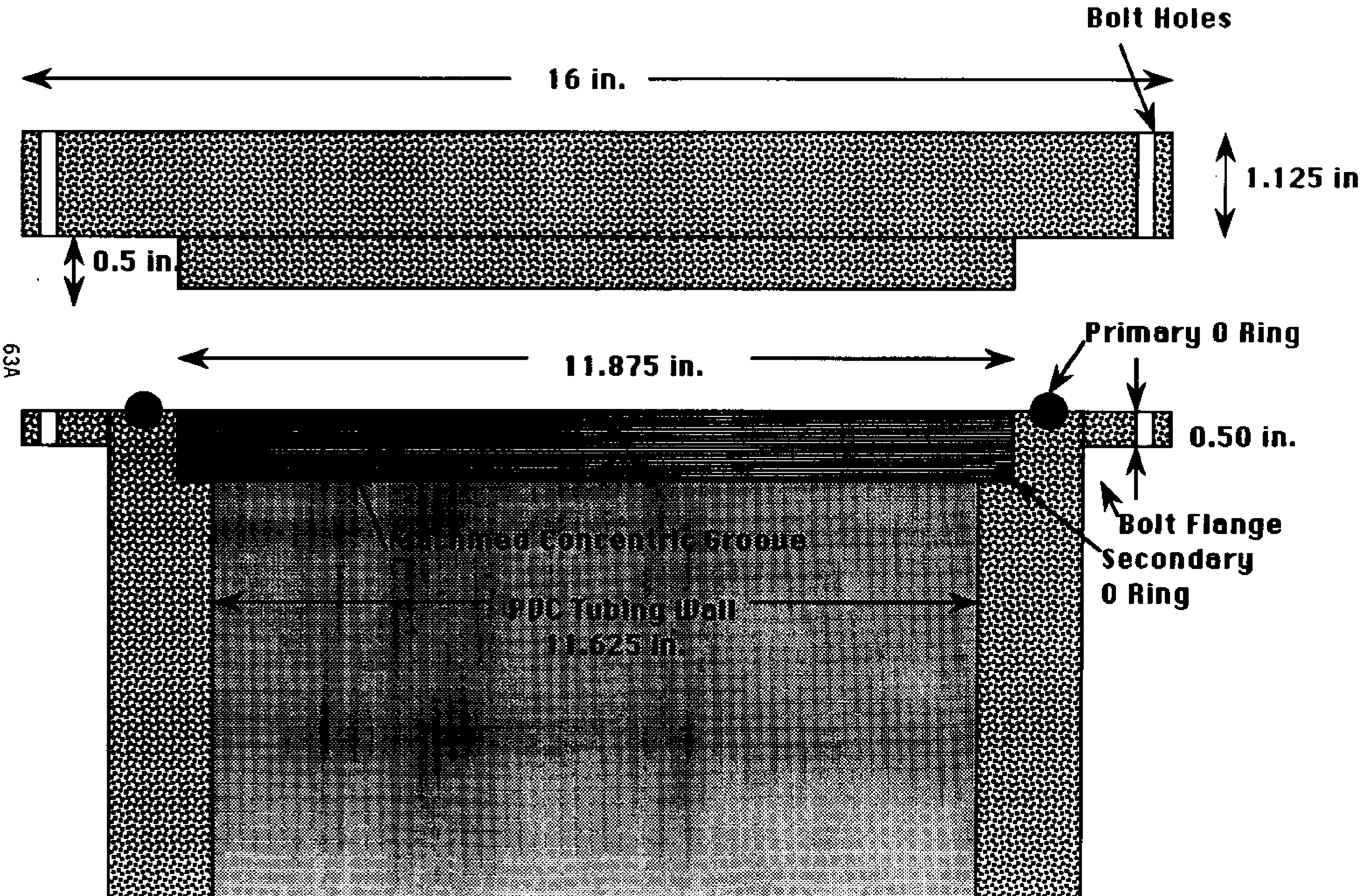


Figure 23

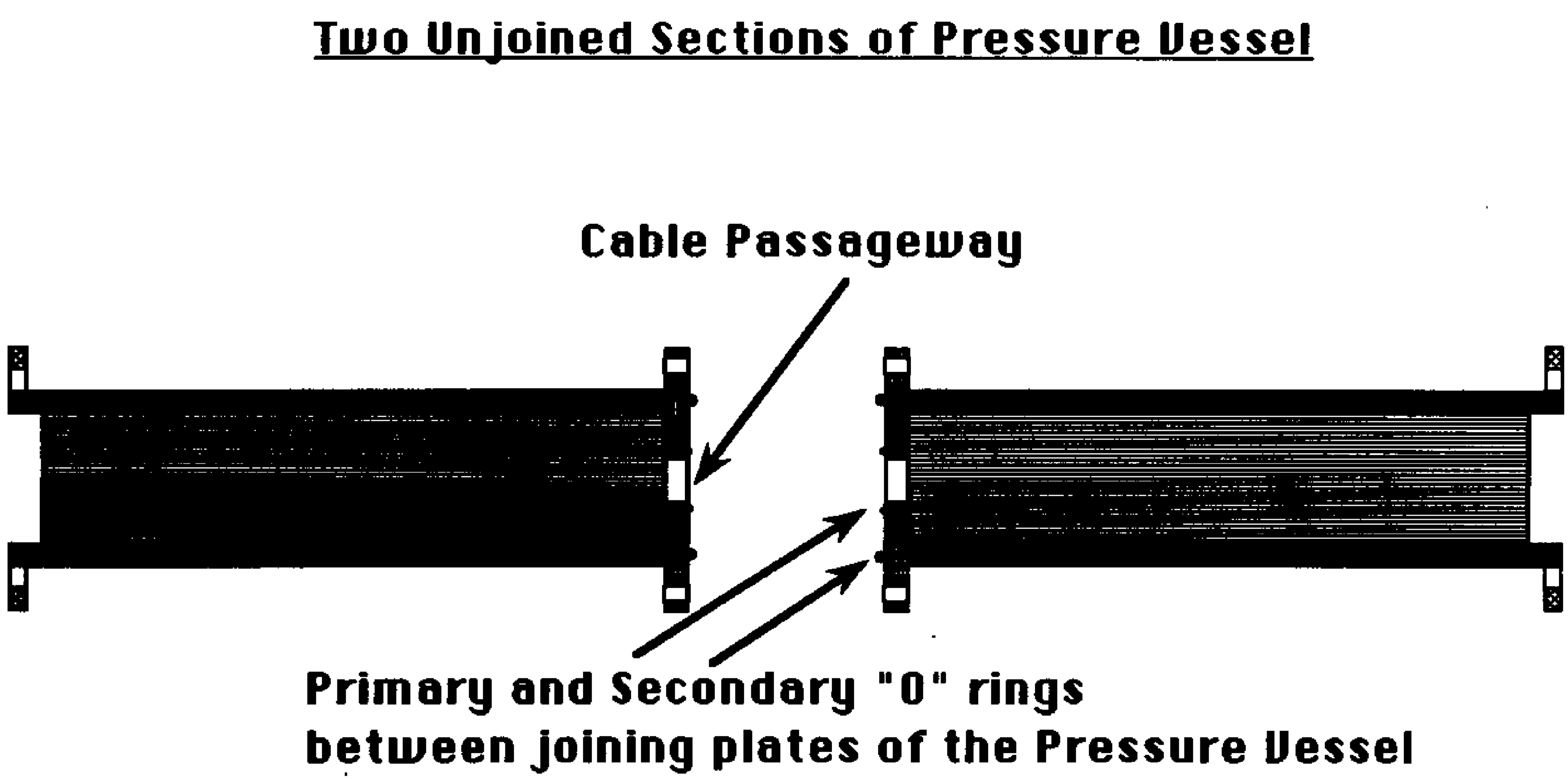

Two Unjoined Sections of Pressure Vessel
Cable Passageway
Primary and Secondary "O" rings
between joining plates of the Pressure Vessel

Figure 24

expected to support were those of holding the end caps on and the flanges together while above the surface. These loads would diminish once under pressure. Two "O" rings were then placed in machined grooves at an 8 inch and 10 inch diameter between the two flanges. Once again two were used for the sake of redundancy. The two halves could then be bolted together.

TAIL FRAME

I. System Description:

The initial design of the control system for the vehicle called for two DC stepper motors to turn the vertical and horizontal control surfaces attached to two aluminum rods. In order to support not only the weight of this system but the torque of the motors, an internal tail frame became necessary. This frame needed to be as strong and light as possible while staying within the outer hydrodynamic fairing.

II. Design Selection:

Material Selection

Here their were two choices. Titanium gives the best strength to weight ratio but is very expensive and not easy to work with. Aluminum has a somewhat lower strength to weight ratio but is much cheaper and fairly simple to work with. Therefore Aluminum was selected.

Tail Design

The frame was designed to approximate a right circular frustrum with a major diameter of 16 inches and a minor diameter of 2 inches. The major diameter was set at 16 inches so that it could bolt up flush to one end cap of the pressure vessel using the same bolt pattern. The minor diameter was dictated by the confines of the conical fairing. It was fabricated in the UNH machine shop from four flat bars and a single 16 inch square sheet of aluminum. Four concentric doughnuts were cut from the sheet and notched every 90°. The doughnuts were then spaced and the bars were attached through the notches. The general layout may be seen in figure 25.

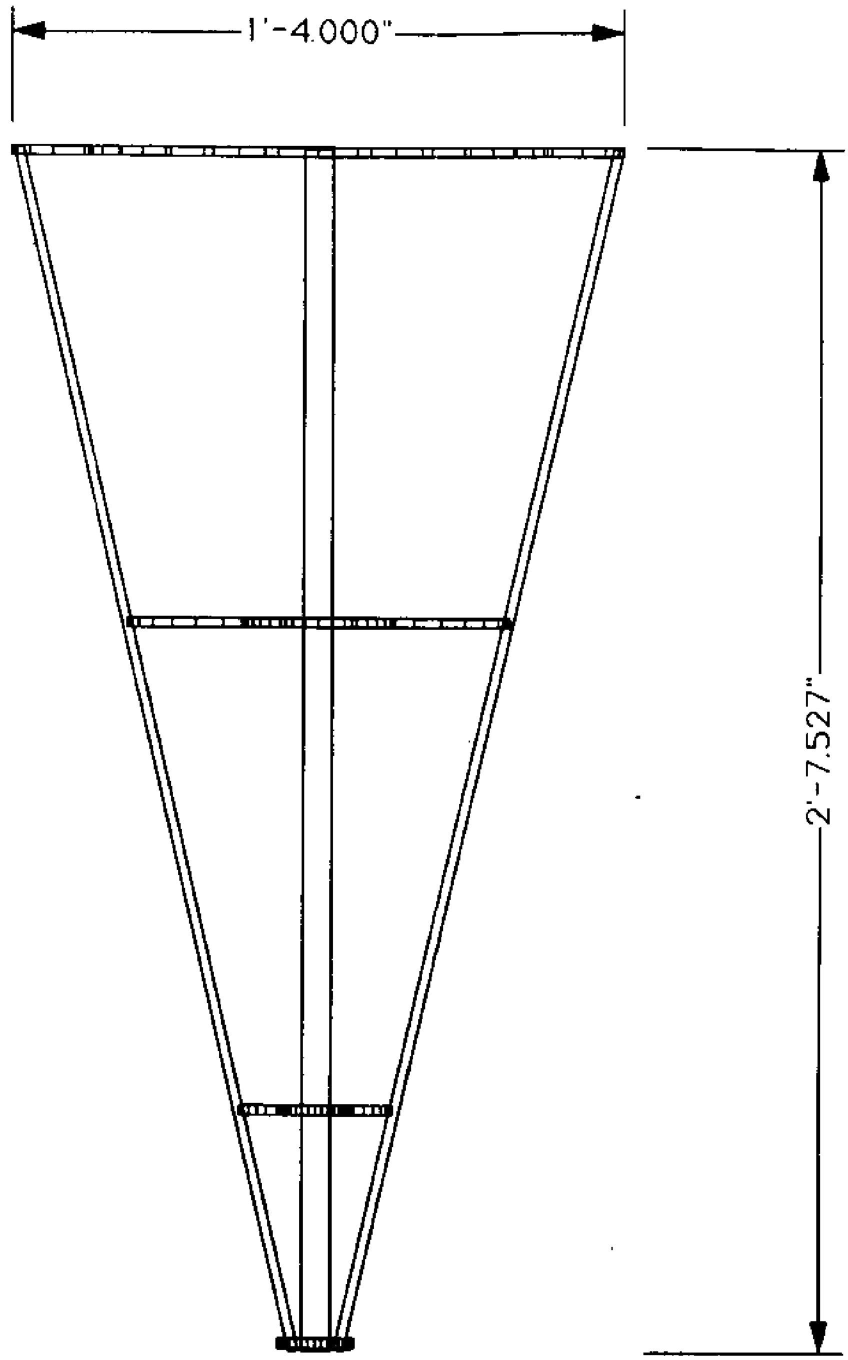


Figure 25

Assembled Frame of Tail Section

Mechanical Engineering-UNH

Frederick Murdock

Scale 1:5

HULL/FAIRING

I. Design Criteria:

1. Lightweight
2. Hydrodynamic
3. Length to Diameter Ratio: $5 \leq L/D \leq 9$

II. System Description:

The mission requirements stated that the vehicle should be able to maintain a speed of 5 knots and cover a range of 200 nautical miles. With this in mind, hull drag becomes highly important. A smooth shape with no sharp discontinuities is required to limit turbulent flow.

III. Alternative Designs:

Theoretical Design Considerations:

The ideal shape for such applications is that of the previously mentioned Series 58 hull developed by the U.S. Navy and implemented on the USS Albacore and the MIT human powered submarine Icarus [13] (see figure 26). Using the ITTC convention for calculating bare hull resistance, this shape was found to have the least resistance due to its low coefficient of residual resistance.[14]

Practical Design

Due to time and financial constraints something approximating a series 58 hull was selected. This was done with the intent that what was selected could be later built up to more closely approximate a series 58 hull. What was selected was the basic torpedo shape. This may be seen in Figure 27. This outer fairing may be broken down into three major pieces; the parabolic nose cone, the cylindrical parallel midbody and the conical tail section.

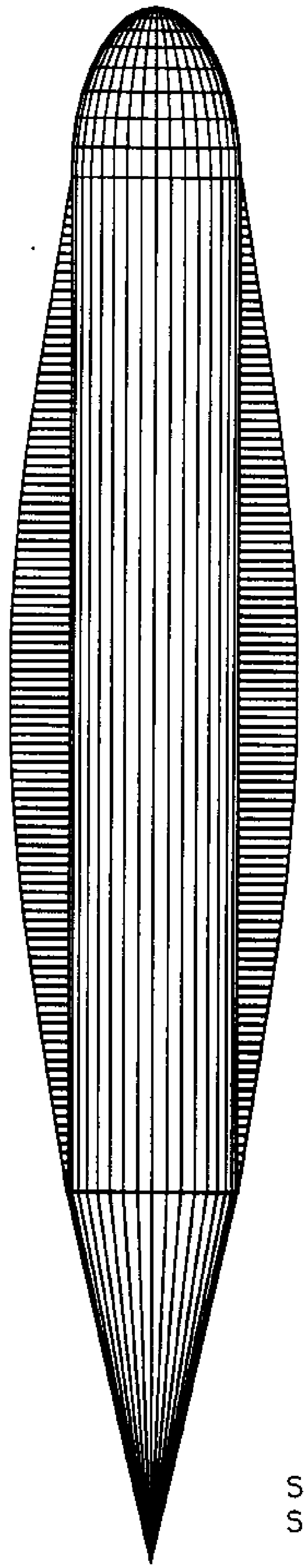


Figure 26

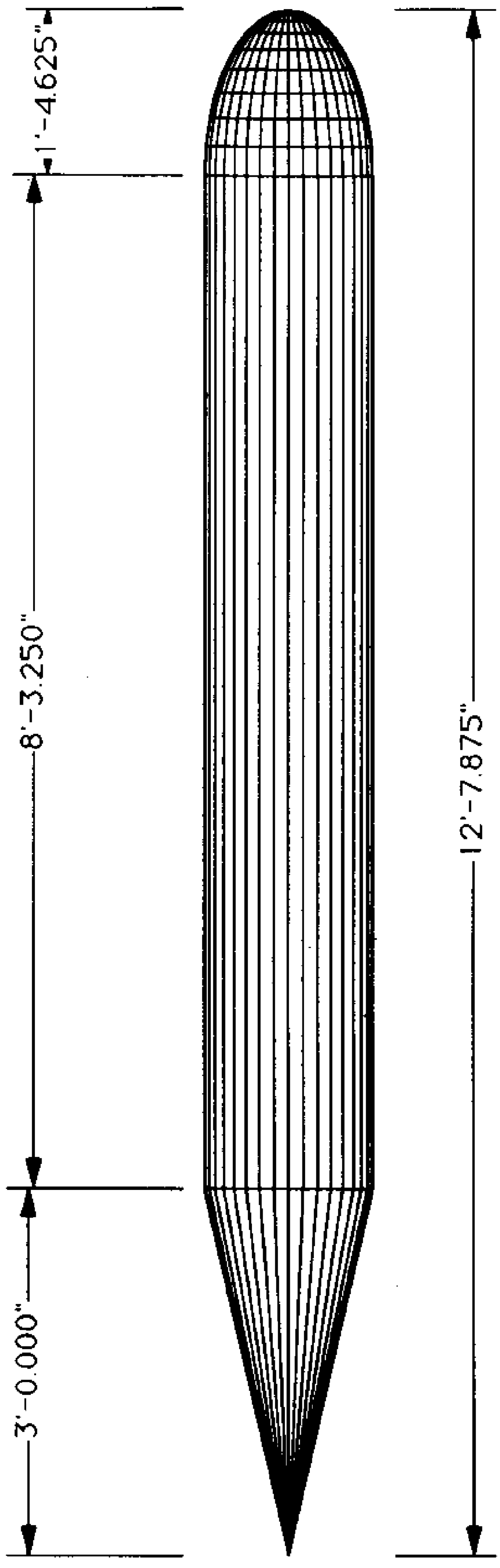


Figure 27

Scale 1:17

IV. Design Selection:

Parabolic Nose Section

The nose cone dimensions were dictated solely by the vehicle diameter, which in turn was dictated by the size of the flanges on the pressure vessel. The height of the parabola was set equal to the diameter. Accounting for a thickness of 0.5 inches, 0.125 inches for play, the height and diameter were determined to be 16.625 inches. This size nose cone allowed for the easy installation of a frame to support the data collection transducers.

Parallel Midbody

The parallel midbody was selected for its ease and speed of manufacture. The dimensions were set by the size of the pressure vessel. In the case where we could not find a fiberglass cone 16.625 inches in diameter and 8 feet long, a cone of the proper diameter could be quickly assembled by simply wrapping a rectangular sheet of Lexan around the flanges of the pressure vessel. This spacing between the pressure vessel and the outer fairing also allowed for the installation of ballast/buoyancy material between the pressure vessel wall and the inner wall of the fairing.

Tail Cone

The greatest requirement in this section of the hull was that it not produce any sharp discontinuities and thereby create additional turbulent flow. For this reason, the angle at the top of the cone was limited to 26°. This angle and the midbody diameter determined the length (2 feet 8 inches) of the tail cone.

V. Design Implementation:

Mr. Edward Briggs at Southwest Research Institute in San Antonio, Texas agreed to fabricate this fairing for us out of fiberglass at no cost. A dimensioned drawing of the assembled fairing may be seen in Figure 27. At the time of this writing the fairing has yet to arrive. Any further complications with this design are as yet unknown.

CENTER OF BUOYANCY/GRAVITY-BALLASTING

Design Considerations

The volume of air contained within the pressure vessel would have supplied a buoyancy force of approximately 367 pounds. The center of this buoyancy force can be thought to act along the longitudinal axis of the vehicle due to symmetry. In order to determine the center of gravity the mass and volume as well as location relative to the keel and the bow of each individual piece placed aboard the vehicle must be known. In order for the vehicle to remain neutrally buoyant, all of the combined weight of each part must be equal to the equivalent weight of water displaced by the vehicle. It is also important that the center of gravity be placed as low as possible below the center of buoyancy. This is necessary for transverse stability. The greater the moment arm (the distance between the center of gravity and center of buoyancy) the greater the stability. Next, the weights and moment arms of each part about the center of gravity must be calculated. In order for the vehicle to be stable longitudinally, these must sum to zero. All of the parts have not arrived so a complete ballasting for this vehicle was not finished.

FAILURE ANALYSIS

I. Description of Failure:

On Thursday April 12, 1990, testing of the pressure-hull design was begun in the UNH hyperbaric chamber. The vessel was placed horizontally on the chamber floor. A dive computer which was used to record the internal pressure of the vessel was placed inside the hull. The o-ring seals were checked for proper placement and the end caps were fastened to the vessel. The testing began with internal atmospheric conditions of 71°F and 1 atm. The camber was sealed shut and the pressure was increased at a moderate rate of 4 psi/min. At exactly 60 psi (a simulated depth of 141 feet under water) catastrophic failure occurred. The vessel walls had imploded. Due to the brittle nature of PVC (polyvinyl-chloride) , the vessel walls fragmented as opposed to yielding which is observed with most metallic failures. With the exception of the two end-caps and the mid-way connection disk, the largest remaining piece of a hull whose original surface area was over 38000 in^2, was measured to be only 40 in^2.

At this point the testing was immediately stopped, and decompression of the chamber begun. Upon entering the chamber, the group members took care to photograph and document the aftermath. This event was totally unplanned for and took the entire group by surprise. An emergency meeting was held the next day to determine our options and future plans.

Testing of the vehicle was entirely out of the question without a pressure hull. The group agreed to temporarily halt construction of the remaining sections and concentrated on doing a complete and extensive analysis on the failed hull.

II. Analysis of Possible Causes of Failure:

The failure investigation may be broken down into three primary areas of consideration. The first area involves the material itself. The second involves a failure due to a flaw in the design. The third area involves a problem caused by improper manufacturing.

Material Failure Analysis

This category is the most difficult area of investigation as the material was completely destroyed in the implosion. Areas under consideration are as follows:

- Cracks in the cylinder wall
- Bubble in the cylinder wall
- Axially prestressed during cylinder extrusion
- Cylinder out of round
- Material was too brittle for this application

A problem such as a crack or bubble in the cylinder wall is impossible to prove at this point. We have requested new samples of the same material from Eptam plastics. Once this arrives, tests for Young's modulus, Poisson's ratio and Yield Strength will be made.

Design Failure Analysis

According to Comstock [12], there are three principle modes of failure of a pressure hull. They are failure by buckling of shell, failure by yielding and failure by general instability. Failure by buckling is characterized by the formation of dimples around the vessel. Yielding is identified by the 'accordion' effect located between ring stiffeners. General-instability is recognized by large dished-in shape deformations traversing the length of the hull. A drawing of these deformations can be seen in figure 28.

Since the pressure vessel completely shattered upon failure, analysis on the cause, or even the type of failure, has become quite difficult. Comstock suggests that the optimal design allow

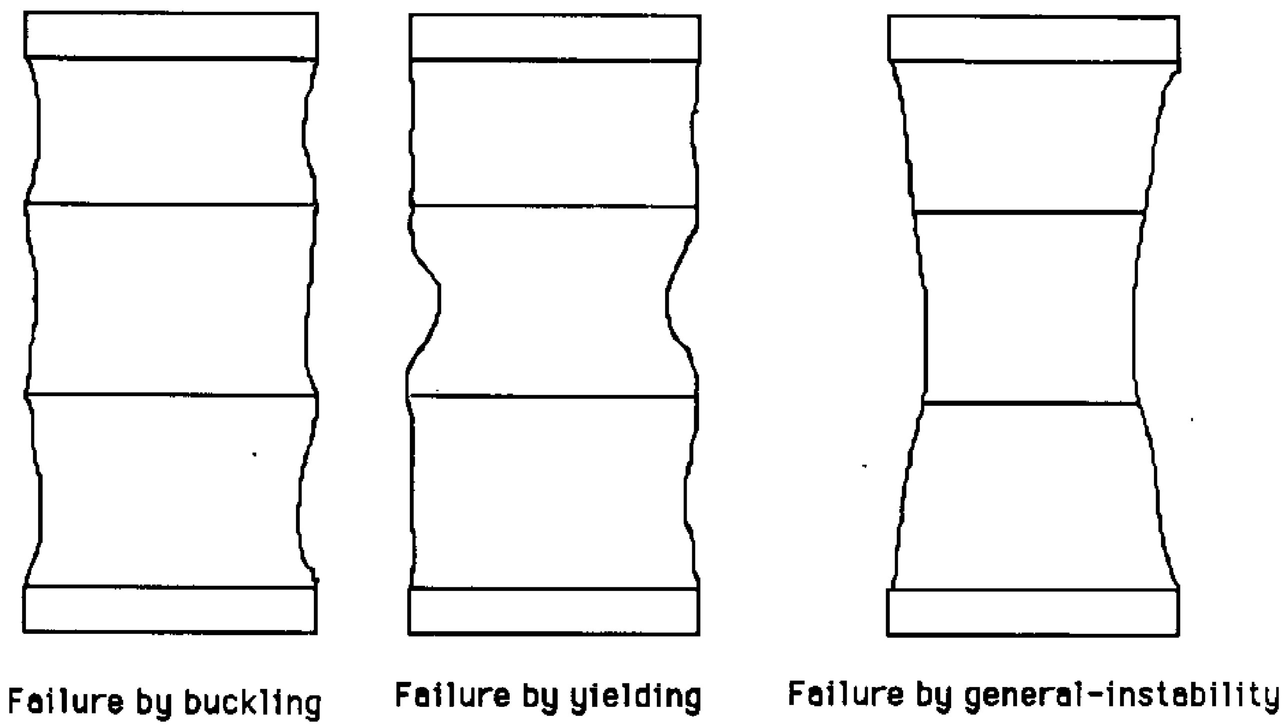


Figure 28: Three typical modes of failure.

the vessel to fail by yielding, with internal supports just large enough to prevent failure by general-instability.

To calculate whether a vessel will fail by yielding, the following equation for the pressure factor is used.

$$\psi = \frac{1.30}{\lambda^2}$$

where the slenderness ratio λ is

$$\lambda = \left[\frac{L/D}{(t/D)^{\frac{3}{2}}} \left(\frac{\sigma_y}{E} \right) \right]^{\frac{1}{2}}$$

L = length of stiffened shell between supports

D = diameter of shell

t = thickness of shell

σ_y = yield stress of shell material

E = modulus of elasticity

A plot of the pressure factor versus the slenderness ratio can be seen in Figure 23. Entering the calculated L value shows immediately that the failure did not occur by yield of buckling. "Both of these modes of failure assume that the frame has sufficient cross sectional area and stiffness to enable the actions to take place. If the frame is deficient, failure will take place in a general-instability mode at a pressure lower than that indicated by Figure 29. The general-instability mechanism of failure is associated with overall body collapse [12]. At this point in time, judging from the magnitude of destruction of the hull, the overall body collapse through general instability seems like the most plausible mode of failure.

To calculate the collapse pressure of the vessel, the following equation is used.

$$P_c = \frac{2.42\,E\,(t/D)^{\frac{5}{2}}}{(1-\mu^2)^{\frac{3}{4}} \left[L/D - 0.45\,(t/D)^{\frac{1}{2}} \right]}$$

P_c = collapse pressure, psi

E = Young's modulus of material, psi

μ = Poisson's ratio

D = diameter to midplane of shell, in.

L = unsupported length of shell plating

Another factor which significantly affects the P_c, is the bulkhead spacing. The bulkhead spacing is defined as the axial distance between the two end caps. The general rule of thumb is to keep the spacing between supports or stiffeners within 1 to 2 diameters. Our vessel used a spacing of 4 diameters.

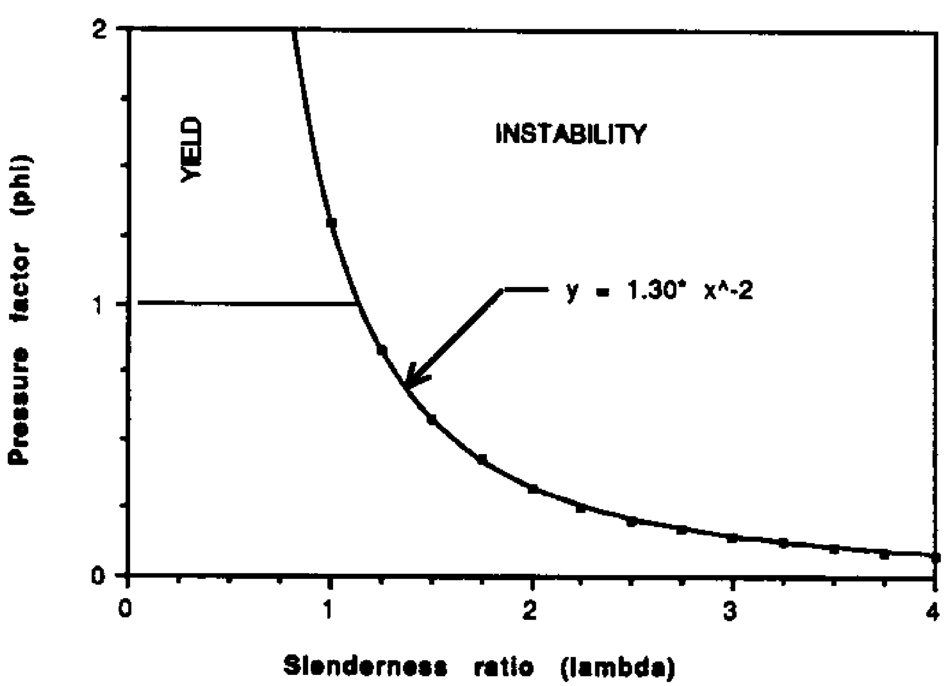


Figure 29: Pressure factor vs. slenderness ratio

Our vessel failed at a pressure 25% less than the calculated collapse pressure. However, eccentricities such as imperfect circularity with nonuniform thickness and induced stress concentrations from the manufacturing process (see figure 30) may have accounted for this difference. These sharp corners have been know to significantly decrease both yield and ultimate strength. These sharp corners are necessary in order to insure a proper O-ring fit.

Although the exact cause of the failure may never be known, the above results show that the most likely cause of failure was buckling under general-instability. This raised the question, 'could this accident have been avoided?'

In hind sight, the answer is an unequivocal yes. The following are some possibilities of things that might help in avoiding future failures.

(a) Place strain gages on vessel and orient them in such a way as to detect the deflection of the vessel walls. When a significant deflection is measured, testing should be stopped.

(b) Analyze specimens of vessel material before manufacturing. Determine the exact values for the specific gravity, Young's modulus of elasticity in flexure, compressive strength, compressive yield strength and Poisson's ratio. After these values have been obtained, the appropriate strength analysis should be done. (However, the specimen testing procedure can be avoided if the properties of the chosen material are well documented).

(c) The addition of shell stiffeners will reduce the chance of failure by general-instability. However, expenses such as fabrication and machining may prove impractical.

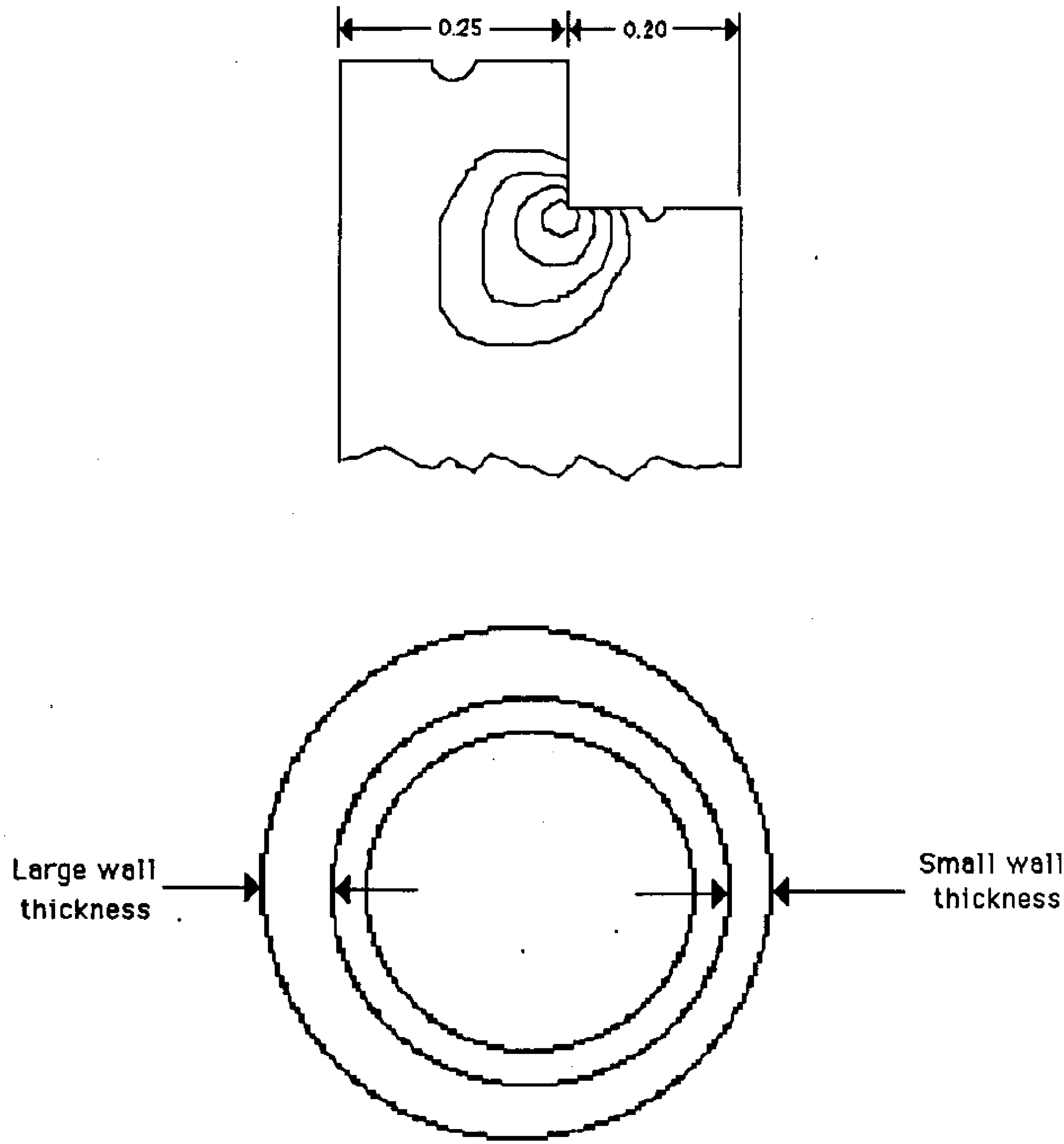


Figure 30: Pressure Vessel eccentricities

Manufacturing Failure Analysis

Areas under consideration include:

-Induced stress caused by the mounting of the end cap flanges

-Brittle Welds

-Notches cut to lay welds in.

Evaluation of these areas involves the observation of how Eptam assembles cylinders and flanges. A visit to the manufacturer will made to evaluate these areas.

CONCLUSIONS

Although still in the early stages of development, autonomous, underwater vehicles have proven to be extremely useful in the collection of data and study of underwater phenomena. By examining the feasibility of this type of vehicle during the academic school year 1989-1990, the members of this group have experienced the entire design process. During the process, we have had to deal with several problems, which proved to be a valuable insight into the total project management and operation process for the team members.

The first problem was to receive approval of a budget. In all projects, management has ultimate control by controlling the amount of money a project receives. Our initial cost estimate exceeded the imposed limit. After cost cuts and performing a line-by-line justification of each item, a budget was approved. Although it was much lower than what we had hoped, it proved to be adequate.

Next, each group member was assigned various tasks relating to a vehicle sub-system. Analysis performed on each sub-system was most likely interdependent on other sub-systems. Therfore, any change in design could conceivably alter any or all of the analytical results. Communication between group members was of paramount importance.

Finally, the old adage states "...you learn from your mistakes." After the implosion of the pressure vessel, we learned how to cope with failure and deal with it in a positive manner. Failures always occur in prototype developments and ours was without exception. The valuable insight we gained from the implosion will undoubtedly better prepare the project team members for future careers.

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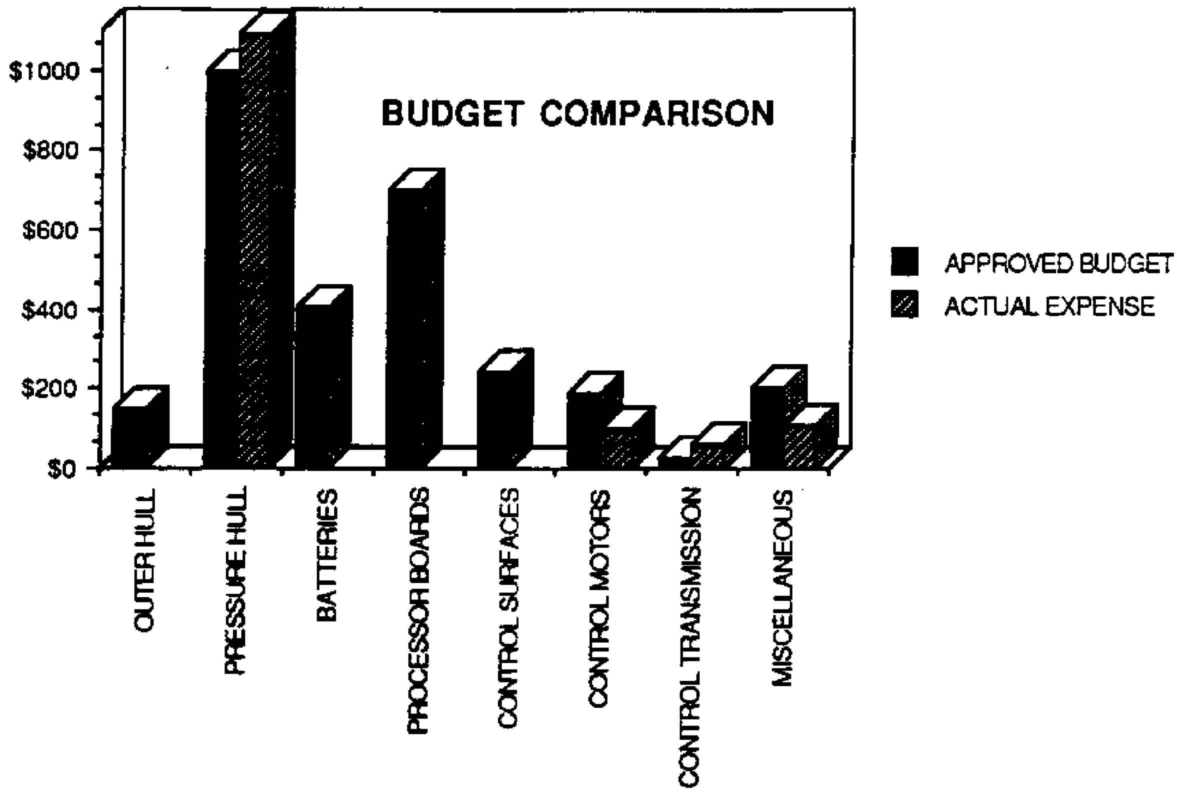
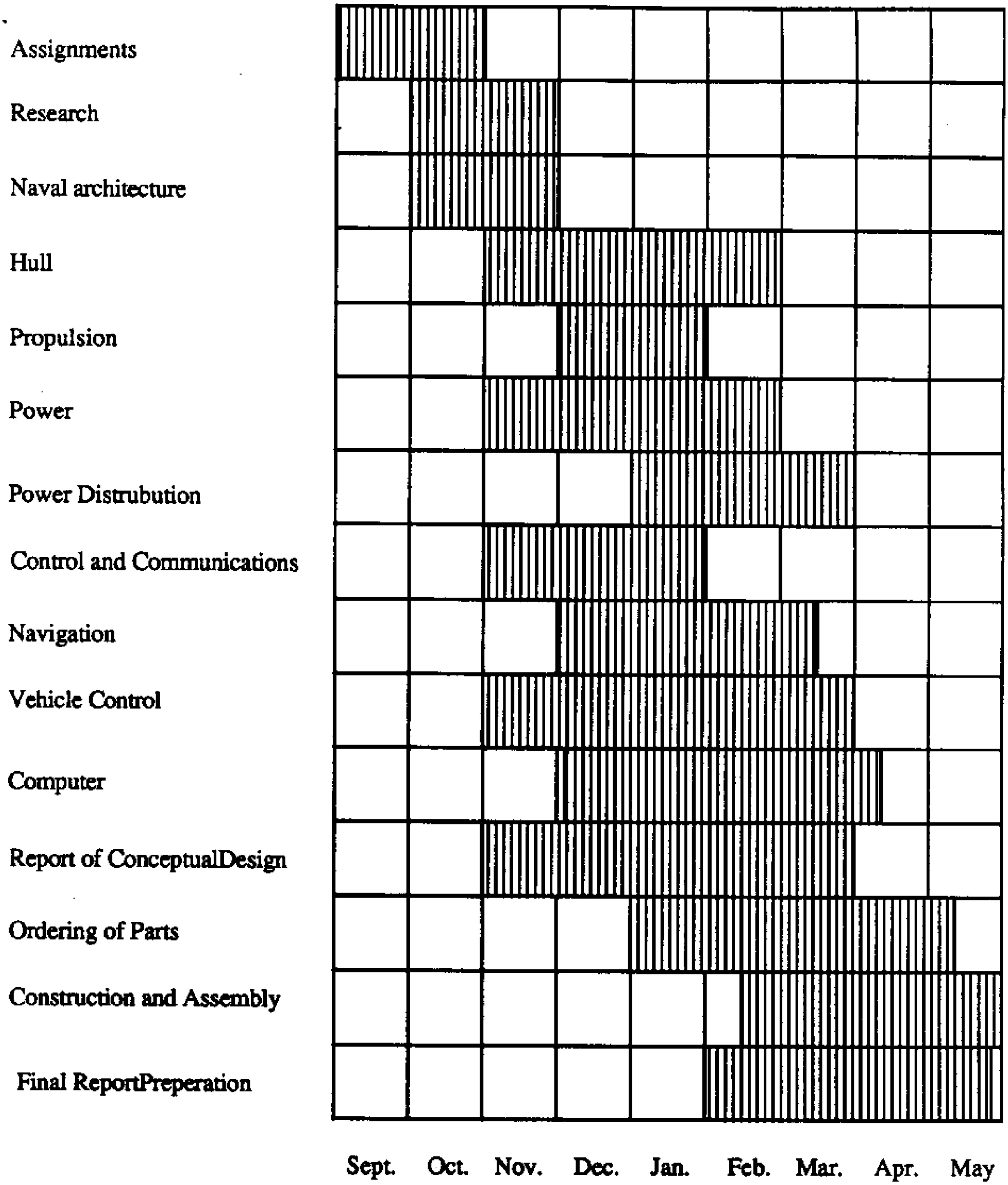
APPENDICES

APPENDIX A



Initial Budget Estimate	$5,600
Approved Budget	2,910
Expended (5/1/90)	1,662

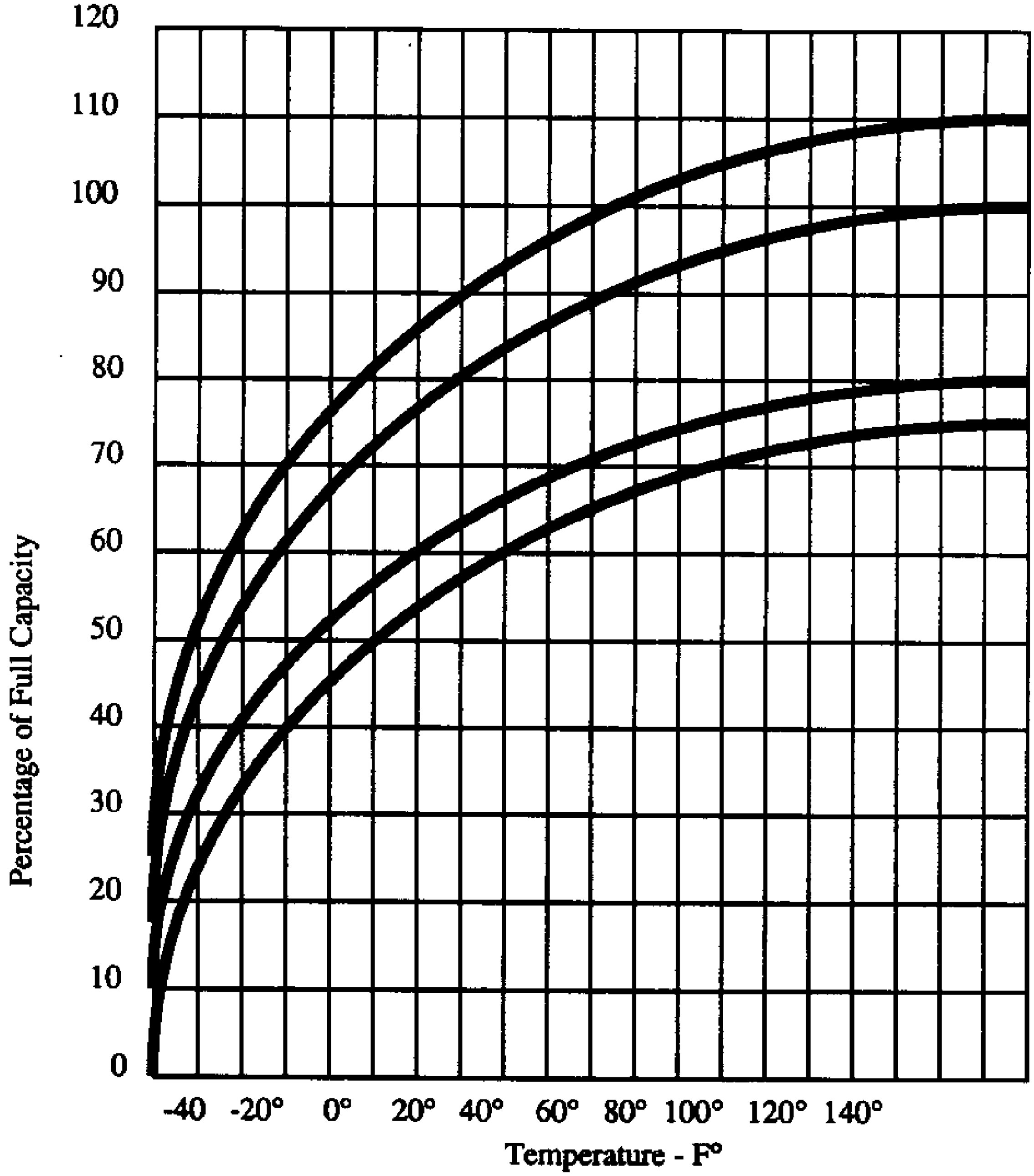
APPENDIX B



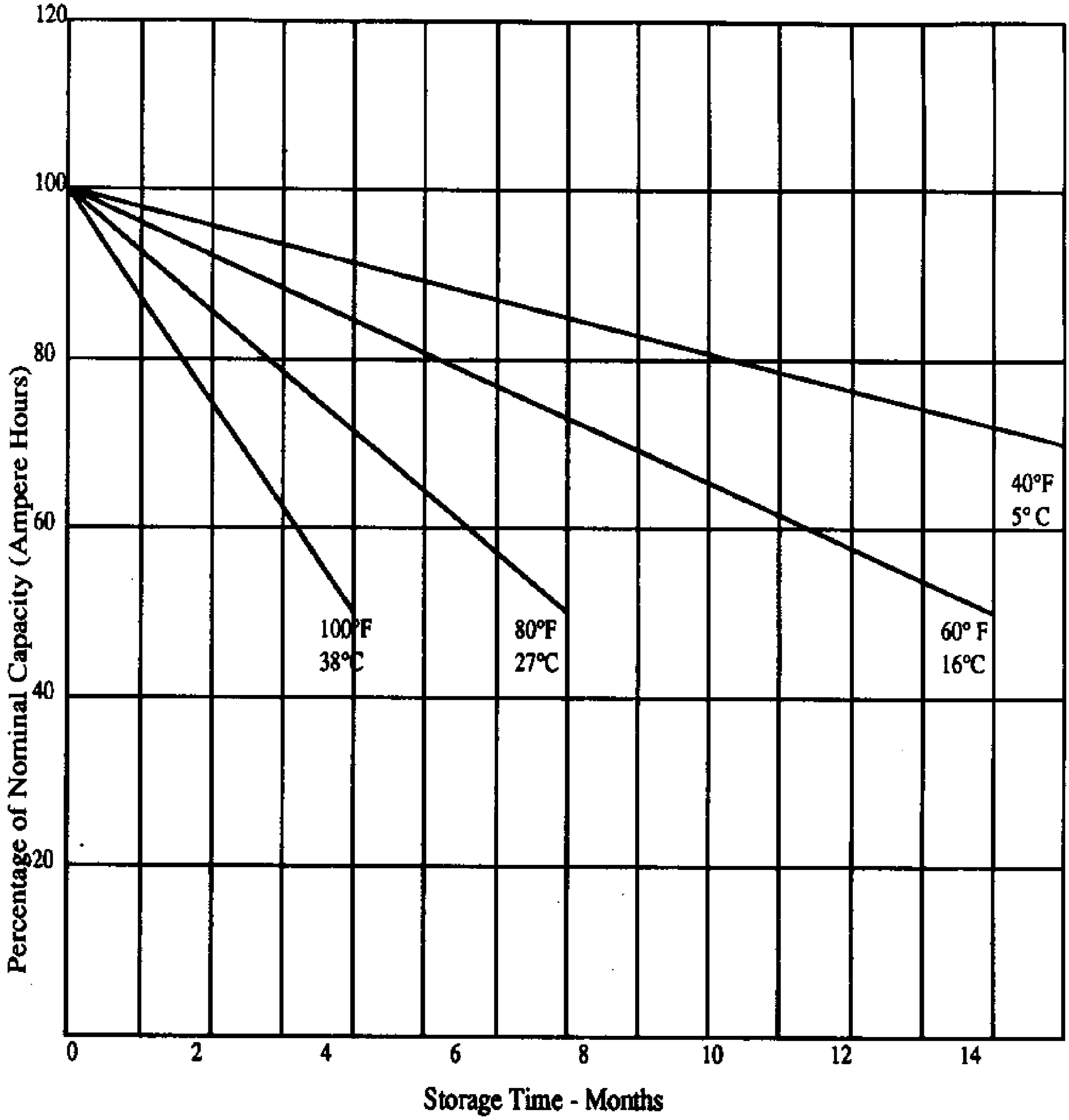
SLAVe project timeline

APPENDIX C

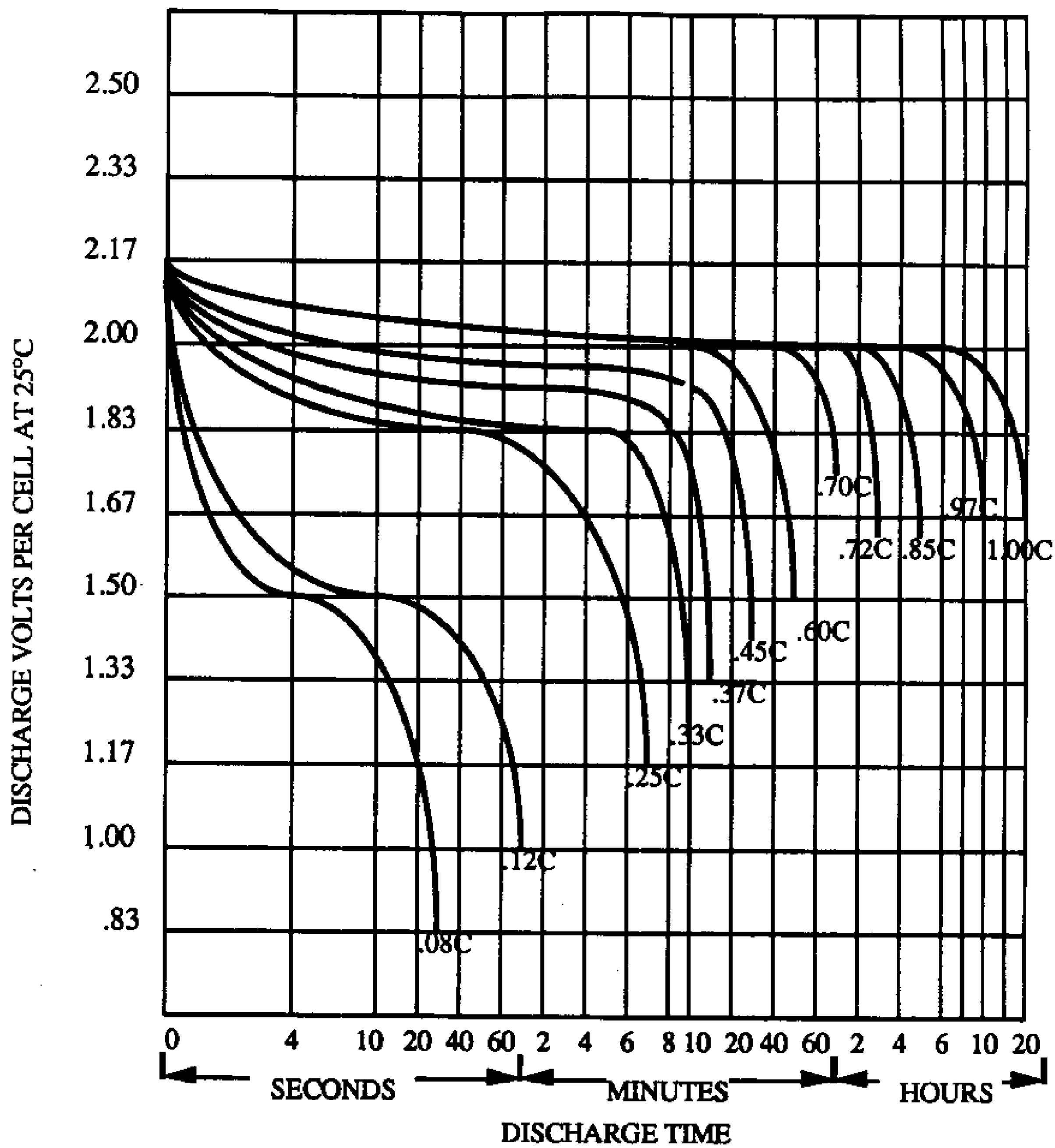
Battery Characteristic



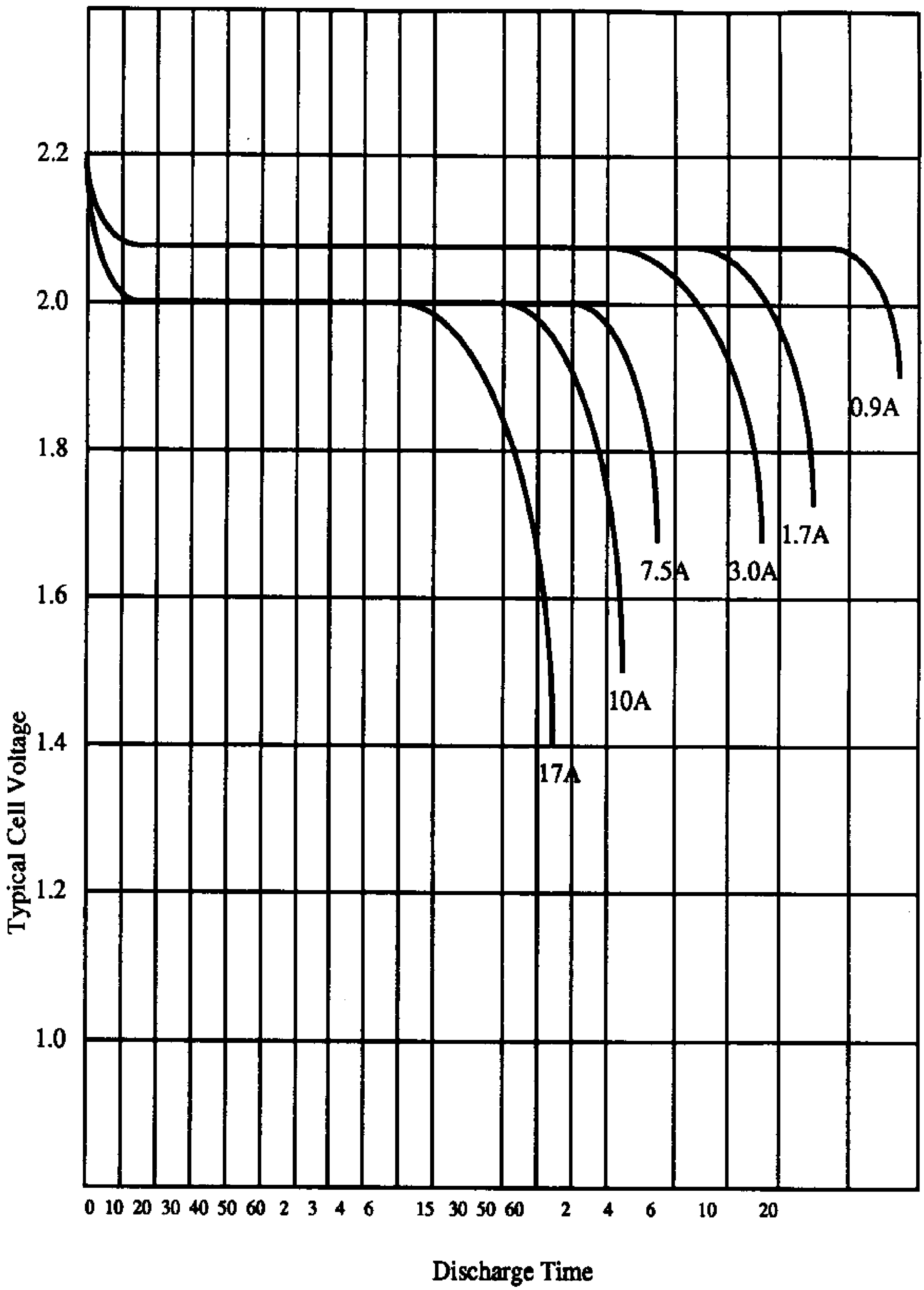
Capacity as Affected By Temperature



Typical Self Discharge Characteristics



TYPICAL CELL CAPACITY RATINGS AT VARIOUS RATES OF DISCHARGE



Typical Voltage Characteristics (70F)

APPENDIX D

WEIGHTED EVALUATION FOR CONTROL SURFACES

Weighted Evaluation for Control

	A	B	C	D	E	F	G	H
1			Appendix D					
2								
3		Evaluation of alternative Maneuvering Concepts						
4								
5								
6								
7			Alternative Solutions					
8								
9								
10					Partially			
11	Design	Weighting	Fixed		Trainable		All-Movable	
12	Criteria	Factor	Surface	QA	Surface	QB	Surface	QC
13	Cost	7	5	35	0	0	4	28
14	Reliability	9	5	45	2	18	4	36
15	Safety	8	5	40	4	32	5	40
16	Fabricability	6	5	30	3	18	5	30
17	Performance	10	0	0	3	30	5	50
18	Control	10	0	0	3	30	5	50
19								
20	Totals		20	150	15	128	28	234
21								
22	Evaluation	0	Least desirable or least effective					
23		5	Most desirable or most effective					

APPENDIX E

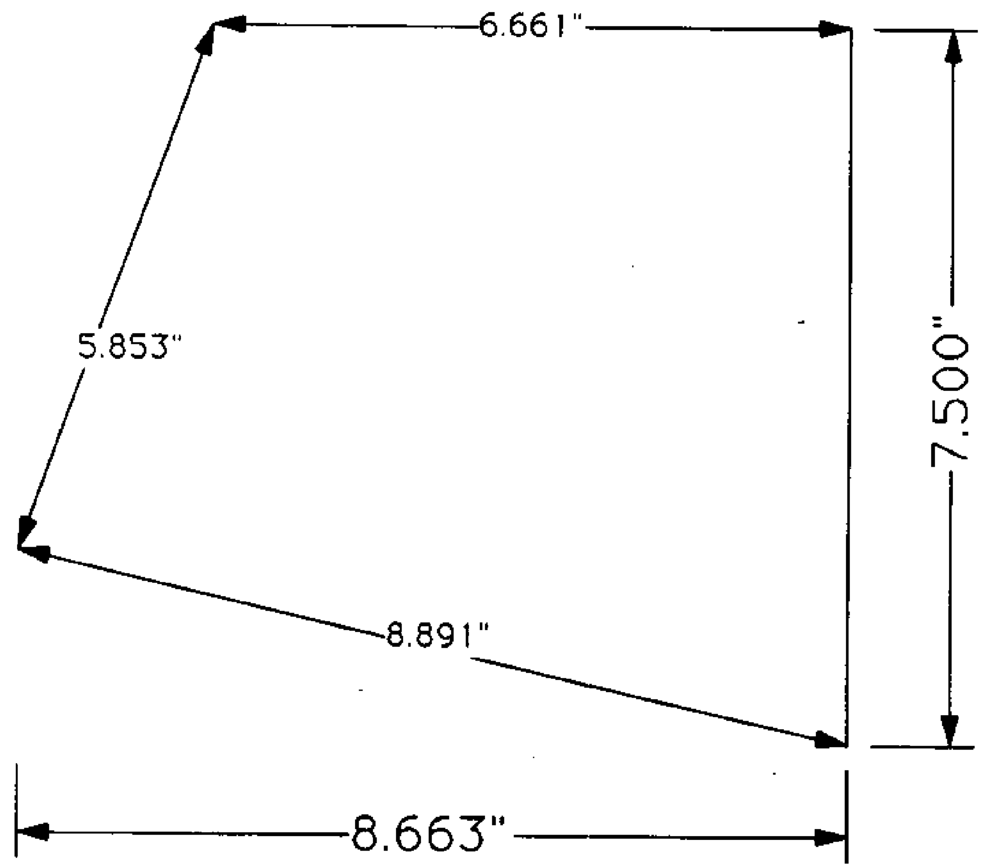
Spreadsheet for Control Surface

Appendix E			
TECH 897			
Revised Spreadsheet containing DATA relating to HORIZONTAL STERN PLANES			
Last Revision:	January,19 1990		
TABLE OF CONSTANTS			
Term		Value	Explanation
a(o)	=	0.098	Section-lift-curve slope
a(e)	=	0.832	Effective aspect ratio, b^2/S
b/2	=	5.500	Semi-span, measured perpendicular to plane of root section (in)
c-bar	=	7.662	Mean geometric chord, (c(t)+c(r))/2, (in)
C(D)	=	xxxxxxxxxxx	Drag coefficient, (Drag/qS)
C(D(o))	=	0.007	minimum section drag coefficient
C(D(c))	=	0.800	Crossflow drag coefficient
C(L)	=	xxxxxxxxxxx	Lift Coefficient, (Lift)/qS
C(L(alpha))	=	0.021	Lift curve slope
C(m(C(L)))	=	0.111	Coefficient given by Hembold's equation
C(m(c-bar/4))	=	xxxxxxxxxxx	Torque of pitching moment coefficient about quarter-chord point of mean geometric chord,
			(pitching moment)/qSc-bar
(CP)c-bar	=	xxxxxxxxxxx	Chordwise center of pressure measured from leading edge at mean geometric chord in
			percent of the mean geometric chord
(CP)s	=	xxxxxxxxxxx	Spanwise center of pressure measured form plane of root section in percent of semispan
M(shaft)	=	xxxxxxxxxxx	Moment exerted about shaft (in-lbf)
e	=	0.900	Oswald efficiency factor
q	=	9950.422	Dynamic pressure, rho*U^2/2
R	=	324323.752	Reynolds number, U*c-bar/v
S	=	50.804	Planform area (in^2)
U	=	8.440	Free-stream velocity (ft/sec)
alpha	=	xxxxxxxxxxx	Angle of attack (deg.)
lambda	=	0.769	Taper ratio, c(t)/c(r)
v	=	1.66E-05	Kinematic viscosity (ft^2/sec)
Ω	=	10.243	Angle of sweep of quarter chord Line (deg.)
L1	=	xxxxxxxxxxx	Distance from leading edge to (CP)c-bar, measured along mean chord line (in)
L2	=	1.916	Distance from leading edge to quarter-chord, measured from root chord (in)
L3	=	2.753	Distance from leading edge to shaft, measured at root chord (in)
rho	=	1.940	Density of fresh water between 37 and 40 Degrees Farenhite (lbf-sec^2/ft^4)
Vel	=	8.440	Velocity of hull (ft/sec)
Diam (ave)	=	16.625	Chosen (in)
L (nose cone)	=	16.625	Parabolic nose cone = Diam ave. (in)
L (midbody)	=	99.250	Chosen such that L/D overall is between 6 and 8 (in)
L (tail cone)	=	36.005	(@ slope of 13 degrees) (in)
L (TOTAL)	=	151.880	Sum (in)
L/D overall	=	9.136	
L/D tail cone	=	2.166	
HORIZONTAL PLANE DIMENSIONS			
a	=	5.500	Chosen such that a+d=Diam/2 (in)
b	=	6.661	(@ 13 Degrees) (in)
c	=	2.002	(@ 20 Degrees) (in)
d	=	2.000	Chosen such that a+d=Diam/2 (in)
L.E.	=	5.500	(in)
T.E.	=	7.500	(in)
Eff. Span	=	5.500	(in)
Tip Chord	=	6.661	(in)
Root Chord	=	8.663	(in)
Mean Chord	=	7.662	(in)
Aspect Ratio	=	0.848	Ratio of effective span to mean chord
A.R. (b^2/A(p))	=	0.832	Ratio of mean span to profile area
Taper Ratio	=	0.769	
L.E. to Shaft	=	2.753	Distance from leading edge to shaft (in)
Hor. Profile Area	=	50.804	calculated (in^2)
Hor. P.A. (desired)	=	49.317	L*D/(2*25.6) (in^2)

PARAMETERS VARYING WITH ANGLE OF ATTACK FOR HORIZONTAL STERN PLANES

C(L)	C(D)	C(m(c-bar/4))	(CP)c-bar	(CP)s	C(M(shaft))	M(shaft)	L1	alpha (deg)
0.000	0.007	0.009	0.125	0.427	0.000	0.000	0.960	0.000
0.031	0.007	0.008	0.141	0.427	0.007	1.290	1.083	1.000
0.063	0.008	0.009	0.156	0.427	0.013	2.406	1.199	2.000
0.098	0.010	0.009	0.171	0.428	0.018	3.390	1.307	3.000
0.129	0.014	0.009	0.184	0.428	0.023	4.246	1.406	4.000
0.162	0.018	0.009	0.196	0.430	0.027	4.981	1.502	5.000
0.197	0.023	0.008	0.208	0.431	0.030	5.610	1.581	6.000
0.232	0.029	0.007	0.218	0.433	0.033	6.141	1.674	7.000
0.267	0.037	0.006	0.229	0.436	0.036	6.582	1.751	8.000
0.303	0.046	0.004	0.238	0.438	0.037	6.942	1.824	9.000
0.340	0.056	0.001	0.247	0.441	0.039	7.226	1.891	10.000
0.377	0.067	-0.002	0.255	0.444	0.040	7.446	1.955	11.000
0.415	0.080	-0.005	0.263	0.447	0.041	7.606	2.013	12.000
0.453	0.094	-0.009	0.270	0.451	0.041	7.716	2.068	13.000
0.492	0.109	-0.013	0.277	0.455	0.042	7.776	2.120	14.000
0.531	0.127	-0.018	0.283	0.459	0.042	7.799	2.167	15.000

APPENDIX F



Horizontal Control Surfaces (Planes)

Mechanical Engineering-UNH

Frederick Murdock Scale 1:2