VARIABLE BALLAST SYSTEM DESIGN

FOR AN UNMANNED SUBMERSIBLE

by

ALF L. CARROLL, III

BSME, Northeastern University
(1980)

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Signature of Author ______________________________
Department of Ocean Engineering
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Certified by ______________________________
Thesis Supervisor

Accepted by ______________________________
Chairman, Department Committee

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ABSTRACT

A high first-cost, low maintenance Variable Ballast system was designed for use on an unmanned submersible. Since the submersible vehicle was to be capable of diving to 1000 ft and operate unattended for a period of one month, the performance requirements on the ballast system are quite robust. Possible types of systems were compared and a seawater hydraulic concept was selected for its reliability and inherent efficiency. The major emphasis of this thesis was focused on selecting components for the system that could meet vehicle control requirements and be compatible with the ocean environment. Some of the components are developmental in nature and merited a substantially detailed discussion.

Thesis Supervisor: Dr. Damon Cummings

Title: Visiting Lecturer, Ocean Engineering Department

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I would like to personally thank Dr. Damon Cummings for the consistently practical guidance he put forth during this design effort, especially during the more perplexing moments which occurred in the conceptual phase. In addition, the valuable insight which Mr. Bernard Murphy offered during his busy schedule always served to sharpen my focus on the unified function of the components as a whole system.

On a more somber note, I would like to dedicate this thesis to my close friend and former co-worker, the late Erik Johannson, who served as a prime impetus and inspiration for much of the early materials applications ideas. He was an extremely practical young engineer with a zealous interest in new developments. I think of him often and wish that he could still be here to talk with me about the challenges of research and development he found so intriguing. May he rest in peace.

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SECTION 1

INTRODUCTION

1.1 General

In order that any vessel may be described as a "submarine" or "submersible," it must (by definition) be capable of developing states of negative, neutral, and positive buoyancy so that it can:

(1) Submerge from and return to the surface.

(2) Dive in a controlled fashion down to any prescribed depth.

(3) Operate at given depths within the ocean thermocline.

Providing this key control of vehicle motion in the vertical plane is the job of the Variable Ballast System. As a submersible changes its operating depth, the buoyant hull and tankage volumes of the ship also vary as a result. Local temperature and salinity variations cause a change in seawater density, and thus in the net upward buoyant force acting upon the vehicle. Any other change in vehicle buoyancy may be affected and/or adjusted to through actuation of the Variable Ballast System.

Simplified early ballast systems satisfied the limited performance requirements of the sub in a reasonably reliable fashion without unreasonable cost. After all, most of the assigned missions of these early submarines were to seek out, find, and sink a particular adversary ship that might be anchored in an occupied harbor. Therefore, it was unnecessary for these early subs to possess the ability to repeatedly ascend and descend during any given mission. As science and technology became more highly developed in years to follow, steel-hulled submarines propelled by steam, diesel, and other forms of power evolved. These advances ushered in concurrent developments in supporting ships's systems technologies, such as steering and diving linkages, actuators, batteries, life support, and ballast systems. Existing performance levels for certain subsystems soon became a limiting factor to the overall capabilities of a particular submarine because of concerns for safety, survivability, maintenance costs, endurance, and other important parameters. As the operating depth capabilities of these modern underwater ships increased, so did the risks associated with:

(1) Designing pressure hulls and other "free-flooded" equipment which would safely resist the increased hydrostatic pressures.

(2) Incorporating a ballast system design that would reliably and repeatedly respond to commanded changes in the state of buoyancy, depending on the design specifications and operational modes of particular submersibles.

Variable ballast systems all accomplishing similar tasks have been developed that vary quite markedly from one another in terms of cost, complexity, reliability, and functional form.

1.2 Design Approach

When first beginning the design of a critical submersible subsystem such as variable ballast, it is quite advisable for one to perform a review of the ballast systems on other submersibles already in service. Of course, the number of vessels in existence indicates a definite need to decrease the breadth of search such that it is only within a certain class of vehicles. Those having similar mission and performance requirements are obvious candidates for "first-order" levels of evaluation, but careful attention should also be given to the variety of design approaches taken for other vessels focusing on quite a different set of performance attributes. This methodology can become quite essential in many design selection procedures, particularly when a unique combination of design parameters may suggest a mixed selection of system features. The engineering effort must involve an assessment of the significant parameters in each ballast system design (and their relation to parameters in other concepts) so as to obtain a resultant configuration that is an optimal compromise of all relevant variables imposed by the ambient environment, performance specs, etc. Alternatively, technological advances in other fields may also combine with fairly standard types of ballast control systems to produce a design suited to perform in much more extreme conditions than a current state-of-the-art version of the particular system design. The final system selected and designed in this thesis is an example of the later case, where several technologies have been advantageously applied to reduce the constraints on a conventional hard-tank seawater system so that this approach can become a reliable mode of buoyancy control for a small, unmanned, untethered submersible. Discussion of the details of relevant developed and existing technologies will appear later on; first however, the author presents the section, "Background and System Selection," which compares the virtues and shortcomings of various systems, making tradeoffs applicable to a submersible which is designed to be suitable for long endurance underwater applications. Information from these comparisons will be quite useful, both as a learning process and as a means by which the final design can be justified.

SECTION 2

BACKGROUND AND SYSTEM SELECTION

2.1 Variable Ballast System Criticality

A point that received major emphasis in the previous discussion was the importance of variable ballast and other ships' systems in realizing an efficient, reliable vehicle design. It should be noted, however, that the relative criticality of each system is greatly dependent upon what overall requirements are imposed on the submersible design. For example, a simple floodable tank (to submerge and dive) ballast system that can drop off lead weights (to ascend) after sinking to the ocean bottom might be entirely satisfactory for a vehicle intended only for a "dive once and retrieve" salvaging mission. But, despite the desirable trait of simplicity, a system like this would certainly not be suited to a vehicle which required extensive and periodic changes in buoyancy over a lengthy mission without maintenance.

Many authorities in the field of ocean hydraulics would agree that a reliable variable ballast system is perhaps the most singularly important element in the completion of any deep sea vehicle work mission (Reference 1). If the vehicle is also unmanned and untethered, the reliability and robustness of the system becomes even more critical in the design. Without a crew, on-board maintenance and manual overrides to accommodate a partial system failure are impossible to obtain, while untethered vehicles might be lost if enough positive buoyancy cannot be obtained when the sub tries to surface. A high degree of reliability is thus essential to the success of any "free-swimming' submersible.

2.2 Impact of Vehicle Requirements on Variable Ballast System Design

The variable ballast system which is described in this thesis was intended to be a high reliability, low-maintenance design that would be suited to an underwater vehicle like the MIT Robot II submarine. However, the design specifications for this new vehicle are similar in concept, but not in extent, to the Robot II, one of the first successful free-swimming submersibles to use an on-board computer for navigation and control functions. Notice Table 2-1 which compares the key parameters of the Robot II and those of the vessel discussed herein that shall be dubbed High-Performance Underwater Vehicle (HPUV).

Table 2-1

Comparison of Parameters between Robot II and HPUV.

	MIT ROBOT II	HPUV
Length	Approx. 8 ft	Approximately 14 ft
Diameter	Approx. 10 in.	20 in.
Maximum Operating Depth	350 ft	1000 ft
Maximum Endurance	10 hours	1 month
Energy Source	Lead-acid or gel-type batteries	Silver-zinc or other high energy density batteries
Vessel Control Mode	Preprogrammed computer	Acoustically commanded or preprogrammed computer

Of the characteristics listed in Table 2-1, the operating depth and endurance are the two which have the greatest impact on the variable ballast system. Although many unmanned vehicles are currently used extensively in the offshore oil and gas industries, the aggressive specifications of the HPUV make it capable of a variety of missions that these conventional designs could never perform. For example, salvage operations in normally inaccessible areas, uninterrupted mining or rescue search operations near the ocean bottom, or subbottom mappings are all possible missions for the HPUV. Therefore, the designed characteristics for a VB System in this vehicle could be generally categorized as follows:

(1) The volume and weight of all components comprising the total VB System package must be held to a practical minimum to allow space in the vehicle for other subsystem's components and preserve the maximum possible mission payload capacity. In general, a minimum dependence on other ships' systems is also desired.

(2) Since the extra weight and space of high-pressure air or hydraulic systems are not feasible for such a relatively small vessel, it was assumed that only electrical power is available to drive motors, pumps, etc. Therefore, the amount of battery energy required to perform ballasting evolutions should be minimized to retain extra power for mission payloads and to optimize vehicle endurance.

(3) A capability to reliably perform enough operational cycles to support the completion of a particular mission and in any partially-failed vehicle modes is essential. So, a system that is simple in design and/or uses simply functioning components becomes implied.

(4) The system should be normally operable throughout the full range of hydrostatic pressure (i.e., from the surface down to design depth) and any components mounted in the exposed free-flood areas (within external bow or stern fairings) should be tolerant of the ocean environment.

(5) The noise radiated by the VB System when it is operating should not be excessive enough to interfere with sonar navigation, acoustic communications with a surface vessel, etc.

2.3 Comparison of System Alternatives

With the previous parameters in mind, a tradeoff study was performed that compared five general types of ballast systems for this application. Each type of system is briefly described below:

(1) Expandable Oil-Bladder* - This approach uses a conventional oil-hydraulic pump to move oil in or out of an elastic bladder located in the free flood areas, thus changing net buoyant volume of the vehicle.

(2) Movable Piston Concept* - A method which changes the net buoyant volume of the hull by moving a sealed piston in or out of the pressure hull.

(3) Fixed Weight Dispensing System - The vehicle obtains an increase in net buoyancy by dropping lead shot, thus reducing its weight. It was assumed that another system must be designed so that the vehicle can also decrease net buoyancy (like tank flooding).

(4) Seawater Ballast System with a High Pressure (HP) Air Flask for Emergency Blow - This is the typical design used on fleet submarines; it simply pumps water in or out of hard ballast tanks to effect a change in displaced volume, and thus a change in net buoyancy. The High Pressure Air Flask would only be included so ballast tanks could be "blown out" at depth and the vehicle recovered if the ballast pump or other critical components should fail in operation.

(5) Seawater Ballast System, HP Air Regulated - The same principle as (4) above, except air pressure is used to help discharge water from the tank at depth and/or maintain a regulated differential pressure inside the ballast tank relative to the ambient hydrostatic ocean pressure.

*(1) and (2) are both "closed-loop" type systems. Others are "open-loop."

A brief description of the major advantages and disadvantages for each type of systm has been prepared (see Tables 2-2 through 2-6). Although the final selection of the appropriate system may seem obvious* after reviewing the "pros and cons" of the candidates in light of vehicle design parameters, the concluding summary at the end of this section states specifically the reasons for selecting a conventional unregulated seawater ballast system. One other general tradeoff comparison should be made pertaining to the possible use of a trimming system which is independent from the Variable Ballast Systems, before the final system selection is discussed.

2.4 Independent Vehicle Trim Systems

For this submersible design, it is apparent that fore and aft weight distribution changes would be most reliably accomplished as a joint task by independent Forward and Aft Variable Ballast Systems via a control system function which monitors inclinometer sensors, inputs these readings into vessel control (mission) logic, and then issues an appropriate ballasting/trimming command. Considering a separate trimming system as an alternative, first note than an additional system inside the pressure hull (like coiling a line or cable between two "wind-up" spools displaced from each other along the ship's axis) will complicate the location of other components, and hinder maintenance and assembly procedures. Mercury transfer systems require oil-hydraulic components and boundary membranes to affect a shift in the mercury volume thus adding too much complexity to a relatively simple design. Also, mercury is a toxic and dangerous fluid that can never be brought aboard a vessel without special handling procedures. The last concept,

*It should be noted here that most all vehicles, whether manned or unmanned, which have performance requirements similar to those of HPUV, utilize conventional seawater ballast systems. However, the operational modes and individual components of the system may vary greatly from one another depending on the specific needs of each design.

Table 2-2. Expandable bladder system.

ADVANTAGES	DISADVANTAGES
1) Oil as cooling fluid thus, conventional components for pumping oil can be selected 2) Is a "closed-loop" system, having no dynamic interfaces with saltwater, a contaminant. Therefore, it is less susceptible to internal biofouling or corrosion	1) Hydraulic components are required, and protection from the ambient environment is needed 2) Unrestricted free-flood space is required to allow for expansion and contractions of bladder(s) which effect a change in buoyant volume 3) Oil is extra weight 4) Oil could leak out of system through seals, resulting in environmental pollution and reduction in reservoir volume 5) Catastrophic failure possible if bladder ruptures, therefore, a resultant loss of buoyancy control. Also, there isn't room in free-flood spaces for redundant backup bladders 6) Requires a hard external tank or other form of storage reservoir which would have to be located in the already limited free-flood space so that extra hull penetrations are avoided

Table 2-3. Movable piston concept (same concept as bladder, with following exceptions).

ADVANTAGES	DISADVANTAGES
1) Needs no oil 2) No pumps or valves 3) Simple, in terms of the number of moving parts	1) Piston would have to penetrate both hemispherical heads of pressure hull, since the vehicle midbody is an unacceptable location for cavities or protrusions in the hydrodynamic sense; therefore, hemispheres would require major reinforcement modifications to compensate for sharp increase in local stresses near the penetration. Result - a large increase in structural weight and difficulty in fabrication 2) Since the interior of the pressure hull must be maintained dry, the following must be considered: a) A dynamic seal at pressure hull hemi-head is very risky and difficult to design with the non-lubricated (seawater) axially sliding surface b) Seal forces to be maintained would be high and significant diameter piston has large seal area. This results in great amounts of work required to change volume, particularly at depth. Large, heavy electric motor required 3) Uniquely designated interior and free-flood space required for piston penetrations in the hemispheres (probably along main axis of vessel). Wasted space in most roomy areas of vessel

Table 2-4. Fixed weight dispensing system.

ADVANTAGES	DISADVANTAGES
1) Simple in concept 2) Few dynamic seals 3) Moving linkages used to dispense weights are not subject to high load/speed conditions	1) Mechanical linkages and/or dispensing hoppers required in the design 2) Possible hydrodynamic complications due to added penetration(s) in the forebody and afterbody fairings 3) Limited number of dives, depending on the amount of lead shot carried on board 4) Additional nonfunctional weight (the lead shot cargo) decreases the energy available for support of payload in favor of battery energy 5) Noncontinuous mass transfer method (i.e., finite buoyancy change per dropped weight) 6) Stored weights waste space in vehicle free-flood areas 7) Fouling of external linkages, hatches, and actuators possible during dropping of weights and inactive periods 8) Still must design a system which "takes on" water so that vehicle may dive again after dropping weights 9) Vehicle endurance is limited by the amount of weights that can be carried and available flood-tank size

Table 2-5. Seawater ballast system (with HP air flask for emergency blow).

ADVANTAGES	DISADVANTAGES
1) No oil and/or mercury reservoirs, therefore, space and weight saved	1) Seawater is both corrosive and abrasive as a working fluid. However, correct use of specially developed materials will reduce risks
2) Can be compact and lightweight in design, if proper components are chosen	2) Filtration of seawater and biofouling are potential problems
3) No hydraulic penetration to pressure hull hemi-head if independent fore and aft systems are used and also, no dynamic seals penetrate the pressure hull, thus lower mission risk. Due to 2) above, redundancy in critical components (valves and pumps) can be easily incorporated in design to improve system reliability	3) Galvanic corrosion considerations in selection of materials for subsystem components are more important due to open-loop type design
4) No pollution from leaking oil or dropped weights	4) Electric drive motor for pump must be protected from seawater by some means (either a hard boundary with magnetic coupling or an oil-pressure compensated design with a dynamic shaft seal)
5) If trimming and ballasting functions are combined and a reversible pump is chosen, the number of valves can be reduced to one per system	5) More pumping power is required against full pressure differential than for an air regulated system
6) Hard tank failure is unlikely, as compared to elastic bladders which are susceptible to punctures and material degradation due to long-term exposure	

Table 2-6. Seawater ballast system, HP air regulated (same as previous alternative with following exceptions).

ADVANTAGES	DISADVANTAGES
1) Reduce power to pump out tanks at depth. Also, a possible reduction in rubbing pressures on pump parts, therefore, long wear life	1) High-pressure air bottle weights reduce payload capacity of the vehicle
	2) HP air bottle and regulator add additional penetrations to the ballast tanks
	3) Number of dives limited by available air volume
	4) Requires another component, an air regulator, which is known to reduce reliability of systems seeing long-term exposure in the ocean environment
	5) Vessel control is lost if the HP air regulated portion of system fails, unless redundant systems are used. However, redundancy of air bottles in design reduces the amount of useful space within the vehicle

that of shifting ballast water between forward and aft tanks, is also undesirable due to additional valves and hull penetrations required. Therefore, controlling the angle of ascent/descent (i.e., pitch angle) should be accomplished by combined use of the vessel's control surfaces and independent Forward and Aft Variable Ballast Systems, rather than adding the complexity of another separate system.

2.5 Ballast System Selection

As a first cut, all systems requiring some form of expendable stores (such as lead shot weights or stored HP Air) to perform ballast evolutions are eliminated due to the long endurance requirements of this vehicle. Too many weights or many large air flasks would be needed to effect a sufficient number of operational cycles - this is unacceptable for a weight-critical vessel design since the vehicle weight devoted to batteries would have to be reduced in order to carry these cargo items. Also, the known failure rates for air regulators exposed for long periods to the ambient seawater environment (which are also of reasonable cost) indicate unreliability. Thus, both the fixed-weight dispensing and air-regulated (i.e., pressure compensated) ballast systems are unsuitable choices for this design.

For the bladder concept, the need to make efficient use of space and weight other than that associated with the payload is the key issue. Separate free-flood spaces would have to be allotted for both the bladders and oil reservoir tanks, possibly necessitating a change in hull shape to accommodate a redundant bladder. For much smaller vehicles, the change in buoyant volume required for vehicle control would have been much less; then perhaps, this concept would be suitable. However, the current design requires a variable buoyant volume on the order of 1000 cubic inches; thus, this concept is also not suited to design requirements.

With critical hardware such as the control computer and energy source to be kept dry, the movable piston appears to be an extreme risk to vehicle integrity. Also, to be capable of a 1000 cubic inch volume change, the pistons would be impractical in both size and in seal drag forces. Stress analysis and structural reinforcement in the pressure hull end caps housing these penetrations would become extremely critical, since dimensional tolerances on the order of 0.005 inch must be maintained around the piston seals through all hydrostatic loads.

Finally, there is left only one remaining choice - a conventional seawater ballast system. That is, to say, conventional in the sense of system operating theory. Very detailed attention was given to the search for components that would be needed to implement the approach effectively. Most of the submarines that were reviewed had variable ballast systems of one sort or another which utilize seawater as a pumped fluid. Some examples are: DSRV, Alvin, Deep Quest, Sea Cliff, Robot II, Leo, Pisces, and naval submarines. However, all of the vehicles mentioned are manned (some with ballast components inside the pressure hull), except for the Robot II, in which the VB System was changed to eliminate the complexity and high power consumption from a set of solenoid valves and a pump. (See Reference 2.) Data relevant to this particular design was obtained from some of the other vehicle systems design in the form of materials used to fabricate components, component-types, flow control methods, etc.

This seawater variable ballast system which evolved during this design effort is comprised of either "special applications" components or others of conventional "high-tech" design modified to extend their operability under an extremely stringent reliability requirement. The resultant system combines the use of these developed components and current design practices to produce desired vehicle performance levels over the duration of exposure to the usually harsh ocean environment.

SECTION 3

SYSTEM DESIGN

3.1 Assumptions

A number of assumptions (some of which impose restrictions on the design) have been established so that the extent and scope of this thesis is discretely defined. Without specifying particular constraints, this treatment of design practices for unmanned submersibles' ballast systems would rigorously address all related interfaces. For example, a buckling analysis on both ballast tanks (including penetrations) under hydrostatic loading would have to be performed; also, a detailed discussion of vehicle control in the vertical plane using different ballasting schemes would be needed. Since this thesis is intended to address only the implementation of hardware exposed to the ocean environment, a set of assumptions is definitely necessary.

The listing of assumptions below have either been specified by individuals in other design ares or by overall project requirements. Although the list is certainly not exhaustive, the issues pertinent to hardware design have been noted:

(a) No hydraulic or mechanical penetrations into the pressure hull are allowed. Specialty marine electrical penetrators will be used to transfer power outside the hull.

(b) Direct current electric power (voltage regulated) only; no ship service hydraulic or air systems are available for use in the vehicle.

(c) Required ballast tank volumes are 500 cu. in. each for forward and aft tanks due to predicated hull compression and the greatest variations in seawater density.

(d) Adjustments to the state of buoyancy shall only be performed when the vehicle has changed depth by more than 200 feet or experiences an equivalent change in buoyancy due to seawater salinity and temperature variations.

(e) The final system components should be able to function maintenance-free during the one-month endurance period.

(f) Control algorithms don't require a tight bias on seawater flow rates, but do require a minimum flow rate equal to 0.25 gal/min.

(g) No flow sensors are to be used in system. However, tank level pressure transducers may be used to monitor ballast weight (see Section 3.7).

(h) The vehicle is intended for use only in "open sea" areas, not in immediate proximity to coastal areas whose waters contain high amounts of entrained sand and silt.

(i) All components must be standard or modified designs for reliability concerns.

The use of these assumptions will appear during the design discussions to follow. Some are obviously applicable to specific components, while most actually affect the total system design.

3.2 Major Considerations in Design

Since nonelectrical hull penetrations are forbidden, the entire variable ballast system must be mounted in free-flooded areas, as shown in Figure 1 (two independent systems located forward and aft). Thus, the most important concern in designing a reliable system is its tolerance to the ocean environment. In addition to the problems that can occur with abrasive particulates in this pumped fluid, metals used in the various "seawater hydraulic" components are subjected to corrosion of varying forms. A great many references have been compiled and reviewed to ensure that proper materials were selected for use in this

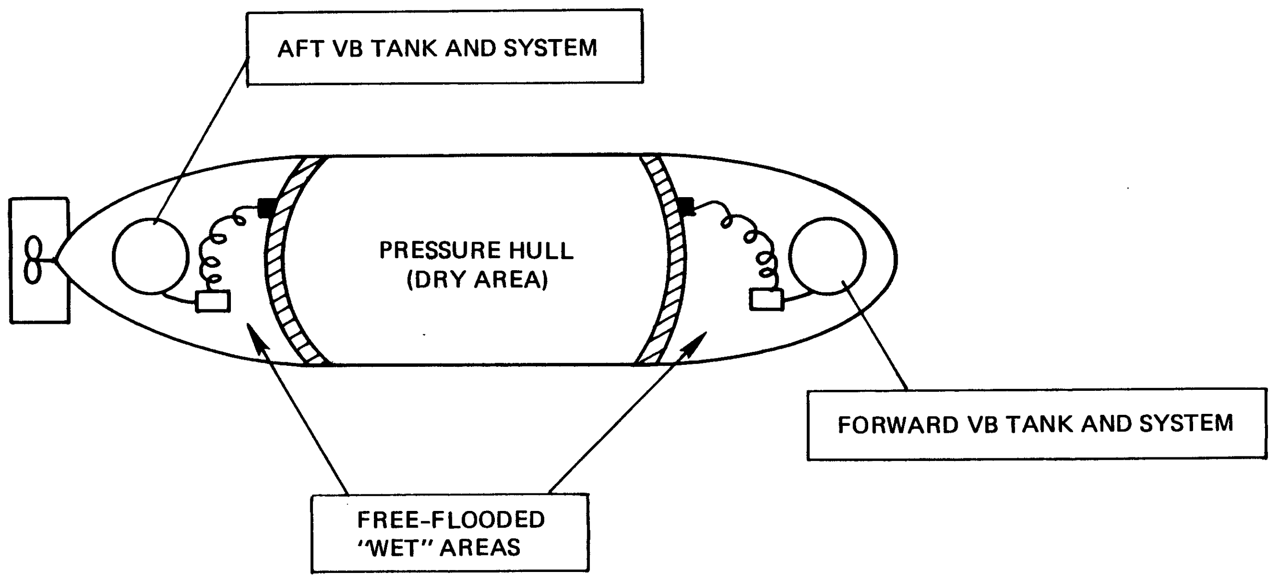


Figure 1. Sketch of variable ballast systems location in submersible.

design (see References 3 through 11). Alloys such as stainless steels, cupro-nickels, bronzes, and more exotic varieties have all been used in different kinds of seawater systems for submarines, surface ships, and other coastal and offshore applications. So a careful study was done to determine the reasons that some materials are a good choice for certain types of service, but not for others.

The main criterion on which to base the selection of materials is almost always cost. Those metals like carbon steel, aluminum, and zinc which are low in the galvanic series in seawater (Appendix A-1) are relatively inexpensive, but corrode at higher rates than do the more noble ones containing copper, nickel, chrome, or other alloying constituents. So, like many things in life, a "cheap" system needs replacement sooner than one built with more expensive materials. In view of the current market prices for nickel, copper, chrome, columbium, and other precious alloys, the issue of justifiable costs is even more critical in engineering decisions like this. A simplified "life-cycle cost" analysis has been performed on this system to show the general techniques that are employed by industry and the military to estimate the total cost of any system over its lifetime (see Appendix B). Although this design efort took costs into account, the requirements of operation placed on the system dictated the use of expensive materials for the "state-of-the-art" variety of system.

Corrosion rate is only one facet of the problems facing a seawater hydraulic system. Actually, local rates of corrosion depend on many factors like seawater flow rate, temperature, and contamination, as well as growth of fouling organisms and the physical shape and stress state of the components. A list of rules that should be generally followed in design appears in Table 3-1. Specific choices made for the system components will be covered by the individual discussions on each one that follow in Section 3. However, a general comment can be made about the use of steel - because low-cost, iron-based alloys corrode and foul at high rates, they are not suitable for dynamic components where surface finish is important (like a pump or valve), and the system is

Table 3-1. Corrosion "Rules of Thumb" for Seawater Service Designs.

1. Where possible, construct equipment from one metal. This eliminates galvanic coupling effects.
2. When not possible or desirable to construct from one alloy, make certain that the key components are more noble, i.e., cathodically protected.
3. Expect and allow for increased corrosion on the less noble metal by providing a large area or heavy wall to support the increased corrosion that will occur.
4. Consider carefully the galvanic effect before painting or coating less noble materials. It is frequently more desirable to coat the more noble material and leave the less noble one bare - a reversal of the more conventional procedure. This reduces the exposed cathodic area and thus the amount of current flowing between the couple, lowering the general wasting rate of the anode. Unless an anode coating can be made "holiday-free" (no pinholes or cracks) or be frequently maintained, local corrosion rates around the exposed areas will be greatly accelerated, inviting possible "surprise" pitting or blistering failures.
5. Avoid the use of alloys that are known to experience selective corrosion. Examples are: "dezincification" of certain brasses and bronzes, "graphitization" of irons, and integranular corrosion of austenitic stainless steels.
6. Ensure that the flow rates in the system are controlled to reasonable limits so that cavitation impingement damage is precluded. In a similar way, sharp corners and edges should be faired in valves or other components to reduce localized cavitation damage.
7. "Oxygen-concentration" cells should be avoided in any case, particularly when using stainless steels. Threaded areas should be sealed, and attachment of barnacles should be prevented from sensitive metals (deep pitting damage has occurred under barnacles in many metals). Fouling-resistant materials should be used for "periodic flow" applications.
8. Selection of fastener and pipe fittings should be done carefully. When noble fasteners are used on less noble bodies, a nonmetallic washer should be placed between the two in order to increase the resistance to current flow between the coupled materials. (See Appendix A-2.)
9. The relative areas of exposed metals with different galvanic potentials should be closely evaluated.
10. Copper or nickel-based alloys (or 6A1-4V titanium, if costs are justified) should be used to prevent the occurrence of corrosion fatigue in cyclically-stressed parts.

installed on a vehicle that will see repeated service. Even though higher cost stainless sheets (like 304 or 316L) have shown good performance in some constant flow services, they are entirely unsuitable for this design due to:

1. Their sensitivity to local oxygen and stress concentrations in a given area, and
2. A tendency to pit in stagnant flow conditions (see References 12 through 14).

3.3 Seawater Pump

The heart of any seawater ballast system which does not use pressurized air or other auxiliary medium to force water out of the hard tanks is its seawater pump. Since this system was selected not be pressure regulated (for a simpler, more reliable design), the pump unit needs to be capable of delivering the minimum required flow rate when working against the full hydrostatic pressure head at operating depth. In other words, the seawater pump specifications are based on the "worst case" delivery parameters. No "precharging" of the ballast tanks' air was included in the design for safety reasons; in fact, the air volume that will always remain in the tank (see Section 3.7) will remain within 80 $lb/in.^2$ of atmospheric pressure at all times. Overall System Operation (3.8) covers this in more detail, but the tank pressure is obviously needed to determine the actual pressure requirements for the pump. Assuming a maximum pressure of drop of 50 $lb/in.^2$ for control valve, filter, and piping which comprise the rest of the system, and a specified maximum weight of 7 lb from the naval architects, the pump's key requirements can be summarized as follows:

(a) Maximum presure head - 500 $lb/in.^2$
(b) Minimum flow rate - 0.25 gal/min
(c) Maximum weight (in air) - 7 lb
(d) Material compatibility with seawater

An exhaustive study of all commercially available pumps which could meet or be modified to meet the design parameters of the ballast system was done. Most were found to be unsuitable in general for the following reasons, arranged according to the type of pump:

(a) Any centrifugal-type pump of reasonable size and weight was not nearly capable of delivering the maximum pressure head required.

(b) Gear pumps which could begin to approach the required pressure head were found to wear out quickly even in finely filtered seawater due to the tight meshing tolerances required to develop such a head. Also, a reasonably high flow rate could not be delivered.

(c) Diaphragm pumps meeting flow and pressure requirements were also heavy (roughly 50 to 100 lb).

(d) Screw (Archimedes) pumps were all very large in size and equally as heavy.

(e) Multiple ceramic piston pumps with Hastelloy cylinder liners would be capable of meeting the flow and pressure parameters, although they weigh almost 20 lb each. In addition, these reciprocating devices are typically quite noisy and would at least require special enclosures to attenuate the noise and vibration. Also, their crankshaft bearings are heavily loaded and require oil lubrication.

In addition to commercial sources, oceanographic and military industries with submersible experience were also researched. Findings from this portion of the quest were also not satisfactory for this aplication, i.e., the pumps were too heavy, too noisy, or incapable of developing the pressure head within the desired endurance limit. Finally, a special seawater hydraulic motor designed to power hand tools for Navy divers was encountered and considered as a possibility. This motor produces power from high pressure (1200 $lb/in.^2$) seawater supplied to a vane motor which uses high strength "TORLON" plastic vanes and Inconel 625 alloy rubbing parts to gain superior performance levels without any lubricants other than seawater.

Now entering its fifth year of development and testing at the Naval Civil Engineering Laboratory in Port Hueneme, California, the seawater motor has proven to be quite successful and reliable during testing. It is also very compact, lightweight (23 cu in. and 5 lb, respectively) and had an overall efficiency of 70% in testing. If the role of the unit is reversed (i.e., drive the unit's output shaft with an electric motor and pump seawater), it would be most suitable for this application, possessing such desirable features as reversiblity (therefore requiring only one control valve in the ballast system design) and dynamic balancing (potentially much quieter than a piston pump, and gives the TORLON shaft bearings a lower, more uniform load). Details of the motor development program, design features, lubricating and cooling theory, etc., are discussed in great detail in References 15 and 16. Similar work has been performed by the Japanese (see Reference 17).

This unit functions similarly to conventional oil hydraulic vane motors, except that the rotor shaft bearings and vanes made from TORLON (manufactured by Amoco Chemicals) are self-lubricating due to teflon and graphite fillers. Heat is removed from the running surfaces by the internally-ported flow of seawater leakage. Figure 2 shows a simplified sketch of the theory of operation; however, the sketch doesn't show the Egiloy alloy springs which hold the tips of the vanes in contact with the Inconel cam ring. These springs are the primary cause of failure in the motor testing to date as a combined result of the abrasion and fatigue corrosion induced by salt water. Work is continuing in this area at NCEL to improve this situation.

A brief discussion on the choice of wearing materials is proper at this point. First of all, Inconel 625 is probably one of the most useful superalloys being applied in critical, "high first-cost" designs requiring high strength, surface hardness, wear resistance, inertness, and good fatigue strength in seawater. Many U.S. Navy submarines are in the process of being backfitted with Inconel parts for some critical propulsion shafting components. Its high nickel and chromium content

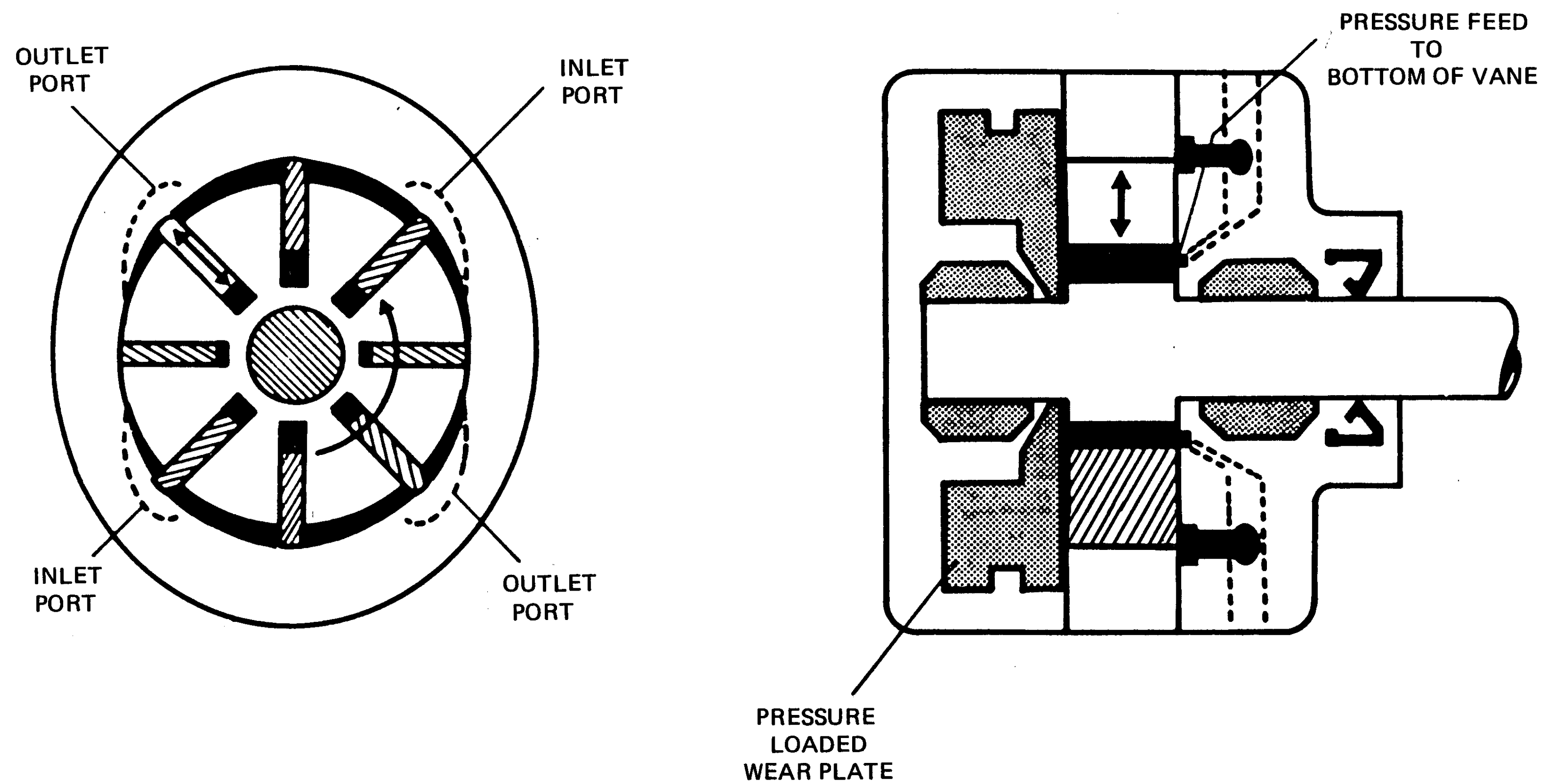


Figure 2. CEL balanced vane concept.

gives the alloy an R_c hardness of 35 (annealed) and a galvanic potential near 0.0 volt in seawater. Reference 18 verifies excellent resistance of this alloy to corrosion fatigue and pitting under endurance test conditions. Also, Reference 19 advises the use of high nickel content alloys for more efficient, low maintenance pumps - this makes Inconel 625 an obvious choice for a small, reliable seawater pump.

The choice of TORLON to be used as a vane and bearing material came as a result of friction and wear testing of selected plastics in seawater performed by Mechanical Technology Inc. (see Reference 20). The material specimens were run in both continuous-sliding and reciprocating modes under test conditions of high load and speed against Inconel and other candidate materials. TORLON 4301 running against Inconel 625 was finally selected for the NCEL motor application due to the low coefficient of friction (0.025) and low wear rates. Because the material's base resin is a polyamide-imide, it is capable of very high bearing stresses, while the 12 percent graphite-3 percent PTFE lubricant fillers give it the low friction characteristics. Low friction means less heat generation and lower starting torques. Abrasive particles present in the fluid will not embed themselves into the plastic (thus forming a "grinding wheel" effect) as with softer teflon varieties, but rather, will roll between the hard metal and plastic surfaces until "washed out" through leakage slots that are designed into the bearings and reciprocating vanes.

Some preliminary testing of the NCEL unit operating as a seawater pump was performed in December 1982. Appendix A-3 shows the results from those tests which ran the pump through a range of speeds against several pressures from 100 lb/in.^{2} to 1000 lb/in.^{2}, and measured values of torque, r/min, and flow rate. Although the results are very encouraging (considering that no modifications to the basic motor were made), the power required to drive the pump efficiently at 500 lb/in.^{2} (which produced a minimum flow of 2 gal/min) was on the order of 1.5 hp. That would require quite a large motor, larger that the propulsion motor, in fact. Therefore, a minor redesign effort has been initiated which is aimed at reducing the cam ring eccentricity, and thus the length of vane stroke. The resultant goal established is to reduce the displacement (flow rate) of the unit to one-eighth its current value at 1000 r/min. Power consumed by the pump should then be around 1/4 hp, which is satisfactory for all those concerned. A beneficial byproduct of the redesign is that sliding wear of the vanes is reduced. Also, the spring forces can be reduced and alternating stress levels in fatigue are lower, thus improving the fatigue life of these "primary failure" items. Including improvements like a plastic "wear sleeve" around each spring to increase its life, the final design of this pump is expected to have a mean operating life greater than 500 hours and an approximate efficiency of 60%.

Filtration of the incoming seawater will be discussed in a later section, but the issue of fouling must be mentioned here. Beause there is a "mechanical cleaning" effect that occurs during each pumping cycle, the initial films that can form on materials and "grow" into biological fouling during stagnant periods are destroyed several times each day. Thus, no concern to internal fouling needs to be given here, as opposed to the case of filters and piping where low water velocities cannot be relied upon as a cleaning mechanism.

Additional operating data is to be obtained for the lower flow redesign during the summer of 1983, when the pump will be run in both directions - pumping out against hydrostatic pressure and as a flow restrictor (water brake or motor) flooding pressurized water into the tank. The unit should be able to keep the flooding rate in the piping within velocity limits (see Section 3.5). If all goals for this redesigned seawater pump are achieved, this component will serve quite satisfactorily as the "workhorse" of this particular seawater variable ballast system.

3.4 Flow Control Valve

In order to ensure that no water will leak into the ballast tank when the pump is inactivated, a flow control valve has been included in the system design. Leakage through bearing slots and around vanes in the pump will vary as a function of depth (ambient hydrostatic pressure). Therefore, the valve that was needed had to be capable of opening against a maximum hydrostatic pressure of approximately 450 $lb/in.^2$ while still providing a good seal under very nearly atmospheric pressures. (Section 3.8 will discuss the operating sequence of the valve in more detail.) Of course, the valve must use electrical power only for actuation and be tolerant of the seawater's corrosive and abrasive nature per the previously stated assumptions. Very few manufacturers sell valves suitable for all the particular requirements of this application, but one outfit, Vacco Industries, has had a good deal of experience designing and building submersible valves. A Vacco design that is very similar to a "Deep Quest" valve was found to be ideal.

In keeping with the overall vehicle design goals of low weight, compactness, and reliability, the valve-type that was selected had to be one which included a lightweight, watertight direct solenoid-operation package. The Vacco Industries' valves which have been made for other customers, like Lockheed and the U.S. Navy, all have stringent reliability standards which compare to those desired for the solenoid- operted poppet valves needed in this design. Since the design is proprietary, it cannot be discussed in detail. However, the main features making this the best choice over other alternatives can be discussed briefly.

First, it is obvious that a solenoid-operated design is desirable from weight, space, and power standpoints as long as the coil is designed to withstand many cycles in the ambient environment without shorting. This is the case with the Vacco valve that uses specially insulated coils in the solenoid that will be isolated from both ambient and pumped fluids. In addition, the valve will fail shut (preventing further flooding of the tank and loss of buoyancy) via a return spring if electrical power to the solenoid is lost. Other valve types like globe (ball) and gate flow controls would need a much larger, heavier motor actuator. Gate valves are also quite sensitive to contaminants, and usually cannot seal well at high pressures. Spool valves used in oil hydraulic systems would certainly gall or seize in seawater use because the working fluid is used to pilot-operate the spool. The poppet design applied here is of the "pressure-balanced" variety, meaning that the solenoid pull force on the poppet doesn't vary with line pressure drop across the valve seat (i.e., as depth of vehicle changes). Appropriate high performance seat materials such as Viton or other polymers will be used to ensure reliable sealing in aggressive environment. The whole valve weighs approximately 5 lb.

Due to concerns similar to those discussed in the previous section, 70-30 copper nickel was specified as the valve body and plunger material for pitting, corrosion-fatigue, and other resistant properties. The dynamic wear situation is far less severe than with a pump, so more exotic (and more expensive) high-nickel content materials like Inconel and Monel were not necessary. A 3/8-in. nominal line size was selected to match the pipe size chosen in Section 3.5 and the maximum pressure drop through the valve will be 10 $lb/in.^2$. A valve of this size will corrode slightly in the presence of the Inconel pump.

3.5 Seawater Intake Filter and Piping

Unlike the internal surfaces of the pump and valve, the intake filter and piping are subject to possible growth of a biofouling film, barnacles, and other forms of marine life without the advantageous "wiping" effect of moving parts. Clogging of the filter should not be a problem (if biofouling growth can be controlled), since the 150-micron mesh screen will be "backflushed" each time seawater is pumped out of the ballast tank. The 150-micron mesh size was selected by comparing the typical distributions of particulate matter in the sea to the filter sizes and applications for other in-service saltwater systems (see References 21 and 22). An excellent paper (Reference 23) on "Marine Biofouling of Synthetic and Metallic Screens" describes two biofouling tests performed over several years in Wiscasset, Maine and Woods Hole, Massachusetts. The results showed the excellent performance of 90-10 copper nickel screen in resisting the growth of biofouling, even over many months of exposure. This resistance is owed to the cuprous oxide corrosion product film that forms on copper-nickel alloys in seawater. Copper-containing bottom paints function on the same principle.

Other general research sources on fouling were reviewed to better understand the process and occurrence of fouling (see References 24, 25, and 26). Some general conclusions can be stated from the review of the literature:

(a) All unprotected materials will foul in seawater. However, the copper alloys (70-30 and 90-10 copper nickels) will usually support little total growth even after several years of exposure.

(b) Aside from using copper alloys, various degrees of biofouling control can be attained by the use of antifouling coatings (like cuprous oxide or tributyl tin oxide), mechanical cleaning of the surface, or environmental control (like chlorination, elevated temperature, or high water velocities). Some of these methods may have undesirable effects on corrosion behavior though.

(c) Sacrificial anodes (like zinc) used for corrosion protection are not desirable because they negate the resistance of copper alloys' resistance to biofouling. This is attributed to the suppressed solubility of copper ions (the active fouling toxins) by the galvanic couple formed with the anodes.

For this system design, mechanical cleaning is not feasible and the inner diameter of small pipes cannot be coated; a water heater would certainly be a waste of energy since an auxiliary heat source would be required. Finally, since the system's operation is infrequent and high flows cannot be maintained, the remaining choice to be made is whether to use 70-30 or 90-10 copper nickel as a piping and filter screen material.

The tradeoff to be made here is a subtle one. Due to high market prices for nickel, the 70-30 alloy is more expensive than 90-10, but is also more noble in the seawater galvanic series because of higher nickel content. The galvanic potential is an important issue since a sizable area of the cathodic Inconel 625 alloy is exposed to the seawater "electrolyte." [Note: Only the internal valve and pump metal need be considered as a cathodic area, since an antifouling paint will be applied to the external surfaces of these two components. Also, cost may be neglected for this design, since very little piping (2-3 ft) is used in each ballast system and will not affect the overall cost of the system too much.] Therefore, since all literature on the subject emphasizes the superior fouling resistance of the 90-10 copper nickel (References 27 and 28), it was chosen as the pipe and filter material. A "heavy-duty" grade of pipe with maximum wall thickness will be used to account for increased corrosion due to coupling affects with the roughly equal area of Inconel (the small area of 70-30 CuNi in the flow control valve will have a minor affect on the couple). Note that the filter will be of cylindrical shape to minimize the screen flow velocities.

Reference 29 is an excellent guide for the use of 90-10 CuNi piping and lists a maximum peak flow velocity of 6 ft/s to be allowed in the piping for preserving the protective corrosion film. Based on this mean flow velocity, a 3/8-in. pipe size (thick-walled tubing actually) was chosen to suit the expected pump volume flow rate of 0.25 to 1 gal/min. This ensures a reasonable Reynolds Number for the flow if smooth bends and other standard installation practices are used, as well as a minimal pressure drop through the piping.

3.6 Electric Drive Motor

As noted earlier in Section 2.5, the motor that is needed to drive the seawater pump must be protected from the ocean environment by either a "pressure-compensated," oil-filled housing with a dynamic rotary shaft seal, or a rigid, lightweight housing with magnetic shaft coupling. The major drawback of the pressure-compensated approach for any extended operating mission is that an extra reservoir of pressurized hydraulic fluid needs to be provided to replace oil leakage that will inevitably occur through the shaft seal. The internal windings of the rotor must be reliably insulated from the oil as well. By contrast, a rare-earth magnetic coupling can be effectively implemented in this design without need for pressure compensation inside the housing due to moderate hydrostatic pressures. Several dc brush motors have been built and tested by Giannini Petro-Marine (San Diego, Calif.) for use as propulsion motors and actuators on a number of submersibles. However, a dc brushless motor was selected for this application to eliminate problems due to brush wear, fluid arcing, and resulting wear particle contamination of the fluid. Also, a brushless motor with a magnetic coupling only requires maintenance when shaft bearings wear out. These motors are also lighter and more efficient than their brush-type counterparts.

An excellent article, "Brushless dc Motors Improving Underwater Propulsion," discusses the advantages of this type of design, as well as the motor control theory which utilizes "Hall Effect" devices to electronically commutate the moving rotor (see Reference 30). Several manufacturers currently offer a line of brushless motors and the task of obtaining a custom-built unit through Giannini Petro-Marine has been assigned to the propulsion motor and drives group. Per discussions and meetings with Giannini, motor specifications for "pumping out" direction of rotation are as follows (assuming a 60% efficient pump):

Shaft horsepower	=	0.25 hp
Shaft speed	=	1000 r/min
Motor weight	=	4 in. diameter
Motor size	=	5 in. length

The speed requirements for the motor operating in the reverse direction (i.e., tank flooding mode) will be determined when relevant pump performance data is obtained from pressure vessel testing in July 1983. The housing material shall be 6 Al-4V titanium (externally coated for galvanic protection of other components) to ensure lightweight, low maintenance and stress corrosion resistance.

Note: Electrical power connections for both valve solenoid and drive motor shall be made with neoprene-jacketed underwater cables and connectors that are rated for the 1000-ft operating depth.

3.7 Ballast Tank

From the hydrostatic calculations done by the naval architects, the total floodable tank volume of 1000 $in.^3$ was split into two equal 500-$in.^3$ tanks to be located forward and aft (this was shown schematically in Figure 1). Therefore, a spherical tank 10 in. in diameter was designed by the structural group under a hydrostatic buckling load criterion and 6 Al-4V titanium alloy was specified as the material for construction due to its many suitable properties. This alloy is widely used in commercial submersibles for similar applications. Basically, the fabrication procedure is to TIG weld two hydro-formed hemispheres to a reinforced split flange. A static seal and piping penetration are an integral part of this flange assembly, which may be unbolted during maintenance periods to facilitate inspection. A neoprene-rubber coating containing a TBTO biofouling toxin shall be applied to the inner and outer surfaces of the tank so this large cathodic area will be galvanically decoupled from the other VB system components (the toxin, of course, inhibits the growth of fouling organisms). Use of this product* on submarine sonar domes, ship hulls, etc., has proven that an 80-mil thick layer will provide good fouling protection for five years. Due to the excellent pitting and corrosion resistance of the alloy, no problems are expected with the materials system, even if the coating is damaged while the vehicle is operating. Repairs are simple and quick.

As was mentioned previously in the assumptions, this thesis has addressed the environmental and operational issues that impacted the design of a VB system for unmanned submersibles. Therefore, no attempt has been made to show any structural analyses on tank, penetrations, or welds. The schematic diagram shown in Figure 3 illustrates the primary operational features of the tank. A comparison of the readings from pressure transducers 1 and 2 (which are sensor inputs to the control computer) will facilitate an accurate determination of tank water level when the vehicle is operating in the horizontal plane. The author has added a weighted snorkel/flexible bellows assembly which can pivot about

*"No-Foul Rubber, B. F. Goodrich Company

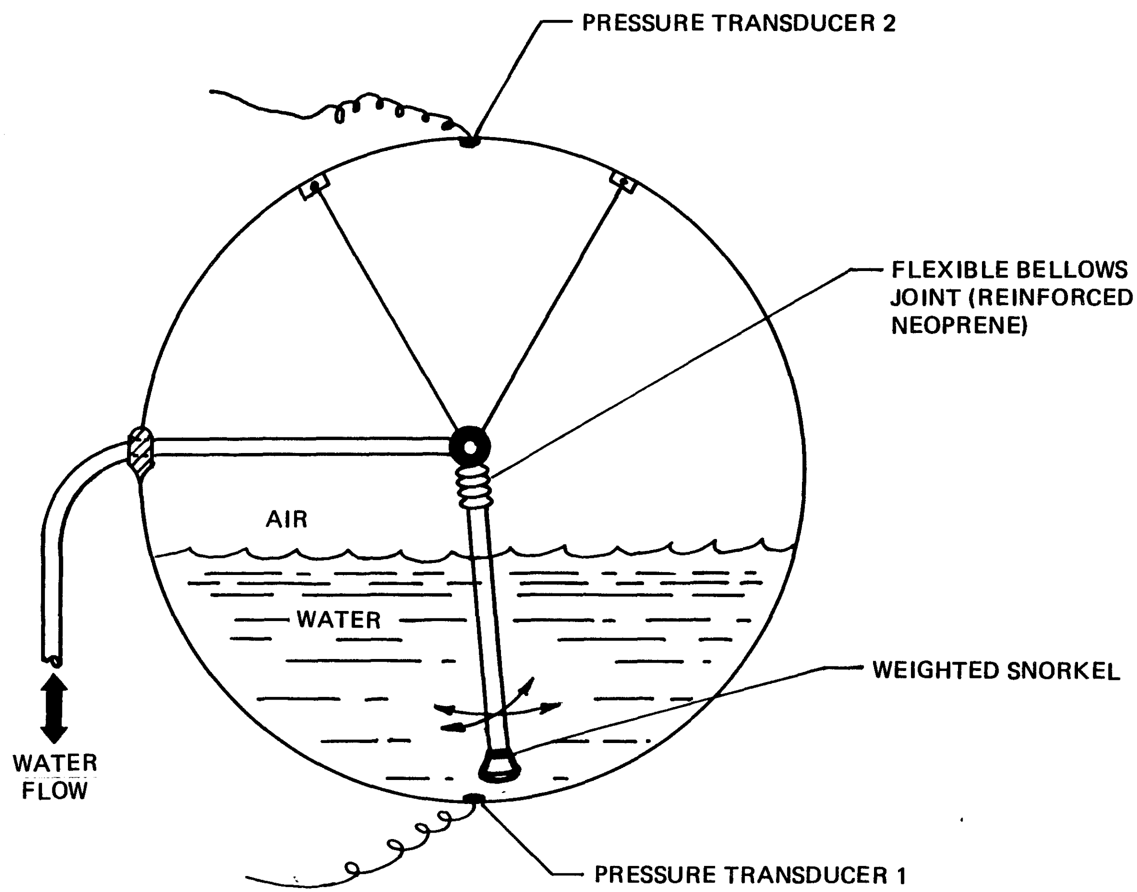


Figure 3. Ballast tank schematic diagram.

the tank origin in two degrees of freedom. As the vehicle may assume various attitudes of pitch and roll during operation, this concept will ensure that only water will be pumped out of the ballast tank when water levels are low. Losing the volume of air would result in a loss of vehicle - this disastrous possibility must be precluded. The neoprene joint is internally reinforced to withstand the slight suction force developed (when pumping out) that would otherwise tend to collapse it. Once again, functional simplicity and reliability are the determining factors behind the selection of this concept over a water-filled bladder approach.

3.8 Overall System Operation

The volume of water in either (or both) forward or aft tanks is varied to maintain an approximate (within vehicle limits for minimum energy consumption) condition of neutral buoyancy for the entire vehicle. This condition is maintained during submerged operation in two independent vehicle operating modes - constant depth, and diving/ascending. Each of these modes is discussed below.

Constant Depth Mode - During normal cruising modes where the automatic control logic does not commany any major changes in operating depth, the Variable Ballast (VB) system will typically remain inactive; any depth control required due to a change in seawater density (i.e., change in vehicle buoyancy) will be accomplished with fin effectors. However, if the energy consumed in fin actuation and drag becomes excessive, the vehicle control system will stop forward propulsion and monitor any changes in depth from gauge sensors to estimate the amount of ballast water necessary to be added (or removed) from the ballast tank(s) to reestablish proper vehicle trim and/or neutral buoyancy. Note here that water absorption of nonmetallic hull materials, accumulation of foreign debris, fouling, etc., could cause the vehicle to be more buoyant aft than forward (or vice versa) thus requiring a trim adjustment. A command is then issued to forward and/or aft VB systems to pump the estimated quantity of ballast water into or out of the tank.

Figure 4 is referred to here so that the system operation can be diagrammatically illustrated. Upon receipt of this command, the solenoid valve opens and the pump is activated for a specified amount of time* to transfer water. As the water volume inside the tank changes, the air volume present inside the "hard" tank will also change inversely; this of course changes the vehicle's net buoyancy. Pumping will cease and the control valve closed shut after the prescribed lapse of time; then depth sensors are again monitored to check if buoyancy is within the desired limits. If not, the process is repeated; if near-neutral buoyancy is obtained, the propulsion motor is again activated and the cruising mode is resumed. Tank level pressure transducers are also monitored as redundant sensor information to correct pump rate bias errors.

Diving/Ascending Mode - Although most of the previous description also applies to this mode in general, there is one essential difference between the two - changes in the vehicle state of buoyancy at constant depth are infrequent and dependent on the environment, while ballast adjustments are necessary each time the vehicle performs a major ascent, descent, or surfacing operation, (in this case, any required change in depth greater than 25% of the vehicle design depth). Thus, a commanded dive from the surface to design depth would involve four separate operations on the way down to compensate for the predicted hull deflections under hydrostatic load. Each ballasting evolution must be performed individually during a dive in a stepwise fashion so the rate of descent can be reliably controlled. Calculations of pressure hull volume changes supplied by those designing this part of the vehicle indicate that the hull is more compressible than the ocean water column. Therefore, water must be periodically pumped from the ballast tanks to obtain added buoyancy in order for the vehicle to maintain stability when diving. The converse is also true when ascending, due to the undesirable shock levels that might be imparted to the vessel if a broaching maneuver occurred. Most of the time the VB system will be activated only when changing depth.

*This time will be dependent on the redesigned pump flow rates at 1000 r/min as a function of ambient pressure.

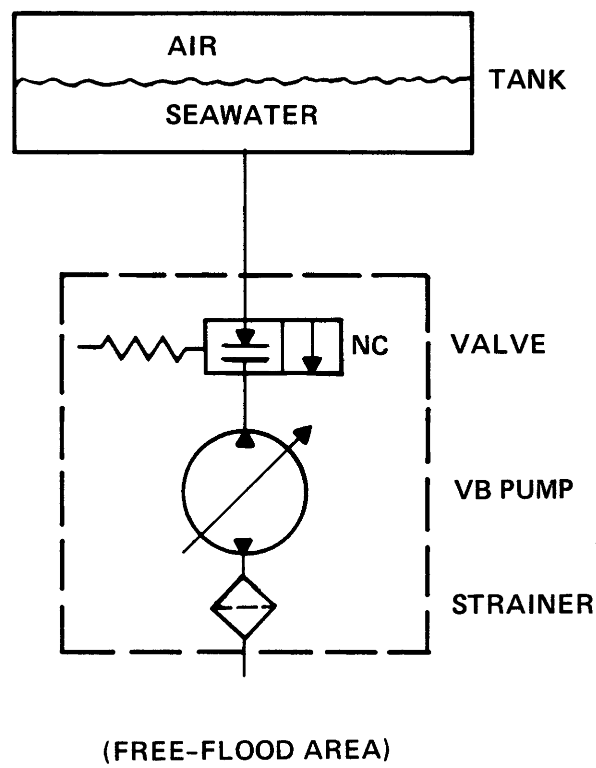


Figure 4. Seawater hydraulic system diagram, HPUV variable ballast baseline design.

If no depth changes are ordered in a given day, the system will be briefly cycled in both directions so that fouling growth inside the pump and valve is prevented. A schematic diagram of the system components has been drawn as a concluding illustration for the System Design section, with the exception of the ballast tank shown previously. Although Figure 5 is not drawn to scale, it gives a clearer picture of the individual components than does the hydraulic system diagram in Figure 4.

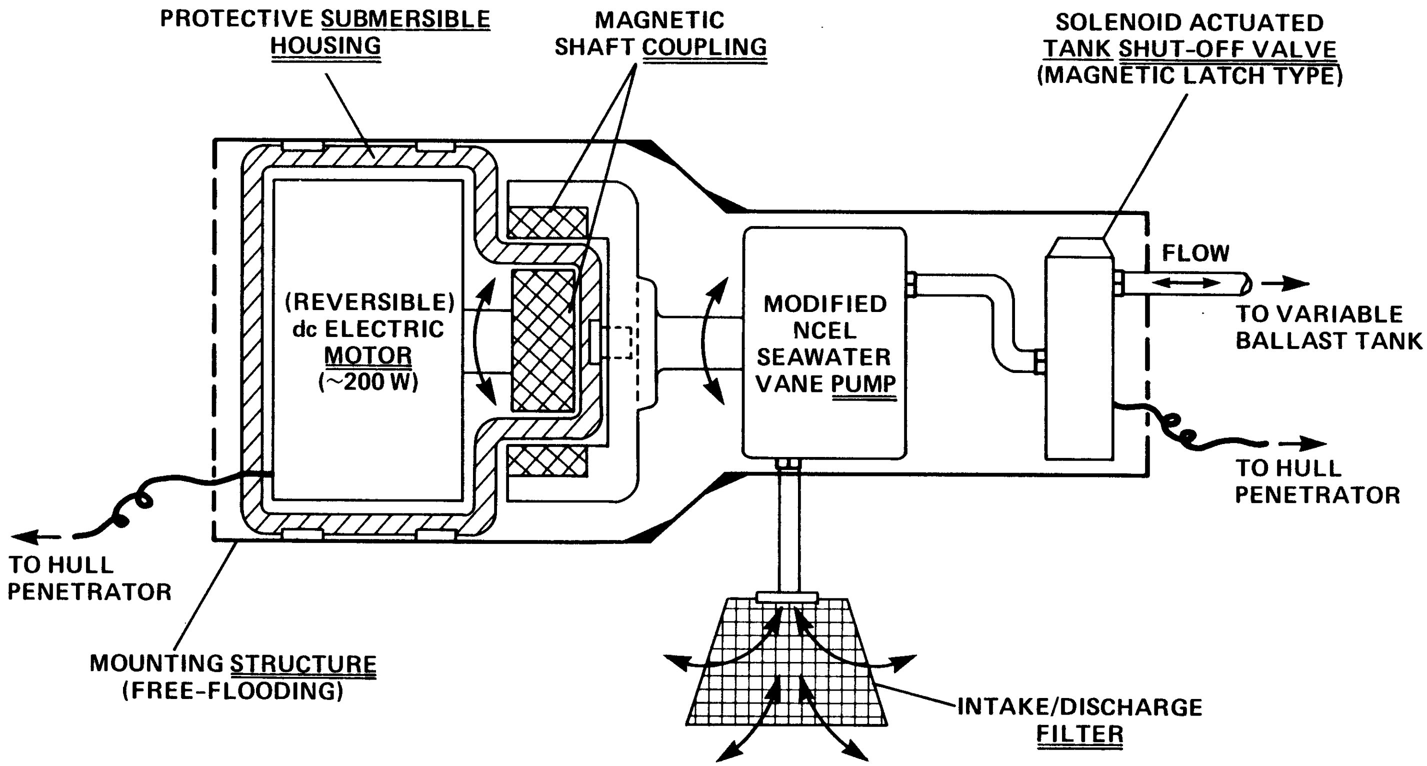


Figure 5. Representative variable ballast system schematic diagram.

SECTION 4

CONCLUSIONS AND AREAS FOR FUTURE WORK

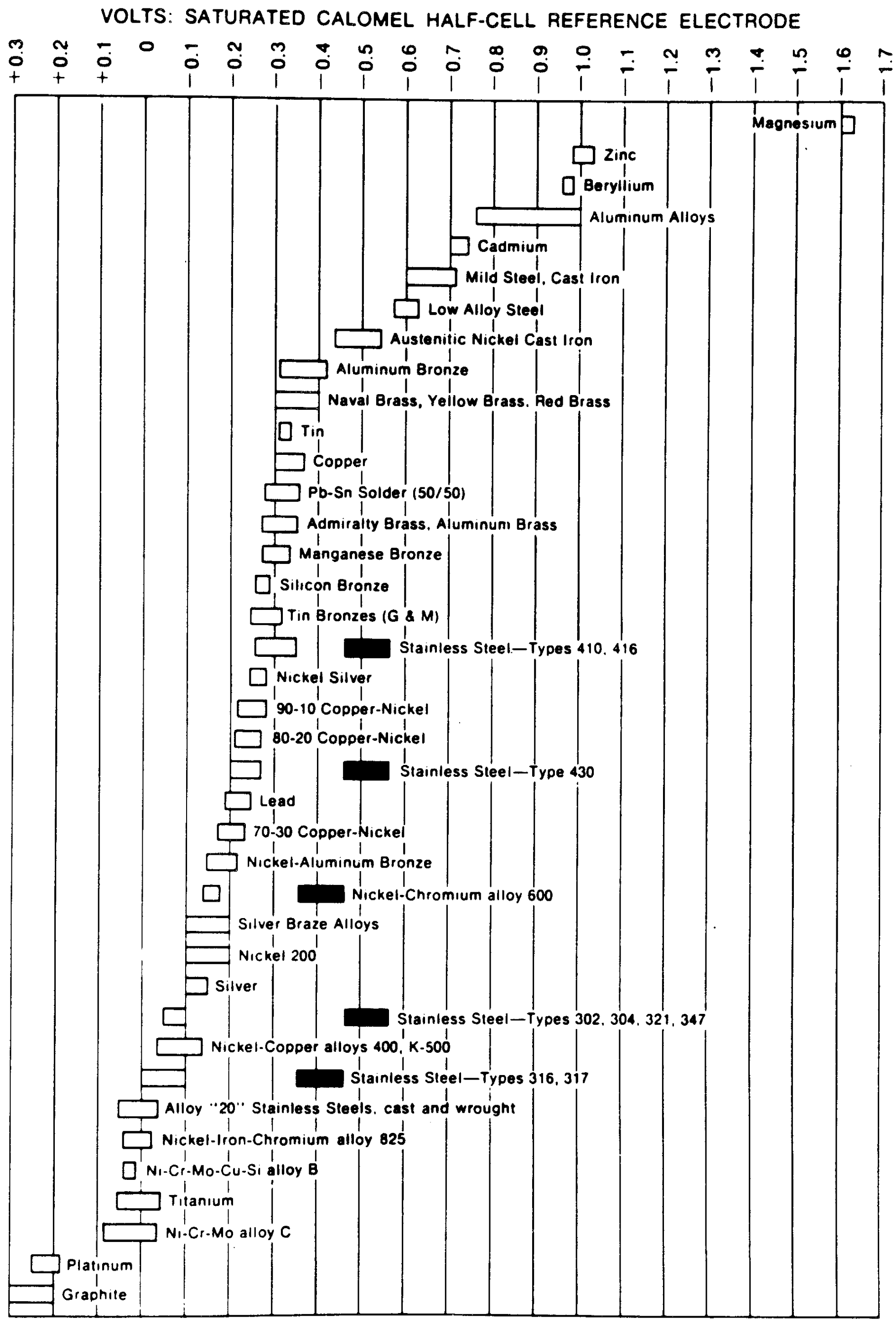
The components which are described by the previous sections comprise a system that is certainly a high first-cost, low maintenance approach to the variable ballast system design for this unmanned submersible. All components were selected and/or modified primarily on the basis of functional simplicity and previous operational tests or applications. In addition, the required maintenance items to be performed over the life of the submersible can be easily accomplished by trained personnel. Expected reliability for the system is 98% under the one month endurance requirement and series of demonstration tests in the open ocean should verify that the overall capabilities of the VB system are proposed.

In order to protect the control valve, piping, and seawater intake filters from erosive cavitation damage, the static leakage testing of the pump in a submerged, hydrostatically pressurized condition must be carried out. If the results of the testing do not show a sufficient degree of flooding control (i.e., pressure drop) through the pump, a fixed-orifice flow restrictor will be added to the system so that maximum flow velocity within the piping is kept below 6 ft/s at all submerged depths. Future work in developing a longer-lived set of pump vane springs could increase the useful life of the pump unit significantly and the work which continues at NCEL along these lines should be closely followed.

APPENDIX A-1

CORROSION - POTENTIALS IN FLOWING SEAWATER

CORROSION — POTENTIALS IN FLOWING SEA WATER
(8 TO 13 FT./SEC.) TEMP. RANGE 50°-80°F



Alloys are listed in the order of the potential they exhibit in flowing sea water. Certain alloys indicated by the symbol: ■ in low-velocity or poorly aerated water, and at shielded areas, may become active and exhibit a potential near −0.5 volts.

APPENDIX A-2

GALVANIC COMPATIBILITY (SEAWATER FASTENERS)

BASE METAL ↓	FASTENER → Aluminum[1]	Carbon Steel	Silicon Bronze	Nickel	Nickel-Chromium Alloys	Type 304	Nickel-Copper Alloy 400	Type 316
Aluminum	Neutral	Comp.[2]	Unsatis-factory[2]	Comp.[2]	Comp.	Comp.	Comp.[2]	Comp.
Steel and Cast Iron	N.C.	Neutral	Comp.	Comp.	Comp.	Comp.	Comp.	Comp.
Austenitic Nickel Cast Iron	N.C.	N.C.	Comp.	Comp.	Comp.	Comp.	Comp.	Comp.
Copper	N.C.	N.C.	Comp.	Comp.	Comp.	Comp.	Comp.	Comp.
70/30 Copper-Nickel Alloy	N.C.	N.C.	N.C.	Comp.	Comp.	Comp.	Comp.	Comp.
Nickel	N.C.	N.C.	N.C.	Neutral	Comp.[3]	Comp.[3]	Comp.	Comp.[3]
Type 304	N.C.	N.C.	N.C.	N.C.	May Vary[4]	Neutral[3]	Comp.	Comp.[4]
Nickel-Copper Alloy 400	N.C.	N.C.	N.C.	N.C.	May Vary[4]	May Vary[4]	Neutral	May Vary[4]
Type 316	N.C.	N.C.	N.C.	N.C.	May Vary[4]	May Vary[4]	May Vary[4]	Neutral[4]

(1) Anodizing would change ratings as fastener.

(2) Fasteners are compatible and protected but may lead to enlargement of bolt hole in aluminum plate.

(3) Cathodic protection afforded fastener by the base metal may not be enough to prevent crevice corrosion of fastener particularly under head of bolt fasteners.

(4) May suffer crevice corrosion, under head of bolt fasteners.

NOTE: Comp. = Compatible, Protected. N.C. = Not Compatible, Preferentially Corroded.

APPENDIX A-3

PRELIMINARY PUMP PERFORMANCE CURVES

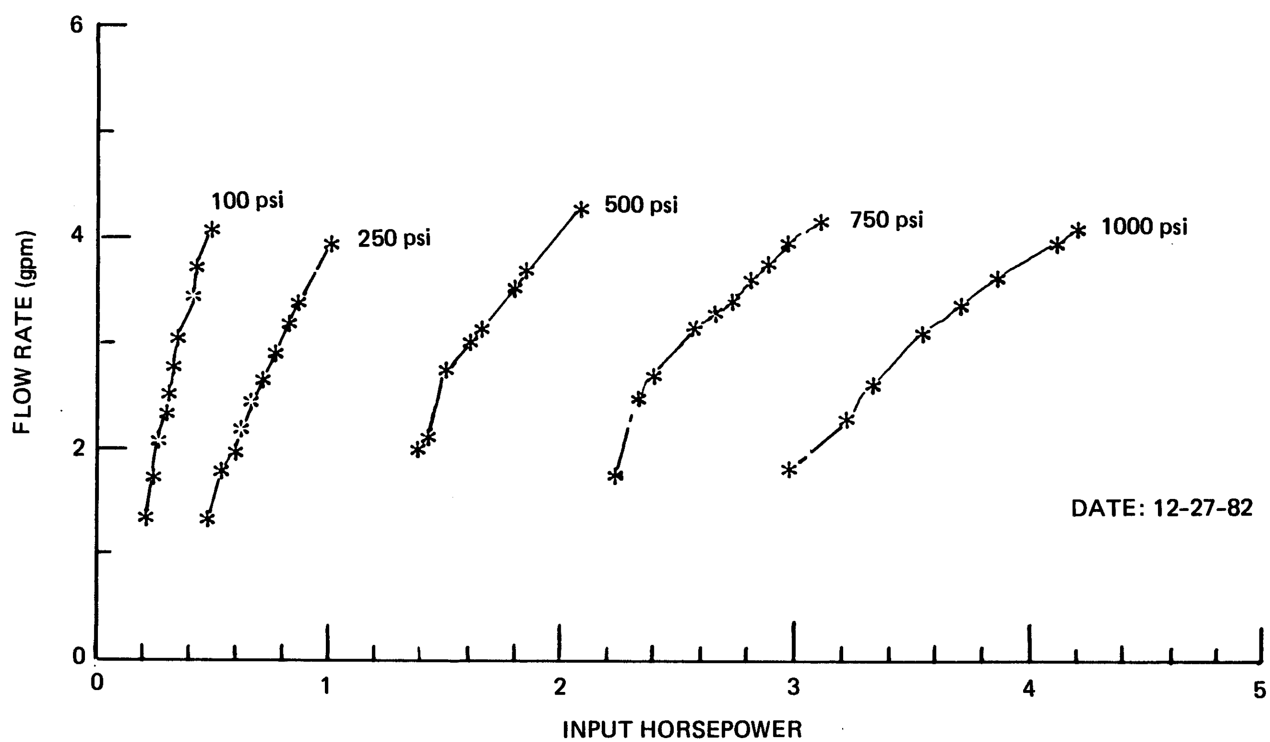


Figure A-1. Sea water pump test - zero suction head.

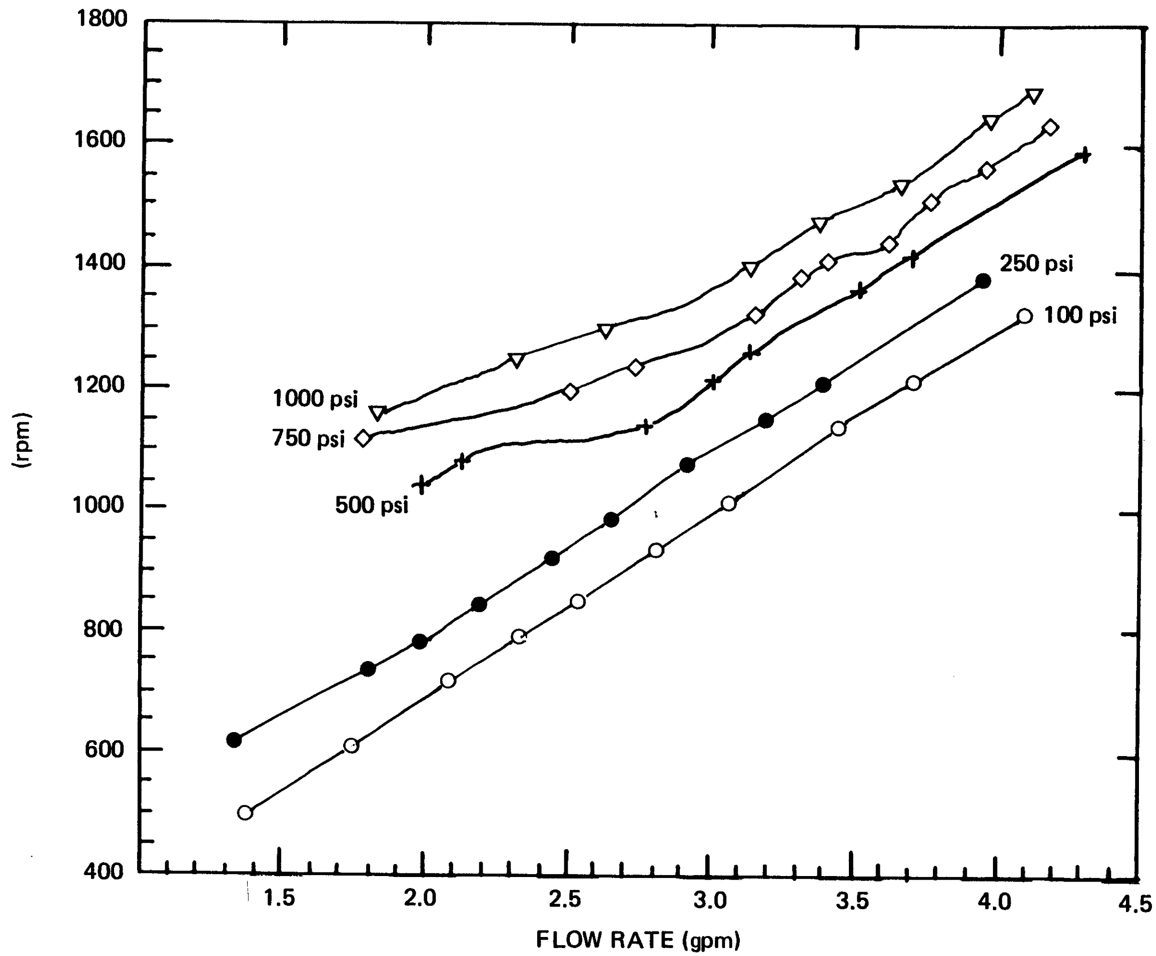


Figure A-2. Sea water hydraulic pump tests preliminary results - rpm.

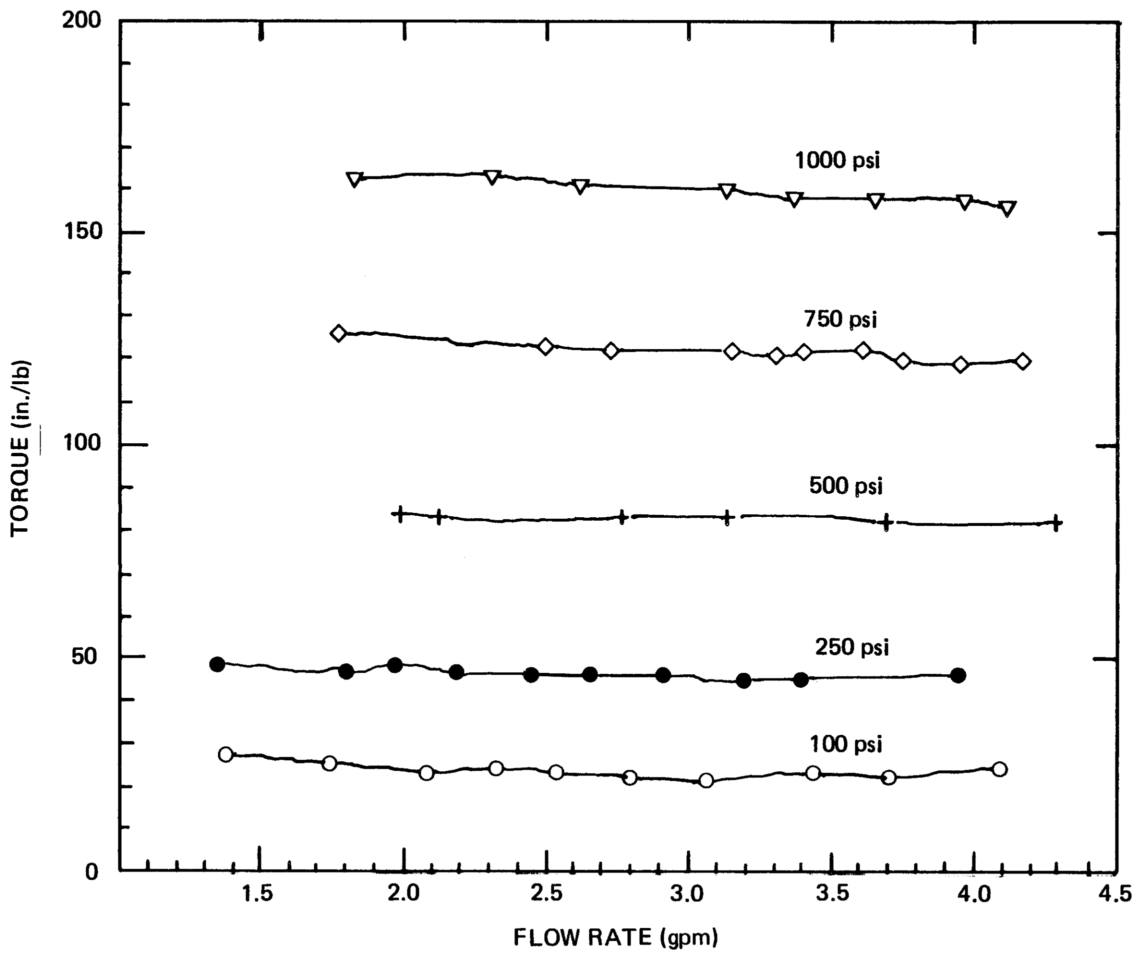


Figure A-3. Sea water hydraulic pump tests preliminary results - torque.

APPENDIX B

A SIMPLIFIED LIFE-CYCLE COST ANALYSIS OF THE VARIABLE BALLAST SYSTEM

Purpose

To provide a current estimate of the total cost for the Variable Ballast system over the 3-year (assumed) life span of the submersible.

Assumptions

1. Vehicle will be used for two assignments per year.
2. The costs of energy (dc power) to operate the drive motor and solenoid valve are covered by vehicle power source recharging and/or replacement estimates, since the VB system uses a minor percentage of the vehicle's energy.
3. Rate of inflation is assumed to be 10% annually (this is conservative).
4. All fabricated parts costs are assumed to increase in accordance with inflation.
5. Labor costs will increase annually at a rate that exceeds inflation by 5% and the base rate (1983) is $25 per hour.

Procedure

1. From the cost breakdowns on Page 54, showing the estimated periodic maintenance items for each component, a total annual material cost will be compiled. Constant 1983 dollars can be assumed since the price of fabricated parts will increase in accordance with inflation.
2. The annual costs of labor shown will be increased to reflect net actual costs relative to constant 1983 dollars.

3. Initial purchase prices of all components are summed to yield invested capital.

4. After all three items above are totaled up to reflect an expected life-cycle cost for one individual system, a factor of two will be applied to the expected cost in order to complete the total cost estimate for the vehicle. (There are two independent systems, one forward and one aft.)

5. Labor costs shown include disassembly, cleaning, parts replacement (when applicable), inspection, reassembly, and testing.

Initial Component Purchases (1983 prices):

Pump	- \$ 5,000
Valve	- \$ 4,000
Motor	- \$ 5,500
Filter	- \$ 2,000
Piping	- \$ 500
Tank	- \$60,000
Connector Cables (4)	- \$ 4,000

\$81,000 = Total Capital per system

Labor Cost Adjustment - use a compound amount factor for 5% relative increase in price for three payment periods (years) = $(F/A)_3^{0.05}$ = 3.153

Maintenance Cost Totals are = 3 (parts cost/yr) + 3.153 (labor cost/yr)
= 3 (2355 + 1150 + 100 + 200 + 500 + 1700 + 700)
+ 3.153 (1125 + 1000 + 500 + 300 + 750 + 750 + 250)
= \$20,115 + \$14,750 = \$34,855 per system

Therefore,

Estimated Life-Cycle Cost = (\$81,000 + \$34,855) X 2 systems

d \$232,000 in 1983 dollars

ANNUAL MAINTENANCE COSTS

System Component	Replacement Parts and Frequency/Year	Fabricated Parts Cost	Total Annual Parts Costs in 1983	Total Annual Labor Costs in 1983
Seawater Pump	Vanes (20), twice/yr Shaft Bearings (4), twice/yr Springs (40), twice/yr Bearing Side Plates (4), twice/yr Inconel 625 Cams (2), twice/yr	\$ 330 \$ 125 \$ 40 \$ 280 \$1600	= \$2355	45 hours × (\$25/hr) = \$1,125
Solenoid Flow Control Valve	Seats and Seals, twice/yr Plunger, once/yr Solenoid Coil, once/yr	\$ 250 \$ 400 \$ 500	= \$1150	40 hours × (\$25/hr) = \$1,000
DC Drive Motor	Shaft Bearings (2) Static Seals, both once/yr	\$ 90 \$ 10	= \$ 100	20 hours × (\$25/hr) = \$500
Seawater Intake Filter Assembly	Screen Element, once/yr	\$ 200	= \$ 200	12 hours × (\$25/hr) = \$300
Seawater Piping	Replace once/yr	\$ 500	= \$ 500	30 hours × (\$25/hr) = \$750
Ballast Tank	Suction Pipe, once/yr Seals, once/yr	\$1200 \$ 500	= \$1700	30 hours × (\$25/hr) = \$750
Miscellaneous Items	Bolts, Nuts, Fittings, Paint, Coatings	\$ 500 \$ 200	= \$ 700	10 hours × (\$25/hr) = \$250

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